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Free and forced convection on the outer surface of vertical longitudinally finned tubes

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ABSTRACT

The paper presents the results of experimental studies of the heat transfer process on the outer surface of longitudinally finned tubes. Experimental values of heat transfer coefficients under free and forced convection conditions were calculated for low air flow velocities. Two air flow patterns were analysed, i.e. airflow parallel and perpendicular to the axis of the pipe. The mean values of heat transfer coefficients obtained during the study amounted to 2–8.5 ($W\,m^{-2}\,K^{-1}$) under free convection conditions, 4.5–19 ($W\,m^{-2}\,K^{-1}$) for forced transverse flow and 4–11 ($W\,m^{-2}\,K^{-1}$) for forced convection along the axis of the tube. Experimental values of heat transfer coefficients for transverse flow around the tube were 22% higher than heat transfer coefficients for airflow along the axis of the tube. The paper lists the dimensionless relationships which can be used to calculate the values of the heat transfer coefficient under free and mixed convection conditions, with transverse and longitudinal air flow around the outer surface of vertical longitudinally finned tubes.

1. Introduction

Heat exchangers with enhanced surfaces can be used as heaters, coolers, air coolers, and cross-current regenerative plate heat exchangers. The exchangers are commonly used in cooling and air-conditioning technologies, as well as in the automotive industry and power engineering. A heat transfer surface enhancement on the fluid side with low heat transfer coefficient values and relatively low flow resistances offsets that side's low values of overall heat transfer coefficients in heat exchangers with a moderate increase of fan pumping power and while maintaining the compact structure of the heat exchanger. The heat transfer surface may be enhanced in a variety of ways and the exact method results from the structure and operating conditions of the exchanger. In cooling and air-conditioning heat exchangers, transversally finned tubes are used if the air flows perpendicularly to the tube axis. Straight fins are used in plate exchangers to delineate the channels through which gas flows among the plates of the exchanger [1]. Longitudinally finned tubes, as well as membranous surfaces are used in power engineering, in steam air heaters, convection steam heaters and water heaters [2]. In such exchangers, tubes are arranged horizontally and the fluid flows transversely to the axes of the tubes.

Flat (continuous) fins on array of tubes are also used as heat pump evaporators, with external air as the cold heat source. Such exchangers most often consist of horizontal tubes and external airflow is forced by fans. This structure of the evaporator guarantees compactness; however, it leads to intensive frosting of the exchanger and therefore to decreased thermal efficiency of the heat pump (due to energy loss resulting from fan operation and defrosting of the exchanger).

The forced flow of gas transversally to the axis of individually finned or lamellated (continuous fins on an array of tubes) tubes makes heat transfer coefficients on the gas side relatively high. For transversally lamellated tubes, the heat transfer coefficients fall within the range of 10–80 $W\,m^{-2}\,s^{-1}$ (or front velocities of 0.25–3 $m\,s^{-1}$). For lamellated tubes they are ca. 12% lower than for tubes with circular fins (according to the equations included in [3], assuming the same front velocity, geometrical parameters and the 2.6 ratio of substitute fin height). In the case of longitudinal flow around membrane tubes, the heat transfer coefficients amount to 10–100 $W\,m^{-2}\,s^{-1}$ (for average flow velocities of 1.2–7 $m\,s^{-1}$, depending on the tube distance scales) [4]. For the same flow velocities and tube distance scales, the heat transfer coefficients in the case of a staggered tube arrangement are ca. 20% higher than in the case of an in-line arrangement.

Thanks to the loose vertical arrangement of longitudinally finned tubes [5] in the heat pump evaporator, no fan needs to be installed. In this case, air flow is only dependent on external conditions which may reflect free or forced convection conditions.

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Nomenclature

A,B,C,D,E	coefficients in dimensionless Eqs. (16), (19) (–)
$A_{\text{“indeks”}}$	surface area (m^2)
A_{fin1}	surface area of the cross-section of a fin $A_{fin1} = s_1 \cdot L(\text{m}^2)$
A_{o1}	surface area of a smooth fin at the base of the fin in relation to a single fin $A_{o1} = (\pi \cdot d_{o1}/n_{fin} - s_1) \cdot L(\text{m}^2)$
c_p	specific heat (J kg^{-1} K^{-1})
c_{pa}	specific heat of wet air (J kg^{-1} K^{-1})
d	diameter (m)
F_1,F_2	constants in Eq. (16)
g	gravitational acceleration (m s^{-2})
G	density of the mass stream of a coolant (kg m^{-2} s^{-1})
Gr	Grasshof number, $Gr = \frac{\beta \cdot g \cdot d_{o1}^3 \cdot \Delta T \cdot \rho^2}{\mu^2} (-)$
h	specific enthalpy (J kg^{-1})
h_{fin}	fin height (m)
h_g	specific enthalpy for saturated water vapour (J kg^{-1})
i	specific enthalpy (J kg^{-1})
k	thermal conduction coefficient (W m^{-1} K^{-1})
L	tube length (m)
Le	Lewis number, Le = Pr/Sc
L_p	length of the measurement channel (m)
n_{fin}	number of fins (–)
Nu	Nusselt number, $Nu = \frac{\alpha \cdot d_{o1}}{k} (-)$
Pr	Prandtl number, $\Pr = \frac{\mu \cdot c_p}{k} (-)$
p_w	partial pressure of water vapour in the air (Pa)
R	heat transfer resistance (m^2 K W^{-1})
Ra	Raylegh number, $Ra = Gr \cdot \Pr = \frac{\beta \cdot g \cdot d_{o1}^3 \cdot \Delta T \cdot \rho^2}{\mu^2} \cdot \frac{\mu \cdot c_p}{k} (-)$
RCJ	degree of process openness (–)
Re	Reynolds number, $\text{Re} = \frac{W_{\max} \cdot d_{o1} \cdot \rho}{\mu} (-)$
$\dot{Q}$	heat capacity (W)
$\dot{Q}_s$	sensible heat capacity (W)
$\dot{Q}_V$	visible heat capacity (W)
s	fin thickness (–)
S_q	tube distance scale (m)
T	temperature (°C)
T_f	end of fin temperature (°C)
T_{walloi}=T_{woi}	temperature of the outer wall of the tube, temperature at the base of the fin at the “*i*” sensor installation point
U	overall heat transfer coefficient (W m^{-2} K^{-1})
V	volume (m^3)
$\dot{V}$	volumetric flux (m^3 s^{-1})
W_a	maximum air velocity (m s^{-1})
W_{ain}	front air velocity (m s^{-1})
$\overline{X}_a$	mean air humidity degree for air at a mean temperature of $\overline{T}_a$ (kg kg^{-1})
$\overline{X}_{wall}$	mean degree of humidity of saturated air at a mean temperature of $\overline{T}_{wallo}$ (kg kg^{-1})
$\Delta T = \overline{T}_a - \overline{T}_{wallo}$	temperature difference in the definition of Grasshof number

Symbols

Δ	increment
α	heat transfer coefficient (W m^{-2} K^{-1})
α_a	heat transfer coefficient of dry air (W m^{-2} K^{-1})
β	volumetric expansion coefficient (K^{-1})
β_α	mass transfer coefficient (kg m^{-2} s^{-1})
ε	fin efficiency (–)
ϕ	diameter of the measurement channel (m)
μ	dynamic viscosity coefficient (Pa s)
η	condition line coefficient (W W^{-1})
ρ	density (kg m^{-3})

Indices

1	fin base
2	fin tip
a	air
Al	aluminium
calc	calculated
Cu	copper
cz	front
fin	fin
HEAT	heater
i	inner
in	inlet
k	forced convection
loc	local value
m	mean
mII	method II
o	outer
out	outlet
R	coolant (refrigerant or water)
s	free convection
t	smooth tube
w	water
wall	wall
–	mean value

A key issue related to designing such evaporators is calculating the air side heat transfer coefficient, which determines the overall heat transfer coefficient and the heat transfer surface area. In the case of lamellated or transversely finned horizontal pipes there are various equations which make it possible to calculate the air side heat transfer coefficient, including those by Schmidt [3], Norris and Spofford [6,7] Wang and Chi [8] Stasiulevisius and Survila [9] and Briggs and Young [8,10] Ganguli and Yilmaz [8]. In the case of longitudinally finned tubes, it is necessary to differentiate between tubes fitted with single ribs or membrane tubes and tubes where ribs are situated centrally and symmetrically along the circumference of the entire tube. For both types, various reference works propose equations for transverse air flow around a bank of tubes. Papers [2] and [11] present the methods used for calculating local and mean heat transfer coefficients for transverse air flow around horizontal membrane tubes or double-finned tubes under forced convection. Papers [4] and [12] describe the sublimating naphthalene method used to determine the heat transfer coefficient for forced transverse flow of exhaust gas around a horizontal triple-ribbed bank of tubes. Paper [13] presented the results of experimental studies on the impact of the location of the fins (at the front or back of the tube), as well as of fin thickness and the geometry of their tips on the intensity of the heat transfer process and the values of flow resistances in the case of transverse air flow around longitudinally-finned tubes. Similar solutions applicable to tubes with an elliptical cross-section (30mm × 15 mm) were discussed in [14]. The study was carried out to investigate single vertical tubes with a height of 125 mm, on which 20 mm fins were mounted at the front or the back, as well both at the front and at the back. The research made it possible to develop dimensionless equations for heat transfer coefficients and flow resistances for transverse air flow under forced convection conditions.

The influence of a single longitudinal straight fin located on the outer surface of a horizontal cylinder on the natural convection heat transfer process was analysed in [15]. Tolpadi and Kuehn [15] found a slight influence of the fin on the averaged value of the heat transfer coefficient with respect to a smooth tube. The results of their studies indicated differences in the measured local