Effect of Flow Bypass on the Performance of a Shrouded Longitudinal Fin Array

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Abstract
Theoretical and experimental studies were carried out to investigate the effects of duct velocity, fin density and tip-to-shroud clearance on the flow bypass and its impact on the pressure drop across a longitudinal aluminum fin array and its thermal performance. The clearance was varied parametrically, stating with the fully shrouded case and variations of the channel height giving partially shrouded configuration of different clearance ratios were also carried out. The flow bypass was found to increase with increasing fin density and insensitive to the air flow rate. That effect of fin density decreased as the clearance increased. The calculated total pressure was greatly affected by fin density. For fully-shrouded fin array, with H_f/S equals to 8 and 12.72, the pressure drop increased by a factor of 4.3 and 20 of that with H_f/S equals to 3.4, respectively. The total pressure drop and the average convective heat transfer coefficients corresponding to the fully and partially shrouded fin array of H_f/S=3.4 were compared. Going from fully to partially shrouded one of the largest clearance ratio (C/H_f = 0.89), the total pressure drop reduced by about 50%. For clearance ratios equal to 0.36, 0.56, and 0.89, the average heat transfer coefficients were reduced by about 12, 17, and 30% for those of the fully shrouded configuration at Re of about 3 x 10^5. That percentage reduction in heat transfer coefficients decreased with the increase of air flow rate.

Keywords; Forced convection; Fin arrays; Cooling of electronic equipment;

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Nomenclature

A  area, m²
BF  bypass factor
Dₜ  hydraulic diameter, mm
f  friction coefficient, dimensionless
h  heat transfer coefficient, W/m²K
H  height, mm
k  thermal conductivity, W/mK
Kₑₑ  entrance loss coefficient
Kₑₑₑₑ  exit loss coefficient
L  longitudinal length, mm
m  parameter, equation (21)
Nₑₑ  number of fins
Nₑₑₑₑ  Nusselt number, dimensionless
P  pressure, N/m²
p  perimeter, mm
Q  rate of heat transfer, W
Re  Reynolds number, dimensionless
S  fin spacing, mm
T  temperature, K
t  thickness, mm
V  volumetric flow rate, m³/s
v  velocity, m/s
W  base plate and fin array width, mm

Greek Symbols

ε  Surface roughness, mm
ηₑₑ  fin efficiency, dimensionless
ν  kinematic viscosity, m²/s
ρ  density, kg/m³
Δ  difference

Subscripts

a  air
av  average
B  by pass
bp  base plate
D  duct
eₐ  entrance
ex  exit
exp  experimental
F  free flow
Fr  frontal
f  fin
fe  fin effective
fp  fin passage
in  input
L  loss
pr  predicted
s  surface
tot  total
∞  bulk

Introduction

Increasing power density of electronic equipment are requiring more effective thermal enhancement to maintain its operating temperatures at a satisfactory level. Despite its relatively poor thermal properties, air through the use of extended surfaces continuous to be used as a coolant for many electronic devices. Design of such surfaces takes the form of longitudinal fin arrays. If the air velocity through the gaps between fins is well approximated, the thermal performance of such fin arrays and pressure drop across them can be determined. Approximating the fin velocity based on the upstream flow rate in the enclosure is difficult, unless the fin array is fully shrouded, which is not practical. In ducting fin arrays, the approach flow redistribute itself and at least a portion of it will take the path of least resistance and bypass the fin array. The performance of such fin arrays is adversely affected by this flow bypass.

Recently, the studies on heat sink design for forced convection have been conducted. Bejan and Sciubba [1992] have predicted the optimal fin spacing for maximum heat transfer from a package of parallel plates that is cooled by forced convection. Knight et al [1992] have developed fin optimization method for heat sink with micro channels by iterative solution of nonlinear equations for Reynolds number, friction factor, Nusselt number, fin length, and pumping power. Since the surrounding flow field of fin array is actually bypass flow over fins, bypass effect should be taken into account.

The effect of flow bypass on the performance of longitudinal fin arrays has been reported by several workers. Sparrow et al [1978], Sparrow and Beckley [1981], and Sparrow and Kadle [1986] investigated the effect of tip clearance on thermal performance of longitudinal plate
fin heat sink. It is reported that the ratio of heat transfer coefficient with and without clearance, to be strongly affected by the tip clearance to fin height ratio, and to be independent of air flow rate and fin height. Experimental results of Butterbaugh and Kang [1995] showed that the thermal resistance is strongly correlated with the pressure drop across the heat sink, and appearing to be independent of the amount of bypass.

Writz et al [1994] studied the effects of flow bypass on the performance of longitudinal heat sinks in ducted flow. It is reported that the values of flow bypass was found up to 60% and its effect was to reduce the overall heat transfer rate. Their study also showed that the design of fin arrays can be optimized for a given flow conditions and shroud configuration. Lee [1995] has analyzed the relationship between the fin geometry and the pressure drop across the fin array taking into account flow bypass from the hydrodynamic analysis and developed an analytical model for optimizing the thermal performance of longitudinal fin arrays in ducted flow.

Thermal characteristics of straight channel longitudinal fin heat sink under forced air cooling were experimentally and numerically investigated using CFD code by Adam and Izundu [1997]. It is reported that much of the augmentation in the heat transfer rates with the heat sink is due to the increased surface area. However, as observed by Writz et al [1994], much of the gain in overall heat transfer rate is offset by the reduction in heat transfer coefficients due to the flow bypass effect which was more pronounced at lower approach flow rates.

Suzana et al [2000] investigated the effects of fin density, air flow rate, and clearance area ratio on flow bypass and its impact on the thermal performance of the heat sink using CFD software. It is concluded that the flow bypass is increased with the increasing fin density and clearance, while remaining insensitive to inlet duct velocity.

Azar and Tavassoli [2003] studied the effect of heat sink dimensions and the number of fins on its thermal performance. It is reported that the selection of heat sink depends not only on its thermal resistance, but also on the number of fins it has and how it is coupled to the board. Simons [2004] has estimated the air flow that actually passes through the fin passages of a ducted heat sink in the presence of flow bypass and investigated the effect of this flow bypass on its thermal performance. It is reported that the flow bypass has a substantial effect on thermal performance of heat sink, and that effect can significantly increase its thermal resistance as the number of fins increases.

Most of the experimental data collected from the previous studies in the above literature did not take the air flow leakage as well as the distribution of air flow through the duct into consideration. An adequate characterization of the thermal performance of such arrays requires a vast amount of data. More experimental work on them and with varying flow regimes is needed. The aim of the present study is to analytically and experimentally investigate the effect of air flow rate, fin density and tip-to-shroud clearance on flow bypass and its impact on the performance of a ducted longitudinal fin array.

**Experimental Investigations**

**Experimental setup**

A wind tunnel is used to investigate the performance of the tested fin array. A schematic of this experimental setup is shown in Fig. 1. It mainly consists of the test section, the power supplies, the flow rate, pressure and temperature measurement devices, a control gate, and the fan. An entry length for the tunnel is
made from wood with enough length to assure a uniform flow upstream of the test section for inlet velocities ranging from 1 m/s up to 20 m/s.

![Fig. 1. Test Rig](image_url)

1- Electric motor, 2- Suction air blower, 3- Air flow control gate, 4- Air stream out, 5- Wooden duct, 6- Temperature probes, 7- Bell mouth, 8- Air in, 9- Test section, 10- Digital temperature recorder, 11- Autotransformer, 12- Manometer, 13- Voltmeter, 14- Ammeter, 15- Blower switchboard.

**Test Section Components and Configurations**

For easy description of the experimental apparatus, reference may be made to Fig. 2, which shows a view of the test section and a cross section of the ducted fin array assembly used in carrying out the experiments when fully shrouded. The wooden test section dimensions of 500×107 mm with adjustable top allow accommodating the tested fin array. It includes an array of seven parallel longitudinal fins, a heated base plate, and a wooden shroud with adjustable top that can be moved up and down to change the amount of clearance between itself and the fin tips. The entire array can be easily mounted among the test section with the assurance of good thermal contact and to expose the fins to a nearly uniform air flow as can be reviewed in Elshafei and El-Negir [2003].

![Fig. 2. Test Section, (a) view of the test section, (b) cross sectional view at AA](image_url)

The fin/base plate assembly is fabricated from aluminum for its high thermal conductivity and low emissivity. An electrical resistance wire, which served
to heat the fin array is sandwiched between two mica sheets and placed in
contact underneath of the base plate and all the fin array assembly is then
fitted into a wooden box filled with polystyrene insulation. The heat
source is completely insulated on all sides in order to direct all heat to the
fin array.

The closure of the fin gaps flow passages surrounded by an insulated
U-shaped cross section wooden shroud with adjustable top, is attached to the
fin array base plate at its outboard edges and sealed. This helps to
minimize heat loss by conduction from the base plate to the shroud and
prevent air leakage. The entire test section is insulated with glass wool
sheet to minimize the heat loss.

As illustrated in Fig. 2-b, the assembly of the tested fin array when
fully shrouded and of 7 longitudinal fins with 6 parallel flow passages
bounded by half width fin spacing between the end side fins and the
shroud.

The wind tunnel is run in suction mode. Air is drawn from the
laboratory through its developing length, and then passed a calibrated
velocity meter into the test section. From there, the air is sucked through a
rectangular cross section duct by a centrifugal fan followed by a control
air flow gate.

**Heating Process and Instrumentation**

The electric power is supplied to the heater by a regulated AC source
via an autotransformer to control the voltage. The voltage settings are
guided by the readings of thermocouples, and both the voltage drop and the current are measured by
Voltmeter and Ammeter respectively.

Nine T-type thermocouples, 0.5
mm diameter, are glued onto the base of the fin array, in nine 1.5 mm
diameter drilled holes in the bottom
surface of the base plate. Another six
thermocouples are positioned along the
longitudinal length of the middle fin, three
at the mid half of its height and the other
three at its tip. The bulk air temperature is
detected by averaging of the readings of
two thermocouples inserted upstream and
that of another two thermocouples inserted
downstream of the test section respectively. Temperatures are recorded by
connecting the ends of all thermocouples via a selector switch to a 6 channels digital
temperature recorder of 0.1 °C resolution
and with an accuracy of ± 0.5 °C.

The air flow rate is controlled by
the graduated gate located at the wind
tunnel exit. A calibrated digital hot wire
anemometer with a range of 0.2-20 m/s, a
resolution of 0.1 m/s, and an accuracy of
± (1%+ 1d) full scale respectively is used
to measure the average velocity of entering
air with the help of a traversing mechanism
located 150 mm ahead of the test section.
Two static pressure taps are located 100
mm ahead and down stream of the test
section and connected to a U-tube water
manometer for the measurements of the
pressure drop across the fin array
assembly.

**The Test Matrix**

The ducted fin array assembly is
shown in Fig. 3. The clearance between the
fin tips and the shroud which is
represented by the top of the housing case
is existed in practice such as in cooling of
electronic components. When a clearance
is present, the longitudinal flow tends to
leak out of the fin passages where the flow
resistance is low. The velocity distribution
becomes more complex than for the no-
clearance, as well as the heat transfer
process.

In the present investigation, the
height of the fins is held constant and the
clearance between fin tips and the top of
the test section can be varied by wooden
smooth spacers with variable thickness that
can be placed between the walls of the
duct and the wooden shroud. The
dimensional values of the fin array
assembly used in the study are summarized in Table 1.

![Diagram of ducted fin array and bypass flow geometry]

**Table 1. Geometrical parameters of the tested fin array.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range Modeled</th>
<th>Material, $k_f = 217 \text{W/m.K}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of fins, $N_f$</td>
<td>7</td>
<td></td>
</tr>
<tr>
<td>Base plate thickness, $t_p$</td>
<td>108 mm</td>
<td></td>
</tr>
<tr>
<td>Base plate length, $L_{bp}$</td>
<td>225 mm</td>
<td></td>
</tr>
<tr>
<td>Fin length, $l_f$</td>
<td>225 mm</td>
<td></td>
</tr>
<tr>
<td>Fin thickness, $t_f$</td>
<td>2 mm</td>
<td></td>
</tr>
<tr>
<td>Fin spacing, $S$</td>
<td>13 mm</td>
<td></td>
</tr>
<tr>
<td>Duct width, $W_D$</td>
<td>125 mm</td>
<td></td>
</tr>
<tr>
<td>Clearance, $C$</td>
<td>0.11, 0.12, 0.18, 0.26, 0.45, 0.4 mm</td>
<td></td>
</tr>
<tr>
<td>Clearance ratio, $C/H_f$</td>
<td>0.22, 0.36, 0.56, 0.69</td>
<td></td>
</tr>
<tr>
<td>Clearance ratio, $C/S$</td>
<td>0.0, 0.76, 1.25, 1.9, 3.0</td>
<td></td>
</tr>
<tr>
<td>Duct height, $H_D$</td>
<td>51.62, 49.7, 79.6, 96.4 mm</td>
<td></td>
</tr>
</tbody>
</table>

Heat transfer and pressure drop results are parameterized by conventional rectangular duct Reynolds number, defined as

$$R_e = \frac{\nu D_h}{\lambda}$$

(1)

Where $\nu$ is the duct air velocity upstream of the fin array and $D_h$ is the hydraulic diameter of the fin array, given by

$$D_h = \frac{4 A_F}{p}$$

(2)

Where $A_F$ is the free flow area and $p$ is the wetted perimeter of the whole fin array assembly, expressed as

$$A_F = (W_D - N_f l_f)H_f + W_D C - W_{bp} t_{bp}$$

(3)

$$p = 2(N_f + 1)H_f + C + W_D$$

(4)

The fin array was installed in its position inside the test section and tested for two different power inputs, 75 W, and 100 W, at various air flow rates in terms of Reynolds number.

To attain a steady state condition, the system was initially run for about two to three hours until constant values of the recorded temperatures for both the bulk air and the array surface were observed.

**Bypass Flow and Pressure Drop Analysis**

The tested fin array is installed in the test section within the rectangular cross sectional duct as shown in Fig. 3. There will be a certain amount of flow bypasses and the velocity through the fin passages is different from the duct velocity approaching the fin array. The amount of bypass flow is related to the cross section geometry and the pressure drop across the fin array passages.

Considering the geometry shown in Fig. 3, and applying momentum and mass balances over the control surfaces 1 and 2, the total pressure drop across the fin array
as well as the by pass passages, $\Delta p$, which is mainly consists of the pressure drop across the each passage due to friction and that due to the entrance and exit effects. This pressure drop is given by

$$\Delta P = \Delta P_{\beta f} + \frac{1}{2} K_{\beta c n} \rho v_{\beta f}^2 + \frac{1}{2} K_{\beta c r} \rho v_{\beta r}^2$$  \hspace{1cm} (5),

$$\Delta P = \Delta P_{\beta f} + \frac{1}{2} K_{\beta c} \rho v_{\beta f}^2 + \frac{1}{2} K_{\beta c r} \rho v_{\beta r}^2$$  \hspace{1cm} (6),

where $\Delta P_{\beta f}$ and $\Delta P_{\beta f}$ are the pressure drop across the fin array and by pass passages due to friction, and $K_{\beta c n}$, $K_{\beta c}$ are the entrance and exit loss coefficients for both the fin array and by pass passages geometries respectively. These coefficients [Kays and London, 1984] are given by

$$K_{\beta c n} = 0.42 \left( \frac{A_{\beta}}{A_D} \right)$$  \hspace{1cm} (7),

$$K_{\beta c} = \left( 1 - \left( \frac{A_{\beta}}{A_D} \right) \right)^2$$  \hspace{1cm} (8),

$$K_{\beta c r} = 0.42 \left( \frac{A_{\beta}}{A_D} \right)$$  \hspace{1cm} (9),

and

$$K_{\beta c r} = \left( 1 - \left( \frac{A_{\beta}}{A_D} \right) \right)^2$$  \hspace{1cm} (10).

Substituting equation (5) into equation (6) gives

$$\frac{f_{\beta c} L_{\beta f}}{D_{\beta}} + K_{\beta c n} + K_{\beta c r}$$

$$v_{\beta f}^2 = \frac{f_{\beta c} L_{\beta f}}{D_{\beta}} + K_{\beta c n} + K_{\beta c r}

\hspace{1cm} (11)$$

Applying the continuity equation across the control surfaces, thus

$$v_{\beta} = v_{\beta f} \frac{A_D - v_{\beta f} A_{\beta}}{A_D}$$  \hspace{1cm} (12)

where $A_D$ is the duct cross sectional area approaching the fin array ($W_D \times H_D$), expressed as

$$A_D = A_{\beta} + A_{\beta f} + A_{\beta r}$$  \hspace{1cm} (13),

$A_{\beta}$ is the flow by pass area ($W_D \times C$), $A_{\beta f}$ is the flow area between fin array passages ($W_D - N_f i_f \times H_f$, and $A_{\beta r}$, $(N_f H_f i_f + W_{\beta f} t_{\beta f})$ is the frontal area of the fin array.

Substituting equation (12) into equation (11), the flow velocity through the fin array passages ($v_{\beta f}$) can be described as a function of the duct flow velocity approaching the fin array ($v_{\beta}$) as follows

$$v_{\beta f} = -\frac{b \pm \sqrt{b^2 - 4ac}}{2a}$$  \hspace{1cm} (14)

where

$$a = \left( \frac{A_{\beta}}{A_D} \right)^2 - \frac{f_{\beta c} L_{\beta f}}{D_{\beta}} + K_{\beta c n} + K_{\beta c r},$$

$$b = \frac{-2 A_D A_{\beta f} v_{\beta}}{A_D^2},$$

$$c = \left( \frac{A_D}{A_D} \right)^2 v_{\beta f}^2,$$

$$D_{\beta} = \left( \frac{4(W_D - N_f i_f H_f)}{N_f (2H_f + i_f)} \right),$$

and $D_{\beta}$ is equals to

$$\left( \frac{2W_D \times C}{W_D + C} \right)$$

are the hydraulic diameters of the fin array and the bypass passages, respectively.

The values of $v_{\beta f}$ can be determined from equation (14), substituting the value of the duct velocity ($v_\beta$) with the assumption of preliminary values for the friction coefficients for both the fin array and by pass passages as a fully turbulent flow. The obtained value of $v_{\beta f}$ is then