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Thermal optimization of vertically oriented, internally finned tubes in natural convection

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ABSTRACT

The thermal performance of vertically oriented, internally finned tubes in natural convection is optimized analytically. The total heat transfer rate is selected as the objective function for the optimization under a constraint of given base-to-ambient temperature differences. In order to predict the total heat transfer rate, a new correlation of the heat transfer coefficient is developed using the asymptotic method. In order to validate the new correlation, experiments are performed for internally finned tubes with different geometries. With the new correlation, the three parameters of fin geometry, i.e. fin thickness (t), fin height (H), and number of fins (n_{fin}), are optimized under given tube sizes of tube diameter (D) and tube length (L). Finally, closed-form equations for the optimal fin geometry are proposed as design guidelines for internally finned tubes with various sizes: $t_{\mathrm{opt}} = 5.44 \times 10^{-3}D$, $H_{\mathrm{opt}} = 0.388D$, and $n_{\mathrm{fin,opt}} = 0.588(D/L)\mathrm{Pr}^{1/4}Ra_{\mathrm{L}}^{1/4}$.

1. Introduction

With the development of long-life electronic devices such as LED lamps, cooling performance and high reliability are required for electronic cooling devices. Natural convective cooling devices are widely used in these applications due to their simplicity and high reliability [1,2]. Internally finned tubes are a promising solution when the heat sources are located on the outer surface of a tube and the fins cannot be directly attached to the outside. As the number of fins increases, the rate of heat transfer increases as a result of the additional surface area. If too many fins exist, however, the rate of heat transfer decreases because the flow resistance increases significantly. Therefore, an optimal number of fins exists that can be determined using the two competing factors associated with increased numbers of fins.

While many studies have been performed for internally finned tubes in forced convection [3,4], limited numerical studies are available for internally finned tubes in natural convection. Prakash and Liu [5] reported that internally finned tubes could yield significant heat transfer enhancements compared with finless tubes in natural convection based on their numerical results. According to their results, internally finned tubes exhibited up to five times higher thermal performance than finless tubes as the number of fins increased. However, the optimum fin geometry was not proposed in their study: they focused on the fin geometries in which the thermal performance increases as the number of fins increases. This indicates that only the effect of the surface area increment, which is one of the competing factors associated with the thermal performance of internally finned tubes, was investigated in their study. Therefore, a more reliable analysis that considers both the favorable effects of increased surface area and the adverse effects of flow resistance is required in order to predict the optimum fin geometry for internally finned tubes.

In the present study, vertically oriented, internally finned tubes in natural convection are optimized analytically. There are three parameters of fin geometry: fin thickness, fin height, and number of fins. For the thermal optimization, a new correlation of the heat transfer coefficient, which is expressed in terms of the three fin geometry parameters, is developed using the asymptotic method, and it is validated with experimental results. In the asymptotic method, two limiting cases are assumed: the fully developed limit where the number of fins is large and the isolated limit where the number of fins is small. In order to complete the analysis, the correlating exponent appearing in the composite relation of the asymptotic method is determined using experimental results for the intermediate region between the two limiting cases. Using the proposed correlation for heat transfer coefficient, the fin geometry is optimized under given tube sizes and base-to-ambient temperature differences. Finally, using the results obtained from the thermal optimization, design guidelines for the optimum fin geometry are proposed in closed-form equations, which are expressed in terms of the dimensionless parameters of the tube sizes and Rayleigh number.

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Nomenclature

A	surface area [m^2]
c_p	specific heat [kJ/kg-K]
D	finned tube diameter [m]
g	standard acceleration of gravity [m/s^2]
H	fin height [m]
h	convective heat transfer coefficient [W/m^2 K]
K	permeability [m^2]
k	thermal conductivity [W/m-K]
L	finned tube length [m]
n	number of fins [–]
Nu	Nusselt number [–]
P	pressure [Pa]
Pr	Prandtl number [–]
Q	heat transfer rate [W]
Ra	Rayleigh number [–]
R_{th}	thermal resistance [K/W]
T_∞	ambient temperature [K]
T_b	base temperature [K]
T_w	surface temperature of cylinder [K]
t	fin thickness [m]
u_D	Darcian velocity [m/s]
V	pore velocity [m/s]

Greek symbols

α	thermal diffusivity [m^2/s]
β	volumetric thermal expansion coefficient [1/K]
ε	emissivity [–]
η	fin efficiency [–]
θ_c	angle between adjacent fins [rad]
μ	dynamic viscosity [N-s/m^2]
ν	kinematic viscosity [m^2/s]
ρ	density [kg/m^3]
σ	Stefan–Boltzmann constant [W/m^2 K^4]
φ	porosity [–]

Subscripts

array	array
b	un-finned base of the tube
bottom	bottom of the cylinder
conv	convection
cross	cross section
cyl	cylinder block
f	fluid
fin	fin
ft	finned tube
fully	fully developed limit
heater	heater
isolated	isolated limit
rad	radiation
s	solid
top	top of the cylinder
total	total
w	wall of the cylinder block

2. Correlation of the average heat transfer coefficient

In this study, the asymptotic method proposed by Churchill and Usagi [6] was used to propose a new correlation of the heat transfer coefficient for vertically oriented, internally finned tubes in natural convection. The schematic and design parameters of the problem are depicted in Fig. 1. The heat is supplied to the outer surface of the finned tube. Air flows in the x-direction and removes heat from the internal surface of the finned tube.

For internally finned tubes, there are two limiting cases associated with the number of fins [7]. When the number of fins is large, the fins are highly populated, and the flow generated along these highly populated fins becomes fully developed for most of the region, except a short entrance region. Therefore, this case is referred to as the fully developed limit and is presented in Section 2.1. When the number of fins is small, the spacing between the fins is large. If this spacing is significantly larger than the thickness of the thermal boundary layer, the effect of the adjacent fins on each other is negligible and the fins can be regarded as isolated. Therefore, this case is referred to as the isolated limit and is discussed in Section 2.2. These two limiting cases are analyzed independently, and the correlations for these cases are combined in Section 2.4 to yield a composite relation that is applicable to an entire range between the two limiting cases. The exponent appearing in the composite relation is determined through an experimental investigation, which is described in Section 2.3.

The definition of the average heat transfer coefficient of a fin array is as follows:

$$\bar{h}_{fin} = \frac{Q_{array}}{A_{array}(\bar{T}_{fin} - T_\infty)} = \frac{Q_{array}}{A_{array}\eta_{fin}(T_b - T_\infty)}, \tag{1}$$

where $\bar{h}_{fin}$, Q_{array}, A_{array}, $\bar{T}_{fin}$, T_∞, η_{fin}, and T_b are the average heat transfer coefficient of the fin array, the rate of heat transfer from the fin array, the surface area of the fin array, the average temperature of the fin array, the ambient temperature, the fin efficiency, and the base temperature, respectively. The total heat transfer rate from a finned tube and its thermal resistance are evaluated as follows:

$$Q_{total} = Q_b + Q_{array} = (\bar{h}_b A_b + \bar{h}_{fin} A_{array} \eta_{fin})(T_b - T_\infty), \tag{2}$$

$$R_{th} = \frac{T_b - T_\infty}{Q_{total}} = \frac{1}{\bar{h}_b A_b + \bar{h}_{fin} A_{array} \eta_{fin}}, \tag{3}$$

$$\eta_{fin} = \frac{\tanh(mH)}{mH}, \tag{4}$$

$$m = \sqrt{\frac{2\bar{h}_{fin}}{k_s t}}, \tag{5}$$

where Q_{total}, Q_b, $\bar{h}_b$, A_b, R_{th}, H, k_s, and t are the total rate of convective heat transfer from the finned tube, the rate of heat transfer from an un-finned base, the average heat transfer coefficient of an un-finned base, the surface area of an un-finned base, the thermal resistance, the fin height, the thermal conductivity of the solid, and the fin thickness, respectively. Heat transfer from the fin tips is neglected simply because it is estimated to be less than 2% of the total heat transfer rate.

2.1. Fully developed limiting case

In the fully developed limiting case, highly populated plate fins are modeled as a porous medium. This porous medium has a hollow cylindrical shape with an outer diameter of D and an inner diameter of $D - 2H$. The governing momentum equation for a porous medium, i.e. the Darcy equation, is as follows [8]: