OPTIMUM DESIGN AND SELECTION OF HEAT SINKS

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Abstract

An analytical simulation model has been developed for predicting and optimizing the thermal performance of bi-directional fin heat sinks in a partiaHy confined configuration. Sample calculations are carried out, and parametric plots are provided, illustrating the effect of various design parameters on the performance of a heat sink. It is observed that the actual convection flow velocity through fins is usually unknown to designers, yet, is one of the parameters that greatly affect the overaH thermal performance of a heat sink. In this paper, a simple method of determining the fin flow velocity is presented, and the development of the overall thermai model is described. An overview of different types of heat sinks and associated design parameters is provided. Optimization of heat-sink designs and typical parametric behaviors are discussed based on the sample simulation results.

Introduction

With the increase in heat dissipation from microelectronic devices and the reduction in overall form factors, it became an essential practice to optimize heat-sink designs with least trade-offs in material and manufacturing costs. Only a handful of discrete information is available in the technical literature regarding the optimization of heat sinks. Azar and his co-workers [1] reported a method of design optimization and presented contour plots showing the thermal performance of an air cooled narrow channel heat sink in terms of fin thickness and channel spacing parameters. They employed Poiseuille's equation in relating the channel flow velocity to the pressure drop, and the optimization method was presented, assuming the pressure drop across the heat sink is known.

Sasaki and Kishimoto [2] optimized, with a criterion of fin to channel thickness ratio of unity, the dimensions of water cooled micro-channels at a given pressure loss. An analytical method of optimizing forced convection heat sinks was proposed by Knight et al. [3, 4] for fully developed flow in closed finned channels. They presented normalized non-dimensional thermal resistances as a function of the number of channels, again for a fixed pressure drop. More recently, Wirtz et al. [5] investigated experimentally the effect of flow bypass on the performance of longitudinal fin heat sinks and devised a set of expressions for determining the optimum fin density for different fin geometry and flow conditions. Computational techniques were also employed in investigating the thermal performance of extruded heat sinks [6, 7].

Existing convection heat transfer data in the literature for extended surfaces invariably require the coolant fluid velocity adjacent to the surface be known. In predicting the thermal performance of a heat sink located in a partially confined configuration, such as in, a typical card-mount environment, however, one of the unknown and hard-to-estimate parameters is the actual flow velocity through the fins. Unless the system is fully confined, so that the entire coolant is channeled through the fins, this surface flow velocity can only be determined by considering the amount of flow bypass that results from the hydro-dynamic balance across the heat sink. With this in mind, an analytical simulation tool has been developed for predicting and optimizing the thermal performance of hi-directional heat sinks, given a set of readily available and measurable input parameters. The model computes for the amount of flow bypass and allows users to carry out a parametric investigation to optimize the heat-sink performance by choosing one parameter at a time as the control variable.

This paper is largely divided into two sections, The first section deals with general discussions on various types of heat sinks, and their relative performance ratings and manufacturing costs. The latter section deals with the analytical model. Simulation results are compared with existing experiment al measurements, and discussions are made on the effect of various design parameters on the performance and optimization of a heat sink.

Heat-Sink Categories

One way to categorize heat sinks is by the cooling mechanism employed to remove heat from the heat-sinks. It can be largely divided into five categories:

1 **Passive Heat Sinks** are used in either natural convection applications or in applications where heat dissipation does not rely on designated supply of air flows.

. typical height at heat input: 10 mm to large

- normal load limit: 5 to 50 watts
- cost range at 10,000 pieces: $0.50 to SIO. OO

2. Semi-Active Heat Sinks leverage off *existing* fans in the system.
   - typical height at heat input: ~10 mm
   - normal load limit: 15 to 25 watts
   - cost range at 10,000 pieces: $5.00 to $10.00

3. **Active Heat** Sinks employ designated fans for its own use, such as fan heat sinks in either impingement or vertical flows, This type of heat sinks usually involves mechanically moving components and its reliability depends heavily on the reliability of the moving parts.
   - typical height at heat input: 35 to 80 mm
   - normal load limit: 10 to 160 watts
   - cost range at 10,000 pieces: $10.00 to $20.00

4. **Liquid Cooled Cold** Plates typically employ tubes-in-block designs or milled passages in brazed assemblies for the use of pumped water, oil, or other liquids.
   - typical height at heat input: 10 to 20 mm
   - normal load limit: huge
   - cost range at 10,OOO pieces: $10.00 to $100.00

5. **Phase Change Recirculating System** includes two-phase systems that employ a set of boiler and condenser in a passive, self driven mechanism, Heat pipe systems incorporate either no wicks in a gravity fed arrangement or wicks that do not require gravity feeds. This category also includes solid-to-liquid systems but those are usually used to moderate transient temperature gradients rather than for the purpose of dissipating heat.
   - typical height at heat input: 5 to 10 mm
   - normal load limit: 100 to 150 watts
   - cost range at 10,000 pieces: $15.00 to $500.00

Heat-Sink Types

Heat sinks can also be classified in terms of manufacturing methods and their final form shapes. The most common types of air-cooled heat sinks include [8]

1. **Stampings:** Copper or aluminum sheet metals are stamped into desired shapes. They are used in traditional air cooling of electronic components and offer a low cost solution to low density thermal problems. Suitable for a high volume production, and advanced tooling with high speed stamping would lower costs. Additional labor-saving options, such as taps, clips, and interface materials, can be factory applied to help reduce the board assembly costs.

2. Extrusions: Allow the formation of elaborate two-dimensional shapes capable of dissipating large wattage loads. They may be cut, machined, and options added. A cross-cutting will produce omni-directional, rectangular pin fin heat sinks, and incorporating serrated fins improves the performance by approximately 10 to 20% at the expense of extrusion rate. Extrusion limits, such as the fin height-to-gap aspect ratio, minimum fin thickness-to-height, and maximum base to fin thicknesses usually dictate the flexibility in design options. Typical fin height-to-gap aspect ratio of up to 6 and a minimum fin thickness of 1.3 mm are attainable with a standard extrusion. A 10 to 1 aspect ratio and a fin thickness of 0.8 mm can be achieved with special die design features. However, as the aspect ratio increases, the extrusion tolerance needs to be compromised.

3. Bonded/Fabricated Fins: Most air cooled heat sinks are convection limited, and the overall thermal performance of an air cooled heat sink can often be improved significantly if more surface area exposed to the air stream can be provided even at the expense of conduction paths. These high performance heat sinks utilize thermally conductive aluminum-filled epoxy to bond planar fins onto a grooved extrusion base plate. This process allows for a much greater fin height-to-gap aspect ratio of 20 to 40, greatly increasing the cooling capacity without increasing volume requirements.

4. **Castings:** Sand, lost core and die casting processes are available with or without vacuum assistance, in aluminum or copper/bronze. This technology is used in high density pin fin heat sinks which provide maximum performance when using impingement cooling.

5. Folded Fins: Corrugated sheet metal in either aluminum or copper increases surface area and, hence, the volumetric performance. The heat sink is then attached to either a base plate or directly to the heating surface via epoxying or brazing. It is not suitable for high profile heat sinks due to the availability and from the fin efficiency point of view. However, it allows to obtain high performance heat sinks in applications where it is impractical or impossible to use extrusions or bonded fins.

Figure 1 shows the typical range of cost functions for different types of heat sinks in terms of the required thermal resistance.

The performance of different heat sinks ranges dramatically with the air flow velocity provided through the heat sink. To quantify the effectiveness of different types of heat sinks, the volumetric heat transfer efficiency can be defined as:

$$\eta = \frac{Q}{\dot{m}c\Delta T_{sa}} \tag{1}$$

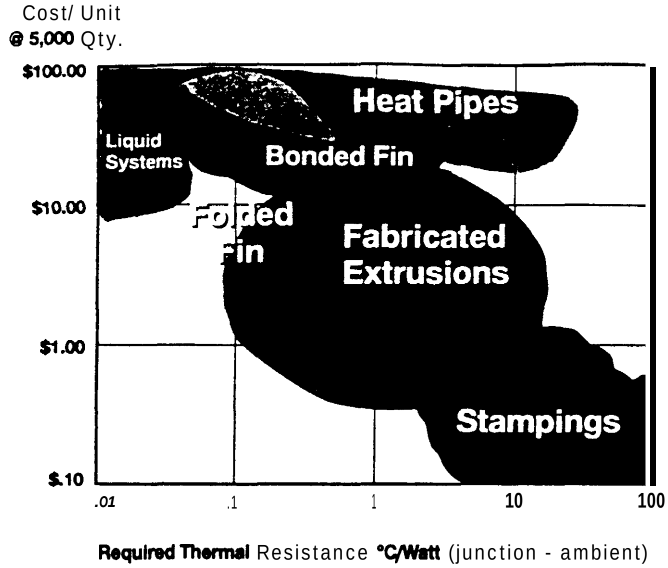

Figure 1: Cost versus Required Thermal Resistance

where, Q is the total heat dissipated, m is the mass flow rate through the heat sink, c is the heat capacity of the fluid, and ΔT_{sa} is the average temperature difference between the heat sink and the ambient fluid. The heat transfer efficiencies have been measured for a wide range of heat-sink configurations, and their ranges are listed below.

Table 1: Range of Heat Transfer Efficiencies

Heat Sink Type	η Range
Stampings & Flat Plates	10/18%
Finned Extrusions	15/22%
Impingement Flow Fan Heat Sinks	25/32%
Fully Ducted Extrusions	45/58%
Ducted Pin Fin, Bonded & Folded Fins	78/90%

The improved thermal performance, shown in the table, is generally associated with additional costs in either material or manufacturing, or both.

Design Parameters

In designing or selecting an appropriate heat sink that satisfies the required thermal and geometric criteria, one needs to examine various parameters that affect not only the heat-sink performance itself, but also the overall performance of the system. Option of choosing a particular type of heat sink depends largely on the thermal budget allowed for the heat sink and external conditions surrounding the heat sink. In any type of heat sink, one of the most important external parameters in air cooling is the flow condition which can be classified as natural, low flow

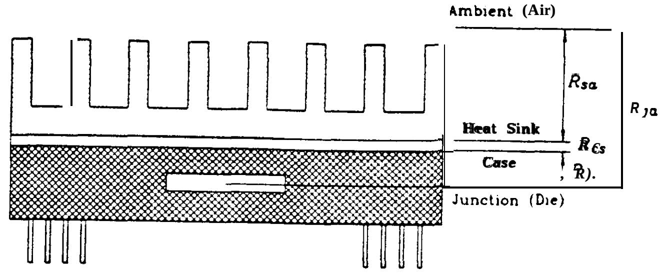

Figure 2: Thermal Resistance Breakdown

mixed, and high flow forced convection. There is no clear definition or consensus on the flow velocity that separates the mixed and forced flow regimes. It is generally accepted in applications, however, that the effect of buoyant force on the overall heat transfer diminishes to a negligible level (under 5%) when the air flow velocity exceeds beyond 1.5 to 2 m/s.

Consider a package with a heat sink shown in Fig. 2, The overall thermal performance of the system can be measured in terms of the total thermal resistance which can further be decomposed into three classical components: the junction-to-case resistance, R_{jc}, the case-to-sink resistance, R_{cs}, and the sink-to-ambient resistance, R_{sa};

$$R_{total} = Rja = R_{jc} + R_{cs} + R_{sa} \quad (2)$$

Here, R_{sa} represents the heat-sink resistance, and is the one most sensitive to the flow velocity. Although the other two are also somewhat dependent on the flow rate, they can be considered relatively a weak function of the flow velocity and may be assumed constant relative to R_{sa}. The thermal optimization of a heat sink addresses to minimize R_{sa}, given a set of design constraints.

A list of design constraints for a heat sink may include parameters, such as

- induced approach flow velocity
- available pressure drop
- cross sectional geometry of incoming flow
- amount of required heat dissipation
- maximum heat sink temperature
- ambient fluid temperature
- maximum size of the heat sink
- orientation with respect to the gravity
- appearance and cost

Given a set of design constraints, one needs to determine the maximum possible performance of a heat sink within the envelope of constraints. The parameters, over which a designer has a control for optimization, typically include,

- fin height

- fin length
- fin thickness/spacing
- number/density of fins
- fin shape/profile
- base plate thickness
- cross-cut patterns
- heat sink material
- *etc.*

Characterization and Optimization of Heat Sinks

In view of achieving an optimum thermal performance, most of the parameters discussed in the previous section are interdependent of the others. It is often true that the impact one parameter has on the performance of a heat sink cannot be generalized, or even foreseen without concurrently considering the consequences exhibited in the other parameters. For example; a longer fin height provides additional surface area for greater heat dissipation and improves the overall thermal performance. However, if the available volumetric flow rate is fixed, the overall performance may deteriorate with the fin height; if the available pressure drop is fixed, a longer heat sink in the direction of flow may have an adverse effect on the performance by decreasing the actual velocity over the fin surfaces, and; an option of having more fins is generally viewed as a way to improve the performance. This is a very dangerous generalization, because, in most cases, having excessive fins induce a higher pressure drop across the heat sink, resulting in a severe reduction in flow velocity and/or a significant increase in flow bypass over the heat sink.

In order to demonstrate and examine the relationship between various parameters and design constraints, let us consider a typical situation where a hi-directional heat sink is placed in a rectangular flow duct of (W x H), as depicted in Fig. 3. From an analytical modeling stand point, one can use a numerical method which typically requires in the order of 50,000 to 100, OOO discretized elements and hours of computation per each case [7]. The benefit of using a numerical method includes the ability to obtain detailed local information over the entire problem domain, and the possibility of achieving a better accuracy. However, in most applications, the ability to optimize and quickly narrow down the parametric ranges of a particular design is extremely valuable, especially in the earlier stage of design iterations, An analytical design tool that utilizes the bulk analyses to obtain average, global performance characteristics usually suffices such needs.

Besides the heat-sink material and fluid properties, the model described herein requires nine input parameters, as indicated in Fig. 3. Although the analysis is not limited to a specific material and coolant fluid, all the cases presented in this paper assume aluminum as the material and air for the coolant. ordinarily, the approach flow velocity, U_d, is a result of the balance between the fan capacity and the system pressure loss, and is not known a priori, However, once the model is developed, the approach velocity can be varied to produce the pressure drop versus flow velocity relationship for a given system which, in turn, can be used in conjunction with a particular fan curve to determine the operating condition of the system,

In a partially confined configuration, where the flow duct is larger than the heat-sink cross sectional profile, there will be a certain amount of flow bypass, and, as a result, the flow velocity through the fins may be substantially different from the approach velocity, The amount of flow bypass is strongly related to the cross sectional geometry and the pressure drop across the heat sink, and this must be determined from the hydro-dynamic analysis before the thermal analysis can be carried out.

Flow Bypass

Using U_f to represent the average flow velocity through the fin region and ignoring the friction loss along the duct surfaces, one can obtain the following expression for the unknown U_f by simultaneously satisfying the mass and momentum balances over the control surfaces surrounding the system:

$$(2a_f - 1)u_f^2 - 2a_f u_f + 1 = (1 - a_f)^2 \frac{\Delta P_f}{\rho U_d^2/2} = 0 \quad (3)$$

where $a_f = (w \times h)/(W \times H)$ is the fin area ratio, $u_f = U_f/U_d$ is the normalized fin flow velocity, ΔP_f denotes the loss due to the friction through the fin region, and ρ is the fluid density.

Since ΔP_f can be described as a function only of U_f,

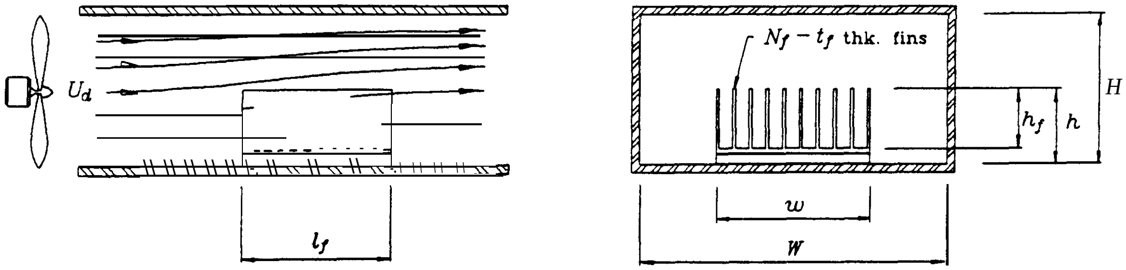

Figure 3: Problem Geometry

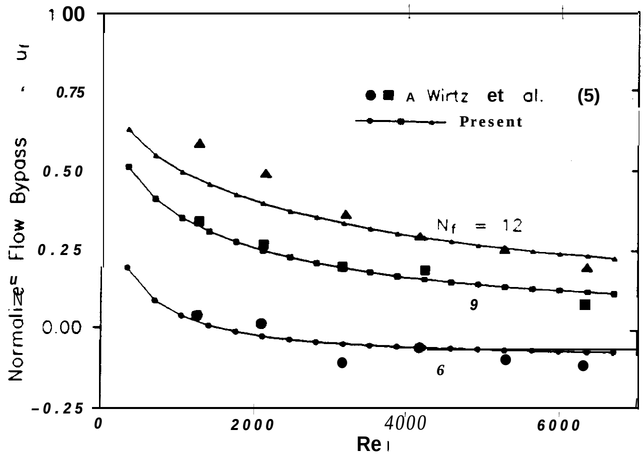

Figure 4: Comparison of Flow Bypass for Different Number of Fins

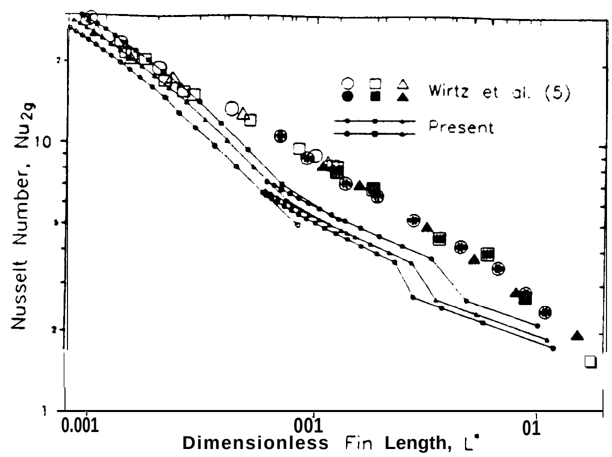

Figure 5: Comparison of Channel Heat Transfer for Different Number of Fins and Fin Heights

the above expression is an implicit equation for U_f, and, therefore, requires an iterative procedure. With an assumed initial value of U_f, one can readily determine the pressure drop using correlations or graphical data available in the literature. In the present study, ΔP_f is determined by interpolating the friction charts provided in reference 9 for different channel aspect ratios. The entrance and exit pressure losses are estimated from reference 10.

Only a limited data are available in the literature that quantify the effect and amount of flow bypass. Wirtz et al. [5] reported a set of flow results that were backed out from the thermal measurements on longitudinal heat sinks. Their test geometry consists of 13 rows and 7 columns of square packages placed in an open circuit wind tunnel with a heat sink mounted on a central element of the seventh row. Although the test geometry, where the measurement sample was placed in the wake of the upstream packages, is not identical to the single element case considered herein, the behavior of flow bypass was expected to be similar, and the results are compared in Fig. 4 for different fin densities. Readers are advised to consult reference 5 for the other dimensions. In the figure, $(1 - u_f)$ represents the normalized amount of flow bypaas over the heat sink, and $Re_l = U_d l/\nu$, where ν is the fluid kinematic viscosity. As can be seen from the figure, the agreement is excellent.

Thermal Analysis

Upon determining the amount of flow bypass and the average fin flow velocity, the Colburn j factor can now be determined, again in this study, by interpolating the graphical data provided in reference 9. Noting that the heat transfer coefficient, h_j, used in the definition of Colburn j factor is based on the log-mean-temperature difference, the effective heat transfer coefficient, h, based on the temperature difference between the heat sink and the inlet fluid temperatures can be computed using the following relationship:

$$h = \frac{\dot{m}c}{A_s}\left[1 - e^{-(h_j A_s)/(\dot{m}c)}\right] \quad (4)$$

where A_s is the total exposed surface area including the area of the base plate exposed between fins.

The parallel plate Nusselt number, defined as

$$Nu_{2g} = \frac{h\,2g}{k} \quad (5)$$

where g is the fin spacing, has been computed for the cases examined by Wirtz et al. [9], Two different fin densities and three fin heights were tested. Figure 5 shows a comparison of the present predictions with the measurements as a function of the dimensionless fin length, $L^* = l_f/(2gRe_{2g}\,\mathrm{Pr})$. Pr is the fluid Prandtl number, and $Re_{2g} = U_f 2g/\nu$ is the parallel plate Reynolds number. The figure reveals good agreement for low values of L^* which corresponds to high flow velocities. The prediction becomes conservative when the flow velocity is decreased at high L^* values. These differences observed at high L^*, or at low flow velocities, are expected to be due to a combination of the buoyancy effect and radiation heat transfer.

Case Studies and Discussion

To examine the effect of various design parameters, let us consider the case depicted in Fig. 3 with the following dimensions as the default case.

$$\begin{aligned} U_d &= 1.0 \text{ m/s} \\ W \times H &= 100 \text{ mm x } 50 \text{ mm} \\ w \times h &= 50 \text{ mm x } 25 \text{ mm} \\ N_f,\ t_f,\ h_f,\ l_f &= 10,\ 1.25 \text{ mm}, 20 \text{ mm},\ 100 \text{ mm} \end{aligned}$$

The parametric behavior, responding to a chosen variable with all others unchanged, is investigated. Due to the

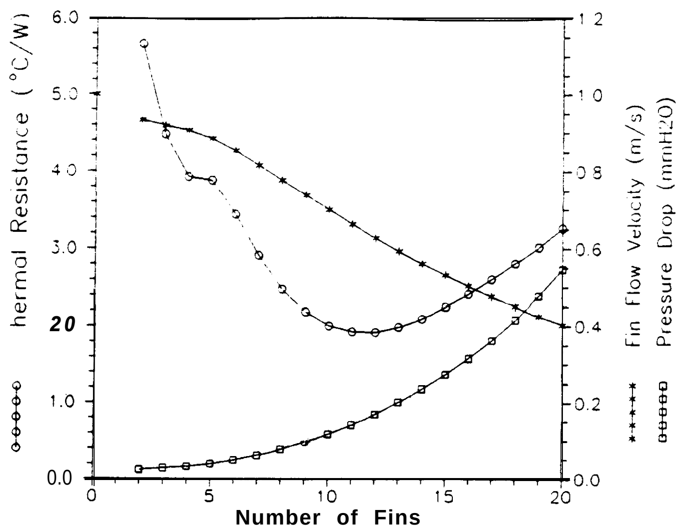

Figure 6: Parametric Performance as a Function of Number of Fins

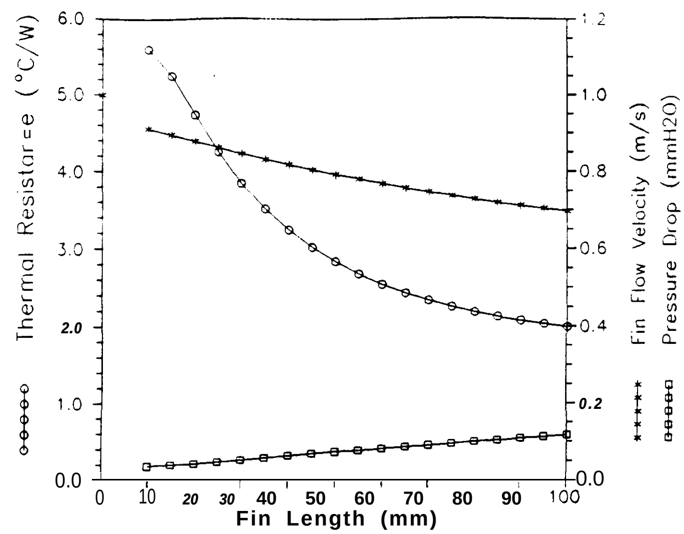

Figure 7: Parametric Performance as a Function of Fin Length

lack of space, only a few calculations are presented and discussed. Figures 6 to 8 show the resulting sink-to-ambient thermal resistance R_{sa}, fin flow velocity U_f, and the pressure drop across the heat sink-assembly ΔP_f as a function of the chosen variable. The thermal resistance is defined as

$$R_{sa} = \frac{\Delta T_{sa}}{Q} = \frac{1}{\eta_f h A_s} \tag{6}$$

where η_f represents the fin efficiency. Each parametric run takes a few seconds using a personal computer.

As expected, Fig. 6 reveals the existence of an optimum number of fins. As the number of fins increases, the total convective surface area increaaes and, as a result of the greater pressure drop, the fin flow velocity simultaneously decreases. Initially, the performance gain from the increase in surface area is greater than that lost from the reduction in fin flow velocity. The net effect is an improvement in thermal performance. However, as the number of fins continue to increase, the net effect reverses, and the performance decreases beyond the optimum number of fins. Although the optimum number of fins suggested by the simulation is 12, the recommended number of fins should be slightly less (i.e. 10). The thermal gain expected from 10 to 12 fins is so small in this case that having two additional fins would not likely justify the additional cost incurred by extra material and, more importantly, the different manufacturing method that must be employed to produce the heat sink with more than 11 fins. The fin height-t-gap aspect ratio of this heat sink with more than 11 fins exceeds the maximum ratio of 6 attainable with a cost effective, standard extrusion method.

The change in flow regime from turbulent to laminar may occur as the number of fins increases and the fin spacing becomes smaller. This phenomenon is reflected in the figure as a momentary lag in the performance improvement between 4 and 5 fins.

The performance behavior as a function of fin length is shown in Fig. 7. Initially, the performance improves as the total convective surface area increases with the fin length. However, "the rate of return on investment" diminishes as the length becomes longer. This is due to the temperature rise in the air stream between fin surfaces combined with a reduction in flow velocity that resulted from a higher pressure drop across the heat sink. This diminishing return on investment is also observed in Fig. 8 which shows the behavior as a function of approach velocity U_a.

Finally, Fig. 9 reveals the importance of flow management. This shows identical case shown in Fig. 6 except the duct dimensions are reduced down to the cross sectional dimensions of the heat sink. This case simulates a fully confined configuration where the entire flow is channeled through the fins with no flow bypass. As compared to the case shown in Fig. 6, the thermal performance has improved significantly and, unlike the case where the flow bypass was allowed, the thermal performance continues to improve with the number of fins. In this case, however, the penalty is in the increase in the pressure drop, requiring a stronger fan to maintain the same approach flow velocity: note that the scale of the right-side vertical axis has been doubled to accommodate the large increase in the pressure drop. In any of the above cases, the actual operating point with a given fan can be found by superimposing the fan performance curve and locating the point of intersection along the pressure curve.

Summary

Different types of heat sinks are examined, and their relative performance and cost ranges are presented. An analytical simulation model and a method of computing the

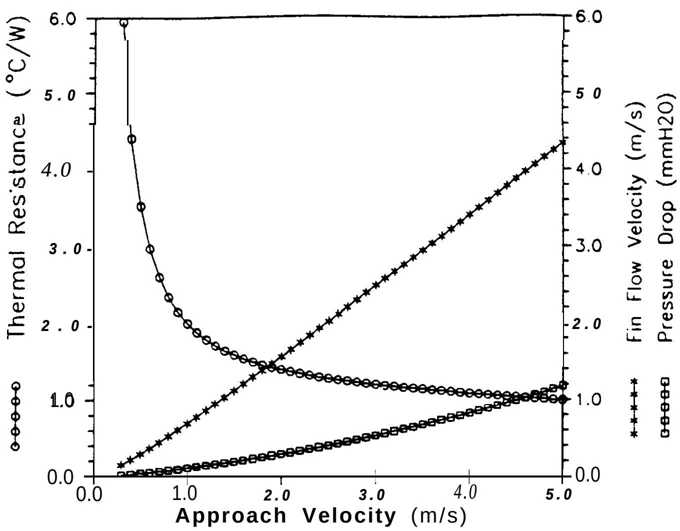

Figure 8: Parametric Performance as a Function of Appreach Velocity

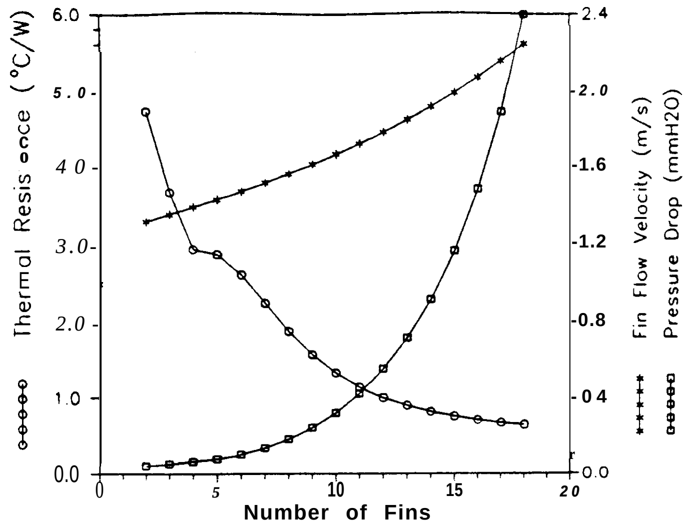

Figure 9: Parametric Performance as a Function of Number of Fins in Confined Configuration

thermal performance ofa heat sinkin a partially confined configuration are described. The model is validated by comparing the results with existing experimental data, and sample cases are presented with discussions on the parametric behavior and optimization of-hi-directional heat sinks.

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