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Mixed convective heat transfer from vertical fin array in the presence of vortex generator

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ABSTRACT

Vortex induced non-Boussinesq laminar mixed convective heat transfer in a shrouded vertical non-isothermal fin array has been investigated numerically taking into account the variable transport properties. Computations are performed for the parametric range: Grashof number = $4.3–8.4 \times 10^5$, Reynolds number = 1280–2667, dimensionless fin spacing = 0.2–0.5 and dimensionless clearance = 0.0–0.25. Computations demonstrate inclusion of vortex generator results in a rise of pressure drop across the duct length to a value as high as 150% which is compensated by the substantial enhancement in heat transfer, as high as 200% locally and 20% globally compared to without vortex generator. Finally, overall Nusselt number is well correlated with the governing parameters of the problem.

1. Introduction

To release heat rapidly and efficiently from the engineering equipment thermal management demands special attention. On one hand, failure of engineering devices (e.g., electrical transformer, air-cooled engine, nuclear reactor, miniaturized electronic equipment) depends highly on the limiting temperature. On the other hand, rapid and efficient heat removal from a system like solar heater/nuclear reactor makes the system more effective. Therefore, there remains an ever-growing need for the rapid and reliable heat removal from the miniaturized system to achieve the desired goal. This invites researchers to look for a better thermal management solution.

Fin/extended surfaces are often used as an additive to the base surface to enhance the heat transfer. Rectangular fins find common application for its simplicity in fabricating the same and therefore following literatures are based on rectangular fin. There exist several literatures [1–15] on extended surface, which contribute significantly to natural convection heat transfer. Natural convection heat transfer process is prevalent in all the systems. But, limited transport of heat due to natural convection pushes researcher to opt for forced convective heat transport as a viable solution. Hydraulically and thermally developed laminar forced convective heat transport over shrouded non-isothermal fin array is computationally visited by Sparrow et al. [16]. Sparrow and Kadle [17], and Kadle and Sparrow [18] report both computational and experimental finding of forced convection turbulent heat transport from shrouded fin. The computational study is however limited to 2-D fully-developed flow. Significant flow by-pass through clearance was indicated by the authors. Writz et al. [19] and El-Sayed et al. [20] reported experimental finding of turbulent heat transfer over fin array. Heat transfer result by El-Sayed et al. [20] is culminated in the form of correlation. Recently, Elshafei [21] experimentally review the issue of flow by-pass from shrouded fin array.

Miniaturization of engineering system introduces a remarkable rise in heat generation. Under such circumstances, the effect of natural convection coupled with forced convection may play pivotal role. A computation of fully developed mixed convective heat transport from a shrouded vertical isothermal plate finned array glued to a horizontal base is reported by Acharya and Patankar [22] for a limited case. Maughan and Incropera [23–24] delineate the limitations of the results reported by Acharya and Patankar [22]. Dogan and Sivrioglu [25–27] further extend the work of earlier authors [22–24] both computationally and experimentally for the case of entry region flow. Recently Das and Giri [28] reports fully-developed mixed convection from downward projecting fin numerically. There exists a series of computational studies [29–32] on mixed convective heat transport from a shrouded vertical fin array on a vertical base. Both Zhang and Patankar [29], and Al-Sarkhi et al. [30] highlight the case of fully developed mixed convection, while Giri and Das [31] and Das and Giri [32] investigate the case of the entry region mixed convection.

To cope up well with the demand, additional resort may be identified other than the use of extended surface. The secondary

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Nomenclature

A_c	cross-sectional area of fin geometry, $S(H+C)$ (m^2)
C	fin tip to shroud clearance (m)
C^*	dimensionless tip clearance, C/H (dimensionless)
$C1$	gap between the tip of vortex generator to base (m)
C_1^*	dimensionless gap between the tip of vortex generator to base, C_1/H (dimensionless)
g	gravitational acceleration (m/s^2)
H	fin height (m)
Gr	thermal Grashof number, $g\beta(T_w - T_o)H^3/\nu^2$ (dimensionless)
h	heat transfer coefficient (W/m^2K)
k	conductivity (W/m-K)
L	fin length (m)
L^*	dimensionless fin length, L/H(dimensionless)
L_1	distance from entrance to first vortex generator along axial direction (m)
L_1^*	dimensionless distance from entrance to first vortex generator along axial direction, L_1/H (dimensionless)
L_2	distance between two vortex generators along axial direction (m)
L_2^*	dimensionless distance between two vortex generators along axial direction, L_2/H (dimensionless)
Nu	Nusselt number (dimensionless)
p	total pressure defect, $p_0 - p_s$, (Pa)
p_{av}	average pressure defect over the cross section (Pa)
p_s	static pressure (Pa)
p_0	ambient pressure, $\int_0^z \rho_0 g \partial z$(Pa)
P^*	dimensionless axial pressure defect, $pH^2/\rho_0\nu^2$
P_{av}^*	dimensionless average pressure defect over the cross section
Pr	Prandtl number, ν/α (dimensionless)
Re	Reynolds number, $W_{in}H/\nu$ (dimensionless)
S	fin spacing (m), $S = 2S_1 + t_v$
S^*	dimensionless fin spacing, S/H (dimensionless)
S_1	distance of vortex generator from right/left fin along x-direction (m)
S_1^*	dimensionless distance of vortex generator from right/left fin along x-direction, S_1/H (dimensionless)
T	temperature (K)
t_v	width/thickness of vortex generator (m)
t_v^*	dimensionless width/thickness of vortex generator, t_v/H (dimensionless)
u, v, w	velocity component in x-, y- and z-direction (m/s)
U, V, W	dimensionless velocities in X-, Y-, and Z- directions, uH/ν, vH/ν and wH/ν
x, y, z	cross stream and axial coordinates (m)
X, Y, Z	dimensionless cross stream and axial coordinates, x/H, y/H and z/H

Greeks

α	thermal diffusivity (m^2/s)
β	thermal volumetric expansion coefficient, $-(1/\rho_0)(\partial\rho/\partial T) = 1/T_0$ (1/K)
ΔT	scaling temperature difference, $T_w - T_0$ (K)
ν	momentum diffusivity (m^2/s)
Ω	dimensionless fin conductance parameter, $(k_{fin}t)/(2kH)$
ρ	density (kg/m^3)
θ	dimensionless temperature, $(T - T_0)/(T_w - T_0)$

Subscript

b	bulk
f	fin
in	inlet
s	shroud
w	base
0	ambient/reference

Superscript

*	dimensionless quantity

flow being one of such examples finds several applications to enhance the heat transfer. The horseshoe vortex is one its kind usually appears surrounding the protrusion, which extends out of the hydrodynamic boundary layer. An exhaustive review of small protuberances in boundary layer flows is well documented in Sedney [33]. It may be noted that the three-dimensional surface bump introduce qualitatively similar effects for both laminar and turbulent boundary layers. Formation of secondary flows depends nominally on the shape of protuberance, but the location and the height of it in a flow field are critical parameters. Protrusions, designed to generate strong vortices, are well-known as 'vortex generators'. Use of vortex generator in a channel is seen in Biswas et al. [34], Biswas and Chattopadhyay [35], Deb et al. [36]. Spanwise average local Nusselt number is predicted to enhance as high as 34% even at a dimensionless channel axial length of 8.4 [35]. A review on the heat transfer enhancement using longitudinal vortices was presented by Jacobi and Shah [37]. Ahmed et al. [38] report a review of heat transfer enhancement using vortex generator (VG) and nano-fluid. Vortex generators introduce transverse and longitudinal vortices. It is highlighted that longitudinal vortices are more efficient for heat transfer enhancement than transverse vortices. Enhancement of heat transfer is found to depend on angle of attack, geometry, VG tips spacing, rectangular, or circular array arrangement of VGs. Introducing a row of vortex generators placed on the fin surface available in between the tubes, Joardar and Jacobi [39] indicated a rise of heat transfer ranging from 16.5–44% for the imposed Re in a range of 220–960 (i.e., 0.7 < V < 1.8 m/s). The vortex generators invites an accelerated flow caused by the passage like a nozzle in the vicinity of the tube walls, which helps to mitigate the size of the tube wake (O'Brien and Sohal [40]). Chang et al. [41] examine the effect of vortex generator placed in a channel formed by flat tube fin. But, the results are not compared without vortex generator. A combined experimental and computational study of fluid flow and heat transfer is reported by Leu et al. [42] from the heat exchanger made of plate–fin and circular tube with vortex generator positioned behind the tubes. It is shown that angle of vortex generator, $\beta = 45^o$ renders the best augmentation in heat transfer with a rise of 8–30% in Colburn factor for Re in a range from 400 to 3000, which causes friction factor to rise from 11 to 15% only. It is highlighted that the use of vortex generator provides the greatest area reduction ratio as high as 25%. Li et al. [43] reports enhanced heat transfer with relatively lower increase in pressure drop at low Reynolds numbers in a range Re = 5000–10,000 by placing vortex generator before the rectangular fin array. But at higher Reynolds numbers (i.e., Re = 20,000–40,000), vortex generators induces nominal increase in heat transfer as compared to the increase in pressure drop.

It is revealed from the reported literatures that mixed convection heat transport from shrouded vertical fin arrays on vertical base is studied in a number of articles [29–32]. Zhang and Patankar