Proceedings of the ASME 2011 Pacific Rim Technical Conference & Exposition on Packaging and Integration of Electronic and Photonic Systems
InterPACK2011
July 6-8, 2011, Portland, Oregon, USA

InterPACK2011-52174

ASSESSMENT OF THERMAL PERFORMANCE OF ELECTRONIC ENCLOSURES WITH RECTANGULAR FINS: A PASSIVE THERMAL SOLUTION

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ABSTRACT

Passive heat transfer from enclosures with rectangular fins is studied both experimentally and theoretically. Several sample enclosures with various lengths are prepared and tested. To calibrate the thermal measurements and the analyses, enclosures without fins ("bare" enclosures) are also prepared and tested. Surface temperature distribution is determined for various enclosure lengths and heat generation rates. Existing relationships for natural convection and radiation heat transfer are used to calculate the heat transfer rate of the tested samples. The theoretical results successfully predict the trends observed in the experimental data. It is observed that the contribution of the radiation heat transfer is on the order of 50% of the total heat transfer for the tested enclosures. As such, a new correlation is reported for calculating optimum fin spacing in uniformly finned surfaces, with rectangular straight fins, that takes into account both natural convection and radiation.

INTRODUCTION

Passive cooling, technologies or design features without power consumption, has been used in thermal solutions for various applications such as power electronics, telecommunications, microelectronics, and HVAC systems. Two significant heat transfer modes involved in passive cooling systems are natural convection and radiation. Convection heat transfer from an object, determined using Newton's cooling law ($Q_{conv} = hA(T_S - T_\infty)$), can be enhanced by increasing the heat transfer coefficient, h, and/or by increasing the surface area, A, if the temperature difference is fixed [1].

Extended surfaces, which are popularly known as fins, are extensively used in air-cooled heat exchangers and electronic cooling. Although fins increase the surface area, they lead to an increase in frictional resistance resulting in a lower air flow rate in the vicinity of the finned surface, and at the same time, add more thermal resistance to the conduction path from the base to the ambient. As such, the heat transfer coefficient for non-interrupted finned surfaces is lower than the value for the base plate without fins. Therefore, the overall heat transfer rate of a finned plate could either increase or decrease in comparison with the base plate without fins. These competing trends clearly indicate the need for an optimization study to establish an 'optimum' fin design for air cooled enclosures. The effect of fin-spacing on natural convection has been investigated by several researchers. Design information of natural convection for fin arrays can be extracted from experimental studies of Starner and McManus [2], Welling and Wooldridge [3], Harahap and McManus [4], Jones and Smith [5], Donvan and

Roher [6], Van de Pol and Tierney [7], Yuncu and Anbar [8], and Guvenc and Yuncu [9]; and the theoretical works of Bar-Cohen and Rohsenow [10], Baskaya et al. [11], Haddad and Bany-Youness [12], and Dialameh et al. [13]. As a result, several correlations have been reported for the optimum fin spacing and the natural convection from heat transfer rate as a function of geometrical parameters.

Radiation heat transfer from a surface located in an ambient of temperature T_∞ can be calculated from [1]:

$$Q_{rad} = \frac{F_{s\infty} A \varepsilon \sigma \left(T_s^4 - T_\infty^4 \right)}{F_{s\infty}(1-\varepsilon)+\varepsilon} \quad (1)$$

where $F_{s\infty}$ is the surface view factor, ε is the surface emissivity coefficient, and σ is the Stefan–Boltzman constant $5.67\times10^{-8}\ W/m^2K^4$. Although fins increase the surface area, they can block part of radiation from the base plate and adjacent fins which reduces the surface view factor in Eq. (1). As such, similar to natural convection, radiation heat transfer also depends on the spacing and other geometrical parameters of the finned surface.

Radiation heat transfer plays an important role in heat transfer from fin arrays. This has been shown by Edwards and Chaddock [14], Chaddock [15], Sparrow and Acharya [16], Saikhedkar and Sukhatme [17], Sparrow and Vemuri [18,19], Azarkish et al. [20], and Rao et al. [21]. It has been reported that the radiation heat transfer contributes between 25–40% of the total heat transfer from fin arrays.

The listed studies have been focused on heat transfer from one single finned plate. Cha and Cha [22,23] investigated laminar steady gravity-driven flow around a single isothermal cube in an infinite medium by employing a control volume finite difference technique. Natural convection from isothermal cuboids and other geometries has been studied by Radziemska and Lewandowski [24,25] and Jafarpour and Yovanovich [26] both experimentally and numerically.

In spite of numerous existing studies on natural convection, our literature survey reveals a lack of experimental and theoretical studies on finned enclosures with conjugate natural convection and radiation heat transfer. To cover this ground, passive conjugate heat transfer from enclosures with rectangular shaped fins has been investigated theoretically and experimentally in this study. A custom-designed testbed has been built to perform the experiments. Several existing finned enclosures, currently being used by our industrial partner, have been prepared and tested to assess their thermal performance. The test bed allows temperature and input power measurements needed to assess the steady-state heat transfer from the enclosures. Existing correlations in the literature are used to predict the overall heat transfer coefficient and the average temperature of the enclosures under natural and radiative heat transfer. The experimental data were in reasonable agreement with the theoretical results. The theoretical analysis is then used to draw conclusions on the effect of fin spacing on the overall heat transfer rate of the systems with conjugate heat transfer.

NOMENCLATURE

A	=	Surface area, m^2
$F_{s\infty}$	=	View factor
g	=	Gravitational acceleration, m^2/s
h	=	Heat transfer coefficient, W/m^2K
H	=	Enclosure height, m
k	=	Thermal conductivity, $W/m\,K$
L	=	Enclosure length, m
Nu	=	Nusselt number
Pr	=	Prandtl number
Q	=	Heat transfer rate, W
Ra	=	Rayleigh number
S	=	Fin spacing, m
t	=	Fin thickness, m
T	=	Temperature, K
W	=	Enclosure width, m
Greek symbols		
α	=	Exponent in Archie's law, Eq. (9)
β	=	Coefficient of volume expansion, $1/K$
σ	=	Stefan–Boltzman constant, W/m^2K^4
ε	=	Emissivity
ν	=	Kinematic viscosity, m^2/s
Subscripts		
conv	=	Convective
fin	=	Related to fin
rad	=	Radiative
total	=	Radiative plus convective

EXPERIMENTAL APPROACH
Tested Samples

Six electronic enclosures were prepared and tested in this study; the electronic enclosures were made of aluminum by extrusion process. Three enclosures were finned while the others had the fins milled off down to the level of the bottom of the fin, see Figs.1 and 2 for more details. During the milling process, no alterations were made that affect the enclosures' length. The enclosures' lengths were 25.40, 30.05, and 40.08 *cm*.

The enclosures' surface is not uniformly finned; as it can be seen in Fig. 1, they are comprised of un-finned (bare) surfaces and regions with uniformly distributed fins (with 2.5 *mm* spacing). The width of the bare regions is large enough, larger than 30 *mm*, such that the surface can be modeled as bare plate. The dimensions of the tested enclosures are listed in Table 1.

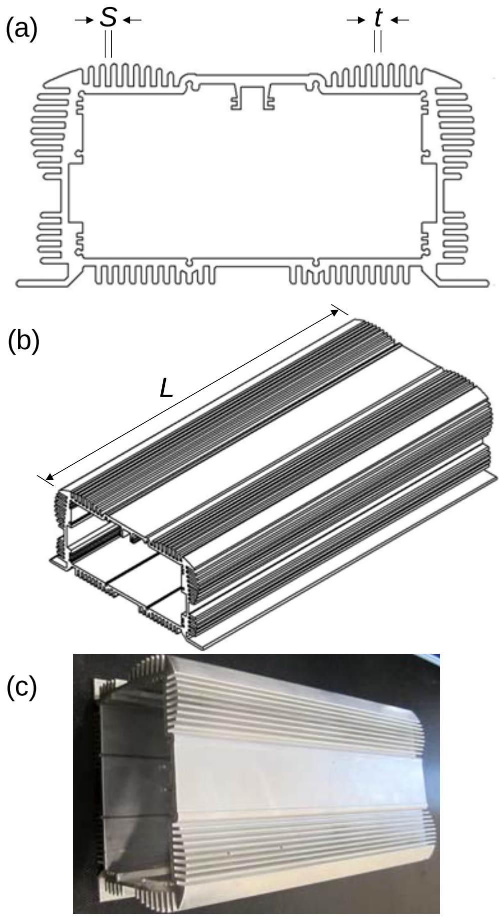

Figure 1: Schematic of the finned enclosures: a) cross-section view, b) isometric view; c) an actual enclosure.

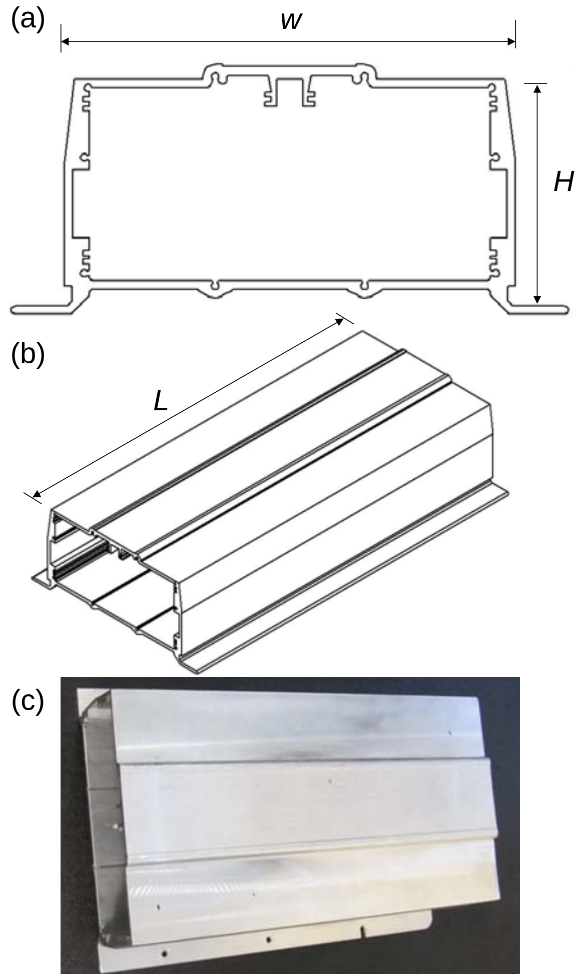

Figure 2: Schematic of the bare enclosures without fins: a) cross-section view, b) isometric view; c) an actual enclosure.

Table 1: Dimensions of the tested enclosures.

Enclosure name	Enclosure type	Length (*m*)/(*in*)	Fin spacing (*m*)	Finned area (m^2)	Bare area (m^2)
B10	Bare	0.254/10	0.0	0.0	0.085
B12	Bare	0.305/12	0.0	0.0	0.102
B16	Bare	0.408/16	0.0	0.0	0.137
F10	Finned	0.254/10	0.0025	0.079	0.078
F12	Finned	0.308/12	0.0025	0.095	0.094
F16	Finned	0.408/16	0.0025	0.127	0.126

Experimental Procedure

Figure 3 shows the schematic and the actual testbed. A wooden board is used to mount the enclosures vertically. The gap between the wooden board and the enclosure surface is filled with thick, high density foam for thermal insulation.

Enclosure ends are capped with 8 *mm* thick Poly(methyl methacrylate) (PMMA) plates to provide an airtight seal to the inside, as well as limiting heat transfer from the end regions. End caps are cut to match the profile of the extrusion so as not to disturb the gravity-driven airflow.

Chromalox strip heaters of 20 cm length are purchased from Omega (Toronto, ON) and are installed inside the enclosures to simulate heat generation from MOSFETs. Maximum rated power of each heater was 150 W. The heaters are powered by an adjustable AC power supply and the consumed power is monitored during the experiments. Ten self adhesive T-type copper-constantan thermocouples with uncertainty of $\pm 1\ °C$ are installed in various positions on the surface of the enclosures, see Fig. 3b. All thermocouple are adhered to the inside surface of the enclosure to prevent disturbing the outside buoyancy-driven air flow. An additional thermocouple is used to measure the ambient room temperature during the experiments. Each thermocouple is plugged into a TAC80B-T thermocouple to analog converter supplied by Omega (Toronto, ON).

The enclosures are tested in a windowless room free of air currents. The room temperature is kept constant at $20\,°C$. Each enclosure is tested at various power levels ranging from 20 to $110W$. The heaters are turned on and the surface temperature is monitored until a steady state condition is reached, i.e., the power input is equal to the heat transfer from natural convection and radiation. Once the variation of the surface temperature is less than $0.1\,°C/hr$, temperatures of all thermocouple are recorded and used in the analysis.

Figure 4 shows the maximum and minimum surface temperatures for enclosure F10 at various input power (heat transfer rates). As expected, the thermocouple closest to the lower edge of the enclosure shows the minimum temperature, T_1; the enclosure temperature increases at higher elevations due to thermal boundary growth and the maximum temperature occurs on the top of the enclosure, T_5.

THEORETICAL CALCULATIONS

Enclosures without Fins

Our experimental measurements showed that the maximum temperature difference along the tested enclosures was less than $8\,°C$. As an example, Fig. 4 shows the maximum and minimum surface temperatures for enclosure F10 at various input power (heat transfer rates). As such, the surfaces of the enclosure are assumed isothermal and the arithmetic average of the thermocouple readings is used in the analysis. The air properties are considered to be constant and are calculated at the film temperature as per the Nusselt number correlations used. The Nusselt number for natural convection from a plane vertical surface (bare enclosure) of length L can be calculated from the following equation [27]:

$$Nu = \frac{hL}{k} = \left\{ 0.825 + \frac{0.387\,Ra_L^{1/6}}{\left[1 + (0.492/\mathrm{Pr})^{9/16}\right]^{8/27}} \right\}^2 \tag{2}$$

where k is the thermal conductivity of air and Ra_L is the Rayleigh number based on the enclosures length:

$$Ra_L = \frac{g\beta(T_S - T_\infty)L^3}{\nu^2}\mathrm{Pr} \tag{3}$$

where g is the gravity acceleration, β is coefficient of volume expansion, ν is kinematic viscosity of air, and Pr is Prantdl number. Nusselt number (and the heat transfer coefficient) is calculated from Eq. (2) and calculate the natural convection heat transfer rate is calculated using the Newton's cooling law [1]. The radiation heat transfer is calculated from Eq. (1) where the view factor $F_{s\infty}$ is assumed to be 1. Total heat transfer from an enclosure can be calculated from:

$$Q_{total} = Q_{rad} + Q_{conv} \tag{4}$$

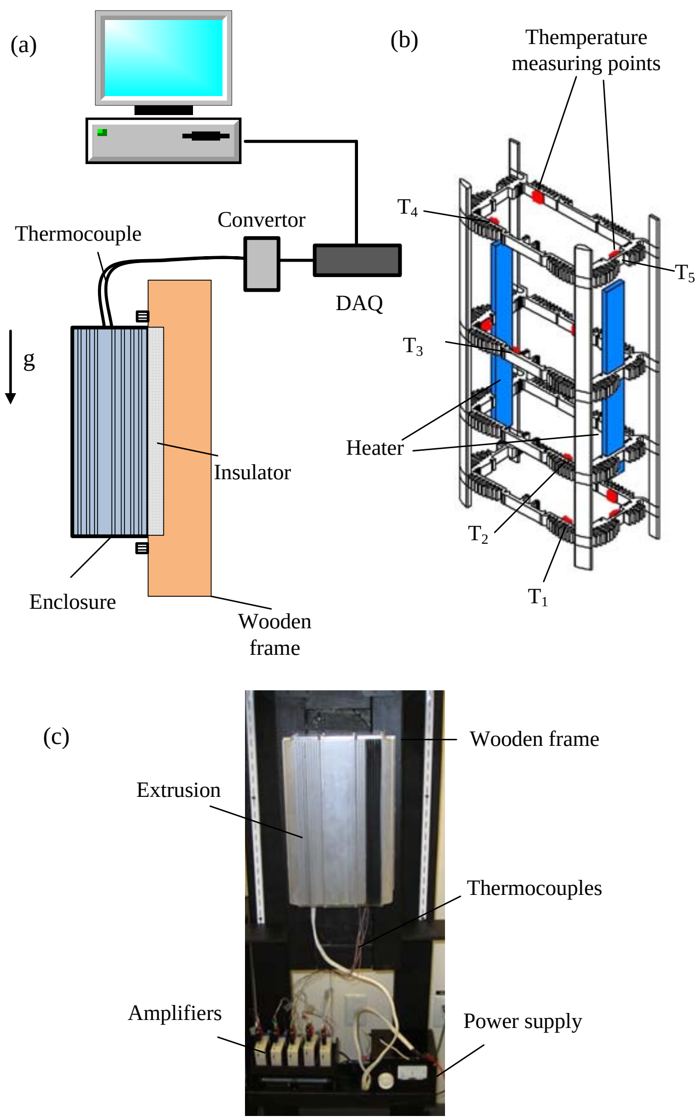

Figure 3: Experimental test bed; a) schematic; b) distribution of temperature measuring points and the location of the heater inside the enclosures; c) actual test bed.

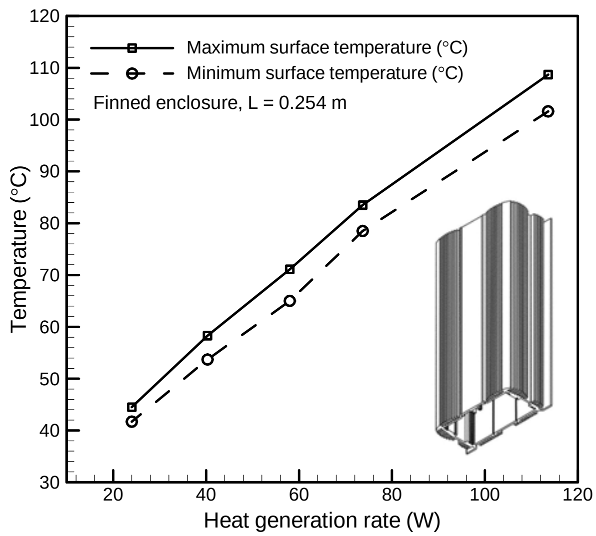

Figure 4: The maximum and the minimum measured temperatures on the surface of the F10 enclosure.

From comparison of the experimental data with the total heat transfer rate, one can estimate the surface emissivity coefficient, ε = 0.75, for the tested enclosures. Therefore, ε = 0.75 is considered in all radiation heat transfer calculations including the finned enclosures

Finned Enclosures

For the finned enclosures shown in Fig. 1, the total area is divided into two regions; i) unfinned region; and ii) finned region. The heat transfer in the unfinned region is calculated similar to the approach described in Section 3.1. Due to a compact spacing and long fins of the tested enclosures, the air flow between the adjacent fins become fully-developed rather quickly; i.e., $L_{\text{fully-developed}} / L_{\text{enclosure}} < 0.01$. As a result, one can assume channel flow, and the Nusselt number for the finned region can be calculated from [27]:

$$Nu_S = \frac{hS}{k} = \left\{ \frac{567}{\left(Ra_S S / L\right)^2} + \frac{2.873}{\left(Ra_S S / L\right)^{0.5}} \right\}^{-0.5} \tag{5}$$

where Ra_S is the Rayleigh number based on the fin spacing in the fin array, see Table 1. The radiation heat transfer is then calculated from Eq. (2) with ε = 0.75. $F_{S\infty}$ for the finned area (see Fig. 5) was calculated from :

$$F_{S\infty} = \frac{S}{H_{fin} \times 2 + S} \tag{6}$$

In the calculation, the average value of fin heights in Fig. 1 is used as H_{fin} .

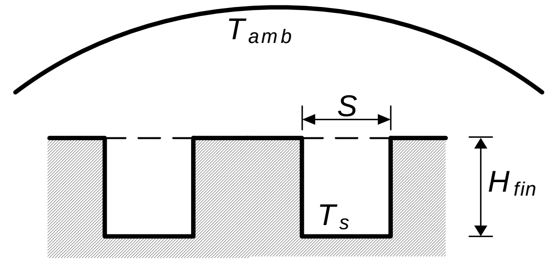

Figure 5: The considered geometry for calculating $F_{S\infty}$.

COMPARISON OF EXPERIMENTAL DATA WITH THEORETICAL CALCULATIONS

The enclosures listed in Table 1 were tested with various heat generation rates under steady state condition as described in Section 2. For convenience, average surface temperatures are used in the present analysis. In Figs. 6 and 7, the experimental data for averaged surface temperature for the tested enclosures are plotted vs. input power (heat generation) and compared with the theoretical predictions.

As can be seen from Figs. 6 and 7, the experimental data for the surface temperature can be successfully predicted by the present model, Eq. (5); the maximum relative difference between experimental and theoretical results is less than 9%. Also, the heat transfer enhancement from finned surfaces is minimal due to small fin spacing used in this particular enclosure design.

The contribution of the radiation heat transfer in the total heat transfer rate from the unfinned (bare) and finned enclosures are plotted in Fig. 8. It can be seen that the radiation portion is larger than 50% for various surface temperatures.

OPTIMUM FIN SPACING

The maximum difference between the averaged surface temperature of the finned and bare enclosures tested, at the same heat generation rate, was less than $4\,^{\circ}C$. The tested finned enclosures had higher surface area by two folds and were 84% heavier than the bare one with the same length. In other words, fins did not contribute to heat transfer enhancement while adding to the manufacturing cost significantly.

According to Bar-Cohen and Rosenhow [10], a fin spacing in the range of 7-12 *mm* (depending on the enclosure surface temperature) should result in the maximum natural convection for the tested samples,. However, the fin spacing in the tested samples is 2.5 *mm* which is much smaller than the recommend spacing by [10]. As a result, the heat transfer coefficient is significantly lower than the one for a bare plate. Moreover, the compactness of the fins leads to smaller surface view factors in the finned region.

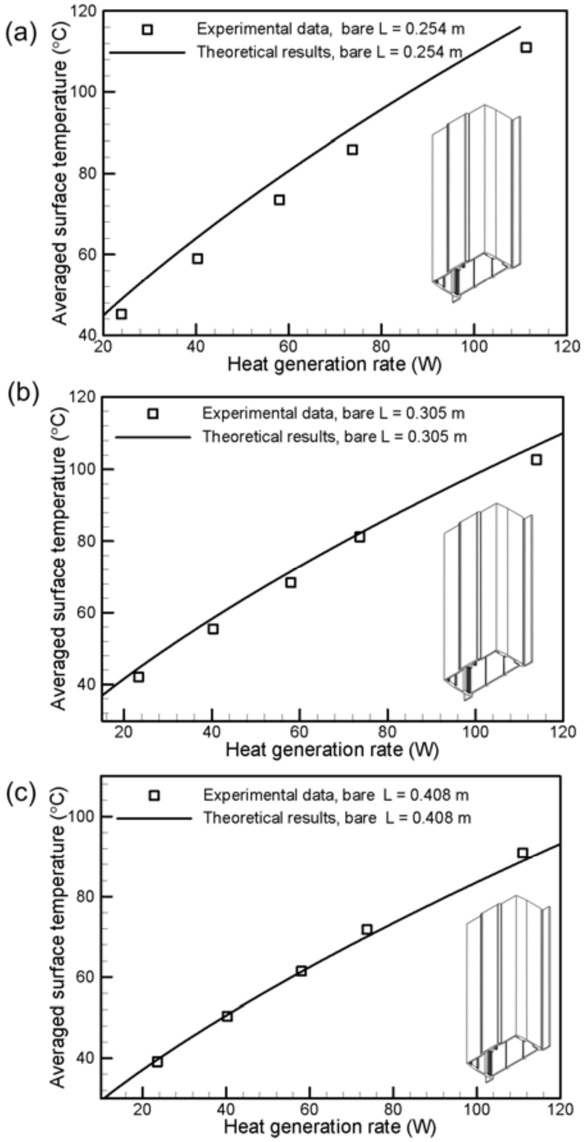

Figure 6: Comparison of the experimental data with theoretical predictions for bare enclosures; a) B10, b) B12, c) B16.

The theoretical model, Eqs. (1-6), which is verified through comparison with experimental data, is used to investigate effects of fin spacing on the overall heat transfer rate from the finned enclosures uniformly finned. Figure 9 shows the calculated results for various surface temperatures. Moreover, the optimum fin spacing values are compared with the values proposed by Bar-Cohen and Rosenhow [10] in Table 2. The optimum fin spacing can be calculated from the following relationship:

$$S_{opt} = 0.0231 \times \left(T_S - T_\infty\right)^{-0.236} \tag{7}$$

Table 2: Optimum fin spacing for the maximum conjugate heat transfer from uniformly finned plates (the present study) and for natural convection from surfaces [10].

T_∞	T_S	S_{opt} present study, Eq. (7) (mm)	S_{opt} [10] (mm)
20	40	11.4	10.1
20	50	10.4	9.2
20	60	9.6	8.6
20	70	9.2	8.2

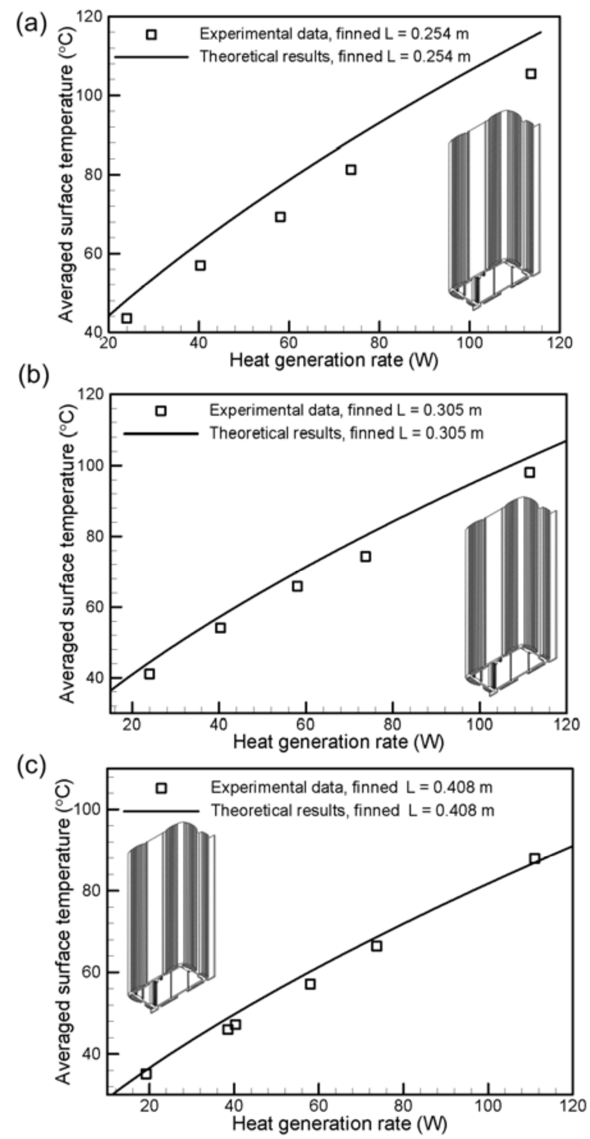

Figure 7: Comparison of the experimental data with theoretical predictions for finned enclosures; a) F10, b) F12, c) F16.

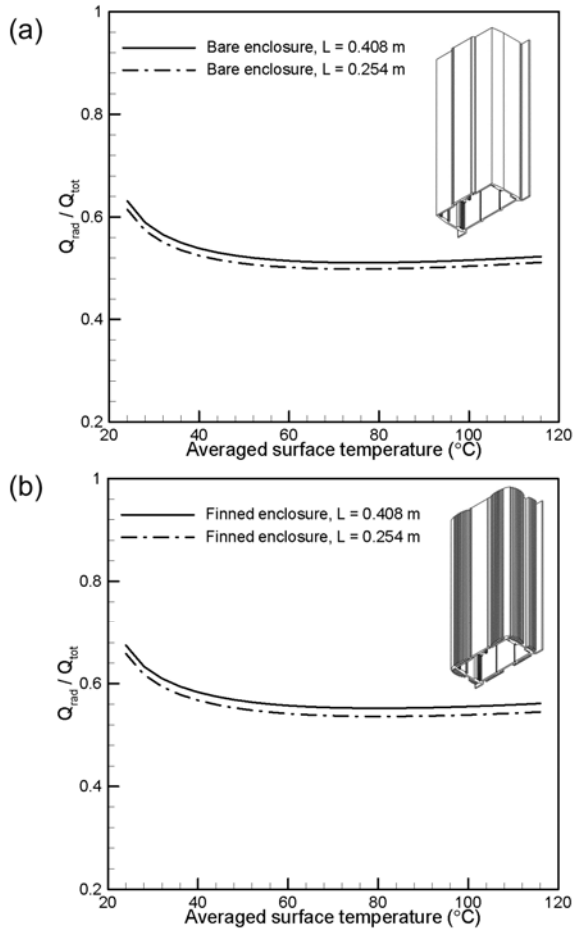

Figure 8: Contribution of radiation in the overall heat transfer; a) bare enclosures, b) finned enclosures.

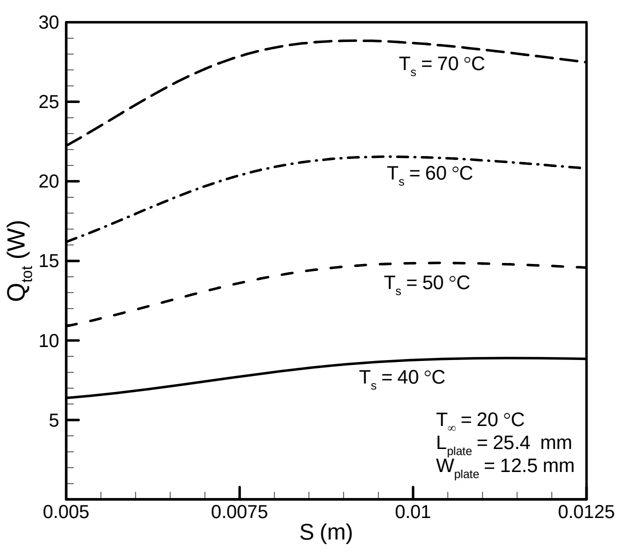

Figure 9: Effect of fin spacing on conjugate heat transfer from a uniformly finned surface.

CONCLUSIONS

Heat transfer from 6 different finned and bare enclosures was studied experimentally. The averaged surface temperature was measured for various heat generation rates. The arithmetic average surface temperature was also calculated using existing relationships for natural convection and radiation heat transfer. The theoretical results were in good agreement with the experimental data. The calculation showed that more than 50% of the overall heat transfer for the tested samples is due to radiation. Moreover, the fin spacing had a significant impact on the cooling capacity. Due to small fin spacing in the tested samples, fin did not affect the surface temperature while increasing the production cost by 84%. In addition, optimum fin spacing for the maximum conjugate heat transfer from a uniformly finned enclosure was reported for various surface temperatures.

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