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Low profile fan and heat sink thermal management solution for portable applications

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Abstract

The increasing heat flux densities from portable electronics are leading to new methodologies being implemented to provide thermal management within such devices. Many technologies are under development to transport heat within electronic equipment to allow it to be dissipated into the surroundings via conduction, natural convection and radiation. Few have considered the approach of implementing a forced convection cooling solution in such devices. This work addresses the potential of low profile integrated fan and heat sink solutions to electronics thermal management issues of the future, particularly focusing upon possible solutions in low profile portable electronics. We investigate two heat sink designs with mini-channel features, applicable to low profile applications. The thermal performance of the integrated fan and heat sinks is seen to differ by approximately 40% and highlights the importance of designing an integrated thermal management solution at this scale rather than fan or heat sink in isolation.

Keywords: Electronics cooling; Low profile; Small scale fan; Thermal management; Portable electronics; Mobile cooling

1. Introduction

Improved performance and smaller scales are two features that the electronics industry continually strives to achieve to gain competitive advantage in the market place. Both of these targets result in increased heat flux densities within electronic devices. These trends are particularly important in the portables sector, where consumer demand for increased functionality, which results in increased heat dissipation, and smaller products are the key market driving forces. Unless technologies are developed to remove this heat at acceptable temperatures the implications on future generations of products such as mobile phones and palmtop computers will be twofold. Firstly, rules of thumb such as those derived from the Arrhenius model suggest that the mean time to failure halves for every 10 °C increase in component temperature, so reliability will be reduced. Secondly, and perhaps more importantly is the effect on user comfort. Depending on the material, case temperatures of 45 °C could cause discomfort to the user [1], and so case temperature should be below this value. By simply taking the surface area of a typical high end state of the art mobile hand set, and assuming typical natural convection heat transfer coefficients, one can estimate that 45 °C case temperatures will be reached at a device heat dissipation of 3 W in an ambient temperature of 30 °C. Given that many such devices are striving towards laptop performance which dissipates an order of magnitude more power, it is clear that thermal management of such devices is impeding the enhancement of performance and the need for active cooling solutions is evident.

The thermal design of any electronic system, including portable electronics, is performed so that the system operates at a sufficiently low temperature to allow acceptable levels of reliability, performance and user comfort. The choice of cooling methodology used to maintain this temperature varies from system to system, and is dependent upon a number of factors, which include the heat dissipated, operating environment, maximum allowable component temperatures, available space, cost and maintenance required.

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Nomenclature

E	Given function	
Q	Volumetric flow rate	$m^3 s^{-1}$
R	Local radius	m
R	Outer radius of heat sinks	m
R_{th}	Thermal resistance	°C W^{-1}
RPM	Revolutions per minute	
T	Temperature	K
W_E	Uncertainty in the result	
W_n	Uncertainty in independent variables	
x_n	Independent variables	
δ	Thermal boundary layer thickness	m
θ	Angle	

Subscripts

amb	Ambient conditions
local	Local conditions
measured	Measured conditions
W	Wall conditions

In general, when heat dissipation is low, acceptable temperatures are maintained using natural convection. Until now, natural convection, coupled with conductive heat spreading techniques, has been the method used to transport heat from within handheld devices, to the ambient air. It is anticipated that in the near future, elevated heat flux densities will mean that natural convection heat transfer coefficients will be insufficient to maintain acceptable temperatures in many high end handheld devices.

In larger scale electronic systems with moderate heat dissipation levels, forced convection air cooling is generally employed, with fans used to force air over heat sources, and carry heat from the source to the ambient air. This technique can result in relatively high heat transfer coefficients, particularly if a fan is assembled with a heat sink and mounted on a heat source. Such fanned heat sink assemblies are commonly employed in larger scale electronic systems to provide cooling for components such as the processors in PCs, servers and laptops. Much work has been done in recent years to characterise and optimise the performance of fanned heat sinks ([2–5] amongst others). However, this work has focused on large scale applications, with no effort to date focused on scales appropriate to handheld electronic devices.

For higher heat flux situations, and/or where insufficient space is available at the heat source to mount a fanned heat sink, liquid cooling can be employed, with liquid recirculated from the heat source to a secondary air cooled heat exchanger, mounted where sufficient space is available, which transfers the heat to ambient. Numerous liquid based cooling schemes are described in the literature including those by Prasher and Mahajan [6], Valenzuela et al. [7] and Eason et al. [8]. Because of the necessity for a secondary heat exchanger to dissipate heat to ambient, liquid coolers are not appropriate to thermal management of handheld devices, as the solution would require a primary heat exchanger, pump, piping and secondary heat exchanger, the combined volume, mass and cost of which would be excessive.

When the advantages and disadvantages of the technologies discussed above are considered, forced convection air cooling is the most appropriate to thermal management of future handheld devices. Relatively high heat transfer coefficients are achievable. Integrated fans and heat sinks are relatively easy to implement, as they require no special packaging considerations. Air, the heat removal medium, is abundantly available, and is noncorrosive to the cooled equipment. However, little work has been done to date on the development of forced convection cooling schemes at scales appropriate to handheld devices. Some studies have addressed piezoelectric based fans [9, 10], but their low flow rate combined with the requirement for high voltages makes them unlikely competitors in the portable electronics market place. Recently, studies addressing the aerodynamics effects of geometrical scaling in both axial fans [11] and radial fans [12], has lead to the development of low profile rotary fans for potential use in electronics cooling applications.

Other technologies under development for thermal management of portable electronic devices are phase change materials [13], micro-heat pipes [14] and high conductivity materials. However such technologies focus upon heat spreading and transport within a device rather than active removal of heat from the device. For example, the phase change materials store heat to be dissipated over time, while heat pipes and high conductivity materials only provide paths of reduced thermal resistance to the flow of heat within devices. Therefore, at present this heat is finally being removed by some combination of natural convection, conduction and radiation in many devices. However, the importance of such technology should not be underestimated, as they may be used to move heat from its source to some form of forced convection air cooling device in future generations of handheld devices.

This paper addresses the integration of miniature fan technology with heat sink designs aimed at low profile portable electronics with the complete thermal management solution being less than six millimeters in height, such as palmtop computers. Two finned heat sinks were manufactured and tested, with features essentially scaled down from conventional macro scale heat sinks, while also considering state of the art design philosophies. It is shown that in low profile applications the use of efficient heat sinks is a critical parameter to minimize the thermal resistance offered by the integrated heat sink and fan thermal solution. It is found that a multi-scale heat sink with non-radial shaped fins performs best, and is a potential solution for portable devices wishing to dissipate in excess of three to four watts.

2. Forced convection radial heat sinks

2.1. Two-dimensional analysis

In this section we will briefly summarise optimisation techniques for heat sinks to obtain enhanced heat transfer densities per unit volume. By considering the essential features of such techniques we developed two heat sinks and characterise the thermal performance of the designs.

In the case of two-dimensional flow for single length scale heat exchangers optimal spacing has been determined for cylinders [15] and parallel plates [16,17] along with a number of additional configurations. These methodologies focus upon the development of thermal boundary layers on the heated surfaces. On each heat generating surface, thermal boundary layers develop on the heat conducting fins. In the case of a radial flow heat sink the problem at hand is to elucidate the optimal angle, θ, that results in maximum heat transfer density for the given length scale. To the authors knowledge this has not been done in the past for radial heat sinks. In Fig. 1(a), where θ is large and the boundary layers between adjacent surfaces do not meet, fluid is pumped through the fins and has no thermal contact with the heat generating regions. The result is regions of “unworked fluid”, as named by da Silva et al. [18] for parallel plates. In Fig. 1(b), where θ is small the boundary layers merge and the fluid between the fins behaves like channel flow. In this instance the fluid is said to be “overworked” [18]. The optimum angle, θ, is achieved when the thermal boundary layers merge at the exit of the heat sink, Fig. 1(c). This optimum angle can be estimated by considering the laminar boundary layer equations [16], and represents the optimum balance between pressure drop and heat transfer from the heat conducting fins.

A recent approach to enhancement of heat transfer densities is the use of multiple scales in heat exchanger design [19, 20]. This approach is in line with the constructal approach to achieve better designs by endowing the flow configuration with the freedom to morph [21]. The multi-scale approach has been demonstrated numerically for both cylinders [22,23] and parallel plates [18,24] in natural and forced convection. Enhanced heat transfer densities per unit volume were achieved with the addition of new smaller length scales in each case, typically heat flux enhancement of order ten percent for a fixed pressure drop are found in the multi-scale designs. The strategy employed by others is to utilise the unworked fluid between adjacent boundary layers by placing a heat generating plate at the inlet between the existing older architecture. The length of the new plate can be found graphically from the intersection of its boundary layers with the boundary layers of the older single scale architecture. Continually smaller length plates may be added to occupy the regions of unworked fluids in a similar manner. This philosophy for a radial heat sink arrangement is demonstrated graphically in Fig. 2(a). However, in the case of the radial heat sink, an additional parameter is added to the optimisation effort since the velocity is inversely proportional to radial position, and therefore the velocity that sweeps each new smaller scale fin is dependent upon its radial location. Hence placing the new smaller plates near the flow inlet, in the regions of high velocity unworked fluid, will result in high friction losses due to shear stress causing a reduced flow rate, leading to an overall reduction in the heat transfer density due to reduced flow rate. A possibility is to place the smaller scale surfaces closer to the flow exit, where the velocity is low and hence frictional losses are reduced. Fig. 2(b) shows the proposed multi-scale architecture. In this architecture the thermal boundary layers of the smaller scales are no longer confined to the areas of unworked fluid, but interact with the least utilised regions of the thermal boundary layers, i.e. the thermal boundary layer regions farthest from the heat generating walls.

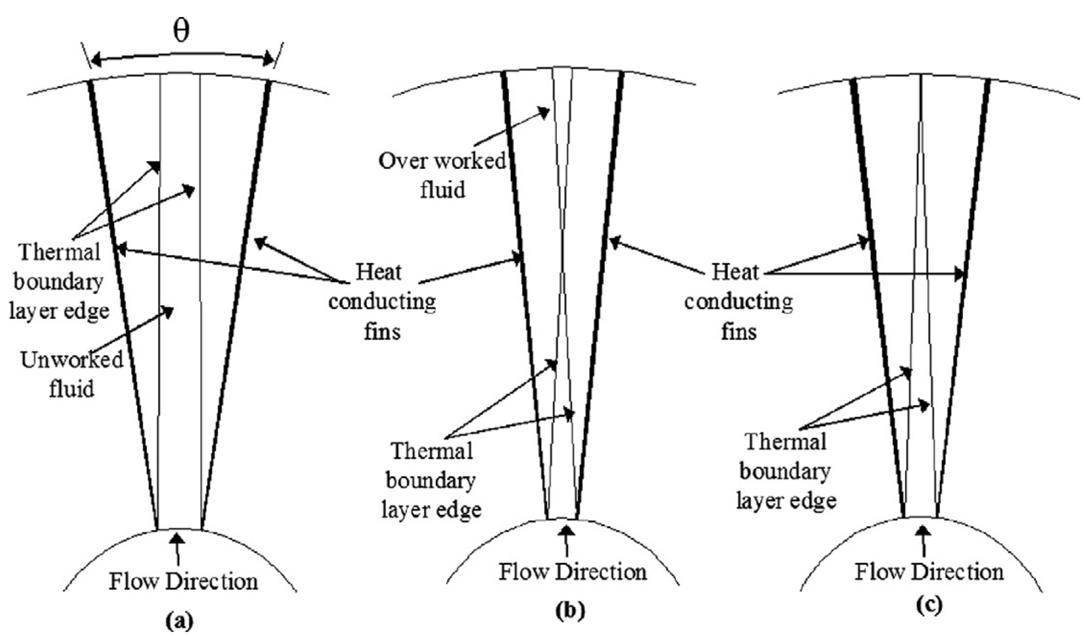

Fig. 1. Optimal angle between plates in radial flow heat exchanger; (a) Spacing greater than optimal, (b) spacing smaller than optimal, (c) spacing at optimal.

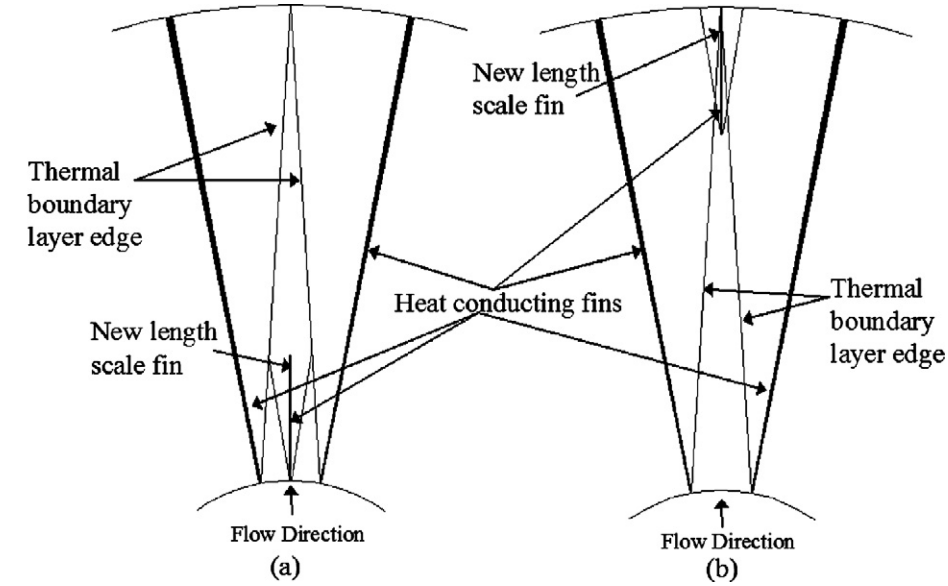

Fig. 2. Placement of smaller plates in radial flow heat sink; (a) Multi-scale structure with smaller scale fins placed at flow inlet, (b) multi-scale structure with smaller scale fins placed at flow outlet.

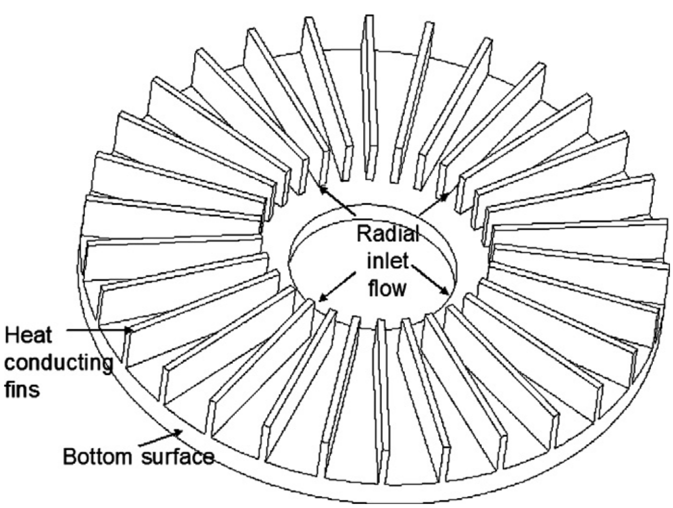

Fig. 3. Schematic of three-dimensional heat sink with upper surface removed.

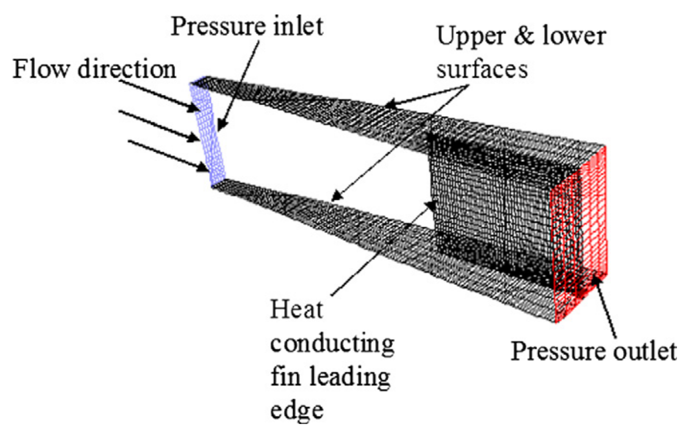

Fig. 4. Simulated model to illustrate the interaction of the upper and lower surface of a radial flow low profile heat sink with the fins.

2.2. Three-dimensional analysis

A key feature of low profile heat sinks is that the distance between the lower and upper surfaces of the heat sink are the same order of magnitude as the length scale between adjacent fins, when optimum spacing based on the two-dimensional analysis is applied. Hence, a three-dimensional interaction between the thermal boundary layers that grows on the fins and on the upper and lower surfaces of the heat sink causes additional heat transfer effects that must be considered. To illustrate this important phenomenon, a 3-dimensional CFD model was constructed utilizing Fluent 6, which uses a finite volume numerical method. A laminar model was used that solves the conservation equation for mass, momentum and energy, the energy equation option required to model the heat transfer aspect of the flow. An illustration of such a low profile single scale heat sink with the upper surface removed for visualisation is shown in Fig. 3. The model, illustrated in Fig. 4, represents an eight-degree segment from a radial heat sink such as that in Fig. 3 (not to scale) showing the boundary conditions and constant temperature heated surfaces of the model. The pressure drop from inlet to outlet was set to 0.1 Pa and the flow direction was normal to the inlet boundary. The wall temperatures were set to a constant 400 K. The Reynolds number based upon the radius and exit velocity of the heat sink is approximately 250, and so the laminar solver was employed. The grid size was 23 400 cells and grid sensitivity was investigated through refinement of the grid size by doubling the cell number in the radial, circumferential and axial directions. This increases the total cell count by a factor of 8. The change in heat flux was less than 0.2% and hence the lower cell number was sufficient to represent the flow in reduced convergence time. Two criteria were applied to ensure good convergence, the residual level of continuity, energy and velocity components was below 1 e^{-6} and the residual had reached a constant value for over 500 iterations. The dimensions of the simulated heat sink were twenty millimetres in

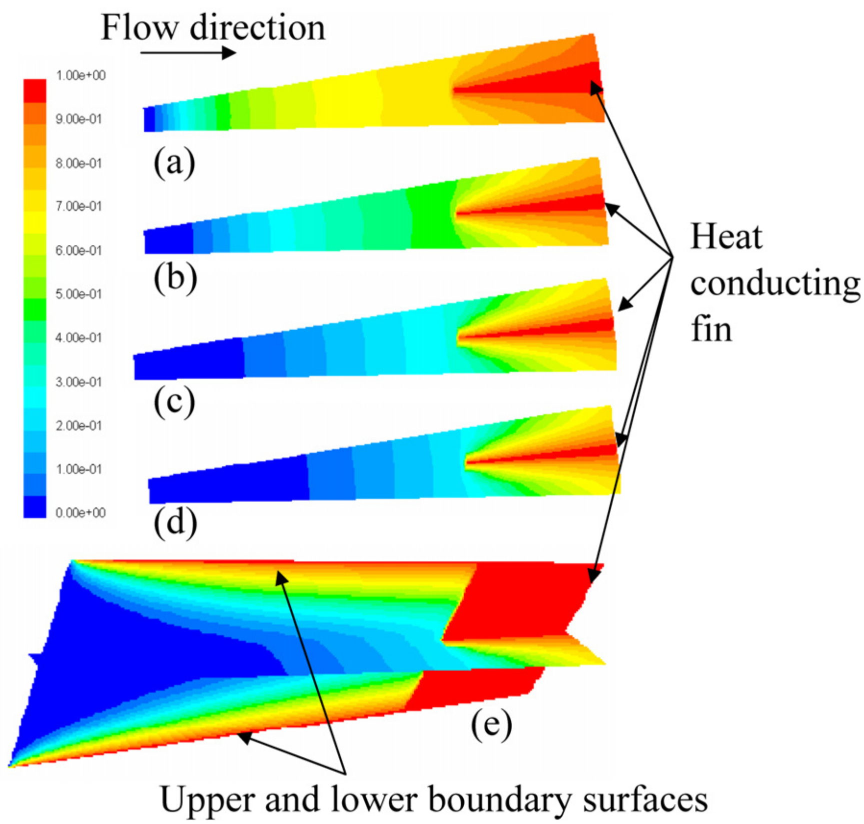

Fig. 5. Predicted iso-surfaces of constant grid in the axial direction (a)–(d), and angular and axial planes (e). (a) Axial plane 0.5 mm above lower surface, (b) axial plane 1 mm above lower surface, (c) axial plane 1.5 mm above lower surface, (d) axial plane 2 mm above lower surface (along center plane in axial direction), (e) center planes of angular and axial positions superimposed. The non-dimensional temperature range scale is also shown.

radius and four millimetres in height. Again, this model is to further demonstrate the unique aspects of the low profile heat sink flow and is not attempting to model any of the experimental investigations and is for qualitative, not quantitative purposes. Fig. 5 shows the predicted non-dimensional temperature distribution along specified planes in the radius–angle plane. The non-dimensional temperature distribution may be expressed as:

$$\widetilde{T} = \frac{T_{\mathrm{local}} - T_{\mathrm{amb}}}{T_{\mathrm{W}} - T_{\mathrm{amb}}} \tag{1}$$

The interaction of the thermal boundary layers between the upper and lower surfaces of the heat sink and the fin is evident. Fig. 5(a) is the closest plane to the lower surface and we see that the fins leading edge is initially contacted by heated fluid due to the growth of the boundary layer on the lower surface. Upon moving away from the surface, (b)–(d), it may be seen that the effect of this interaction is reduced and the fin provides increased heat transfer per unit area, essentially tending toward the two-dimensional case.

Fig. 6 illustrates this point further where a temperature profile in the radial direction along each axial plane presented in Fig. 5 is shown. The profile is taken midway between adjacent fins, i.e. along the edge of the numerical domain. As the heat flux is driven by a temperature difference, the plane closest to the surface will contribute less to the overall heat transfer rate than the centre planes. The end result is that the fin area close to the surface provides a relatively low level of heat transfer, but still contributes to the frictional losses.

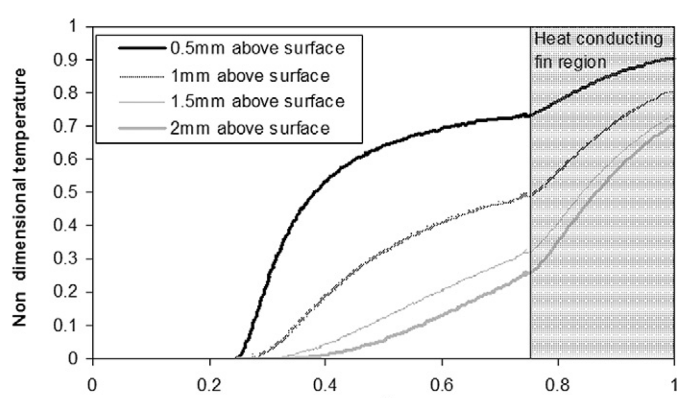

Fig. 6. Non-dimensional temperature along a line midway between the fins on planes 0.5, 1, 1.5 and 2 mm above the surfaces as shown in Fig. 5.

2.3. Experimental apparatus

This work experimentally investigates the hypothesis of utilising different configurations for heat sink in low profile applications. The chosen dimensions of both heat sinks considered between the upper and lower surfaces is 4 mm, with a 20 mm radius, as this represents potential heat sink dimensions for portable electronics such as a mobile phones and palm communicators. It is also anticipated that three-dimensional effects are present in the heat transfer phenomena as a strong interaction between the boundary layers will be present. This is because for the *Re* number range measured, the thermal boundary layers on the upper and lower surfaces will have a significant interaction with the thermal boundary layers that grow on the finned surfaces.

To provide mass flow through the heat sinks we designed and integrated a mini-scale radial fan and motor into the heat sink arrangements. The motor has a variable unloaded rotational speed from 0–60 000 RPM. The eleven millimeter diameter fan was manufactured using rapid prototyping and consists of 10 blades of 2.88 mm in height, with a base thickness of approximately 0.5 mm. Fig. 7 shows photographs of the manufactured fan, with the velocity triangles when the rotor is operated in forward and backward curved blade orientations are also shown.

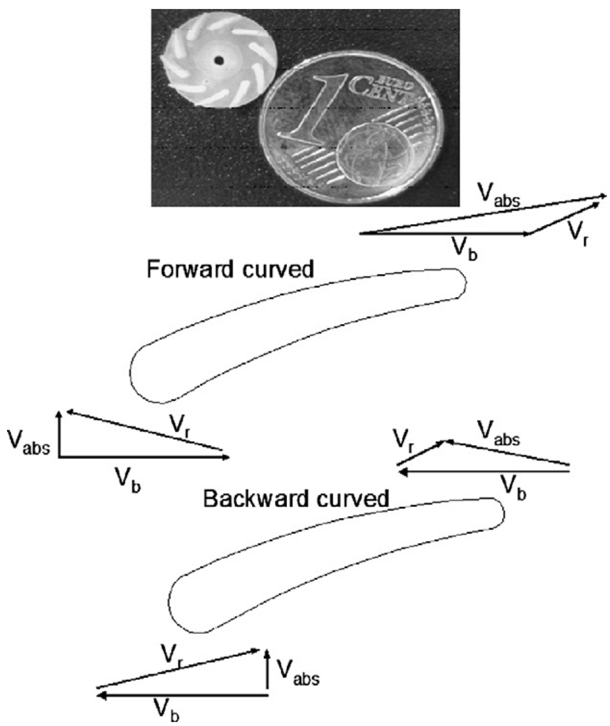

Fig. 7. Fan and size relative to a one cent coin with inlet and exit velocity triangle for both directions of rotation.

This simple velocity triangle analysis is important when interpreting and understanding the results as this affects the aerodynamic performance of the blades. Further details and aerodynamic performance characteristics of the utilised fan and motor were reported recently [12]. The maximum Reynolds number based on the exit velocity and heat sink radius was calculated to be less than 400, indicating that the flow will be laminar.

A number of heat sinks were manufactured from copper, to obtain isothermal surfaces, utilising a micro CNC machine. These prototypes reflect modern design methodologies, as discussed above with respect to fin spacing. Fig. 8 shows the manufactured heat sinks, which have channel dimensions ranging from 1.2 to 4.7 mm. The thickness of all fins was 0.5 mm, with fin lengths of 3.5, 11 and 14 mm for the non-radial small scale fins, radial fin and non-radial fins respectively. The non radial fins were manufactured at forty five degree angles to the radial plane. The mini-fan in Fig. 7 is inserted into the center of each heat sink and held in place with high temperature kapton adhesive tape during the characterisation phase of this investigation. Each heat sink will reach its optimum performance at different fan operating conditions. Varying the fan rotational speed will vary the pressure drop across the heat sink and hence alter the non-dimensional Bejan number, which is used to characterise geometrically similar forced convection heat sinks [20–22,24]. However within the operating envelope of the mini-fan it is expected that each design will pass through its optimum thermal performance conditions. This is important as the aim of the current study is not to design a heat sink for single point optimum performance but rather to investigate the performance of the heat sink designs against each other when integrated with the mini-fan at low profile scales. The current literature lacks such investigations. Note that in normal operation, integrated heat sinks and fans of the type described will have the same pressure at inlet and outlet, and so the flow and heat transfer values presented are representative of operational values.

The pressure-flow characteristics of the integrated fan and heat sinks were measured utilising the facility illustrated in Fig. 9. Air enters the test facility through the orifice plate (1) which measures the flow rate. The flow rate was calculated from the pressure drop across the orifice plate measured at the orifice pressure tappings (2). The relationship between flow rate and pressure drop was previously calibrated by Hanly

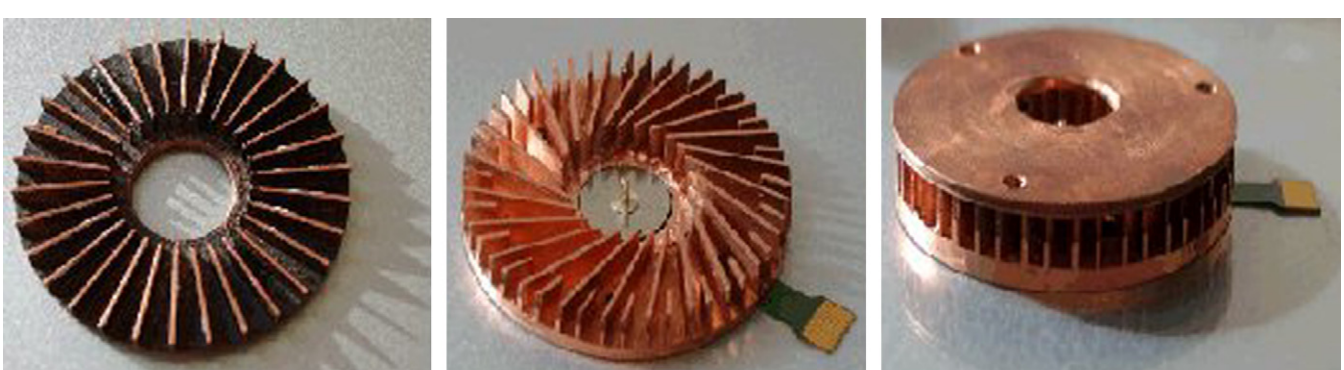

Fig. 8. Manufactured heat sinks. Single scale radial with 30 fins (left), multi-scale tangential with 24 fins of each scale (middle), finned heat sink with cover as used in characterisation (right). Fins have a thickness of 0.5 mm.

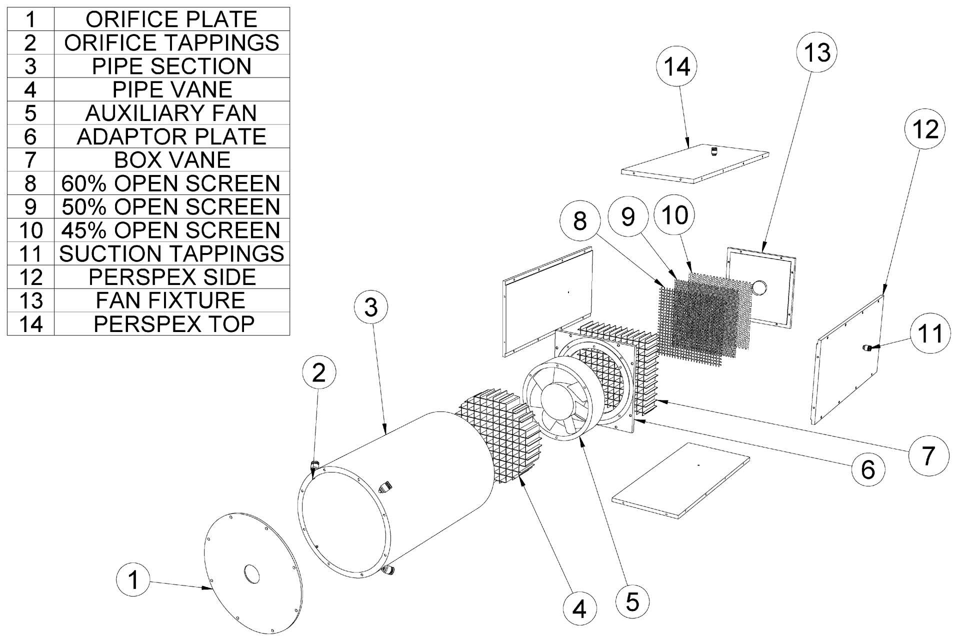

Fig. 9. Flow rate measurement facility for integrated fan and heat sink thermal management solution.

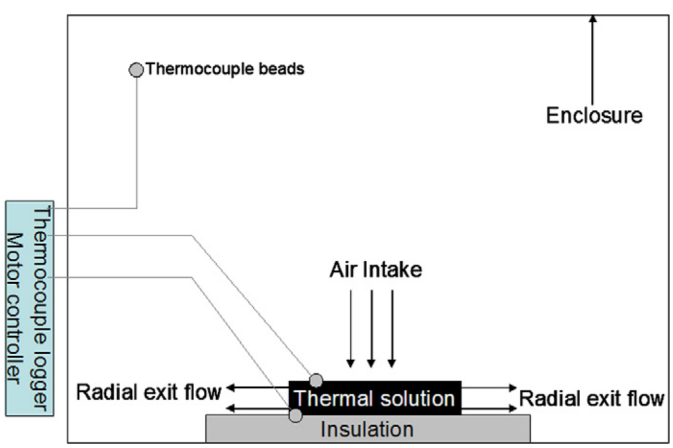

Fig. 10. Schematic of experimental setup for thermal resistance measurements.

et al. [12]. The air then passes through the pipe section (3) and then through the straightening vanes (4) to remove swirl. The air then passed through the auxiliary fan (5), which is used to overcome the pressure loss of the facility and provide atmospheric inlet pressure conditions to the heat sink. Flow then passes through another set of straightening vanes (7). Once air passed the second straightening vane it entered the fan inlet chamber. Three settling plates (8), (9) and (10) ensured uniform velocity in the air passing the fan inlet tappings. The differential static pressure across the heat sink was measured using these tappings. The total length of the facility is approximately 150 mm.

An annulus shaped thin film heater, available from Minco, was attached to the lower surfaces of each prototype to provide the heat source. The experimental setup is illustrated in Fig. 10, where the thermal management solution represents the integrated fan and heat sink solutions. Flow enters axially and exits in a radial direction. The enclosure is necessary to isolate the measurement from any airflow in the larger room. The enclosure volume is several thousand times greater that of the thermal solution and hence its effect on the result is negligible. Thermocouples were set into drilled holes on the upper and lower surfaces of the heat sink and back filled with epoxy, such care is necessary when using standard thermocouples. A thermal paste was used to thermally connect the top plate to the heated heat sink and good temperature uniformity over the heat sink surface was recorded using thermocouple readers. The temperatures were recorded after steady state conditions were reached. A third thermocouple was used to measure the ambient air temperature inside the enclosure. The measured voltage and current to the Minco thin film heaters was recorded and used to calculate the heat flux of the integrated fan and heat sink solutions, typically two watts. From these measurements the thermal performance of the solution can be presented in terms of thermal resistance given by:

$$R_{\mathrm{th}} = \frac{T_{\mathrm{W}} - T_{\mathrm{amb}}}{Q_{\mathrm{measured}}} \tag{2}$$

The thermal resistance is a measure of the temperature increase per unit power dissipated in a device and is the quantity used to describe heat sink performance. It is somewhat misleading, however as it camouflages information about the size of the heat sink, or indeed the size of the fluid mover required in forced convection designs.

3. Uncertainty analysis

The uncertainty in the measurement of pressure, voltage, current and temperature were estimated to be 0.5 Pa, 0.01 V, 0.01 A and 0.5 K respectively. Using these values the uncertainty in thermal resistance and relative volumetric flow rates were calculated using the method of Kline and McClintock [25]. This method uses Eq. (3) to determine the uncertainty in a given variable above and below the actual calculated value, where the uncertainty of the independent variables is known.

$$W_E = \sqrt{\left(\frac{\delta E}{\delta x_1} W_1\right)^2 + \left(\frac{\delta E}{\delta x_2} W_2\right)^2 + \cdots + \left(\frac{\delta E}{\delta x_n} W_n\right)^2} \qquad (3)$$

The maximum uncertainty in the thermal resistance and relative flow rate was less than 5% and 1% respectively. However, the absolute error in the volumetric flow rate was substantially higher than the relative value with the calibration curve for the utilised orifice plate having a scatter of up to 10% [26]. The tachometer has a resolution of 1 RPM but during experimentation the uncertainty in rotation speed was estimated to be approximately 50 RPM as fluctuations were observed in the measurement close to this level over the duration of the tests.

4. Results and discussion

Fig. 11 summarises the relative volumetric flow rate measurements of the integrated fan and low profile heat sinks over a range of rotational speeds. Measurements were performed with the rotor operating in both forward and backward curved blades orientations, i.e. the rotational direction of the motor was varied. The forward and backward curved blades are represented by the open and filled symbols respectively in Fig. 11. For both the radial and tangential heat sinks, an increase in flow rate occurs when the fan is operated with backward facing blades. This can be explained through Fig. 7, where the inlet velocity triangle demonstrated that when the rotor is operated as in the forward curved blade orientation, as illustrated in the middle image in the figure, the relative inlet velocity is at a high incidence angle with the rotor blade. The result of this is a reduction in aerodynamic efficiency of the blade design, where a large separation region can be expected to exist in the leading edge region due to the large adverse pressure gradient on the suction surface. The result is a reduction in the mass flow. When operated in the backward curved direction, the inlet flow has a low incidence angle and hence higher aerodynamic efficiency, resulting in a higher mass flow rate. For both rotational directions, the flow rate is higher in the tangential than the radial heat sink. This higher flow rate in the tangential heat sink was expected when the fan was operating in the forward curved direction as the fins were shaped to match the exit flow angle from the rotor, and thus reduce the pressure drop across the heat sink. In the backward curved direction we also note a substantial increase in mass flow rate for the tangential heat sink. Initially this was unexpected; as the flow leaving the rotor will impinge upon the tangential fins causing larger regions of separation than in the radial finned case resulting in increased pressure drop. However, this was not the case and it is most likely the result of the number of fins being reduced in the tangential heat sink thereby reducing the blockage effect close to the blade exit. Although speculative, it appears that additional aerodynamics losses due to the higher impingement angle in the tangential finned case when compared to the radial finned case is not excessive.

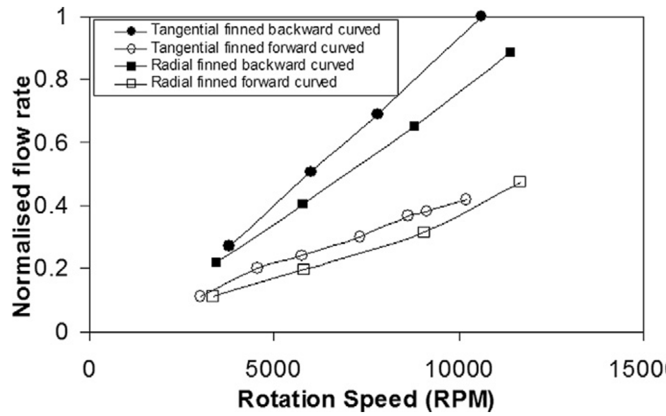

Fig. 11. Measured flow rate for integrated mini-fan and radial finned, and shaped finned heat sinks over a range of rotational speeds.

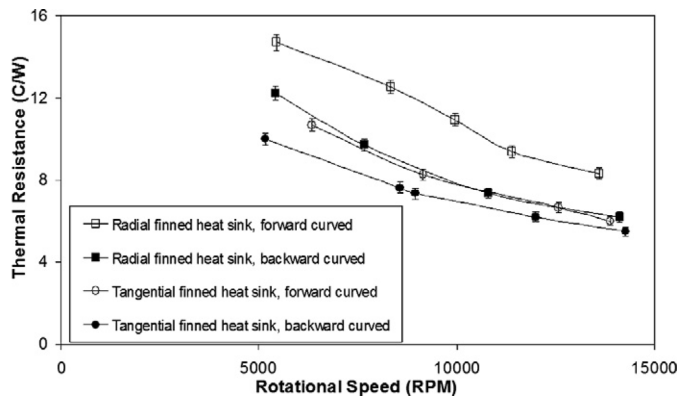

Fig. 12. Measured thermal resistance for integrated mini-fan and radial and tangentially finned heat sink over a range of rotational speeds. Error bars calculated from Kline and McClintock [25].

This trend in flow rate is reflected in the thermal resistance measurements for both heat sinks as shown in Fig. 12, where the highest flow rate configuration results in the best heat sink thermal performance, although the difference is not linearly proportional to flow rate. It is interesting to note that although the radial heat sink with backward curved blades results in a large increase in flow rate over the tangential heat sink with forward curved blades, this increase is not reflected in the thermal resistance plots of Fig. 12, where both give a similar thermal resistance, highlighting the importance of heat sink design. An important design parameter for such an arrangement is that the rotational speed of the fan to be as low as possible to achieve the required thermal resistance in a given volume. Benefits arising from low rotational speeds include reduced acoustic noise, reduced power consumption and increased reliability. From the data presented in both Fig. 11 and Fig. 12 it is evident that the design of efficient thermal management solutions for low profile applications requires the design of the full thermal solution and not simply the heat sink or fan in isolation. Design methodologies should begin by addressing integrated solutions. This is important and demonstrates the need for considering more than simply the heat sink and fan in isolation when designing heat sinks combined with fan flow fields.

5. Conclusions

Two heat sinks with radial and tangential shaped fins, integrated with a mini-fan, have been investigated for the purpose of developing low profile heat sinks for portable electronic applications. Despite the limited research in this field to date and with the emergence of high heat generating portable devices, it will be the preferred method of cooling such portable devices. This paper has highlighted some design issues that are unique to low profile scales, primarily the fact that the temperature field will be three dimensional in nature with reducing profile, where classical approach to thermal management optimisation may no longer be valid. The experimental results demonstrate the need to consider the fan and heat sink in an integrated manner and further demonstrate the importance of efficient heat sink design at this scale, where the multi-scale tangential finned heat sink influenced the resultant thermal performance when compared to the more conventional radial finned case.

Acknowledgement

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