Development of effective thermal management strategies for LED luminaires

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Development of effective thermal management strategies for LED luminaires

By

James R. Pryde

A Doctoral Thesis
Submitted in Partial Fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University

Wolfson School of Mechanical and Manufacturing Engineering

September 2017

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Abstract

The efficacy, reliability and versatility of the light emitting diode (LED) can outcompete most established light source technologies. However, they are particularly sensitive to high temperatures, which compromises their efficacy and reliability, undermining some of the technology’s key benefits. Consequently, effective thermal management is essential to exploit the technology to its full potential. Thermal management is a well-established subject but its application in the relatively new LED lighting industry, with its specific constraints, is currently poorly defined. The question this thesis aims to answer is how can LED thermal management be achieved most effectively? This thesis starts with a review of the current state of the art, relevant thermal management technologies and market trends. This establishes current and future thermal management constraints in a commercial context. Methods to test and evaluate the thermal management performance of a luminaire system follow. The defined test methods, simulation benchmarks and operational constraints provide the foundation to develop effective thermal management strategies. Finally this work explores how the findings can be implemented in the development and comparison of multiple thermal management designs. These are optimised to assess the potential performance enhancement available when applied to a typical commercial system. The outcomes of this research showed that thermal management of LEDs can be expected to remain a key requirement but there are hints it is becoming less critical. The impacts of some common operating environments were studied, but appeared to have no significant effect on the thermal behaviour of a typical system. There are some active thermal management devices that warrant further attention, but passive systems are inherently well suited to LED luminaires and are readily adopted so were selected as the focus of this research. Using the techniques discussed in this thesis the performance of a commercially available component was evaluated. By optimising its geometry, a 5 % decrease in absolute thermal resistance or a 20 % increase in average heat transfer coefficient and 10 % reduction in heatsink mass can potentially be achieved. While greater lifecycle energy consumption savings were offered by minimising heatsink thermal resistance the most effective design was considered to be one optimised for maximum average heat transfer coefficient. Some more radical concepts were also considered. While these demonstrate the feasibility of passively manipulating fluid flow they had a detrimental impact on performance. Further analysis would be needed to conclusively dismiss these concepts but this work indicates there is very little potential in pursuing them further.
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Nomenclature

Units

As per / derived from the International System of Units, SI, wherever possible

$S$ Dollar (unit of monetary value expressed in currency of United States of America)

°C Degree Celsius (unit of temperature relative to 273.15 K, $\Delta 1 ^\circ C = \Delta 1 K$)

Ω Ohm (unit of electrical resistance, $1 \Omega = 1 \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-3} \cdot \text{A}^{-2}$)

A Ampere (unit of electric current, SI base unit)

cd Candela (unit of luminous intensity, SI base unit)

Hz Hertz (unit of frequency, $1 \text{Hz} = 1.\text{s}^{-1}$)

J Joule (unit of energy, $1 \text{J} = 1 \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-2}$)

K Kelvin (unit of temperature, SI base unit)

kg Kilogram (unit of mass, SI base unit)

lm Lumen (unit of luminous flux, $1 \text{lm} = 1 \text{cd} \cdot \text{sr}^{-1}$)

m Metre (unit of length, SI base unit)

N Newton (unit of force, $1 \text{N} = 1 \text{kg} \cdot \text{m} \cdot \text{s}^{-2}$)

Pa Pascal (unit of pressure, $1 \text{Pa} = 1 \text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-2}$)

rad Radian (unit of plane angle, $2\pi \text{rad}$ in a circle)

s Second (unit of time, SI base unit)

sr Steradian (unit of solid angle, $4\pi \text{sr}$ in a sphere)

t Tonne (unit of mass, $1 \text{t} = 1000 \text{kg}$)

V Volt (unit of electrical potential, $1 \text{V} = 1 \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-3} \cdot \text{A}^{-1}$)

W Watt (unit of power, $1 \text{W} = 1 \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-3}$)

Wh Watt hours (unit of energy, $1 \text{Wh} = 3.6 \times 10^3 \text{J}$)

Symbols

$\mu$ Dynamic viscosity ($\text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$)

$\rho$ Density ($\text{kg} \cdot \text{m}^{-3}$)

A Area ($\text{m}^2$)

D Diameter ($\text{m}$)

g Gravitational acceleration ($9.81 \text{m} \cdot \text{s}^{-2}$)

h Average heat transfer coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)

P Power ($\text{W}$)

R Thermal resistance ($\text{K} \cdot \text{W}^{-1}$)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re</td>
<td>Reynolds number (dimensionless parameter)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (K)</td>
</tr>
<tr>
<td>v</td>
<td>Velocity (m.s(^{-1}))</td>
</tr>
<tr>
<td>W</td>
<td>Wetted perimeter (m)</td>
</tr>
</tbody>
</table>

**Superscript**
- ° Degree (unit of plane angle, 1° = (π / 180).rad)

**Subscript**
- θ Theta (denoting a thermal parameter)
- amb Ambient (referring to ambient environment)
- c Convective (referring to convective heat transfer)
- e Electrical (denoting an electrical parameter)
- h Hydraulic (denoting a hydraulic parameter)
- high High (referring to an upper reference point)
- low Low (referring to a lower reference point)
- r Radiative (referring to radiative heat transfer)
- ref Reference (referring to a specified reference point)

**Prefix**
- $ Dollar (monetary value expressed in currency of United States of America)
- € Euro (monetary value expressed in currency of European Union)
- £ Pound (monetary value expressed in currency of Great Britain)
- Δ Delta (difference between two values)
- μ Micro (multiplication factor, x10\(^{-6}\))
- c Centi (multiplication factor, x10\(^{-2}\))
- k Kilo (multiplication factor, x10\(^{3}\))
- M Mega (multiplication factor, x10\(^{6}\))
- m Milli (multiplication factor, x10\(^{-3}\))
- n Nano (multiplication factor, x10\(^{-9}\))
- T Tera (multiplication factor, x10\(^{12}\))
Suffix
bn  Billion  (multiplication factor, short scale definition, \(x10^9\))

Acronyms and Abbreviations
AlGaInP  Aluminium Gallium Indium Phosphide
AP  Acidification Potential
ARD  Abiotic Resource Depletion
CAD  Computer Aided Design
CFD  Computational Fluid Dynamics
CFL  Compact Fluorescent Lamp
COB  Chip On Board
CRI  Colour Rendering Index
CSP  Chip Scale Packaging
DC  Direct Current
DMM  Digital-Multimeter
EDP  Ecosystem Damage Potential
EP  Eutrophication Potential
EPD  Environmental Product Declaration
EUR  Euro (see €)
FAETP  Freshwater Aquatic Ecotoxicity Potential
FR4  Flame Retardant 4 (glass-reinforced epoxy laminate sheet designation)
GaN  Gallium Nitride
GaP  Gallium Phosphide
GBP  Great British Pound (see £)
GLS  General Lighting Service
GWP  Global Warming Potential
HDDC  High Density Die-Casting
HTP  Human Toxicity Potential
HWL  Hazardous Waste Landfill
LED  Light Emitting Diode
LU  Land Use
MAETP  Marine Aquatic Ecotoxicity Potential
MCPCB  Metal Core Printed Circuit Board
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Full Form</th>
</tr>
</thead>
<tbody>
<tr>
<td>NHWL</td>
<td>Non-Hazardous Waste Landfill</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone Depleting Potential</td>
</tr>
<tr>
<td>OLED</td>
<td>Organic Light Emitting Diode</td>
</tr>
<tr>
<td>PC</td>
<td>Phosphor Converted</td>
</tr>
<tr>
<td>PCB</td>
<td>Printed Circuit Board</td>
</tr>
<tr>
<td>POCP</td>
<td>Photochemical Ozone Creation Potential</td>
</tr>
<tr>
<td>QD</td>
<td>Quantum Dot</td>
</tr>
<tr>
<td>RWL</td>
<td>Radioactive Waste Landfill</td>
</tr>
<tr>
<td>SiC</td>
<td>Silicon Carbide</td>
</tr>
<tr>
<td>SLS</td>
<td>Selective Laser Sintering</td>
</tr>
<tr>
<td>TAETP</td>
<td>Terrestrial Ecotoxicity Potential</td>
</tr>
<tr>
<td>TIM</td>
<td>Thermal Interface Material</td>
</tr>
<tr>
<td>USD</td>
<td>United States Dollar (see $)</td>
</tr>
<tr>
<td>UV</td>
<td>Ultra Violet</td>
</tr>
<tr>
<td>VIA</td>
<td>Vertical Interconnect Access</td>
</tr>
</tbody>
</table>
Chapter 1:
Introduction

As will be discussed, LEDs are revolutionising general lighting, but impose a number of physical challenges. In particular their thermal operating conditions have a critical impact on performance. Consequently, there is a need to ensure they are deployed effectively. This thesis explores the topic of their thermal management to define what constitutes an appropriate strategy and how it can be implemented to greatest effect. This chapter discusses some of the topics and principles necessary to the understanding of this subject. The justification and objectives of this research follow. Finally, a broad overview of the research is provided.

1.1 Fundamentals of light and its perception

To establish some context for the work to follow, it is worth understanding the nature of light, its importance to society and the development of associated technologies through history. This helps define a number of important criteria to support the following research.

Visible light is a form of electromagnetic radiation. It arises from electromagnetic interaction, one of the four known fundamental interactions occurring between elementary particles, and propagates as an electromagnetic wave. Wave-particle duality simultaneously describes electromagnetic radiation as made up of discrete particles called photons. (Clugston, 1998). The wavelength of the emitted radiation relates to the energy it carries. The potential range of energies result in a spectrum of waves with different wavelength (Fig. 1-1).

This phenomenon gives many organisms the ability to perceive and interact with the environment. Of particular interest are the wavelengths of light within the range of human perception (the visible spectrum). To facilitate this we, along with many other species, have evolved the eye. As described by Hubel (1995), the eye contains specialised cells containing photoreceptive complexes. In the presence of light these cells undergo a series of changes that results in an electrical signal sent to the nervous system. Interpreting these signals allows the brain to build an understanding of our environment. Beside stimulus / no stimulus, humans have
the ability to discern detail such as the intensity, contrast, texture, spatial characteristics and spectral composition (colour) of a light source or light reflecting surface (Caelli, 1981).

Fig. 1-1: Corresponding wavelengths, frequencies and energies of the electromagnetic spectrum (National Aeronautics and Space Administration, 2013)

The human eye contains two main types of photosensitive cell, known as rods and cones. Rods are extremely sensitive to low intensity stimulus but only across a limited range of wavelengths. Several types of cone cells are each sensitive to particular ranges of wavelengths, thereby enabling us to distinguish different colours, but our sense of colour is restricted by the types of photosensitive cells. This gives rise to the property of metamerism whereby spectrally-different stimuli can result in equivalent stimulation of photosensitive cells (Wyszecki and Stiles, 1982). This allows objects to appear identical under illumination sources with vastly different spectral compositions. However, the effect is not necessarily equivalent for all observers and can also lead
to unanticipated variation in perception when viewing objects under different conditions. This creates a particular challenge when assessing the quality and consistency of different light sources, but also provides opportunities to compensate for variations by exploiting this effect (Berns, 2000).

There are other photosensitive cells that play different roles not necessarily related to vision forming. For example, Wicks et al. (2011) report photoreceptive mechanisms in skin cells that trigger the production of melanin to protect from UV radiation and Hattar et al. (2002) report photoreceptive ganglion cells in the eye appear to play a role in regulating circadian rhythms in response to day / night cycles of light / dark. Gradually, we are beginning to understand the powerful influence light has on the human body beyond vision. For example, a review of the impact of light in buildings and its impact on human health by Boyce (2010) identifies a number of effects that are attributed to the effects of light. It concludes light has a significant impact on the body, both positive and negative, with effects including tissue damage, eye strain, potential links to increased incidences of cancer and aiding in the treatment of seasonal affective disorder. Mills et al. (2007) report how lighting has been shown to have a positive influence on vitality, depressive symptoms, alertness, task performance and sleep quality. They go on to note that particular types of lighting can even enhance productivity in commercial environments. As knowledge in the field develops we are beginning to understand how we can exploit and manage the aspects of human response through artificial lighting.

Discrete wavelengths of light are perceived as colours. However, many light sources emit a range of light wavelengths. A common definition of chromacity, derived from the spectral power of each wavelength of light multiplied by a weighting factor based on its stimulus value with respect to human vision, was developed by the International Commission on Illumination (Commission Internationale de L’Eclairage, CIE). This allows different colours to be presented in a colourspace chart as shown in Fig. 1-2. In the correct combination of magnitudes, different wavelengths produce what we perceive to be white light. However, there are a range of hues that fall within this category. To define a particular shade of white the temperature of an ideal light source (known as a black body radiator) emitting electromagnetic radiation solely through incandescence is used. This property is referred to as the colour temperature and measured in degrees kelvin (K). The profile of the ideal light source and the reference temperatures are marked on the image in Fig. 1-2. This is known as the Plankian locus. For light sources that do not fall exactly on this locus, the correlated colour temperature (CCT) is used.
The co-ordinates for the emission from an incandescent filament bulb are almost perfectly on the plankian locus, but other common light sources, such as fluorescent and LED, do not. This is because different light source types emit light at differing wavelengths and intensities. For example, the typical spectra of a traditional incandescent filament lamp, a halogen filament lamp, and a fluorescent lamp are shown in Fig. 1-3.
These three sources are perceived to produce white light, yet each has a different spectral composition. An object viewed under these light sources will reflect differing amounts of each wavelength of light, creating small differences in how it is perceived and its position on the colourspace reference chart. Metamerism can help to compensate for the disparate spectra, but a degree of variation often remains. The CCT does not define the spectral composition of the light source or its deviation from the ideal plankian locus. Consequently, means of defining the quality of the spectral composition have been proposed. One of the most common used in the lighting industry is the Colour Rendering Index (CRI). As explained by Hunt (1987), the CRI describes the average variation in the appearance of prescribed colour samples under the light source with reference to the nearest equivalent ideal illuminating source. There is now a consensus that this method is inadequate, owing to the out of date metrics and colour space definitions. There is also evidence that these criteria are unable to evaluate the distinct spectral composition of LED light sources on a basis consistent with other light sources and as a result underestimate their colour rendering properties (Luo, 2011). The ultimate goal remains a harmonised definition of the light
source across a complex and varied spectrum relevant to a diverse range of observing criteria. For the foreseeable future this is likely to remain the focus of considerable development. In the meantime the standard CRI model provides a simple reference value through which a light source’s accuracy in rendering colours can be readily quantified.

As understanding of light’s properties developed, there came the need for definitions and measurement methods. There are two categories of quantification: photometric and radiometric. Radiometric units relate to specific wavelengths of light; photometric criteria define the visual effect of a broad spectrum of light on the human eye. Photometric criteria are most relevant to this study of white light sources used in general illumination.

The candela is a measure of the luminous intensity of a light source. It was defined in 1979 under the international system of units (Le Systeme international d’unités, SI) as the luminous intensity of a monochromatic source with radiation frequency of $540 \times 10^{12} \text{ Hz}$ (corresponding to a wavelength of 555.016 nm) with a radiant intensity in that direction of $1 / 683 \text{ W per steradian (sr)}$ (National Physics Laboratory, 2010). The candela references a single wavelength of (green) light to which the eye is most sensitive. To account for other light wavelengths, a correcting function, as put forth by the CIE under joint ISO / CIE standard BS ISO 23539:2005 (British Standards Institute, 2005), can be applied. This quantifies the relative stimulus of different wavelengths of light, distinguishing between high and low light conditions using different functions. It is limited in that the visual response of an individual observer can significantly differ from the standard correcting function as a result of physiological and psychological variation. However, much has been done to develop accurate and typical standard observer models (Wyszecki and Stiles, 1982), enabling photometric properties to be calculated with reasonable confidence.

Expanding on the definition of the candela comes the definition of luminous flux measured in lumens (lm). It has the following definition:

$$1 \text{ lm} = 1 \text{ cd} \times 1 \text{ sr}$$

(Hunt, 1987)

This essentially defines the total quantity of useful light being emitted. However, as it is based on the candela it shares the same flaws.
These metrics offer the means to compare and evaluate the performance of different lighting technologies. Of particular interest is efficacy: the useful white light generated from the supplied power. Efficiency is not appropriate as white light is comprised of a spectrum of wavelengths offering varying degrees of stimulus which must be accounted for. This raises the issue that the photometric theoretical maximum efficacy is dictated by the observer, illumination regime (high or low light levels use different photoreceptors of the eye) and spectral composition. For a single monochromatic source emitting light at the human eye’s peak sensitivity wavelength (defined as 555 nm exactly for the CIE standard photometric observer), under photopic conditions (high lighting levels), the theoretical maximum luminous efficacy is 683 lm.W⁻¹. Scotopic (low light level) vision has a theoretical maximum of 1700 lm.W⁻¹ at the slightly different wavelength of 510 nm exactly for the CIE standard photometric observer, owing to the different photoreceptive mechanisms employed (Hunt, 1987). However, scotopic vision has little relevance to the study of lighting, which by nature is intended to create conditions to allow photopic vision to occur and monochromatic light sources are also unacceptable for general lighting. For white light, theoretical maxima range from 250 lm.W⁻¹ to 370 lm.W⁻¹ depending on the colour temperature, spectral composition and observing criteria. For a 5000 K, 85 CRI light source the maximum efficacy has been calculated to be 365 lm.W⁻¹ (Murphy, 2012).

There are a number of different processes by which light can be produced and it is not practical or necessary to detail all of them here. Fig. 1-4 provides a basic summary of some of the most relevant mechanisms, with examples of where they occur. The main distinction to note is the difference between incandescence and luminescence. Incandescence refers to electromagnetic radiation as a consequence of atomic motion. This motion is a product of the thermal energy held by a material and requires high temperatures to emit significant amounts of light. Luminescence describes processes where energy is converted to electromagnetic radiation by a transition between energy levels within a material (Clugson, 1998).
1.2 A brief history of lighting

If nothing else, the presence of light is clearly critical to enable us to interact effectively with our environment. For this reason light sources have been a key technology throughout mankind’s history. Williams (1999) and Bowers (1998) both provide excellent accounts of the history of light which are summarised in the following material. Obviously, humans were initially reliant on light from the sun and to a lesser extent sunlight reflected by the moon (star light can be regarded as negligible). When these natural sources of light were unavailable, our sight was severely limited. Driven by needs such as productivity, safety, defence and practicality came development of new methods of generating light. Early man would have been limited to fire, initially from natural occurrences but eventually would have come mastery over the means to create it. The release of energy accompanying combustion provided the energy required to cause incandescence and provide light along with benefits such as heat for cooking. As the understanding of various fuels’ behaviour developed, there came the means to create torches that burn brightly in a compact and mobile form. Over millennia this was refined to become the lamp. Archaeological evidence

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**Fig. 1-4: Summary of light generating mechanisms (Encyclopaedia Britannica, 2016)**

<table>
<thead>
<tr>
<th>Light generating mechanism</th>
<th>Luminescence</th>
<th>Incandescence</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Luminescence</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electroluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conversion of electrical energy to electromagnetic radiation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Examples include:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas discharge lamp</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Arc lamp</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LED</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Photoluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conversion of electromagnetic radiation to different wavelength</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Examples include:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fluorescent materials</td>
<td></td>
<td></td>
</tr>
<tr>
<td>employed in gas</td>
<td></td>
<td></td>
</tr>
<tr>
<td>discharge lamps to convert UV to visible light</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Phosphorescent materials</td>
<td></td>
<td></td>
</tr>
<tr>
<td>employed alongside LEDs to convert monochromatic emission to a broad spectrum</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Notable others include:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chemiluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Bioluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mechanoluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Triboluminescence</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Radioluminescence</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
suggests these developed at least 70,000 years ago and originally consisted of a hollowed out rock filled with an absorbent material such as moss soaked in animal fat (Encyclopaedia Britannica, 2012 a). This made light sources more manageable and easier to position. Lamp designs evolved through the ages, reflecting the materials used and the sophistication of the manufacturing methods. Eventually came the invention of the candle around 3000BC (Encyclopaedia Britannica, 2012 b), using solid wax as the fuel source. Much later came the advent of gas as a fuel source which found widespread application in early street lighting (Encyclopaedia Britannica, 2012 a).

These technologies were steadily refined and improved. Less polluting, more practical and more effective concepts were continually sought, which drove supplementary developments. Different optical controls such as reflectors and lenses helped improve the distribution of light. Similarly, advances in oil and gas lamps to improve the luminosity of the flame, consistency and efficiency were all made over time. Advances in chemistry and physics brought about new developments to improve output such as the gas mantle, the first effective iteration of which was demonstrated in the 1800s (Encyclopaedia Britannica, 2012 c). This contained various compounds, that when placed in a flame would emit an intense bright white light greatly improving illumination. The discovery that heating calcium oxide (quicklime) in an oxyhydrogen flame produced an incredibly intense white light opened up new opportunities. This ‘limelight’ found some application in general illumination but was commonly found in theatre and stage lighting, where the intense and controllable output were extremely useful. However, the apparatus required was not scalable enough to employ in general purpose applications, so it was eventually relegated by alternative technologies.

The light sources discussed so far all relied on combustion as a means of releasing the energy needed to produce light. This releases particulates and other pollutants as a by-product. They also demand a constant supply of fuel and are inherently hazardous. Around the start of the 19th century came the advent of electricity as a power source, which offered new possibilities of circumventing these issues. Sir Humphry Davy offered early demonstrations of the potential for electrically-powered light generation. One method used a current passed between two separated carbon electrodes. An electrical arc would form between them making the intervening air electroluminesce, whilst also causing the electrodes to incandesce from the released heat. At the same time Davy demonstrated the use of electricity to create light by passing it through a thin platinum strip. Resistance to the passage of current resulted in internal heating causing the metal
strip to incandesce. These concepts formed the basis of two major lighting technologies of the past two centuries.

In the 1850s the arc light concept was the first to establish itself as a commercially viable light source. Its high output made it ideal for use in searchlights, floodlights and projectors. According to Bowers (1998), accounts of the time noted examples that would emit 700 candlepower from a 1 horsepower generator (approximately 11.8 lm.W⁻¹), though the accuracy of such claims must be treated with scepticism. The high luminous flux made arc lights well suited to street lighting. Throughout its history it underwent development to improve operating lifetime by minimising consumption of the electrodes and introduction of new materials to improve the luminosity of the source. It was in use as recently as the Second World War in anti-aircraft searchlights and street lighting. However, its potential was ultimately limited by the practicalities of scaling the technology down, its high power consumption, the heat it produced and the associated hazardous emissions.

The alternative incandescent filament bulb took longer to develop but undoubtedly found much wider application. Since its first demonstration numerous names have been associated with its development. The best known were Thomas Edison and Sir Joseph Swan (Encyclopaedia Britannica, 2012 c). In 1881, filament bulbs demonstrated at a Paris exhibition were achieving outputs of 150 - 200 candlepower per horsepower (Bowers, 1998). Subsequent developments led to the filament bulb as it is known today: a fine coiled tungsten filament sealed in an inert gas filled glass bulb. This design is reasonably simple and produces a very clean, practical and consistent light. However, the mechanism of light generation still requires very high temperatures. This makes it inherently inefficient because most of the supplied power is dissipated as heat while only a small percentage (approximately 2 %, see Table 1-1) is converted to useful light.

By sending an electrical discharge through a gas, some atoms of the gas can become ionised. In the presence of an electric field, these ions are accelerated towards an oppositely charged electrode, gaining energy in the process. As the ion travels it may collide with a neutral atom, transferring its charge and returning to a lower energy state. In doing so its energy can be emitted in the form of electromagnetic radiation. This energy can then be emitted as electromagnetic radiation (Encyclopaedia Britannica, 2012 d). The properties of the gas and the electric field influence the nature of the emitted light, providing a number of possibilities. The phenomenon was first recorded by Jean Picard in 1675 (Encyclopaedia Britannica, 2012 e) and now primarily
forms the basis of the gas discharge lamp. As Bowers (1998) notes, early commercial versions showed higher efficacy, and therefore lower operating costs, than filament bulbs of the time (early 1900s). Consequently, they attracted considerable development. Later advances came when it was discovered a small amount of metal could be vaporised along with the gas to modify the emission spectrum. Sodium and mercury became possibly the metals most commonly used for this purpose. Sodium gas discharge lamps can deliver incredibly high efficacies, but produce almost monochromatic yellow light (Eastop and Croft, 1990). These became widespread in street lighting where the quality of light was not a high priority. Mercury based gas discharge lamps found widespread application in general lighting. These emit light almost entirely in the invisible ultraviolet (UV) range of the electromagnetic spectrum. To produce white light they are paired with a material to absorb the UV light and re-emit it at visible wavelengths by means of fluorescence, hence they are commonly known as fluorescent lamps. As Bowers (1998) discussed, adoption of the fluorescent lamp was initially slow owing to the poor quality of the light emitted, its tendency to flicker, and high installation costs resulting from their incompatibility with existing fittings. Gas discharge lamps also undergo a warm up cycle before reaching full output (Eastop and Croft, 1990). Although regulations require current gas discharge lamps to reach 60% of full output within 60 seconds (European Commission, 2009), it still limits their functionality.

A final technology worth noting is a variation of the gas discharge lamp commonly known as the electrode-less plasma or induction lamp to distinguish it from standard gas discharge lamps. An early version of this concept was patented by Nicola Tesla in the late 19th century (Morgan, 2009). Compared to the gas discharge lamp discussed previously, these still exploit electroluminescence of a gas in a sealed container but remove the electrode from that environment. Instead excitation is achieved by applying an electromagnetic field to the gas from an external source. Removing the electrode from the plasma means its degradation is not an issue, so the light source can be subjected to more extreme conditions for improved performance. Morgan (2009) reports one leading manufacturer’s claims of high efficacy (over 100 lm.W⁻¹), long lifetime (up to 40,000 hours) and a compact form.

When we consider the history of these light sources, there has been a clear trend towards higher efficacy (see Table 1-1). However, there is still a considerable gap between the best-performing technologies and the theoretical maximum efficacy. These technologies are also well established in industry. Consequently, they have reached maturity: they have been refined to the point that
there is very little potential for further improvement. To close the gap between the theoretical maximum and realised performance requires a move to new technologies such as the LED.

### Table 1-1: Approximations of different light source efficacy values

<table>
<thead>
<tr>
<th>Light source</th>
<th>Efficacy (lm.W⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Candle¹</td>
<td>0.02 - 0.22</td>
</tr>
<tr>
<td>Kerosene wick lamp¹</td>
<td>0.05 - 0.21</td>
</tr>
<tr>
<td>Incandescent Tungsten filament bulb²</td>
<td>12 - 20</td>
</tr>
<tr>
<td>Incandescent Tungsten filament, halogen bulb²</td>
<td>25</td>
</tr>
<tr>
<td>Compact fluorescent tube²</td>
<td>43</td>
</tr>
<tr>
<td>Fluorescent tube²</td>
<td>64</td>
</tr>
<tr>
<td>High pressure sodium gas discharge lamp²</td>
<td>90</td>
</tr>
<tr>
<td>Low pressure sodium gas discharge lamp²</td>
<td>143</td>
</tr>
<tr>
<td>Monochromatic light source (555 nm wavelength)</td>
<td>683 ³ (Theoretical maximum)</td>
</tr>
<tr>
<td>White light source (CCT of 5000 K, CRI of 85)</td>
<td>365 ⁴ (Theoretical maximum)</td>
</tr>
</tbody>
</table>

¹Eastop and Croft, 1990  
²Mahapatra et al., 2009  
³Hunt, 1987  
⁴Murphy, 2012

### 1.3 The Light Emitting Diode (LED)

LED is sometimes used as a broad term to denote any device employing the light generating technology. However, there are various levels of its integration. For clarity, the terms used in this thesis are:

- “The LED”, “LEDs” or just “LED” in general reference to the class of light source technology.
- “LED die” or “LED chip” denotes the semiconductor structure that converts electrical energy into light.
- “LED component” or “LED package” are used interchangeably to describe the functional component containing the LED die along with the additional electrical interconnections,
optical elements and physical structures that enable their processing by conventional techniques such as automated circuit board assembly.

- The “LED lamp” is a replaceable module that houses one or more LED die or LED components. It can be installed in a lighting fixture (luminaire).
- The “LED luminaire” is a complete fixture with one or more integrated (non-replaceable) LED die or components.

Spring et al. (2015) offer a thorough explanation of the structure, principles and developments relating to the LED. To summarise, the LED die is formed from two contacting semiconductor materials that create an electrical junction. These materials are doped with impurities that give rise to particular electrical properties. The n-type material contains impurities that give the material an excess of electrons that are readily released for conduction. It is paired with a p-type material or quantum well with which the released electrons from the n-type material readily recombine. The transfer of electrons between these materials involves a transition of energy states. Electrical power applied to the semiconductor junction provides the energy necessary for electrons to escape the n-type material. When these recombine with the p-type material, the electron alters to occupy a more stable state which embodies less energy. To satisfy this condition they must release the excess energy they carried which allowed the transfer. This can be realised through electromagnetic radiation, a process known as electroluminescence. The change in energy relates to the bandgap between the free and captive states of the electron and dictates the wavelength of the emitted electromagnetic radiation. By managing the properties of the materials employed in the semiconductor, it is possible to confine this emission to visible wavelengths of light. By its nature, the LED chip is a monochromatic light source. However, in general lighting applications a broad white spectrum is required. For this the LED chip is at a disadvantage and must rely either on multiple emitters in combination, or on phosphorescent coatings to convert its output. To simplify the processing and integration of the LED chip with the surrounding system it is built into a component package. A simplified schematic representation of an LED die and surrounding component packaging is shown in Fig. 1-5.
Bender et al. (2015) summarised the history of the LED’s development as follows. In 1907 Henry Joseph Round noted that certain materials luminesce under an applied electric current. It was not until the work of Nick Holonyak in 1962 that this phenomenon was exploited to create the first practical visible red LED component. These early components found applications in display and indicator devices where reliability, not necessarily intense light emission, was required. Later work to improve output and advances in materials opened up communication and data transfer applications. The work of Shuji Nakamura in 1994 created the blue LED die, which completed the available spectrum of primary colour emitters. The high energy blue light wavelength also permitted the use of phosphorescent materials. These down-convert a portion of the high energy, 450 nm wavelength, light emitted by the LED chip to longer yellow and red wavelengths, creating a broad spectrum white light source (Fig. 1-6).

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**Fig. 1-5:** Schematic representation of a phosphor-converted LED die and component package

**Fig. 1-6:** Spectral distribution of a blue LED die with phosphor conversion to create a white light emitting component (Samsung, 2016 a)
1.4 Market landscape and technological potential

The International Energy Agency (2006) offers a thorough summary of the global lighting industry. They noted in 2006 that an estimated quarter of the world’s population still had no access to electric lighting, a greater number of individuals than when the incandescent filament bulb was commercialised in the 1800s. The reliance on fuel-based lighting as an alternative is expensive, inefficient and the cause of numerous cardiac and respiratory deaths each year. Consequently, global demand for artificial electric light is projected to increase 80% by 2030 with an associated impact on energy consumption as plotted in Fig. 1-7. In 2005 approximately 19% of the world electricity consumption (2650 TWh) was attributable to lighting. Of this, the residential sector consumed an estimated 811 TWh at an average luminous efficacy of 21.5 lm.W\(^{-1}\) while the commercial sector demanded another 1133 TWh at an average efficacy of 52.5 lm.W\(^{-1}\). Further breakdown of this by nation and light source technology is available in the reference material. Increasing global energy consumption can be mitigated by improving the performance of the light source. It has been predicted that a 30 - 50% reduction in total energy consumption would be feasible, with the residential sector contributing 40 - 60% and the commercial sector 25 - 40%. Difficulty obtaining data means residential performance is more uncertain, but it is reasoned that the poorer average efficacy presents the greatest opportunity for improvement.

![Fig. 1-7: Increasing global lighting energy consumption by end use sector between 1995 and 2030 based on current trends and policies (International Energy Agency, 2006)](image-url)
Environmental pressures and sustainability incentives are understood to be significant influences on the lighting industry. The total energy consumed by lighting equates to the release of 1900 Mt of carbon dioxide into the atmosphere per year (although this also includes emission caused by non-electrical lighting sources such as paraffin lamps and consumption of fuel to power vehicle lights) (International Energy Agency, 2006). Based on 2014 emissions figures from the International Energy Agency (2016), this would equate to roughly 6 % of global emissions without accounting for any changes in the intervening years. Widespread concern regarding depleting natural resources, increasing energy demand, rising energy costs and consequences of pollution are creating growing incentives to adopt more efficient light sources such as the LED.

Propelled by developing nations, rising incomes and growing populations, the global lighting market value is estimated to increase by €27 bn during 2011 - 2020 to a total of over €100 bn (Fig. 1-8). This combines all lighting technologies and market sectors. Of this, LED systems accounted for just 12 % in 2011 but with predictions estimating that by 2020 they will represent 63 %. Excluding niche applications, such as screen backlighting, LED based products are projected to account for 52 % of worldwide lamp and luminaire shipment volume (McKinsey and Company, 2012). Clearly the potential value of the opportunities presented is considerable.

Fig. 1-8: Value of global lighting markets (Note some values are rounded so do not sum correctly) (McKinsey and company, 2012)
According to prevailing opinion, the tipping point for mass adoption of LED technology is an installed system capital cost of approximately $12.5 \text{ klm}^{-1}$ (United States Department of Energy, 2013). Table 1-2 summarises some typical market prices for different light sources, including Organic-LED (OLED), a sheet based version of the LED chip. The features and properties of LED components, particularly their reliability, diminish the need for them to be a replaceable element of the luminaire (i.e. a lamp). This makes it practical to integrate them directly into the system, providing commercial benefits such as control over their implementation and optimisation to suit the application. LED luminaires thereby exist as a distinct category. While the capital cost of LED devices remains at a significant disadvantage compared to established light source technologies, they have now reached an acceptable price point. As demonstrated by Fig. 1-9, both warm (low CCT \(\approx 2700\) K) and cool (high CCT \(\approx 6500\) K) packaged LED component prices and efficacies are expected to continue improving, which will further increase acceptance and market penetration. It is worth noting that as LED component prices decrease the cost of the surrounding luminaire represents a larger proportion of the total system capital cost. Table 1-2 suggests the luminaire adds approximately one third to the overall capital cost compared to the lamp alone. Therefore, there is a growing incentive to address these costs and a significant opportunity for improvement that would enhance adoption.

Table 1-2: Comparison of typical market prices for various light sources (United States Department of Energy, 2016)

<table>
<thead>
<tr>
<th>Light source</th>
<th>Price ($/\text{klm}^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Halogen Lamp</td>
<td>2.5</td>
</tr>
<tr>
<td>Incandescent Lamp</td>
<td>0.63</td>
</tr>
<tr>
<td>Compact Fluorescent Lamp (CFL)</td>
<td>2</td>
</tr>
<tr>
<td>Dimmable CFL</td>
<td>10</td>
</tr>
<tr>
<td>Linear fluorescent lamp with Ballast</td>
<td>4</td>
</tr>
<tr>
<td>LED Lamp</td>
<td>19</td>
</tr>
<tr>
<td>LED Luminaire</td>
<td>29</td>
</tr>
<tr>
<td>OLED Luminaire</td>
<td>870</td>
</tr>
</tbody>
</table>
The barriers to adoption of efficient lighting systems and the potential impacts such action can have make them a focal point of government policy. These programmes present significant opportunities for new technologies to establish themselves within the market. For example, regulations to phase out the most inefficient light sources (European Commission, 2009) have been gradually coming into force, thereby encouraging a market transition towards more effective alternatives. On a global scale the influence of these policies can be considerable. Between 1990 and 2005, various measures had helped achieve an 8% reduction in energy consumption for lighting. Within 45 years the efficacy of an average lighting system had risen from 18 lm.W⁻¹ to 48 lm.W⁻¹ (International Energy Agency, 2006). Evaluating the direct impact of policies is complex and requires a great deal of data, analysis and careful extrapolation. Even where this is possible, there are still considerable margins for error. Consequently, the conclusions have to be treated with caution and causality cannot be assumed. However, it is clear markets are rapidly moving towards better-performing technologies.

A major drive behind the adoption of LED technology in lighting comes from its versatility. This is complemented by a growing appreciation of how to manage and use light, evidenced by the growing recognition of the subject’s value (for example, professional accreditation programmes (The Institution of Lighting Professionals, 2012)). What is apparent is a need for adaptable,
controllable lighting. As a light source the LED component is highly directional (DiLouie, 2006), enabling greater control of the output with the potential to minimise waste heat. Compact size, simple operation, controllability and easy integration into systems make them suitable for many roles where traditional lighting sources would be impractical. Their properties make it relatively simple to create controllable lighting systems that meet the desired specification. This ability to provide lighting with adaptability and added capabilities help support adoption of LED technology as well as representing a new era of ‘smarter’ lighting.

LED devices represents a key technology to meet the evolving demands of the market. Although there are some limitations, the advantages of the technology clearly justify their development. With such great potential, it is obvious that even minor improvements could have a far reaching and valuable impact.

1.5 Thermal management of electronics and LED components

LEDs are no different to any other electronic component in that, owing to a combination of various physical mechanisms, they release heat as a waste product. If this is not managed it will impact the performance of the system, potentially resulting in unwanted damage and failure. Even when heat is essential to the components function (e.g. an incandescent lamp) an excess can be intolerable or cause damage to surrounding devices. The Arrhenius equation gives rise to the long-accepted rule of thumb that each 10 K increase in an electronic component’s operating temperature equates to a doubling of failure rate (Peck, 1979). For a typical LED component this increase can reduce the projected functional life by 27 % (Tridonic, 2016). These values vary depending on the system and component in question, but this does highlight the importance of thermal management. As Fig. 1-10 shows, demand for greater performance along with increasing component density has historically driven upwards both power consumption and heat flux. The heat flux passing through the microscopic footprint of electronic components such as laser diodes can now exceed several kilowatts per square centimetre (Huddle et al., 2000), making thermal management extremely challenging.

1 Tridonic TALEXmodule STARK FLE GEN1.
Thermal management involves the efficient transfer of heat from its source to a sink without it having an unacceptable impact on the device or system. This is facilitated by the fundamental processes of heat transfer; conduction, radiation and convection. Conduction enables the transfer of thermal energy through a body by molecular interaction. In fluids this is facilitated by collisions and diffusion of energetic particles, in solids through transport by free energy carriers such as electrons and vibrational wave propagation (phonon motion). The rate of conductive heat transfer is dictated by the properties of the conducting medium, its configuration and the temperature differential across it. The thermal conductivity of a material is assigned S.I. units of watts per metre kelvin (W.m\(^{-1}\).K\(^{-1}\)). Although this is a temperature-dependent property, any change can normally be assumed to be negligible, especially within the temperature range of interest to this research. Radiation relies on the same physical process as incandescence, but also includes energy transfer occurring in the invisible range of the electromagnetic spectrum. Radiative heat transfer is described by the Stefan-Boltzmann law which relates its rate to the emitting surface area, the absolute temperatures of the two interacting surfaces, the surface
emissivity and the Stefan-Boltzmann constant. Convection is the transport of heat by the motion of a fluid medium. Passive, or natural, convection arises from thermal energy conducted to a fluid resulting in localised expansion and an associated decrease in the density of that fluid. Consequently, buoyancy forces are established which move the heated fluid up as it is displaced by cooler, denser fluid drawn down under higher gravitational pull. The fluid flow rate and thermal capacity of the transferring medium dictate the magnitude of energy transferred. This process can be quantified by the convective heat transfer coefficient, given in watts per square metre kelvin (W.m\(^{-2}\).K\(^{-1}\)). Natural convection of ambient air, referring to convection arising solely from buoyancy-driven fluid motion, is relatively limited owing to air’s small expansion coefficient (and consequently low flow rate) and low thermal capacity. To increase convective heat transfer the fluid flow can be artificially driven (forced convection) to transfer a greater volume of heated fluid away from the heat source or the fluid can be replaced with a higher thermal capacity alternative (such as a liquid). Table 1-3 compares the typical convective heat transfer coefficients that these conditions can achieve. It also includes phase change processes which introduce additional fluid motion due to agitation (boiling) and additional energy transfer due to the change in state (vaporisation). Fig. 1-11 shows the relationship between heat flux and temperature differential under various convection conditions. These comparisons clearly show natural convection presents a severely restrictive heat transfer mechanism. However, system design can be manipulated to counteract such limitations (i.e. by employing a larger surface area, lower heat flux, alternative flow conditions or by accommodating a larger thermal gradient).

<table>
<thead>
<tr>
<th>Table 1-3: Typical convective transfer coefficients for different fluid types, phase states and flow regimes (Cengel, 2003)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Convection condition</strong></td>
</tr>
<tr>
<td>Natural convection of a gas</td>
</tr>
<tr>
<td>Natural convection of a liquid</td>
</tr>
<tr>
<td>Forced convection of a gas</td>
</tr>
<tr>
<td>Forced convection of a liquid</td>
</tr>
<tr>
<td>Phase change</td>
</tr>
</tbody>
</table>
Fig. 1-11: Comparison of convective cooling capacity under different heat flux, temperature differential and fluid flow regimes. (Image from Cengel, 2003. Data reproduced from Kraus and Bar-Cohen, 1983)

The aim of thermal management is to efficiently transfer excess heat away from sensitive structures to a heat sink (usually the surrounding environment) without excessive temperature rise. It should be noted that this does not simply mean providing sufficient heat transfer to keep components within operating limits. Nor does it mean the highest rate of heat transfer physically possible is required. Implementation must take into account commercial constraints, environmental responsibilities and effectiveness of the thermal management design.

1.6 Aims of this thesis

As the market looks set to make LEDs the dominant light source technology of the future, there is a clear need to ensure they are effectively exploited. To this end luminaire systems must integrate appropriate thermal management strategies. As will be discussed later in this work, the
thermal management of LED components presents an interesting case because they are extremely sensitive to high temperatures whilst imposing a number of unique requirements compared to other electronic devices. Thermal management of electronics is a well-established subject with considerable research and development behind it. However, there is very little assessment of how this can be applied to the relatively new and particular demands of the LED for general lighting applications. This poses the question: how can their thermal management be achieved most effectively? To answer this requires various topics, influences, constraints and techniques to be bought together. This research aims to build upon previous work (Pryde, 2012a) (Pryde, 2012b) (Pryde and Archenhold, 2013) to address gaps in the available literature. The research was conducted from a commercial perspective to maximise its relevance to industry. Whilst LED luminaires are the focus of this work, it also has a wider relevance to electronics in general. As discussed, all electronic systems require some degree of thermal management. Wherever there are shared constraints and objectives, this research’s findings could be extended.

1.7 Research approach
The first part of this research was an evaluation of academic and industrial developments. This did not aim to report the latest benchmark achievements. Such a review would quickly become obsolete and be impossible to complete. Instead, it was intended to be a discussion of the development trends in the field and their potential impact on the subject matter. A literature review was conducted to assess the LED’s technological status and trends which may describe future changes. To complement this, an assessment of pertinent thermal management technologies was undertaken. Market surveys were conducted to assess commercial practices and trends in the implementation of LED technology in general lighting applications. This established the requirements and constraints which directed the subsequent investigation. The middle part of the research was concerned with defining and verifying the means to develop effective thermal management strategies. This included the definition of suitable test methods and criteria to evaluate the performance of systems; benchmarking of simulation models to assess their ability to accurately reproduce the behaviour of a real system; and characterisation of the physical changes in the system resulting from exposure to the environment. This provided the foundation to develop more effective thermal management concepts and evaluate their performance. The final part of the research was an exploration of how the thermal management of LED luminaires can be realised most effectively. This was achieved with the use of
computational simulation to aid in the analysis and optimisation of various concepts. Optimisation was pursued with reference to practical system configurations. This identified several potential performance enhancements in answer to the objectives of the research.

1.8 Thesis structure

Fig. 1-12 summarises the structure of this thesis. Chapter 1, this introduction, has provided some context for the work that follows; its aims and the approach taken. The body of research is then split into 3 segments. The first of these, encompassing Chapters 2, 3 and 4, establishes the current state of the art and its commercial implementation. Each chapter addresses a different element in a parallel fashion. Chapter 2 is a traditional literature review concerned with developments that may influence the future of LED thermal management, while Chapter 3 explores relevant thermal management technologies and Chapter 4 examines the current implementation of these technologies in industry. This establishes the current requirements and expected evolution of effective thermal management. The second segment of this work is concerned with defining the analysis methods and parameters needed to develop a suitable solution. Chapter 5 outlines the methodology used to test and evaluate a typical luminaire system. These techniques are then used to provide benchmark data for a series of simulation case studies in Chapter 6. The benchmarked case studies enable simulation techniques to be applied with confidence in the subsequent investigation. Chapter 7 reports a separate study using these analysis techniques to evaluate what effect exposure to an environment has on the systems operating characteristics, something absent from the reviewed literature. This captures the effect of any changes to ensure they can be accommodated during the development of the system. The final segment goes on to explore how the knowledge gathered can be applied, and its potential value. Chapter 8 combines the benchmarked simulation boundary conditions, environment effects and criteria identified from the initial research to analyse a range of appropriate concepts and select the most effective. The results of this selection are then taken further in two parallel optimisation studies. One, in Chapter 9, is based on a series of practical geometric constraints to determine what impact can be realised by applying the findings of this research to an existing product. The other, Chapter 10, is conducted for the same system but without as many constraints in order to assess the potential enhancement that could be realised by departing from conventional practice. The thesis closes with a chapter containing a summary of the findings of the research (Chapter 11), discussion of its potential impact and some suggestions for further study.
Fig. 1-12: Thesis map

1. Introduction
2. Literature review
3. Technical review
4. Trends in commercial implementation
5. Test methodology and evaluation of luminaire performance
6. Validation of simulation boundary conditions
7. Characterisation of operating environment's effects
8. Analysis of heatsink concepts
9. System optimisation (Constrained)
10. System optimisation (Unconstrained)
11. Discussion of results and further study

Establishing the status of research and industry
Defining and evaluating criteria to develop more effective thermal management concepts
Implementing findings to develop more effective thermal management concepts
Chapter 2: Literature review

A comprehensive review of semiconductor technology and electronics would be impractical, so the scope of this review has clear parameters. The following chapter contains a brief overview of the most relevant topics relating to the development and performance of the LED luminaire. It covers: trends in the luminous performance of the LED and the consequences for its operating conditions; the sensitivity of the LED component to high temperatures and how this is changing; constraints and developments in the thermal behaviour of the LED package that must be accommodated; the reliability and lifecycle performance of LED devices that must be supported by the wider system; improvements in material properties and structures that may increase or mitigate thermal management requirements; new concepts that could disrupt the adoption of LEDs in general lighting; and the impact of control systems. Extrapolating the potential impact of developments was avoided as they may never reach commercial viability. The following offers a summary of the current state of the art and the known consequences regarding effective thermal management.

2.1 Advances in LED performance

The most fundamental performance criteria relevant to lighting is the LED chip’s ability to generate light. This has two aspects; the luminous flux being produced and the efficacy of its generation.

2.1.1 Luminous flux

The output from an individual LED chip is often lower than can be achieved with most other lighting technologies. For instance, a metal halide gas discharge lamp\(^1\) can emit 34,000 lm (Osram, 2016) while a single, commercially available Samsung LH351B LED package, one of the most powerful commercially available, can only emit 525 lm (Samsung, 2016 a). By this metric LEDs are

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\(^1\) Osram HQI-BT 400 W/D PRO.
severely handicapped compared to alternative light sources and so multiple die must be employed in parallel to compete. This has clear cost implications, so the LED chip is often driven at its maximum output to minimise the number of components required. However, this also has thermal implications, as will be discussed later.

The luminous flux and cost of LED lamps follow a very clear long-term trend, which is a 10x decrease in cost per lumen generated and a 20x increase in maximum luminous flux each decade (Fig. 2-1) (Haitz and Tsao, 2011). However, the increase in luminous flux of white LEDs has exceeded the rate of increase demonstrated by red LEDs. These trends are facilitated by the improvements occurring at the LED chip and component level.

One factor enabling the LED to deliver greater luminous flux and lower cost is increasing operating power density. Noted in the Optoelectronics Industry Development Association (OIDA) 2002 roadmap update, in 2002 a typical LED chip operated at approximately 100 W.cm⁻². This high power density is in part due to the small size of the LED chip, which typically measures just 1 mm² (Wright, 2013). OIDA predicted an intermediate case would see 2012’s generations of LED chips operating at power densities of 500 - 750 W.cm⁻² increasing to 600 - 1000 W.cm⁻² in 2020. LED chips have recently been produced that operate at 300 W.cm⁻² (Henry, 2013), lower than OIDA predicted but still demonstrating significant growth.
As a small side note regarding luminous flux, there is an upper limit to mass market demand. Most general purpose lighting systems emit approximately 3000 lm from a single luminaire (Christensen and Graham, 2008). There is, of course, a wide variety of systems with different requirements, but this serves as a general guideline. Unfortunately, the single LED chip is currently incapable of delivering this luminous flux, and so compensating strategies are necessary.

2.1.2 Luminous efficacy

Separate from luminous flux is the light source’s efficacy. As previously noted, the theoretical maximum efficacy of a white light source is approximately 365 lm.W\(^{-1}\) (Murphy, 2012). Haitz and Tsao (2011) expect improvements in efficacy to plateau within the decade, agreeing with projections from the United States Department of Energy (2013) (Fig. 2-2, “pc” denotes phosphor converted white light source, “Qual” refers to qualified data). This suggests the efficacy of white LED components will actually reach around 245 lm.W\(^{-1}\). Current generations of commercially available components\(^1\) are able to deliver efficacies in excess of 180 lm.W\(^{-1}\), showing a significant proportion of energy is still wasted as heat (other losses can be assumed to be negligible).

![Fig. 2-2: Reported and projected white light LED efficacy (United States Department of Energy, 2013)](http://www.samsung.com/global/business/led/products/led-component/mid-power/lm561b-plus)

When this is compared to typical performance efficacies of incandescent tungsten filament bulbs (12 - 20 lm.W\(^{-1}\)) and compact fluorescent tubes (43 lm.W\(^{-1}\)) (Mahapatra et al., 2009), it becomes clear the LED already has a significant advantage. Examining the evolution in luminous efficacy (Fig. 2-3) highlights the relatively gradual improvement in mature light source technologies. Current generations of white LED components have not quite followed the predicted trend, but are already off the scale and still improving fast. Within a period of 14 months, a new generation\(^1\) of an existing component\(^2\) saw a further 27 lm.W\(^{-1}\) (16 %) improvement in luminous efficacy (Samsung, 2016 c) (Samsung, 2016 b). Ongoing efficacy improvements can be expected to continue to support the adoption of LED components while alternative light source technologies struggle to compete.

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\(^1\) Samsung LM561B Plus CRI 80.  
\(^2\) Samsung LM561B.
2.2 Thermal sensitivity of the LED

Table 2-1 summarises a review of known LED component failure modes and their causes conducted by Chang et al. (2012). It demonstrates that thermal and thermo-mechanical stresses influence almost all failure mechanisms. Effective thermal management clearly plays a critical role in ensuring the reliability and longevity of the component. This study seeks to address the means of managing the cause of failure rather than its effects, so a more detailed study of the processes involved was not pursued within this review. For further information the reader is referred to the original source.

Table 2-1: LED failure modes and contributing factors (Chang et al., 2012)

<table>
<thead>
<tr>
<th>Failure site</th>
<th>Failure Cause</th>
<th>Effect on Device</th>
<th>Failure Mode</th>
<th>Failure Mechanism</th>
</tr>
</thead>
<tbody>
<tr>
<td>Semiconductor (Die)</td>
<td>High Current-Induced</td>
<td>Thermomechanical Stress</td>
<td>Lumen Degradation, Increase in Reverse</td>
<td>Defect and Dislocation Generation and Movement</td>
</tr>
<tr>
<td></td>
<td>Joule Heating</td>
<td></td>
<td>Leakage Current, Increase in Parasitic Series Resistance</td>
<td></td>
</tr>
<tr>
<td></td>
<td>High Current-Induced</td>
<td>Thermomechanical Stress</td>
<td>Lumen Degradation</td>
<td>Die Cracking</td>
</tr>
<tr>
<td></td>
<td>Joule Heating</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td></td>
<td>High Ambient Temperature</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Poor Sawing and Grinding</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Process of p-n Junction</td>
<td>Thermal Stress</td>
<td>Lumen Degradation, Increase in Series Resistance and / or</td>
<td>Dopant Diffusion</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Forward Current</td>
<td></td>
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<tr>
<td></td>
<td>High Current-Induced</td>
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<td></td>
</tr>
<tr>
<td></td>
<td>Joule Heating</td>
<td></td>
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<tr>
<td></td>
<td>High Ambient Temperature</td>
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<td></td>
<td>Poor Fabrication Process</td>
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<td>of p-n Junction</td>
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<td></td>
<td>High Current-Induced</td>
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<td>Joule Heating</td>
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<td></td>
<td>High Ambient Temperature</td>
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<tr>
<td></td>
<td>Thermal Stress</td>
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<tr>
<td></td>
<td>High Drive Current or High</td>
<td>Electrical Overstress</td>
<td>No Light, Short Circuit</td>
<td>Electromigration</td>
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<tr>
<td></td>
<td>Current Density</td>
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<td></td>
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<td></td>
<td>Interconnects (Bond Wire,</td>
<td></td>
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<td></td>
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<tr>
<td></td>
<td>Ball, and Attachment)</td>
<td></td>
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<tr>
<td></td>
<td>High Drive Current / High</td>
<td>Electrical Overstress</td>
<td>No Light, Open Circuit</td>
<td>Electrical Overstress-Induced Bond Wire Fracture</td>
</tr>
<tr>
<td></td>
<td>Peak Transient Current</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Thermal Cycling Induced</td>
<td>Thermomechanical Stress</td>
<td>No Light, Open Circuit</td>
<td>Wire Ball Bond Fatigue</td>
</tr>
<tr>
<td></td>
<td>Deformation</td>
<td></td>
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<tr>
<td></td>
<td>Mismatch in Material</td>
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<td></td>
<td>Properties (e.g., CTEs,</td>
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<tr>
<td></td>
<td>Young’s Modulus)</td>
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<tr>
<td></td>
<td>Moisture Ingress</td>
<td>Hygro-mechanical Stress</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stress Source</td>
<td>Stress Condition</td>
<td>Degradation Effect</td>
<td>Root Cause</td>
<td></td>
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<tr>
<td>---------------</td>
<td>------------------</td>
<td>--------------------</td>
<td>------------</td>
<td></td>
</tr>
<tr>
<td>High Drive Current or High Pulsed / Transient Current</td>
<td>Electrical Overstress</td>
<td>Lumen Degradation, Increase in Parasitic Series Resistance, Short Circuit</td>
<td>Electrical Contact Metalurgical Interdiffusion</td>
<td></td>
</tr>
<tr>
<td>High Temperature</td>
<td>Thermal Stress</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Poor Material Properties (e.g., poor thermal conductivity of substrate)</td>
<td>Thermal Resistance Increase</td>
<td>No Light, Open Circuit</td>
<td>Electrostatic Discharge</td>
<td></td>
</tr>
<tr>
<td>High Voltage (Reverse Biased Pulse)</td>
<td>Electrical Overstress</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Package (Encapsulant, Lens, Lead Frame, and Case)</td>
<td>High Current-Induced Joule Heating, High Ambient Temperature</td>
<td>Electrical Overstress</td>
<td>Lumen Degradation</td>
<td>Carbonization of the Encapsulant</td>
</tr>
<tr>
<td>Mismatch in Material Properties (CTEs and CMEs)</td>
<td>Thermomechanical Stress</td>
<td>Lumen Degradation</td>
<td>Delamination</td>
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<tr>
<td>Interface Contamination</td>
<td></td>
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<tr>
<td>Moisture Ingress</td>
<td>Hygro-mechanical Stress</td>
<td></td>
<td></td>
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<tr>
<td>Prolonged Exposure to UV</td>
<td>Photodegradation</td>
<td>Lumen Degradation, Colour Change, Discolouration of the Encapsulant</td>
<td>Encapsulant Yellowing</td>
<td></td>
</tr>
<tr>
<td>High Drive Current Induced Joule Heating, High Ambient Temperature, Presence of Phosphor</td>
<td>Thermal Stress</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High Ambient Temperature</td>
<td>Thermomechanical Stress</td>
<td>Lumen Degradation</td>
<td>Lens Cracking</td>
<td></td>
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<tr>
<td>Poor Thermal Design</td>
<td></td>
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<td></td>
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<tr>
<td>Moisture Ingress</td>
<td>Hygro-mechanical Stress</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High Current-Induced Joule Heating, High Ambient Temperature</td>
<td>Thermal Stress</td>
<td>Lumen Degradation, Broadening of Spectrum (Colour Change)</td>
<td>Phosphor Thermal Quenching</td>
<td></td>
</tr>
<tr>
<td>Mismatch in Material Properties / Thermal Cycling Induced High Temperature Gradient</td>
<td>Mechanical Stress, Cyclic Creep and Stress Relaxation, Fracture of Brittle Intermetallic Compounds</td>
<td>Lumen Degradation, Forward Voltage Increase</td>
<td>Solder Joint Fatigue</td>
<td></td>
</tr>
</tbody>
</table>
At a specific operating power, an LED chip’s luminous efficacy (and thus the luminous flux emitted) is known to decrease with an increase in temperature. Component manufacturers often provide data to predict the change in output with regard to the LED’s semiconductor junction temperature (e.g. Fig. 2-4). There has been significant research into the cause of this relationship and numerous models developed to describe the response. It is generally attributed to power leakage, delocalisation (Wang et al., 2010) and non-radiative recombination (Meyaard et al., 2011). In simple terms, these inhibit the LED chip’s light generating mechanism and are exacerbated by high temperatures. The result is that the achievable peak luminous flux does not necessarily correspond with the manufacturer’s specified maximum power (Fig. 2-5, “\(P_d\)” denotes power supplied to device, “\(R_{hs}\)” refers to the resistance to heat transfer through the accompanying heatsink) (Hui and Qin, 2009). Excessive input power can compromise the LED chip’s luminous efficacy to the extent that any increase in electrical power actually reduces luminous flux emitted. Regardless of cause, the effects and evidence highlight the need to manage the LED component’s operating conditions to optimise its output.

Fig. 2-4: Resulting decrease in luminous flux emitted at higher LED junction temperature (Samsung, 2016 a)
It is worth noting that over the past decade tolerable LED junction temperature has not increased as rapidly as expected. Manufacturers’ current datasheets typically claim the maximum allowable LED junction temperature is around 423 K, while the OIDA roadmap (2002) anticipated that allowable chip temperatures would be in the range of 448 - 498 K by 2012. The manufacturers’ specification may include a large safety margin, but this also suggests that there has been slower than predicted improvement in the thermal tolerance of the component. This may also explain the lower than expected improvement in operating power density.

2.3 Internal thermal resistance

The physical properties of the LED chip and its surrounding packaging impose a barrier to the transfer of thermal energy. This resistance, the reciprocal of thermal conductance, is defined by the resulting temperature difference between two points for a given heat flow rate and area (McNaught and Wilkinson, 2006). For a specific situation with a defined area the absolute thermal resistance can be used (expressed using units of kelvin per watt, K.W⁻¹). Thermal resistance is analogous to electrical resistance, with multiple resistances acting in parallel or series fashion.

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The physical properties of the various components and contact interfaces along the path of heat transfer combine to establish a system’s total thermal resistance. A simplified thermal resistance network is shown in Fig. 2-6. The power dissipated, maximum allowable LED junction temperature and ambient environment temperature typically dictate a luminaire’s maximum total thermal resistance. However, within this constraint it is possible to offset an increase in thermal resistance at one point with a corresponding reduction elsewhere in the system, providing a degree of flexibility in thermal management design.

As verified by Choi and Shin (2011), thermal resistance is inextricably linked to the performance of the LED. Greater thermal resistance prevents the removal of waste heat from the LED and thus correlates with a reduction in performance as well as increased degradation. Consequently, as Fig. 2-7 shows, the thermal resistance of LED components has been rapidly decreasing to enable greater performance.
At present, commercially LED packages designed to be surface mounted on a circuit board\(^1\) can achieve thermal resistances below 4 K.W\(^{-1}\) between the semiconductor junction and package base. For such a component dissipating 2 W of heat, a 4 K.W\(^{-1}\) thermal resistance would equate to a junction temperature 8 K higher than the component’s base. As these components can operate with a junction temperature of 423 K, and presuming a typical ambient environment temperature of 298 K, the total permissible junction temperature rise above that of the ambient environment would be 125 K. It is clear that while the junction temperature rise attributable to package thermal resistance is significant enough to attract research interest, the thermal resistance of the surrounding system plays a far larger role in dictating the component’s junction temperature and offers more scope for development.

LED package thermal resistance has been shown by Yang et al. (2006) to vary with operating power and temperature. The most likely cause for the observed change appears to be the effect of joule heating at higher power leading to increased temperatures within the component packaging. This would be supported by their findings that showed very little change in thermal resistance at low power when joule heating would have had less impact. Material properties and

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structural degradation are also likely to have some temperature dependence so could be expected to affect component package thermal resistance. However, their significance and permanence are poorly represented in the reviewed literature and so their potential influences are difficult to judge. Jayasinghe et al. (2006) also identified the non-linear variation of LED thermal resistance when subjected to different operating conditions and the lack of associated literature. Their work showed that if component thermal resistances are measured in an appropriate manner they can be translated to alternative operating conditions. This enables the respective rise in LED junction temperature to be derived from thermal resistance properties and a reference point’s temperature with an uncertainty of just 5%. However, it is difficult to ensure manufacturer’s supplied data is measured to the same standards to achieve similar accuracy in practice. There are methods to assess the LED’s thermal resistance in situ (e.g. Lin et al., 2011) but these tend to be complex and consequently their commercial implementation is limited. This difficulty obtaining reliable thermal resistance data makes it necessary to allow for some uncertainty when they are used. However, the current literature provides no clear guidance regarding what allowance is necessary.

The LED’s thermal and electrical resistances have been shown to increase as the component ages (Trevisanello et al., 2008). Potential causes of this include optical and mechanical degradation leading to reduced component efficiency (Lu et al., 2016). The consequence of this is an increase in joule heating within the device. The effect of this additional thermal load would also be amplified by the reduced conduction of heat away from the semiconductor junction. Its potentially critical impact should be known and accommodated in the system’s design. Unfortunately the severity of these effects appeared to be unique to each case, meaning no generalisation could be made from the published literature.

2.4 Reliability
The LED chip is an inherently stable and reliable light source. Components such as Tridonic’s TALEXmodule STARK FLE GEN1 can readily achieve an operating lifetime of 60,000 hours (Tridonic, 2016) and some manufacturers even claim to offer components that can exceed 100,000 hours (OSRAM Opto semiconductors, 2012). Applying an average usage estimate of 8 hours per day, this would equate to over 20 and 34 years of service respectively. The reliability and robustness of the semiconductor also offers additional benefits. As discussed by DiLouie (2006), the LED package is mechanically durable and resistant to shock as well as able to function
in ambient environments as cold as 233 K. However, these claims must be evaluated alongside the methods for their measurement, which are often flawed. Owing to the LED chip’s long lifetime, accelerated test methods are necessary to produce timely reliability data. Electrical and thermal stressing factors can be used to accelerate the optical, electrical and mechanical degradation mechanisms of the component (Trevisanello et al., 2008). These techniques clearly reduce the component’s lifetime, enabling them to be assessed rapidly (Narendran and Gu, 2005). However, the various physical properties, the complex interaction between different elements and the potential for stressing factors to affect the failure site at different rates and via various pathways mean accelerated testing can only offer an estimate rather than a definitive measurement (Caruso and Dasgupta, 1998). Despite their widespread use, verification of the accuracy of accelerated test methods was not identified in the literature, highlighting a potential field for further study. The variety of conditions to evaluate is immense. Operational testing under high temperature, low temperature, room temperature, wet as well as high temperature conditions and environmental testing such as vibration, humidity and tolerance to storage are some of the more frequently addressed (Chang et al., 2012). As discussed by Richman (2011), a test method designated IES LM-80 (Illuminating Engineering Society, 2015) was developed by the Illuminating Engineering Society (IES) in an attempt to standardise procedures for evaluating LED component’s photometry. However, it lacks a defined method to extrapolate lifetime performance from the results. For this IES TM-21 (Illuminating Engineering Society, 2011) was developed in association with leading manufacturers. Alongside these IES LM-79 (Illuminating Engineering Society, 2008) has recently been proposed to address assessment of the entire luminaire’s photometric performance (luminous flux, luminous intensity distribution, electrical power consumption and colour characteristics) (Richman, 2011). Although these standards do not overcome all the limitations of the test methods, they do help ensure data is consistently measured and reported so it can be used to make valid lifetime comparisons.

It is important to recognise that system reliability may be dictated by failure of the associated luminaire components (electronic systems, optics, mechanical structures, etc.). Cyclic thermal loading, localised stresses and mechanical constraints of assembly impose considerable demands on the entire system which should be assessed and managed throughout its design. Perpina et al. (2012), for instance, offer their analysis of an LED lamp which showed significant thermo-mechanical stresses were focused around a vertical interconnect access (VIA) within the circuit board. They predicted these could cause mechanical failure after approximately 100,000 thermal cycles between 233 K and 503 K, which may or may not be critical depending on its usage profile.
Some other potential failure mechanisms would include degradation of waterproof seals, corrosion, fatigue of flexible joints, electrical overload (lighting strikes) and impact loads (vandalism). It is also conceivable the installation of a system could subject the LED component to damaging chemical atmospheres or electromagnetic radiation (e.g. ultra violet light from the sun). However, the reviewed literature provides very little guidance on how the reliability of the complete system can be verified. At present, failure mechanisms are poorly described and methods to integrate models are undeveloped (Hamon and van Driel, 2016). As a consequence, it is desirable to eliminate or minimise the possibility of failure occurring in order to maximise system reliability.

2.5 Lifecycle

To develop complementary thermal management solutions that are truly effective it is important to understand the product’s lifecycle performance. The lifecycle behaviour presented here is taken from an assessment of academic literature, manufacturer data and other independent research reports performed by the United States Department of Energy (2012). In order to keep the scope of their assessment manageable, a number of generalisations and estimates (agreed by a panel of manufacturers and industry experts) were applied. This was done with the aim of ensuring a representative assessment of the prevalent practices and anticipated future developments. Although LED chips can be utilised in a number of ways, their analysis was limited to replaceable GLS (General Lighting Service) type lamps as they were the most common in the source literature and represent the largest installed base. The results were based on an operating efficacy of 64 lm.W\(^{-1}\) as a representative value for a typical lamp. The review included an assessment of manufacturing (incorporating resource acquisition, processing and assembly), transport (conveying the completed product from manufacturer to retail outlet) and power consumption during use. Emissions during production were excluded owing to the lack of available data. Data was based on contemporary manufacturing processes which are continually improving. Transport of constituent elements such as raw materials was assumed to be included in manufacturing data. As a result of limited data, transportation impact was calculated

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separately based on estimated energy consumption for parallel sea and land shipping routes to Washington D.C.

As Fig. 2-8 shows, the impact of production tends to be relatively insignificant, even after accounting for multiple lamps required to meet a common 20 million lumen-hour functional unit. The majority of lifecycle energy consumption occurs during the use phase, so the greatest potential for reduction comes from increasing the efficacy of the light source. The average consumption of an LED and CFL lamp was roughly 3900 MJ, approximately one quarter of an incandescent lamp’s consumption. It is worth noting that some manufacturers claim component lifetimes in excess of the relatively short 25,000 hour operating life used in this analysis[^1]. It appears reasonable to conclude longer lifetimes would proportionately increase the impacts of energy consumption during production of competing technologies whilst minimising its significance with LED systems.

![Fig. 2-8: Life-cycle energy consumption of incandescent, compact fluorescent and LED lamps (United States Department of energy, 2012. Part 1)](image)

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The United States Department of Energy (2012) assessment reports the estimated energy consumed during the manufacture of the LED lamp ranges from 0.1% to 26.5% of the lifecycle total. A more detailed breakdown of the LED energy consumption across the range of reported data is shown in Fig. 2-9. The use phase clearly dominates the lifecycle energy consumption in all cases so offers the greatest scope for reduction.

![Fig. 2-9: Breakdown of LED light source energy consumption and comparison of different sources estimates for energy consumption required for production (United States Department of Energy, 2012. Part 1)](image)

The results of a study of the environmental impacts of various light source technologies on a common lumen-hour basis are summarised in Fig. 2-10 (United States Department of Energy, 2012). The study was based on an LED lamp employing a phosphor-converted white LED package and used performance data correct as of 2012 along with predictions for 2017. Again, data was corrected for an equivalent operating life. Even though the incandescent source is lightweight and least complex, resulting in low impact from its production, the poor efficacy meant it has much higher power consumption and consequently higher impact across all environmental criteria. The CFL was slightly more harmful than the LED lamp in all respects apart from hazardous landfill. Part 3 of the lifecycle assessment evaluated environmental testing and found CFL’s mercury content could escape detection. It was unclear if this was taken into account in the summary, but it could have potentially influenced the data. Between indicator LED components in 2007, and illumination LED components in 2012, on a per lumen basis the overall environmental impact has reduced by almost 95%. Projections for 2017 make realistic assumptions that LED lamp efficacy will increase to 134 lm.W⁻¹ and lamp lifetime reaches 40,000 hours. Consequently, LED lamp impact should be approximately half that of 2012 and 70% lower than current CFL sources.
Breaking down the individual environmental impact criteria clearly shows that the use phase still dominates (Fig. 2-11). The next most significant impact arises from raw material consumed, with an average impact of approximately 17%, representing a smaller but still considerable margin for improvement. Aluminium used in heatsinks was highlighted as a major contributor to this impact, so there is an incentive to reduce its use.
Focusing on use or manufacture alone fails to appreciate all the hazards posed by the LED luminaire. End of life disposal also requires consideration. Adequate design for end-of-life recovery and reclamation is difficult to evaluate. The United States Department of Energy (2012) lifecycle assessment was based on virgin material or minimum recycled material content where appropriate. Their assessment also included some chemical analyses. A selection of CFL, incandescent and LED lamps were disassembled, the constituent parts milled and the concentrations of hazardous elements measured. It discovered that according to Californian regulations nearly all lamps exceeded at least one restriction, normally for copper, zinc, antimony or nickel content. The most significant contributors to these failures were the metal screw bases, drivers and ballasts, i.e. the LED was not responsible for exceeding any threshold but the surrounding lamp was. Conservative end of life predictions suggest recycling could mitigate most adverse impacts.

Similar lifecycle assessments of the entire luminaire are relatively rare. One that was available for review (Tahkamo et al., 2013) arrived at a similar conclusion; energy consumption during use dominates the environmental impact but the associated electronic elements and aluminium components still have a considerable effect. Eliminating these wherever possible would therefore be beneficial. The reviewed literature offers very little guidance to balance use, manufacture and end-of-life requirements. Identification of the ideal trade-off would be extremely advantageous to optimise the product lifecycle and would be worthy of further study.

### 2.6 Materials

The materials used to construct the LED determine its electroluminescent behaviour, optical efficiency, thermal properties and robustness. As they are so closely linked to the performance of the LED they are a key research topic, and so some of the more significant developments are summarised here.

#### 2.6.1 Semiconductor materials

The materials used to create the semiconductor junction of the light emitting diode dictate the wavelength of light emitted. As outlined by Kovac et al. (2003), there are a number of materials in common usage. The most significant in relation to this work, with its focus on general lighting, is that used for the manufacture of blue LEDs, i.e. gallium nitride (GaN). Blue light represent the
high photon energy wavelengths of the visible spectrum. This permits the down-conversion to lower energy wavelengths, therefore enabling a broad white spectrum to be produced. Other semiconductors based on aluminium gallium indium phosphide (AlGaInP) and gallium phosphate (GaP) can respectively be used to generate red and green wavelengths of light. By modifying the composition of the semiconductor material it is possible to tune the emitted wavelength to meet specific demands. Consequently, the palette of emission colours available is able to satisfy most requirements. The development of semiconductor materials is, therefore, broadly driven by the same demands placed on LED technology as a whole, i.e. greater efficiency, higher output, improved reliability, better spectral composition or commercial advantages. Developments in these areas (e.g. Der Maur et al., 2012 and Berencen et al., 2012) can be expected to continue established trends towards greater performance. New semiconductor materials, such as those capable of producing near-ultraviolet (UV) wavelengths (see ‘2.8.4 Near-ultraviolet LED’), may offer some unique benefits that enable them to enter the general lighting industry. However, the reviewed literature offers very little evidence to suggest there are any materials that will radically change the thermal management demands placed on the component. Semiconductor material developments are, therefore, considered to have very little influence on the direction of this research.

2.6.2 Phosphorescent materials

As noted in the introduction, white LED light can be created by a combination of LEDs emitting a variety of wavelengths or by use of a secondary phosphorescent material to convert a portion of the LED emission spectrum. The shape, composition, particle size and arrangement of these materials, as well as composition and refractive index of the surrounding matrix can impact the light quality and output (Sommer et al., 2012). A huge number of phosphorescent compounds exist but only a relatively small proportion perform well across the various demands, including insensitivity to high temperatures (Smet et al., 2011). The literature presents no significant obstacles or advances that may interfere with established trends in LED performance improvements.

Yun et al. (2012) proposed improving visual satisfaction as a means of reducing illuminance requirements, and therefore reducing power consumed by lighting. They showed that a light source which accentuated emission in the red portion of the spectrum resulted in greater visual satisfaction for a sample group. This effect was predicted to reduce the demanded power
consumption in a generic office environment by 38%. It is important to recognise that this would not necessarily translate to lower thermal load applied to the individual LED die. Illumination preferences are also very subjective and vary widely (Veitch, 2013). Consequently, there is no evidence to suggest this will impact thermal management requirements.

### 2.6.3 Chip substrate materials

The substrate on which the LED die is formed has a significant impact on its operating behaviour and cost. As demonstrated by Zhang and Liu (2014), there is considerable research into candidate materials, their respective performances and methods of overcoming physical incompatibilities. However, a recent market report (Lux Research, 2013) suggests competing substrates will have limited impact on industry. Sapphire substrates currently dominate the LED market, with a 90% share as of 2013. The use of silicon carbide (SiC), a competing commercial substrate material, is tightly controlled by a single manufacturer\(^1\). Bulk gallium nitride (GaN) theoretically has no material mismatch with the LED semiconductor enabling improved output and higher power consumption but the costs are currently extremely high (typically $2000 - $3000 for a 50.8 mm wafer compared to $350 for sapphire). If material incompatibilities could be overcome, LED die formed on silicon substrates could potentially be produced in larger quantities to enable greater manufacturing throughput and cost as little as one eighth as much as those grown on sapphire. However, the analysis indicates that up to the year 2020, sapphire substrate materials will see the greatest cost reductions. Hence, from a total substrate market value of $4 bn, just 19% is expected to be SiC and 10% silicon. As a consequence, no significant changes in thermal management as a result of transitioning to new substrate materials are anticipated.

### 2.6.4 Package encapsulation materials

Developments in encapsulation materials are being pursued in the name of performance. The encapsulant refers to the material used to encase the LED die, providing physical protection and often a degree of optical control. Epoxy based materials are popularly used as they offer low cost, high transparency, good refractive properties, strong adhesion and mechanical strength. However, thermal ageing effects such as molecular rearrangement and breakdown can result in

yellowing of the clear material and reduction in light transmission efficiency. This tends to occur before mechanical breakdown and in some instances may even induce functional failure (Hsu et al., 2012). Silicone based materials offer superior resistance to thermal degradation but are generally poorer optically and mechanically. As such, work is being conducted to overcome its weaknesses (for example that of Yang et al., 2011, who demonstrate a silicone material with good thermal stability, resistance to yellowing, high hardness, and high refractive index). These developments should gradually improve the robustness of the LED package, but do not promise to eliminate all thermal management demands.

**2.7 Structural configuration**

An LED die, typically being just 1 mm² (Wright, 2013), operates at a relatively high power density of 300 W.cm⁻² (Henry, 2013). For reference microprocessor operating power densities are on the order of 100 W.cm⁻² across an area of 130 mm² (Smil, 2015). Ensuring waste heat is effectively dissipated from such a small source requires considerable attention to the design of the component package and surrounding system’s structure. This is further complicated by the LED chip’s limited luminous flux, which often requires multiple sources to be employed in parallel. While this configuration can produce more light at greater luminous efficacy (Qin and Hui, 2010), overlapping thermal flux fields from each heat source in the array can also increase LED junction temperature compared to a single isolated component (Christensen and Graham, 2008). Fig. 2-12, shows the limits of existing thermal management techniques (the blue region in the lower left corner) ("Tjunction" refers to the temperature of the LED semiconductor junction). There are readily available LED array modules that reflect these limits¹. However, the small size and increasing power density of LED die (see ‘2.1.1 Luminous flux’) mean these limits can be readily exceeded, highlighting the need for thermal management to be properly implemented.

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The array concept can be extended to the combination of multiple LED chips on a single circuit board. These chip-on-board (COB) modules can reduce the number of discrete components in the system and produce a higher combined luminous flux than an individually packaged LED chip, but concentrate thermal loads within a small region. To maintain performance, these consequences need to be managed. Advances within the component packaging, being closer to the heat source and subjected to higher heat flux, have the most direct influence on LED junction temperature. However, increasing separation between heat sources and reducing the overall thermal resistance of the system can still play a significant role in reducing junction temperatures (Wu et al., 2012).

The LED’s structure is the subject of intense research and development. Pardo et al. (2013), for instance, propose an alternative packaging structure to enhance heat transfer from the LED die. By eliminating electrical isolation of the die they were able to reduce thermal resistance between a 1 mm² LED die and package base to 5.5 K.W⁻¹ (compared to typical values of 8 - 24 K.W⁻¹ (Lin et
al., 2012)). Commercial manufacturing processes used to form the LED chip are traditionally based on epitaxial layer growth upon the previously discussed substrate materials (‘2.6.3 Chip substrate materials’). This process results in a layer structure to which electrical interconnections are then bonded. The size of the electrode on the top light emitting surface of the LED chip must trade-off joule heating from constrained current flow, which leads to non-radiative electron recombination (reducing luminous flux), against obstruction of emitted light. Incorporating a current blocking layer, effectively a baffle to redirect electrical flow, has been proposed as a method improving performance by decreasing temperature variation within the chip (Hwu et al., 2009). Flip chip architecture is an alternative concept to the conventional chip structure that overcomes some of its inherent limitations. The concept takes an LED die grown on a substrate by traditional manufacturing techniques and places it top face down (hence ‘flip’) into the component package. By arranging it in this inverted configuration the electrical interconnections can all be positioned on the underside and formed without the need for secondary wire bonding (Fig. 2-13). This enables effective thermal dissipation from the chip and less blocking of generated light, but effective electrical distribution remains a critical challenge when optimising performance (Chen et al., 2007).

Fig. 2-13: Flip chip LED architecture (Carey, 2014)
Flip chip technology promises a step improvement in package thermal resistance with associated performance benefits. As a result it is already being adopted commercially. By attaching the flip chip to a high thermal conductivity base, there is potential for improving the LED chip’s power consumption, junction temperature and output stability (Chang et al., 2009). Highly conductive materials can also minimise hotspots in the LED die, reducing consequences such as darkening of the packaging encapsulant (Arik and Weaver, 2004). Enhancing the transmission of light from the LED package is another strategy that contributes towards ongoing efficacy improvements. For example, the use of a thin film light emitting layer can reduce light emitted from the LED chip’s side, which tends to be an inefficient direction in terms of escape from the luminaire. This can enhance luminous flux by 51% compared to non-thin film flip chips. This can be further enhanced by roughening of the emitting surface to increase light extraction efficiency. When applied to a phosphor converted white LED flip chip, luminous flux can be increased by approximately 45% (Shchekin et al., 2006). Liu et al. (2009 a) provide an excellent review regarding the challenges and constraints of packaging design. Their assessment provides a comprehensive discussion encompassing optical, thermal, reliability and cost developments. They report that arraying multiple LED chips in a single component can often provide exceptionally low thermal resistance because some of the features associated with discretely packaged components can be combined or eliminated. They also note that further improvements in thermal resistance are feasible, but are generally constrained by cost. Their review highlights one source claiming to have developed a package with a thermal resistance of just 2 K.W\(^{-1}\) (Gao et al., 2008). As regards reliability, materials and process controls play a major role in future improvements. Liu et al. go on to assert reliability issues cannot be completely eliminated, only minimised, but as a result of the historical drive for processability and low cost, these factors have been neglected. Chip-Scale Packaging (CSP), is seen as one of the ultimate aims of packaging design. Not only is it compatible with established assembly processes, it also offers superior electrical and physical characteristics, a smaller form factor allowing closer spacing of components, and potentially lower cost (Thompson, 1997). However, reliability can still pose a challenge that hinders the adoption of this packaging style (Liu et al., 2014). The activity and breadth of research demonstrated in the

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literature indicate there are still significant performance improvements to be realised which could help offset thermal challenges. However, it is unclear if this potential will be commercially realised. Even if these improvements are exploited there is no suggestion that thermal management will become irrelevant, only less critical.

As noted by Lin et al. (2012), LED packaging is designed in part to enable automatic treatment in subsequent manufacturing processes. However, this restricts packing density and imposes physical restrictions on the component properties. Alongside efforts to improve the behaviour of the component, alternative configurations which aim to bypass some of the associated issues entirely are being heavily researched. Lin et al. (2012) propose a design which aims to enhance heat transfer from the LED chip to its environment by attaching it directly to a large metal board (Fig. 2-14). The use of highly conductive materials, direct contact and large surface area resulted in a module thermal resistance of just 3.7 K.W\(^{-1}\). It demonstrates the possibility of advancing accepted packaging convention to improve thermal performance. However, the practicalities of these novel concepts are often poorly addressed by the reviewed literature and so they are expected to have very little commercial impact on thermal management design.

As mentioned in the materials review, phosphorescent materials are an essential element of most white LED components. The phosphor’s light emission is non-directional, resulting in a significant proportion (approximately 60 %) directed back towards the LED chip where it is reabsorbed and wasted in the form of heat (Narendran et al., 2005). By separating the phosphor from the LED
die, and modifying the optical properties of the package, it is possible to improve light extraction efficiency (Luo et al., 2005) (Allen and Steckl, 2008). However, this remote phosphor configuration appears to suffer from a greater rate of performance degradation (Hwang et al., 2010). This is speculated to be a consequence of the poor dissipation of heat generated within the phosphor. Remote phosphors also impose additional practical challenges. As discussed by Liu et al. (2012) the low throughput and poor uniformity inherent in their production techniques increase cost and hinder commercial viability. Remote phosphors have the potential to enhance component efficacy and thus reduce thermal management requirements, but there are currently significant challenges to overcome before this can be realised. Addressing these challenges is the subject of component packaging design, which goes beyond the scope of this work. At present this configuration does not appear commercially viable and so its influence on thermal management practice is negligible. Future developments may enable the benefits of remote phosphors to be exploited without penalty, but even so they are not expected to overcome thermal management constraints entirely, and could possibly even contribute to increasing thermal management demands. Therefore, effective removal of heat from the component can be expected to remain essential.

A property that can be overlooked when considering thermal management is the system’s transient behaviour. The nature of the inherent thermal capacitance is a combination of the properties of the elements within the system. This transient response can be exploited to maximise output and performance by increasing the time taken for the LED die to reach peak temperature, thus postponing any negative impacts (Hui and Qin, 2009). Tao and Zhang (2013), plus an earlier investigation by Tao and Hui (2012), explored the time dependency of thermal behaviour. They also considered the role played by the location of the capacitance in the thermal path. In their examples, the thermal capacitance of LED packages, being small and close to the heat source, meant the LED junction reached a stable operating temperature within a few seconds of a change in operating conditions. The capacitance of the surrounding system, and in particular a heatsink structure, was shown to be much larger. The slow rate at which this reached a stable temperature after a change in operating conditions (on the order of several thousand seconds) demonstrated it has far greater potential to delay the LED junction reaching its peak temperature, thereby minimising degradation in system performance and buffering the effects of environmental fluctuations. The authors conclude the system’s thermal resistance should be minimised and thermal capacitance maximised wherever possible to kerb the thermal degradation of the LED die. Transient behaviour is of most relevance to dynamic systems where
thermal loads vary. This bears relevance to general lighting where the sources may be adjusted
dynamically, employed intermittently or environment fluctuation is present (discussed in more
detail in ‘2.9 Control systems’). For this reason, transient behaviour should be optimised as an
objective of an effective thermal management strategy.

The academic literature available highlights the areas undergoing development, but the latest
breakthroughs in production of commercially available components is, understandably,
confidential information. As such, it is not normally published and was unavailable for review.

2.8 New concepts

It is worth considering some potential influences on the future of solid state lighting. While these
may not currently be commercially viable, they have the potential to disrupt established trends
and should be evaluated to understand their prospective impacts. Discussion here is limited to
some of the more refined concepts that are closer to realisation or are receiving greater
attention.

2.8.1 Organic-LED

Organic based semiconductor LEDs (OLEDs) have received huge amounts of publicity over recent
years, but despite demonstrations of their feasibility (i.e. Pellegrino et al., 2015), they are still
struggling to reach widespread commercial adoption. The OLED is based upon the same
electroluminescent principles of the traditional LED, but is constructed by sandwiching a thin
organic semiconductor layer between electrodes rather than epitaxial layer growth on a crystal
substrate. The benefits of this approach are the large light emitting area that can be formed, the
uniformity of light emission and potential to be made flexible (Bender et al., 2015). If projected
improvements can be realised, they may open up new lighting design opportunities with new
thermal management requirements. For example, the OLED could demand thermal management
that can conform to accommodate complex or flexible shapes. However, current commercially
available devices such as Philips Brite 2 FL300, one of the few available, operate with a power
density of just 0.05 W.cm\(^{-2}\) (Philips, 2016) which could potential be low enough to make dedicated
thermal management unnecessary. The performance of OLED devices has steadily improved, but
it still struggles to compete with traditional LED technology in general lighting applications. A
technical roadmap from the United States Department of Energy (2013) expects luminous
efficacy to remain below conventional LEDs for the foreseeable future (Fig. 2-15). Qualified performance data mostly lies below this projection, suggesting the realised performance is actually even lower. The roadmap also highlights unique technical challenges such as susceptibility to water and oxygen that need to be overcome before OLED systems can offer robust, reliable lifetime performance. Additionally, OLED adoption is impaired by its extremely high cost. The few commercial products that can be referenced currently costs about $1500 - $2700 per thousand lumens (compared with typical LED costs of 5 - 15 $.klm$^{-1}$). At present OLED is limited to niche applications such as decorative lamps\(^1\) and flat panel displays\(^2\). They have a lot of potential but current limitations restrict their relevance to industry. As the literature offers no indication that these limitations are about to be overcome, it is believed that OLED will not, for the foreseeable future, disrupt the development of conventional LED devices or their market growth.

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2.8.2 Quantum Dot

Another development relevant to solid state lighting is the quantum dot (QD). These are nano-scale structures that demonstrate fluorescent behaviour. The effects of quantum confinement, arising from their small size, dictate the nature of their fluorescence. By controlling these parameters during synthesis it is possible to tune the wavelength of the resulting light emission. The benefits and state of QD technology were reviewed by Wood and Bulovic (2010). The QDs formation methods allow them to be processed in solution, facilitating low cost, large scale deposition by numerous methods. The composition is also inherently stable, reducing the importance of their thermal management. The QD produces a narrow band of emitted wavelengths that offers very pure, vivid and saturated colour generation making them ideal for use in display applications\(^1\). This controllability and clarity is also very useful in LED lighting for colour converting elements that could potentially replace phosphor materials. A demonstration of this was able to produce better spectral composition and efficacy than a conventional phosphor-converted LED package, although it also benefited from a remote phosphor style arrangement (Bi et al., 2015). A review by Talapin and Steckel (2013) also highlighted the potential to generate light by direct excitation of the QD, thereby enhancing device fabrication and overcoming integration issues. However, as Liu et al. (2016) show, this cannot yet compete with the luminous efficacy of standard LED devices. From the reviewed literature, QD’s appear to promise a number of benefits and so their development can be expected to be pursued. As converter elements their stability means they can be expected to reduce, but not eliminate, LED thermal management demands. Utilising QDs as a light source requires development before it is commercially viable. For these reasons, they are not expected to disrupt established trends or thermal management requirements for the foreseeable future.

2.8.3 Laser diode

The laser diode (LD) is an alternative semiconductor based light source with a number of benefits that could enable it to displace conventional LED components. It employs a modified light emission process that can potentially produce much higher output whilst also being extremely compact and resistant to elevated temperatures. The technology is able to employ remote

phosphors arranged to reflect rather than transmit light, with the advantage that they can be applied directly to a heat sinking structure for enhanced thermal stability. The light generation mechanism is also able to be driven at higher operating input power for greater luminous output because it does not suffer the same performance droop characteristics at high power density as traditional LED chips. However, the power required to initiate the electroluminescent mechanism, being high, results in them suffering from large resistive losses and consequently a lower power conversion efficiency (typically just 30 % compared to 80 % for a blue LED component, (Crawford et al., 2015). So while a practical luminous flux of 380 lm has been achieved, the efficacy was only 70 lm.W\(^{-1}\) (Hashimoto et al., 2012), placing it below that of a conventional phosphor converted LED component. However, the technology does have the potential to exceed the efficiency of traditional LED components (George et al., 2016) and with development could outperform them. Applications where the importance of a high output supersedes that of efficacy (for example automotive headlights) are currently the most promising sectors for this lighting technology. Continuing growth in other applications of LD technology such as communications, materials processing and medical applications (Overton et al., 2016) can be expected to promote its development and may lead to it becoming competitive with established LED technology. However, its long term impact on general lighting is unclear. Even if LD technology does eventually displace existing LED technology, similar thermal management challenges can be expected to remain, although the constraints imposed would be considerably different. The high output and small size would introduce additional practical considerations regarding light distribution that would need to be overcome. Cost is not discussed in the reviewed literature, suggesting it is not yet competitive for general applications. Their poor luminous efficacy would also hinder development and oppose their adoption. They are believed, therefore, to have no imminent bearing on the direction of this research.

**2.8.4 Near-ultraviolet LED**

A near-ultraviolet-emitting LED die can be used in place of the traditional blue emitter to stimulate a phosphor and generate a broad white light spectrum (Choi et al., 2012). The benefit of this configuration is that the temperature dependent shift in the LED die’s emitted spectrum is restricted to the invisible range of the electromagnetic spectrum. While this can potentially mitigate some consequences of excess temperatures it does not overcome all of its damaging
effects. It is, therefore, considered to be unlikely to have any significant short- or mid-term impact on thermal management.

2.8.5 Monolithic white LED
A monolithic white LED generates a broad light spectrum by combining multiple wavelength emitters into a single chip. The benefit of this approach is that it does not require a phosphorescent material, thus offering high efficiency and cost effective fabrication (Ooi and Zhang, 2015). Removing the phosphor from the component package should also aid thermal performance and reliability. This technology was reviewed by Xia et al. (2015). They discuss numerous developments in materials and fabrication but note the technology is restricted to laboratory scale demonstrations. While there is potential in the technology, it does not, as yet, appear to be commercially viable. It may ultimately reduce thermal management challenges, but not all potential weaknesses are removed. Current practices are, therefore, unlikely to be affected by developments in this area and thermal management will remain relevant even if it reaches commercial realisation.

2.9 Control systems
The way in which the LED die is controlled determines its operating conditions and, consequently, is closely linked to thermal management. The technology also affords new opportunities to integrate capabilities into luminaires such as networking and monitoring. For example the electrical characteristics of the LED die change in response to its operating conditions. Methods are being developed to exploit these effects, initially to assess the operating temperature of the LED junction (as per the test method defined in EIA / JESD51-1, 1995) but could potentially be used to provide active feedback and protect against excessive operating temperatures. The development of control systems and their integration with the luminaire is a diverse topic currently undergoing rapid evolution. As they can play a significant role in the definition of suitable thermal management strategies, they are worthy of attention in this investigation.

The control system plays a key role in reducing energy consumption. As DiLouie (2006) discusses, in 1990 the trend towards energy efficiency in lighting applications started a pattern of integrating systems for more efficient performance, initially focussed on dimming but with later advances in occupancy sensing, daylight sensing and lumen maintenance. By integrating sensors
and feedback, the controls are able to monitor environment conditions and activate lighting only when necessary. The reviewed literature all seems to agree this has a positive impact on energy consumption, but quantifying its extent is complex and estimates vary wildly. One source suggests daylight sensing can potentially save up to 61% of energy consumption and occupancy sensors can improve this by a further 1 - 4%, increasing non-linearly as occupancy rate decreases (Roisin et al., 2007). While this does not alter the operating conditions of the LED chip, it does impact the luminaire’s design life and intermittent operation would emphasise the transient behaviour of the system when designing suitable thermal management strategies. There are some clear advantages arising from the use of controls, but the effects on the component and system are poorly defined in the reviewed literature. They can conceivably extend the installed operational life of the system by reducing its utilisation, but also increase the number of thermal cycles the system experiences. This is a topic that would need further investigation in order to evaluate its impact with confidence.

The LED die’s drive current can be adjusted very rapidly, opening up possibilities for dimming or communication. Chen et al. (2012) compared the luminous flux output of an LED package when the supplied drive current was stable and when modulated at high frequency (in the megahertz range). They showed that within the first 50 hours of operation both regimes cause rapid degradation in luminous flux (and hence efficacy) (Fig. 2-16). After this brief initial period, the rate of degradation diminished for both regimes. However, while modulating the drive current initially caused the greater rate of degradation, it subsequently slowed more and so after approximately 175 hours of operation a stable drive current appeared to cause more degradation and offer the shortest projected lifetime. In their comparison, both LEDs were subjected to the same peak drive current. As such, the root mean square (RMS) value of switched drive current was lower than the fixed drive current. They concluded that the initially greater rate of decay in luminous flux under switched conditions was the result of the effects of cyclic thermomechanical stresses within the component, but the lower overall power was responsible for the slower rate of degradation. As an LED package can be expected to function for up to 100,000 hours (OSRAM Opto semiconductors, 2012) the initial change in output is relatively inconsequential. The lowest overall rate of degradation is clearly most desirable and so a switched regime appears to be advantageous under the assessed conditions.
A separate study of lower frequency (kilohertz range) drive current modulation (Buso et al., 2008) suggests the response is a little more complex. Buso et al. showed that, depending on the structure of the component package, degradation in luminous flux can occur at different rates in response to a change in modulation frequency (with constant average current). Again, the rate of degradation was observed to decrease as the LED package aged. Further study is required to determine with confidence how and why these effects were observed. Chen et al. (2012) hypothesise initial differences in the rate of luminous flux degradation are linked to the magnitude of cyclic thermomechanical stresses. They believe slower frequencies result in larger cyclic stresses but, as the LED package ages and thermal resistance increases, cooling decreases so cyclic stresses are reduced. However, they do not appear to consider the additional effect of increased heating. Following an initial period, they believe degradation mechanisms are then
predominantly driven by the thermal load rather than cyclic thermomechanical stresses and so frequency becomes irrelevant. It seems that switching the LED can diminish degradation in luminous flux by reducing the thermal load. However, it also appears to accelerate the initial rate of degradation. The literature seems to suggest pulsing the drive current has an overall beneficial impact on performance, but further analysis would be a valuable avenue for further research.

2.10 Chapter evaluation

- Increases in LED component operating power density are helping to meet ongoing demand for greater device luminous flux. Improving luminous efficacies help to reduce associated increases in waste heat but as this plateaus thermal management seems set to become more challenging.
- Combining multiple LED die in an array is one method of meeting demand for high luminous flux. However, the interaction between multiple heat sources in a densely packed, high power array elevates component temperatures, increasing demand for the effective rejection of waste heat.
- A review of LED component failure mechanisms has highlighted that nearly all are either exacerbated or directly caused by heat. While components have become more robust they are still vulnerable to damage and performance droop at relatively low junction temperatures (typically below 423 K).
- It is clear from the reviewed material that development has resulted in a gradual decrease in the LED package’s thermal resistance and there is no reason to believe this has reached its limits. The surrounding luminaire therefore becomes an ever more significant aspect of the system’s thermal management.
- The thermal resistance of the component package, and consequently the resulting LED junction temperature, was noted to increase as it ages. Failure to account for this, which does not appear to be widely appreciated or accurately quantified, would potentially accelerate degradation of the LED component’s performance and lead to premature failure.
- If correctly managed LEDs promise extremely long service lives (potentially in excess of 100,000 hours). Exploiting this to its full potential would require a surrounding luminaire and thermal management design with a complementary lifetime.
- What is evident in the literature is a focus on LED component reliability, with little attention to the behaviour of the entire luminaire. The limited information reviewed here indicates that
thermal effects are unlikely to induce failure in other parts of the system but this is difficult to verify. This means prevention and elimination of potential failure modes is desirable.

- Owing to the LED component’s long predicted functional life the most significant lifecycle impact occurs during its use phase. The small impact of manufacturing also suggests compromises to this phase are readily offset during service if they allow for improved operating performance. Therefore, maximising system efficiency should be a priority of thermal management design.

- The properties of phosphor, substrate and encapsulation materials have been the focus of intense research and development. The work in this area seeks to improve resistance to damage and performance degradation at high temperature as well as enhance extraction of waste heat (e.g. Yang et al., 2011). The packaging and structure of the LED chip is also being researched with the aim of improving thermal management (e.g. Pardo et al., 2013). This does not, however, indicate that demand for thermal management will consequently decrease. Improvements may be exploited to realise greater component operating power and output rather than reduce thermal management demands. No revolutionary developments appear to be imminent and so thermal management requirements are not expected to change significantly in the near future.

- There are several new light source technologies being developed, but the current state of the art suggest there will be no significant disruption to LED technology or thermal management requirements in the short- to mid-term future.

- Integrating the means to respond to its environment can significantly reduce the required luminous output and utilisation of the LED luminaire, thereby reducing power consumption and maximising its design life. These factors would tend to prioritise reliability and diminish the importance of high power dissipation from the chosen thermal management strategy.

- Increasing the thermal capacitance of the system maximises the time taken for the LED junction to increase in temperature, delaying the damaging impact of heat. The literature reveals that the surrounding system plays a significant role in providing this buffer, adding merit to a thermal management strategy that provides greater capacitance.

This chapter offered a summary of the status of LED development, new concepts affecting the adoption of LEDs in lighting applications, and possible directions for evolution. While further improvement of the LED component’s performance can be expected, thermal management requirements will not necessarily diminish and there are suggestions they will become even more challenging. It is important that the complete luminaire system and the thermal management
strategy complements LED technology in order to maximise its benefits as a light source. Optimisation of the surrounding systems’ properties, how they interact with the LED light source and meet lifecycle requirements along with providing adequate thermal management, all need to be considered but are inadequately covered in the reviewed literature. Establishing best practice for how the LED should be integrated with effective thermal management represents an opportunity to establish new knowledge with industrial and academic value so was the focus of the research presented in the following chapters.
A typical electronic system combines multiple components. Waste heat produced by the system is transferred through these components from its generation sites to the surrounding environment (i.e. Fig. 2-6). Thermal management relates to the manipulation of this heat flow, typically through the provision of a suitably low resistance path, to ensure adequate transfer of heat away from sensitive areas. The methods by which this is achieved are extremely diverse and difficult to evaluate fully within this review. Chapter 2 revealed thermal management at the system level, rather than component package, has been poorly covered in the past and so was chosen as the focus of this research. This chapter offers a brief account of the more notable technologies currently available (summarised in Fig. 3-1). Their main restrictions, advantages and properties are discussed, with the aim of evaluating their effectiveness alongside LED (and similar) technology.

Fig. 3-1: Categorisation of thermal management technologies
3.1 Passive components

For the purpose of this review, passive methods are defined as those which occur naturally in response to the physical conditions established by the operation of the system. They therefore require no additional motivating force or power consumption to facilitate heat transfer, making them inherently reliable and efficient.

3.1.1 Heatsinks

Possibly the most common and basic passive thermal management device is a natural convection cooled heatsink. The function of these devices is to enhance the rejection of heat from a system to the surrounding environment. Although heatsink designs are very diverse, one common feature is an extended surface area to maximise heat exchange with the environment. Some typical examples are shown in Fig. 3-2.

Heatsink design has been extensively studied and there are a number of resources to draw on. For example, Kraus and Bar-Cohen (1995) provide a broad overview of mathematical models, fin profiles, fin array arrangements and optimisation. They acknowledge the considerable research into this subject presented in the academic literature and define a number of fundamental relationships. Bar-Cohen et al. (2006) build on these foundations to develop least energy optimisation models for heatsink design. Their analysis of a natural convection cooled parallel plate fin heatsink (shown in Fig. 3-3) with a base area measuring 0.1 m long (“L”), by 0.1 m wide (“W”), and material thermal conductivity (“k”), of 20 W.m\(^{-1}\).K\(^{-1}\) showed that there is an optimum
fin separation distance ("S"), to maximise average heat transfer coefficient ("h_a"), from the heatsink (plotted in Fig. 3-4).

![Analysed heatsink geometry](image)

**Fig. 3-3:** Analysed heatsink geometry (Bar-Cohen et al., 2006)

![Optimum average heat transfer coefficient](image)

**Fig. 3-4:** Optimum average heat transfer coefficient for polymer based heatsink for different fin aspect ratios (Bar-Cohen et al., 2006)

It can be seen that the maximum average heat transfer coefficient was achieved with an optimum combination of fin separation distance and aspect ratio (fin height, “H”, to fin thickness, “t”). As fin separation decreased, the optimum average heat transfer coefficient was achieved with higher aspect ratio fins. In this study average heat transfer coefficient was calculated in relation to the heatsink base area. Therefore, peak average heat transfer coefficient coincides with peak heat transfer. As fin separation decreased, heat transfer was reduced as a consequence of passive airflow becoming increasingly obstructed, hence the reduction in peak average heat transfer.
coefficient. This reduction in heat transfer as a result of reduced separation can be offset by increasing the surface area of the heatsink (i.e. increasing the fin aspect ratio). They also showed that higher conductivity materials can permit higher average heat transfer coefficients (Fig. 3-5). Copper was able to dissipate 180 W.m\(^{-2}.K^{-1}\) at a fin height of 0.40 m while aluminium fins were able to dissipate 125 W.m\(^{-2}.K^{-1}\) at 0.25 m tall. Pin fin heatsinks were also evaluated and showed similar relationships.

These properties were determined using an isothermal heatsink base condition so the impact of spreading resistance (i.e. the resistance to conductive heat transfer from the small localised heat source across the larger heatsink base area) would need to be evaluated when translating these relationships to other situations. It is clear from Fig. 3-5 that for fins up to approximately 0.05 m tall, the thermal conductivity of the material had very little impact on performance. Taking into account recyclability, aluminium, copper and thermally conductive polymers were estimated to embody 200 MJ.kg\(^{-1}\), 71 MJ.kg\(^{-1}\) and 120 MJ.kg\(^{-1}\) respectively, suggesting that copper is preferable. However, the lower density of aluminium means it embodies less energy per unit mass. It was also shown that, owing to differences in density, aluminium can transfer approximately 60 W.kg\(^{-1}.K^{-1}\) while copper can only transfer 40 W.kg\(^{-1}.K^{-1}\) under forced convection.

![Fig. 3-5: Peak average heat transfer coefficient for optimally spaced, different height plate fins formed from materials with different thermal conductivity (Bar-Cohen et al., 2006)](image-url)
conditions. Polymers are interesting for their low density, low cost and favourable manufacturing characteristics but their thermal conductivity, in the region of 5 W.m⁻¹.K⁻¹, is comparatively low (Cho et al., 2016). Ongoing research and development (e.g. Kovacs and Suplicz, 2013) can be expected to lead to increases in their thermal conductivity. Consequently, their feasibility will improve, which can be accelerated by accommodating low thermal conductivity in the heatsink’s design. The relationships noted here show traditional materials and heatsink designs already offer reasonable capacity for thermal management. This makes them relevant to a wide array of LED lighting applications which are generally low power. However, there are physical constraints that mean increasing surface area is not an open-ended answer for higher thermal loads. This requires the development of alternative strategies to improve performance.

A large proportion of the literature relates to conventional parallel plate fin heatsinks. However, the literature also features a number of alternative concepts and analyses which demonstrate performance benefits. For instance, providing the material has sufficient thermal conductivity, a perforated sheet functions as a lightweight heatsink with excellent airflow characteristics (Ma et al., 2010). Jeong et al. (2015) discussed a perforated heatsink design that exploits this enhanced airflow to reduce thermal resistance and component mass by approximately 30 % compared to a non-perforated design in a horizontal orientation. Zografos and Sunderland (1990) showed that for an equivalent surface area a pin fin heatsink can transfer twice as much energy as a flat plate design, with inline pin fins achieving almost 20 % greater heat transfer than a staggered arrangement. Chapman et al. (1994) demonstrate the behaviour of a die-cast heatsink employing elliptical pins. The concept is proposed to minimise vortex effects arising at the edges of rectangular fins, which increases pressure drop and opposes fluid flow. Their elliptical pin design achieved equivalent thermal resistance to an extruded cross cut fin design despite lower thermal conductivity material and reduced airflow. Sikka et al. (2000) offer a summary review of literature regarding heatsink design and apply this knowledge to develop novel heatsink fin configurations. Their investigation compares the heat transfer performance of natural convection cooled pin fin and parallel plate heatsink designs with wavy and fluted fin configurations (Fig. 3-6), when the heatsink base is oriented horizontally and vertically (Fig. 3-7).
Fig. 3-6: Fin profiles examined by Sikka et al. (2000)

Fig. 3-7: Horizontal heatsink orientation (left), vertical heatsink orientation (right), and test environment evaluated by Sikka et al. (2000)
Under natural convection conditions, their findings demonstrate that the pin fin heatsink offers the greatest performance. It outperformed the parallel plate fin design by 15 - 32 % when in a horizontal orientation and 4 - 6 % when in a vertical orientation. The cross diagonal fluted heatsink design was able to outperform the parallel plate fin design when in a horizontal orientation by 0 - 9 % but was 7 - 10 % worse when aligned vertically. Longitudinally aligned, the wavy fin designs were marginally better than the parallel flat plate design, but slightly inferior when oriented vertically. Therefore, it is concluded that improvements using wavy fin profiles are possible but small and not adequate to justify the normally greater cost and manufacturing complexity. It is worthwhile noting that the inclusion of a chimney structure over a heatsink can enhance convective fluid flow (Fisher and Torrance, 1998). Park et al. (2016) exploited this to draw air more effectively across the surfaces of a heatsink. By incorporating an optimised chimney, they were able to reduce thermal resistance between the heatsink and the surrounding air by 20 % compared to a reference design augmented with a simple vertical tube. Another novel design was evaluated by Tavassoli (2000). They considered a plate fin heatsink but with the fins arranged in a non-parallel fashion (Fig. 3-8). The study has a clear commercial emphasis and several methodological flaws, so the conclusions are treated with caution, but they report an 18 % decrease in thermal resistance over a parallel fin model. This suggests some significant improvements can be realised with minimal evolution of current practices.

![Fig. 3-8: Non parallel fin array heatsink cross-section (Tavassoli, 2000)](image)

Jang et al. (2012) offer an evaluation of a radial heatsink. Following sensitivity analysis and geometry optimisation they arrived at the short fin array shown in Fig. 3-9.
They demonstrated an optimised heatsink design could achieve the same cooling performance as a non-optimised design yet reduce component mass by 35%. They also evaluated the impact of radiative heat transfer on this design. Under the applied boundary conditions, they found it reduced heatsink thermal resistance by 22% compared to the action of natural convection alone, demonstrating it can play a significant role in system thermal management. The effects of radiative heat transfer are well established. The effective heat transfer coefficient of a radiating surface 10 K hotter than the surrounding environment can be a similar order of magnitude to natural convection (Kraus and Bar-Cohen, 1995). With respect to a pin fin heatsink transferring heat to its surrounding environment, radiation has been found to make a 25 - 40% fractional contribution to total heat transfer (the combined effect of natural convection and radiative heat transfer), making its peak contribution when the temperature difference between the heatsink and ambient environment is small (Sparrow and Vemuri, 1985). Experimental validation of the radiative effects of different surface treatments are well reported in the literature. For example, Wankhede et al. (2007) compared the temperature of an aluminium enclosure with different surface treatments. Compared to the natural uncoated aluminium surface which is very reflective, applying a black or white surface finish produced a 25% reduction in system temperature. Shives et al. (2004) report the properties of a bonded fin heatsink employing graphite fin plates. One benefit of a bonded fin design is the opportunity to integrate different materials. The materials used can be selected to enhance thermal management rather than being
dictated by the manufacturing processes, thus enabling performance improvements. By
exploiting this they were able to produce a heatsink with similar thermal resistance to that of a
copper finned heatsink but for a 40% lower weight. It also achieved a lower thermal resistance
than an identical geometry aluminium heatsink.

The range of potential configurations to assess are almost infinite and offer ample ground for
research. Advances in simulation and modelling now make it easier to evaluate heatsink designs
(e.g. Sun et al., 2015) but these techniques are restricted to individual cases rather than general
strategies for their optimisation. The literature demonstrates there are considerable
improvements in heatsink performance that can be realised. These improvements all have the
potential to benefit the cost and performance of LED luminaires which justifies further attention.
These potential improvements can also be used to ensure they remain a viable solution as
thermal management requirements continue to grow.

Despite their significance to thermal management there is no agreed standard to define the
performance of heatsinks. In commercial practice thermal resistance will often take precedence
when specifying a suitable heatsink for an application. This ensures adequate thermal dissipation
can be achieved to maintain appropriate component temperatures but it does not offer any
measure of the heatsink’s effectiveness. Lasance and Eggink (2005) did make an effort to address
this with an experimental method for ranking performance. They related average heat transfer
coefficient to a range of fluid velocities scaled by heatsink mass, weight or height. However, data
from experimental trials are of limited relevance to situations that impose different conditions.
Of particular concern is the nature of the applied heat source. There are models to define and
relate different conditions (Sadeghi et al., 2010), meaning it should be possible to establish a
common and comparable definition of heatsink performance. Despite the potential value there
is no evidence that such a definition is currently in development.

3.1.2 Thermosyphons, heatpipes and vapour chambers

These devices all employ a fluid in a sealed chamber to transfer thermal energy, and so will be
addressed in unison. Heat transferred to the fluid instigates convective heat transfer. Heat
transfer can be further enhanced by selecting a working fluid that undergoes a change of phase
from a liquid to a vapour. In doing so, energy is removed from the heat source. The vapour then
diffuses to cooler regions of the device, where it can condense and release embodied heat before
being transported back to the evaporating sites. In some cases a wick structure is incorporated to enable transport through capillary action (Fig. 3-10 offers a schematic diagram of one such example). These devices offers a high heat flux, high thermal conductivity passive heat transfer structure, but in contrast to the heatsink do not facilitate the rejection of heat from the system.

![Fig. 3-10: Schematic diagram of a typical heatpipe (Gilmore, 2002)](image)

As with any device, performance is difficult to quantify because the properties relate to the particular design and application. Transfer of heat flux densities on the order of 1000 W.cm$^{-2}$ have been reported (Chen et al., 2015). Experimental investigations are also presented in the literature which claim heat transfer rates of up to 310 W.cm$^{-2}$ at a temperature gradient of 18 K (Kang et al., 2010). This represents a relatively high performance benchmark under mild conditions, which makes them well suited to the thermal management of LED packages. However, there is a threshold at which solid materials offer comparable performance. This relates closely to thermal resistance of the material, heat source dimensions, plate geometry and cooling regime (Sauciuc et al., 2002), so would need to be assessed on a per case basis. One case showed a thermosiphon was only beneficial when power transfer exceeded 60 W. Below this, a copper plate offered comparable performance (Zhang et al., 2008). Yang et al. (2012) determined that a heatpipe can potentially perform the same function as pure copper but with 80 % less mass. They also highlighted the need to co-ordinate the material properties with the requirements of the system. If the vapour does not condense at the same rate as evaporation the pipe essentially becomes blocked with vapour while the liquid does not return to the heated surfaces rapidly enough to
facilitate transfer. The result is called a dryout situation where the heat transfer can drop dramatically by an order of magnitude (Shi et al., 2016). Conversely if condensation exceeds evaporation then the device can become flooded at the heat source end. These behaviours restrict the effective operating temperature range of the structure. In addition, incompatibility between the fluid and the enclosing structure materials can lead to corrosion and surface fouling, inhibiting heat transfer. Reaction and generation of non-condensing gasses within the chamber is also a risk. The presence of any non-condensing gas hinders the movement of the transfer fluid vapour. It has been shown that the thermosyphon’s thermal resistance is constrained by the evaporation and boiling boundaries (Zhang et al., 2008). Consequently, there is a considerable amount of research activity seeking to enhance their performance through working fluid and surface structure enhancements. For instance the use of sintered copper surfaces incorporating carbon nanotubes alongside working fluid optimisation has been proposed to increase heat flux by 46 % over a basic reference design (McHale, et al., 2011). It is also important to note these devices can be sensitive to gravitational effects which means the installation orientation will affect performance. This is because the return path of the liquid can be assisted or inhibited depending on the orientation. The optimum orientation employs the device with the heat source at its base. However, more effective wick structures can reduce the influence of orientation (Loh et al., 2005).

The reviewed literature shows significant activity into these devices. As a result there are a variety of designs to effectively redistribute large localised thermal loads. However, there are a number of restrictions that must be acknowledged: they rely on a specific orientation which may preclude their use in certain situations; they have a limited operating temperature range to avoid dryout of the working fluid; and material compatibility can be an issue. These restrictions are likely to improve as a result of research activity but performance advantages must also be weighed against the simplicity, low cost and capabilities of conventional materials. The reviewed literature provided very little information to evaluate long-term behaviour. Although there is nothing to suggest there is an issue, there is also nothing to verify that performance remains stable for the duration of an LED luminaire’s lifetime. The limitations and unknown long-term behaviour restrict their suitability for use in LED luminaires but in circumstances where thermal conductivity must be enhanced they offer a feasible solution.
### 3.1.3 Immersion cooling

As the name suggests this involves immersing the heat source in a liquid to which it transfers waste heat. The liquid, having a greater thermal capacity than air, is able to facilitate a much larger heat transfer coefficient. Baker (1972) studied the thermal management of microelectronic integrated circuits in some detail using this method. The liquid’s properties are clearly closely related to the heat transfer performance. For instance, immersing the source in Freon 113 was found to offer 3 times more convective heat transfer than immersion in air while silicone oil was less effective owing to its higher viscosity opposing fluid transfer. Interestingly, the smaller the heat source’s dimensions the greater the heat transfer coefficient from its surface became. Different fluid conditions also influence the rate of heat transfer. Boiling, alongside small heat sources, demonstrates only minor improvements over natural convection as a result of vapour bubbles obstructing heat transfer to the fluid (Baker, 1973). The relationship between heat source size and heat transfer is particularly relevant to the thermal management of small LED die. Tamdogan and Arik (2015) demonstrated water immersion cooling can increase heat transfer from an LED package by over 80%, but they recognise there are optical and practical issues that need to be addressed. Nevertheless, there has been at least one case of a commercially available product employing liquid immersion cooling (Fig. 3-11). However, the use of a supplementary heatsink reveals the limitations of the technique, acting only to redistribute heat rather than reject it from the system to the environment. As discussed by Shah et al. (2016), immersion cooling can enhance system reliability by preventing failure modes such as corrosion and improving heat transfer. Equally, the fluid can cause components and materials to deteriorate, potentially leading to failure, so its application needs to be carefully considered. The lack of information to support the use of this strategy in LED lighting suggests it is not currently practical and therefore of limited relevance to this work.
3.1.4 Heat storage

Energy storage should be acknowledged as a potential thermal management method. It is feasible that excess waste heat could be stored elsewhere to prevent a damaging build-up within the LED package. However, such a concept would have a limited capacity. Once this was reached, excess heat would still need to be rejected from the system to ensure suitable operating conditions were maintained. This would require supplementary thermal management methods that would probably nullify the need for energy storage. The literature review failed to discover adequate models of a luminaire’s operating profile. There is, therefore, no foundation to define energy storage capacity that is assured of meeting the thermal management requirements. The benefits of thermal capacitance as a buffer mechanism were noted in the preceding literature.
review but there was no suggestion complex energy storage methods are necessary. Storage was not, therefore, considered an appropriate thermal management method for use alongside LED luminaires.

3.2 Active components
Active technologies artificially enhance passive processes to increase the rate of heat transfer. However, to drive their operation they require an external energy input. This is usually electrical in origin and in addition to the power consumed to drive the system’s primary output. Consequently, they may reduce overall system efficiency unless the improved thermal dissipation provides a sufficient improvement in system output.

3.2.1 Electromagnetic fan
The electromagnetic motor driven fan may be one of the simplest and most common devices used to enhance convective heat transfer. By artificially driving the flow of cooling fluid, it enables the rate of heat transfer from the system to be increased. To maximise heat transfer it is often coupled with a heatsink. Walsh and Grimes (2007) studied this configuration. They note the interaction between the elements is not simple, concluding that heatsink and fan optimisation must be considered in an integrated manner. As discussed by Wang and Muller (2000), forced air cooling can provide adequate heat transfer for many demanding applications but its reliability hinders adoption. They go on to assert that limits in cooling performance are not imposed by technological restrictions but by ineffective deployment. The literature reveals considerable study of the optimum combination of fan output and heatsink geometry to maximise heat transfer from a system (e.g. Copeland (2000), Jonsson and Moshfegh (2002), Ning et al. (2008)). However, despite efforts to develop appropriate evaluation methods (e.g. Holahan, 2005), simple performance assessments remain elusive. The sheer number of variables associated with fan-assisted cooling makes theoretical models cumbersome and impractical. Not only does this hinder their integration, it also makes it difficult to assess their effectiveness. One useful comparison of performance was performed by Kaya (2014). Although this investigation did not allow enough time for the test sample to reach a stable thermal condition, a rough extrapolation of the data provided suggests a 2.16 W fan could be used to reduce the junction temperature of a 30 W array of LED chips by about 35 K compared to using a heatsink alone. For a typical LED
component\(^1\) this temperature reduction would equate to about a 5 \% improvement in light output for a given energy input (Samsung, 2016 a) but the fan adds 7.2 \% to the system’s total power consumption. Other devices may perform differently, but this suggests a fan will slightly increase a systems lifecycle energy consumption. There are other LED performance benefits arising from the reduced component temperature, but these are by no means exclusive to an electromagnetic fan cooled system. The same reduction in LED junction temperature could be achieved using a passive device, with the added advantage that the additional lifecycle energy consumption penalty would be avoided.

Using a fan to cool a system with a long design life requires a sufficiently reliable motor. There are commercial products that claim an operating lifetime to match that of an LED system\(^2\) and evaluating fan lifetime is supported by a range of techniques and standards (Jin et al., 2012), so these claims may be considered valid. Therefore, with regard to operating life, the electromagnetic fan can be considered a suitable thermal management technology for an LED luminaire. However, there are other considerations. The literature recognises that noise is an issue (Cattanei et al., 2007), and that it is likely to limit their suitability. The integration of an electromagnetic fan would presumably add to production and operation costs as well as complicating the development of the system. One potential benefit resulting from the enhanced heat transfer they provide would be the ability to employ a smaller heatsink, but it is unclear how valuable this size reduction is in commercial practice. It is feasible that they could be coupled with heat storage to provide intermittent cooling as and when the system reaches capacity, thus reducing their lifecycle impact. However, as previously noted (‘2.9 Control systems’), there are no clearly defined luminaire usage profiles, making it difficult to evaluate the potential benefits of such a strategy. Combining multiple thermal management methods would also complicate system design without necessarily overcoming the other issues with the technology. Despite the fan’s ability to overcome the physical restrictions of natural convection, there is very little evidence to suggest this constitutes an effective technique for cooling of LEDs and similar components. They have, therefore, little relevance to this research.

\(^1\) Samsung LH351B.

3.2.2 Piezoelectric fan

An electrical potential applied to a piezoelectric material causes it to deform. By forcing the piezoelectric material to deform at its resonant frequency, the motion can be amplified and maintained using relatively little energy. This can be harnessed to drive convective heat transfer from a system. Toda and Osaka (1979) have been credited as pioneers of this method of forced convection cooling for electronic equipment. The experimental setup studied by Acikalin et al. (2007) (Fig. 3-12) showed that if effectively designed it can increase average heat transfer coefficient to 100.8 W.m\(^{-2}\).K\(^{-1}\), a 375 % increase relative to natural convection. Their experiment also demonstrated the highly non-linear flow patterns established by the oscillating fan element.

Fig. 3-12: Left: Piezoelectric fan cooler, Right: Fluid flow velocity contours as fan passes starting positon (Acikalin et al., 2007)

Analysing how these devices interact with surrounding structures, and how to effectively integrate them into systems, is a key literature focus (e.g. Liu et al., 2009 b, Ma et al., 2012, Li et al., 2013). These studies reveal the critical influence the device’s configuration has on performance and the numerous localised fluid flow behaviours that need to be managed. The requirement for alternating current at a very specific frequency to match the resonant frequency of the device presents another challenge. An incorrect driving frequency can demand 20 times more power than necessary to provide a particular rate of cooling (Ma et al., 2012). The low energy consumption and significant enhancement of convective heat transfer (in the case of Ma et al., 2012, approximately a 55 % increase in average heat transfer coefficient compared to natural convection, using just 0.022 W) make this device particularly attractive for the thermal management of LED die. Sufian et al. (2014) have already demonstrated their use for this purpose.
and were able to reduce system thermal resistance by over 50% with a corresponding 60 K reduction in LED junction temperature. Performance degradation within 50,000 hours of service has been shown to have a negligible impact on an LED component's temperature (Song et al., 2012). In addition they are practically noiseless (Acikalin et al., 2010). They clearly have a lot of potential as a thermal management device, but integrating them effectively is a challenge that requires further development before the technology becomes a practical option. Consequently, their relevance to typical commercial practice, and therefore to this investigation, is currently limited.

3.2.3 Fluid jet

As described by Etemoglu (2007) a fluid stream impinging on a heated surface disrupts the boundary layer at that surface and thus enhances convective heat transfer. This may be employed as a mechanism for rejecting heat from a system to the surrounding air. It can also be exploited to maximise the transfer of heat to a circulating fluid, thereby enhancing the redistribution of heat within the system. Jet impingement is widely employed to maintain low temperatures where large thermal power densities are present (e.g. lasers and x-ray anodes). The process is generally considered to be aggressive and potentially unsuitable for delicate applications (Mudawar, 2001). Jet impingement used to cool turbines can transfer heat flux on the order of 100 W.cm$^{-2}$, equating to a heat transfer coefficient in the range of 1000 - 3000 W.m$^{-2}$.K$^{-1}$ (Zuckerman and Lior, 2005). Liu et al. (2008) studied a closed loop system using water to remove heat from an LED array and transport it to a heatsink. Their investigation was concerned with the optimisation of the array of fluid jets. The resulting design transferred a total heat flux of approximately 14 W.cm$^{-2}$ and maintained the surface to which the LEDs were mounted at a temperature of 342 K. It is readily apparent from the literature that this technique holds significant promise for meeting the thermal management requirements of high power applications. It offers the potential to use a working fluid other than air which provides greater thermal conductivity and capacity, it generates effective mixing of the fluid to enhance convection, and the cooling can be targeted to the critical regions of a system. However, generation of the fluid flow would clearly have associated cost, reliability and practical implications. If a liquid is used, it will probably need to be isolated from the electronic systems, be collected for recirculation and be carefully selected for compatibility with the surrounding structure. The fluid must also remain free of contaminants which could obstruct its flow. Passive technologies such as heatpipes are able to transfer higher heat flux, so
it seems to offer no advantage as a method of transferring heat within a system. As a method of rejecting heat to the environment, the high heat transfer coefficient it can achieve appears excessive considering the relatively low power of typical LED luminaire. Therefore, high performance fluid jet cooling appears to have limited relevance to typical LED luminaire thermal management.

Using a piezoelectric element to generate the fluid jet is one possibility that could overcome reliability issues (Song et al., 2012) and still significantly reduce the temperature of an LED package using a moderate fluid flow rate (Singh et al., 2014). There are commercially available devices employing this mechanism to enhance the rejection of heat from a system (Aavid Thermalloy, 2015), so it is clearly viable and appears more relevant to this work than higher performance alternatives. But while this is a promising strategy, there are still drawbacks. In particular, effectively integrating it within a system is a complex task that would benefit from further investigation. This technology has the potential to enhance thermal management and there are no major incompatibilities with the subject of this research, but, while it can help improve heat transfer, its necessity and impact remain unclear. The value of its benefits and drawbacks are closely tied to the individual application, so are difficult to define. Further investigation in this regard would be useful but not possible with the resources available. As the reviewed literature presents limited evidence of its commercial value, and examples of its adoption appear relatively uncommon (see ‘Chapter 4: Review of commercial LED luminaires’), this only appears to be a viable solution when other possibilities have been dismissed. Consequently, it promises limited impact on typical thermal management practice and so has little bearing on the direction of this work.

### 3.2.4 Spray cooling

Kim (2007) provides an excellent review of spray cooling technologies. As per their definition, “Spray cooling occurs when liquid forced through a small orifice shatters into a dispersion of fine droplets which then impact a heated surface. The droplets spread on the surface and evaporate or form a thin liquid film, removing large amounts of energy at low temperatures due to the latent heat of evaporation in addition to substantial single-phase convection effects.” The fluid vapour is typically condensed and recycled, requiring a supplementary method of rejecting heat from the system, as spray cooling only acts to transfer heat away from the source. With this method heat transfer of 1200 W.cm\(^{-2}\) has been reported in the literature (Pais et al., 1992). It should be noted
that this is far in excess of the 300 W.cm\(^{-2}\) LED chip power density of reported by Henry (2013). Kim’s 2007 review goes on to state that mechanisms of heat transfer and parameters to achieve maximum effectiveness have been thoroughly researched. These have found spray cooling to be highly sensitive to orientation. If the incident spray contacts at an angle exceeding 40° from perpendicular heat transfer drops significantly. This is because lateral momentum carries rebounded droplets away from the target. Orientation with respect to gravity can also create an issue with flooding. As a thermal management technology, it is very attractive for the low fluid volume required, the uniformity of cooling and the high heat flux it can transfer. Spray cooling has been shown to achieve a heat transfer coefficient of 9375 W.m\(^{-2}.K^{-1}\) from an array of LED packages (Hsieh et al., 2014). One apparent drawback with this technology is the need for a pumping mechanism to create the spray, which consumes energy and has potential reliability repercussions. There are also the practical challenges of containing the coolant liquid. The main advantage of providing high heat transfer is of limited relevance to relatively low power electronic components such as the LED. The drawbacks and lack of benefits therefore appear to make spray cooling poorly suited to typical LED luminaire thermal management.

3.2.5 Electrowetting

Electrowetting refers to the concept of applying an external electric field to modify the surface energy, and therefore wetting angle, of a liquid droplet contacting a surface. This phenomenon can be exploited for the purposes of heat transfer. It can be dynamically controlled to locally manage heat transfer across a surface, it involves no moving parts, requires relatively little power, can be reliably controlled, does not require pressurised fluid and can permit the use of high thermal conductivity liquid metal alloys (Baird and Mohseni, 2008). Kumari and Garimella (2011) demonstrate that an array of droplets moving across a surface is able to transfer 40 W.cm\(^{-2}\) whilst consuming just 0.2 mW. This approach certainly demonstrates considerable potential. However, the commercial manufacture of such a system has not been discussed in the reviewed literature, with studies limited to laboratory scale experimentation and simulations. LED luminaires, which generally operate under steady state conditions, negate one of the core benefits of this method: its dynamic controllability. It is also apparent that a constant supply of cool liquid or supplementary methods of cooling the liquid would be required. The achievable magnitude of heat transfer is also relatively small. For comparison, heat pipes have been reported that passively achieve 1000 W.cm\(^{-2}\) (Chen et al., 2015), far exceeding current electrowetting
capabilities. It therefore has limited relevance to this research, as the dynamic control offers limited benefit, commercial realisation appears to be a long way off, and performance can be exceeded by other means.

3.2.6 Coldplates

Coldplate refers to a structure actively cooled by an internal network of fluid filled channels. The fluid employed will generally be a liquid such as water, which offers high thermal capacity and conductivity, with a pumping mechanism to drive its circulation. Because the fluid can be actively driven through the channel and flow bypass is prevented, coldplates are not restricted by the same fluid stagnation and geometric constraints as extended fin heatsinks. Deng and Liu (2010) show the effective heat transfer coefficient of a water filled coldplate can be up to 3675 W.m⁻².K. Using a liquid metal as a working fluid can offer a heat transfer of 9343 W.m⁻².K. Owing to the high performance, this technology has been considered for the thermal management of LED chips used in automotive headlights. It was shown to adequately transfer 40.5 W of heat from 15 components in a challenging and confined environment (Lai et al., 2009). However, supplementary components for cooling of the working fluid were necessary and it was not made clear what advantage this system offered over simpler alternatives such as heatpipes.

It can be shown that reducing the fluid channel’s hydraulic diameter results in a larger cross-sectional area to perimeter ratio. This translates to proportionally greater surface area for heat transfer, which has resulted in development of micro scale channel structures (microchannels). Shao et al. (2007) studied simple microchannel geometry, developing a component capable of transferring 278 W.cm⁻² from a 6 x 6 mm area with a total thermal resistance of 0.12 K.W⁻¹. The concept is generally attributed to the pioneering work of Tuckermann and Pease, who in 1981 proposed forming microchannels directly into the silicon substrate of an electronic component. The fundamental issues of fluid transport have since been thoroughly studied. New opportunities are offered in the simultaneous exploration of nano-scale design and fabrication (Kandlikar et al., 2013). The potential performance offered by microchannel cooling is clearly very attractive, especially if it can be integrated with component packaging to remove heat directly from the source, but once again this technique appeals to high power density applications rather than LED systems. One issue is that it does not enhance rejection of heat from the system, only its redistribution. A heatpipe offers essentially the same enhancement, but can also accommodate higher power densities and does not require energy to function. Coldplates also appear to share
many of the same challenges faced by the other methods discussed in this review (integration, unknown reliability and so on). Addressing manufacturing challenges and the means to implement this solution present avenues for further research, but the reviewed literature does not give any indication that these are being pursued with respect to the thermal management of LED components. For these reasons the technology is not expected to have any significant influence on future typical LED luminaire thermal management strategies.

3.2.7 Thermoelectric devices
Thermoelectric devices are fundamentally different to the preceding heat transfer technologies. While the previously discussed technologies all act to enhance the conditions needed for heat transfer to occur (for example by increasing cooling fluid flow rate or employing high thermal capacity liquid as a heat carrying medium), thermoelectric devices actively transport thermal energy. This is achieved via electron carrier and can operate in opposition to a thermal gradient, meaning they can develop a surface temperature below that of the ambient environment. There are two main categories of thermoelectric transfer to consider. These are characterised by the nature of the electron transfer. Diffusive thermoelectric transfer occurs via electron carrier passing through a material. Ballistic thermoelectric transfer occurs across a gap between two surfaces when a heat carrying electron is emitted from one and absorbed by the other. By controlling these mechanisms it is possible to actively manage heat transfer.

Diffusive thermoelectric devices are commonly known as Peltier coolers. Peltier coolers can be very small, have no moving components and their temperature can be controlled by regulating electrical input power (Sales, 2002). Devices based on standard materials can achieve a transferred heat flux in the region of 100 W.cm\(^{-2}\) in a standard ambient environment with a temperature difference from hot to cold side of 30 K (Bulman et al., 2006). An investigation by Li et al. (2011) considered Peltier devices for the thermal management of a high-power LED component, building on previous literature which deals with alternative electronics cooling applications. In their experiment they demonstrated the temperature of an LED package could be decreased by 17 K to just 282 K when stacked on a thermoelectric cooler. They identified the rapid reaction of the thermoelectric cooler as advantageous. They also showed that the temperature of the LED junction is directly proportional to the power supplied to the thermoelectric cooler, indicating it can be easily tailored to the requirements of the system. However, the additional energy consumption and its impact on system efficiency are not factored
into their conclusion that this is an effective strategy. In their analysis the thermoelectric cooler added an additional 16% to the systems total energy consumption. For a typical LED component\(^1\), the associated improvement in luminous efficacy and flux due to the achieved reduction in junction temperature would equate to approximately 4% less power required to produce the same output (Samsung, 2016a). Therefore, the net effect would be an overall increase in system power and a detrimental impact on lifecycle performance. The system must also reject the additional heat released by the thermoelectric cooler. The control and ability to create sub-ambient temperatures could be beneficial in some applications, but these advantages do not appear to overcome the significant limitations or additional complexity of their integration. Therefore, they are not foreseen to make any significant contribution to future LED thermal management.

Ballistic transfer (sometimes referred to as thermionic) exploits the principle that a charged cathode with a sufficiently high work function will emit electrons carrying thermal energy. These will then be absorbed by a low work function anode. This was initially proposed as a means of integrated cooling of high power electronic components by Shakouri and Bowers (1997). A core benefit to this mechanism is the lack of solid bridge between the hot and cold side of the device. This hinders heat transfer against the direction of carrier flow by conduction, thereby overcoming a number of the challenges facing diffuse thermoelectric device materials. Heat transfer in the range of 1 - 100 W.cm\(^{-2}\) across a transfer gap of 7 - 10 nm is possible within the current state of the art (Hishinuma et al., 2001). Similar heat transfer has since been demonstrated in a standard temperature ambient environment, but a number of technical challenges remain. These relate principally to forming closely separated emission and collection surfaces. There are also difficulties maintaining the separation while subjected to mechanical and thermal expansion stresses (Hishinuma et al., 2003). Tanielian et al. (2011) present a method for manufacturing a device which should achieve a heat transfer of 1 W.cm\(^{-2}\). They assert this would make the performance commercially viable, but it is far behind alternative technologies and no devices based on this technology have been identified in the market. Research into the performance and manufacturing methods is continuing, but there appear to be no imminent breakthroughs which would have any influence on this investigation.

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\(^1\) Samsung LH351B.
It should be noted that a large proportion of the literature evaluates performance on the basis that the thermoelectric device transfers heat against a thermal gradient, effectively creating a negative thermal conductivity, thus the thermal conductivity of the device materials must be kept to a minimum. If the direction of heat transfer and thermal gradient are synchronised then the thermal conductivity of the device needs to be maximised, opening up new possibilities. Enerdyne Solutions Inc. (Enerdyne Solutions Inc., n.d.) offer a commercially available demonstration of this principle with their Polara product. The active heat transfer is claimed to be greater than conduction through copper for a similar cost. However, no evidence for this is provided. Conductive heat transfer can be adequately facilitated by other means, so this device offers very few advantages with reference to the subject of this work.

### 3.3 Circuit board enhancements

A circuit board provides the means to simultaneously attach multiple components and to establish electrical interconnections. The techniques used to form the conductive interconnections are normally based on a printing process; hence the name Printed Circuit Board (PCB). As part of a system, the circuit board has a significant effect on its thermal behaviour. Heat will typically have to pass through the circuit board as it moves from its generation sites in an attached component through to the heat sink of the wider system and surrounding environment (Fig. 2-6). This is hindered by the fact that the circuit board will generally have to provide some electrical isolation between the components and the surrounding system, which by nature also imposes a degree of thermal isolation as per the Wiedmann-Franz law (Clugston, 1998). It is worth summarising some of the associated constraints and opportunities imposed by this component and how the barrier it imposes can be overcome.

There are two common circuit board substrate materials used in LED systems which are generally known as FR4 (a glass fibre reinforced polymer) and MCPCB (a metal core PCB). MCPCB offers high through board thermal conductivity (for example 7.5 W.m$^{-1}$.K$^{-1}$, The Bergquist Company, n.d. a). As a guideline FR4 thermal conductivity can be taken as approximately 0.343 W.m$^{-1}$.K$^{-1}$ through the thickness of the board and 1.059 W.m$^{-1}$.K$^{-1}$ in the plane of the board (Sarvar et al., 1990). However, if the layout and structure of an FR4 board is effectively designed it can offer lower thermal resistance than a typical MCPCB (Yu et al., 2008). It is even possible for an FR4 based circuit board to achieve lower thermal resistance than a circuit board formed upon a ceramic substrate (Juntunen et al., 2014). Efforts to further enhance the thermal properties of
the MCPCB are also presented in the literature. For example, Wu et al. (2013) significantly reduced the thermal resistance through an MCPCB from 7.2 K.W⁻¹ to 1.3 K.W⁻¹ by including a copper filled VIA through the MCPCB’s dielectric layer and Cho and Kim (2008) roughly halved the thermal resistance through an MCPCB (from approximately 10 K.W⁻¹ to 5 K.W⁻¹) by improving the thermal conductivity of the dielectric layer. There are challenges that would need to be addressed with these enhancements, manufacturing being one. Introducing conductive pathways through the structure may also compromise electrical isolation pathways. However, the improvements are significant, with a clear indication that greater heat transfer through both types of circuit board can be attained if necessary. Therefore circuit board performance is not expected to impose any additional constraint on the future of thermal management relevant to this work.

Aside from thermal conduction properties, it is important to consider reliability. Jakovenko et al. (2013) offer a particularly useful study comparing the mechanical, thermal and reliability characteristics of a range circuit board structures with an attached LED array subjected to cyclic thermal loads. When cycling the system temperature between 233 K and 503 K (based on applicable manufacturing processes and a potential worst case operating environment temperature) they predicted the shortest solder joint lifetime would occur when the LED component was bonded to a MCPCB, with failure occurring after 170,000 thermal cycles. This was only slightly sooner than for FR4 based samples. Much earlier failure was predicted to occur at the sites of thermal VIAs in an FR4 based circuit board (around 95,000 cycles). This appears to be because the VIAs concentrate thermomechanical stresses. These results are in agreement with a similar study by Perpina et al. (2012). However, their experimental evidence also indicates that the type of simulation employed in both these studies typically overestimated the time to failure by an approximate factor of 2. In practice the thermal cycle range is unlikely to be as severe as that considered by Jakovenko et al., but for long lifetime systems such as an LED luminaire, combined with a control system that repeatedly switches the system on and off (for example a luminaire used in conjunction with an occupancy sensor, see ‘2.9 Control systems’) cyclic thermal stresses could still prove to be critical. However, the reviewed literature does not provide sufficient information to evaluate the potential impact. Further study would be valuable but is beyond the scope of this work. All that can be concluded, therefore, is that if the thermal conductivity of an FR4 based circuit board is insufficient, MCPCB appears to be a safer choice than an FR4 based circuit board enhanced with thermal VIAs.
3.4 Thermal interface material enhancements

Heat transfer between two adjacent bodies is hindered by poor contact. A thermal interface material (TIM) displaces the insulating layer of air found between mating surfaces (Fig. 3-13) which typically arises from micro-scale surface imperfections and waviness (Sarvar et al., 2006). It is useful to evaluate how these materials can enhance or constrain the performance of a system.

Fig. 3-13: Schematic magnified representation of the contact between two surfaces (right) and the role of a thermal interface material (left) (Andrews and Leather, 2009)

Guideline thermal resistances for interfaces augmented with a thermal grease, a filled polymer or with phase change materials are 0.2 - 1 K.cm².W⁻¹, 1 - 3 K.cm².W⁻¹ and 0.3 - 0.7 K.cm².W⁻¹ respectively (Otiaba et al., 2011). Sarvar et al. (2006) reviewed the state of the art of TIMs, discussing the general properties of different material categories, selection criteria and technical developments. Four categories of TIM were considered; greases, phase change materials, filled polymer matrices and carbon based materials. The limitations of greases, phase change and filled polymer materials are summarised in Table 3-1. The phenomenon of “pump-out”, whereby the thermal expansion and contraction associated with repeated thermal cycling results in migration of the grease from the interface thereby causing a progressive increase in thermal resistance, is particularly noteworthy. One study reports this can lead to a 4 - 6 fold increase in interface thermal resistance after 7500 cycles between 273 K and 373 K (although when subjected to a slightly less severe transition, between 273 K and 353 K, there was negligible change after 2500
cycles). Clearly the operating conditions and material need to be carefully matched to ensure adequate performance throughout its anticipated operating lifetime.

| Table 3-1: Comparison of thermal interface material properties (Sarvar et al., 2006) |
|-----------------------------------------------|-----------------------------------------------|
| TIM                                          | Advantages                                   | Disadvantages                                      |
| Thermal Grease                               | • High thermal conductivity                  | • Thermal cycling can result in pump-out and phase separation |
|                                               | • Thin joint with minimal attach pressure therefore low thermal resistance | • Can be messy and in a manufacturing environment can pollute assemblies and reflow baths |
|                                               | • No curing required                         | • Dry-out over time reducing reliability           |
|                                               | • Delamination not an issue                  | • Thickness difficult to control                   |
|                                               | • Low cost                                  | • Excess grease can flow out beyond the edges      |
| Phase Change Materials                        | • Increased stability and less susceptibility to pump-out   | • Lower thermal conductivity than greases         |
| Polymeric                                    | • Easier application and handling compared to greases | • Surface resistance can be greater than greases. Can be reduced by thermal pre-treatment |
|                                               | • No cure                                   | • Constant pressure required which can cause mechanical stresses |
|                                               | • Delamination not an issue                  | • Voids can result with thermal cycles and subsequent phase changes that cannot be refilled |
|                                               | • No dry-out                                |                                                     |
|                                               | • Lower thermal resistance than greases     |                                                     |
| Low Melting Alloys                           | • Easy to apply                             | • Dry-out causing voids at interfaces             |
|                                               | • All metal path                            | • Intermetallic growth at the interface           |
|                                               | • No cure required                          | • Oxidation / Corrosion at elevated temperature cycles |
| Filled Polymers                              | • Not messy                                 | • Curing required                                |
|                                               | • Easy to handle                            | • Thermal conductivity lower than grease          |
|                                               | • Eliminates problem of applying exact amount of grease with each application | • Delamination can be a problem                   |
|                                               | • Conforms to surface irregularity before cure | • Do not flow freely                             |
|                                               | • No pump-out or migration                   | • Permanent clamping needed                     |
|                                               | • Resists humidity and other harsh environments | • Higher cost than grease                      |
|                                               | • Good dielectric properties                |                                                     |
|                                               | • Low modulus (stress)                      |                                                     |
|                                               | • Can be easily cut to size of mounting surfaces |                                                     |

The review goes on to assess carbon based TIMs. Multiple walled carbon nanotubes have thermal conductivities as high as 3000 W.m⁻¹.K⁻¹ (Kim et al., 2001) but using these as a filler materials has
proved challenging, with reported TIMs achieving thermal conductivity of only 15 W.m\(^{-1}\).K\(^{-1}\) (Yang et al., 2002). Carbon fibres applied to an adhesive substrate reportedly achieve a thermal conductivity as high as 200 W.m\(^{-1}\).K\(^{-1}\) (Seaman and Knowles, 2001). While thin sheets of graphite impregnated with polymers can offer thermal conductivity through the sheet of 8 W.m\(^{-1}\).K\(^{-1}\) at 0.13 mm thick (HALA Contec, 2014). The literature exhibits a clear focus on nano-scale carbon based thermal interface compounds. For more information the reader is referred to the work of McNamara et al., (2012) who discuss these in a more focused review. Following the general consensus of the literature, Sarvar et al. (2006) conclude that nano-structure carbon TIM compounds show significant promise for enhanced thermal performance compared to conventional bulk materials, but they are currently unable to deliver the anticipated performance. Further research is needed to study the alignment, dispersal and interfacing of nano-materials with the carrying matrix. The field of thermal interface materials is very active, with a large number of published works discussing various material breakthroughs. These promise future performance improvements which could help offset thermal management challenges. However, the anticipated timescale and cost effectiveness of realising these improvements are not discussed and so they cannot be relied upon to alter the thermal management strategies considered in this investigation.

The degradation of the thermal interface’s performance is a critical factor, yet there is a lack of published work regarding how it can be assessed and managed. One analysis (Skuriat et al., 2013) compared the behaviour of silver-filled polymer paste / grease, tin based solder, and silver and tin foils. Samples of each were installed between copper discs and stored at a temperature of 443 K under a clamping pressure of approximately 0.5 MPa. At set intervals the thermal resistance through each of these samples was measured while clamped with a pressure of 1.4 MPa and transferring 200 W of heat. The resulting thermal resistances of each type of sample over the course of 90 days are plotted in Fig. 3-14. This seemed to show that interface thermal resistances gradually increase. Without any TIM, this trend appeared to begin to reverse after a brief time. This was attributed to the formation of an interfacial oxide layer. The effect ultimately resulted in a lower thermal resistance than for most of the TIM samples. Solder and grease offered comparable performance, initially lower than the foil interfaces or no TIM. The thermal grease, which claimed to offer stable properties at higher temperatures than applied in this analysis, showed fluctuating behaviour calling its reliability into question. It is also important to note that initial performance assessments and published material data are unlikely to accurately represent the interface’s properties throughout service. As a result the design of a system’s thermal
management should leave a considerable safety margin. Assuming these results are typical, and there is no long-term change, then allowing for a 300 - 500 % increase in thermal impedance across an interface appears reasonable. However, it is unclear how representative these results are. For example, the effect of repeatedly changing the clamping force and periodic heating may have had an unintended effect on the interface material’s properties or forced it to be expelled from between the copper plates. It is difficult, therefore, to draw any reliable conclusions from this work with regard to a typical application. Further study to characterise accurately the thermal interface material’s evolution in situ and to reduce any unnecessary safety margin would be extremely valuable.

Fig. 3-14: Comparison of different thermal interface materials’ thermal resistance across a period of 90 days (Skuriat et al., 2013)

The thermal conductivity of grease material typically has a much higher bulk value than its apparent conductivity in situ. This apparent conductivity is a function of material thermal resistance, interface thickness and contact resistance at the surface. For a thin interface (0.01 mm) the apparent thermal conductivity can be as little as 8 % of the bulk material value (Chiu et al., 1997). The wide variety of surface types, mechanical conditions and thermal profiles make
characterisation of this behaviour difficult. Issues associated with existing test methods include: uncertain or unrepresentative contact pressure; transient development of interface conditions (ageing of TIM, warpage of surfaces, reduction in contact pressure); non-uniform heating; inadequate thermocouple positioning; presence of contaminants; and difficulty taking sufficiently accurate measurements. Consequently, vendor data often significantly (by a factor of 2) overestimates the performance of an interface material (Lasance et al., 2006). As a result, predicted performance has to be treated with extreme caution. Unless the interface behaviour can be verified in a representative application, a significant margin of error must be incorporated into the system’s design.

Metallic solder joints as interface materials offer high thermal conductivity alongside good mechanical stability and electrical conduction, making them especially useful in electronic assembly. A study of appropriate test methods (Bai et al., 2005) offers some indication of the properties of a typical solder interface (see Fig. 3-15). Compared to silver filled grease, solder bonding offers greater mechanical strength, a thinner bond line and lower total thermal resistance in Bai et al.’s test case (1.13 K.W⁻¹ compared to 1.74 K.W⁻¹) but grease remains more cost effective (Chung et al., 2012). As a result, they are not expected to replace more conventional materials.

![Graph](image)

*Fig. 3-15: Thermal conductivity of a soldered interface with reference to bondline thickness (Bai et al., 2005)*
Defects and intermetallic compounds formed at the interface surfaces of solder joints can affect thermal impedance. Under certain circumstances, the interaction between the metallic solder and interface play a more significant role in defining its behaviour than the material conductivity. The interface conductivity can be as low as 7% of the bulk material value (Yoon and Park, 2009). A comprehensive review of solder metallurgy cannot be conducted here, but it is important to be aware that there are some additional failure mechanisms. In particular, mismatched coefficients of thermal expansion between bodies and repeated temperature cycling results in sliding at the metallic grain boundaries. This is generally acknowledged as a major cause of crack initiation and propagation in solders (Ma et al., 2013). Effective thermal management is therefore essential for long-term reliability.

3.5 System integration

So far, a number of distinct technologies and technical developments have been reviewed in isolation. When these concepts are employed within a larger system it is important to understand how the different elements interact. Models to determine appropriate electrical and thermal parameters for maximum output from the LED component are reported in the literature (Bender et al., 2013). These allow the system thermal management requirements to be defined to achieve maximum output or the appropriate operating parameters to suit a given system’s thermal management capability. However, ensuring the system’s thermal management is effectively implemented is poorly represented in the literature, so presents an opportunity for further investigation.

A frequent feature in the literature is the direct integration of thermal management technologies with the component parts of a system (i.e. not employing distinct parts operating in tandem but combining them into a single module with the functionality and benefits of both). For example, Huang et al. (2010) proposed applying circuit interconnections and LED packages directly to a vapour chamber (Fig. 3-16). This had a much higher cost than traditional circuit board assembly but resulted in greater thermal conductivity than a traditional PCB with a thinner structure.
Wits and Vaneker (2010) present a similar concept of forming a heatpipe within the laminar structure of a circuit board. To minimise costs, they restricted themselves to traditional circuit board manufacturing techniques. In a physical prototype measuring just 4 mm in thickness they achieved a thermal conductivity over seven times greater than solid copper. Implementing thermal management technologies to improve the performance of a component can provide greater thermal conductivity, overcome design restrictions and eliminate barriers to heat transfer. These are all valuable benefits. However, the manufacturability and implementation of these concepts presents a common challenge. To realise the potential benefits often requires a departure from established practices which hinders adoption and can compromise other properties. With regard to the focus of this work, the commercial justification to develop novel thermal management technology and component integration is not immediately obvious. Therefore, these concepts are not expected to lead to any significant shift in common thermal management strategies for the foreseeable future.

In 2004 Solbrekken et al. designed and tested a prototype system to explore the potential to drive a cooling fan with energy harvested from the system by a thermoelectric element. Their system achieved a combined thermal resistance between the heat source of a packaged semiconductor component and the ambient environment (representing a thermal gradient of 50 K) of 2 K.W$^{-1}$ (for comparison the thermal resistance of a typical LED component package$^1$ alone is 4 K.W$^{-1}$).

For a system dissipating 25 W of heat, the thermoelectric element generated power on the order of 100 mW, which was sufficient to drive a fan. These properties are expected to improve with developments in materials and manufacturing to enhance the performance of the thermoelectric element. While this strategy does offer exceptional thermal management performance, it relies on both thermoelectric devices and electromagnetic fans. It therefore combines the limitations of both. The complexity and associated costs would also be likely to act as significant barriers to adoption. It is worthwhile noting that this concept establishes a negative feedback loop (increased heat source temperature generates more power, in turn leading to providing more cooling from the fan). This could be a useful feature where the thermal environment varies and other control systems cannot be applied. However, this is believed to represent a very narrow opportunity and so energy harvesting is not expected to play any significant role in upcoming system design.

When assessing novel heat transfer structures, the value of theoretical models and simulations is readily apparent. The rapid assessment, with minimal investment of resources compared to physical prototyping and testing, is extremely beneficial. The literature shows benchmarking of theoretical models against experimental evidence to be a common practice to establish confidence in such models. Simplified thermal simulations have been shown to be representative of performance, typically reporting component temperatures to within 2.5 K of the true value (Xiaogai et al., 2011). The benefit to this approach is its ability to rapidly and accurately direct the design of an effective thermal management system by identifying potential issues during early stages of development.

In contrast to the generally accepted view presented in the literature, which suggest the limits of air cooling are being approached, Rodgers et al. (2005) assert that air cooling capacity achieved with current technology is comparable to liquid cooled systems of the past, and that to assert that air cooling has reached technical limits misrepresents the state of the industry. Their review recognises recent work that has shifted from parametric design methodologies towards a more case-specific development approach, allowing new performance benchmarks to be set. They also note that thermal design usually takes into account an accumulation of worst-case scenario assumptions which do not represent the typical operating environment. Consequently, many systems are poorly optimised for their role. Better monitoring and control of the environment would provide the basis for leaner systems that are still capable of meeting the applications needs. They raise concerns regarding the issue of surface fouling reducing the effectiveness of
heatsinks and heat transfer surfaces, especially in fine pitched finned surfaces. However, the impact of this is poorly reported in the reviewed literature. Its potentially detrimental effect on thermal management performance needs to be better understood and requires further study. Finally Rodgers et al. summarise the need for standardisation within the industry, particularly regarding characterising thermal interface materials. They conclude that there are significant challenges facing thermal management, but also point to numerous opportunities for performance improvements to be realised. This can be aided by improved definitions of the objectives of thermal management and constraints imposed by the application.

3.6 Chapter evaluation

- Conventional heatsinks’ inherent reliability and absence of need for power to function makes them very well suited to the thermal management of highly efficient, long design lifetime systems such as LED luminaires. The literature offers a substantial repository of knowledge regarding conventional heatsink designs, but exploring the performance of alternatives offers numerous opportunities for further research.
- The reviewed literature offers very little regarding the evaluation of a heatsink’s effectiveness (i.e. how efficiently the materials and surfaces are utilised for heat transfer). Neither do the effects of exposure to an environment appear to have been studied in any significant detail. Integrating these parameters into heatsink development could benefit long-term system reliability and reduce unnecessary cost, so is worth further study.
- Thermosyphons, heatpipes and vapour chambers are ideally suited to applications where geometric constraints prevent the heat from high power density sources being dissipated locally. However, solid copper has been shown to offer comparable thermal conductivity in a number of cases. With respect to LED lighting, there seem to be very few benefits to justify their use.
- Immersion of the heat source in a liquid can potentially realise far greater heat transfer than immersion in air. However, there are a number of practical issues to overcome and little evidence of any commercial benefit at present, so the process has very little relevance to this research.
- Energy storage would provide the means of buffering transient temperature fluctuations. However, the unpredictability of operating regimes and the finite heat capacity available
makes it unsuitable as a sole thermal management mechanism, so again it is expected to have very little bearing on this research.

- Active technologies offer high capacity for heat transfer but tend to be complex, to contradict the benefits of a high efficiency system or to offer very little evidence of their commercial advantages. There do not appear to be any imminent breakthroughs that would challenge this conclusion, so they are anticipated to have limited influence on the short-term future of thermal management strategies appropriate to this investigation.

- The thermal behaviour of circuit boards can be considerably enhanced using established manufacturing techniques. However, commercial factors would also be likely to restrict their adoption. These improvements cannot be expected to eliminate thermal management challenges.

- Stresses induced within a circuit board during cyclic thermal loading can lead to system failure. The literature offers very little discussion regarding typical luminaire usage patterns and so it is difficult to evaluate the risk this poses. It is assumed that in most cases systems operate under steady conditions so reliability under cycling conditions is of no concern. However, providing additional penalties are not incurred, minimising stresses is advisable, further promoting effective thermal management.

- It is clear that there are numerous thermal interface material (TIM) performance improvements to be exploited. However, there are no definitive timescales for when these enhancements will be commercially realised. They cannot, therefore, be relied upon to overcome current thermal management challenges.

- There is concern regarding the accurate assessment of TIMs. The literature also indicates that the properties of the interface can significantly alter during service. Until properties and lifetime degradation are better defined it is wise to incorporate a substantial margin of safety. Unfortunately, it is not clear what margin would be appropriate. For now, thermal management design should instead focus on ways to minimise undesirable consequences (e.g. eliminating interfaces and selecting more consistent materials).

- Integrating multiple thermal management technologies into a single system, or even a single component, can offer superior thermal management performance. However, it can also add to the system’s complexity and combine the limitations of the individual technologies. Unless the associated issues can be justified by the demands of the application there appears to be no clear commercial incentive to pursue this strategy.
• Theoretical models and computer-aided simulations are commonly employed in the literature. If such models are correctly defined they can accurately reproduce the system’s behaviour, enabling rapid development of concepts.

• The literature suggests that conventional thermal management technologies, and particularly passive forms, are restricted by technical constraints rather than physical limits, so there are opportunities for further improvements to be realised.

• For maximum effect, current developments tend to focus on specific cases, but the reviewed literature provides very few guidelines to support the implementation of this approach. Further work to establish effective design strategies and assess the potential improvements would have considerable value.

From this chapter, it is clear that there are opportunities to enhance the performance of established technologies. Passive devices, and particularly heatsinks, are inherently well-suited to the thermal management of LED systems and similar technologies. The potential to develop these devices provides an attractive and potentially valuable topic for further study. Assessing the impact of refining their design and defining methodologies to exploit their full potential was, therefore, chosen as the focus of this research.
Chapter 4: Review of commercial LED luminaires

The advantages offered by LED components over incumbent technologies have led to their rapid proliferation in the general lighting market. The result is a wide array of component package styles and performance categories which each offer unique benefits and limitations. The reviewed literature deals poorly with questions such as: which of these packages are most relevant to industry practice; how does industry practice appear to be evolving; and how this can be expected to influence the luminaire’s thermal management design. To address this gap a survey and analysis of commercial products across a two year period was conducted. The survey captured published data from a selection of luminaire manufactures regarding which types of LED component they employ in their products and how their thermal management is catered for. The results were analysed to identify industry trends in luminaire design and the commercial implementation of LED technology, how the industry is transitioning towards new strategies, and where the expected demand for future enhancements can be expected.

4.1 Methodology

A range of manufacturer’s products were assessed in an attempt to capture a broad overview of industry practices. Their selection was performed without any commercial influence or conscious bias towards particular organisations apart from the availability of data. No assumptions regarding product specifications were made, information was only recorded when explicitly provided in the manufacturer’s published literature (i.e. datasheets, product leaflets, catalogues). The only exception to these conditions was in regard to data on Thorlux Lighting’s products, which was supplied directly by the company’s technical manager.

The properties of interest were the luminaire’s total luminous flux, LED component package type, LED component package power consumption, luminaire thermal management strategy, forming processes employed and material composition. Luminaires were categorised according to the LED component type employed, classed as either low power (< 0.1 W), medium power (0.1 < 1 W), high power (> 1 W) or an array of LED chips in a single package, often referred to as a chip-on-board (COB) array. In the context of this survey, and following industry convention, COB array
refers exclusively to the package style shown in Fig. 4-1 rather than defined by a particular power consumption range. This array configuration allows a wide range of operating powers, with readily available components such as Tridonic’s TALEXmodule STARK FLE GEN1 consuming as much as 175 W (Tridonic, 2016).

Fig. 4-1: An example of a COB array (Tridonic, 2016)

To evaluate how the implementation of LED technology is evolving as the industry matures surveys were conducted at two distinct points, the first during October 2013 and the second during October 2015. Future surveys to expand the timeframe of this investigation would be extremely valuable, but unfortunately are not possible within the bounds of this research. The first survey drew from each company’s entire catalogue of available products. To establish how commercial trends evolved from here the second survey only included new products launched in the interim period. The first survey sample size was 75 and the second was 76. In cases where multiple specifications of the same luminaire existed, only the highest power consumption, most compact model was recorded. For consistency, the preferred sample emitted light at 4000 K and with a colour rendering index (CRI) > 80. In cases where a matching luminaire was not offered the
closest alternative was recorded. The difference in efficacy between a 4000 K, > 80 CRI and a 5000 K, > 70 (but < 80) CRI LED component is approximately 23 % (Samsung, 2016a). While this would significantly impact the luminous flux emitted, as this only accounts for about 25 % of total power dissipated (United States Department of Energy, 2009) the difference in waste heat is approximately 6 %. As the sampled luminaires were the closest models to the target 4000 K, 80 CRI output the difference was not believed to be as extreme and so it was assumed this did not significantly compromise the results of this work.

### 4.2 2013 Survey results

The full database of sampled products can be found in appendix A, Table A-1. The total luminous flux of each sample luminaire is plotted as shown in Fig. 4-2, with the data sorted by output and grouped according to the LED component category employed.

![Fig. 4-2: 2013 survey results for luminous flux emitted by luminaire differentiated by LED component category](image)

With respect to the range of luminous flux output by each luminaire (100 – 20,000 lm), 80 % of products were spread across a relatively narrow range (300 – 3000 lm output). This appears to satisfy the demand of most applications. There were a few products offering higher and lower output, but these tended to be niche applications. Luminaires developed around high-power LED

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1 Samsung LH351B.
packages dominated the survey, but there were several mid-power and COB array based products which offered comparable luminous output. There were no cases of luminaires employing low-power LED components captured in the survey.

The thermal management method employed by each of the sampled luminaires is summarised by Fig. 4-3. There were three methods observed; passive (natural convection) heatsinks (a dedicated structure with a large surface area to enhance convective heat transfer to the environment); body redistribution (no evidence of dedicated thermal management features beside the inherent heat transfer properties of the luminaire’s mechanical structure); and active (forced convection) cooling (e.g. heat transfer from the luminaire, which may or may not incorporate a dedicated heatsink, augmented with additional systems such as electromechanical fans). These were sorted according to the LED component category they were employed alongside.

![Graph](image.png)

**Fig. 4-3: 2013 survey results for distribution of thermal management methods employed by each LED component category**

The thermal management methods employed by luminaires utilising high-power LED components showed an even split between passive heatsinks and body redistribution. There was one example of an actively cooled, high power, LED system. Heatsinks were typically formed from extruded or die-cast aluminium. It was observed that all surveyed luminaires had black, grey, white or reflective surfaces. Half of the surveyed luminaires designed around mid-power LED components employed no dedicated thermal management. Because these chips operate at lower power it is believed that waste heat released by each component was small enough to circumvent
any requirement for dedicated management. However, this survey did not verify such a strategy offers acceptable operating conditions. At this stage of the survey only one COB array based luminaire employed no dedicated thermal management. These LED modules are generally compact and incorporate multiple LED die, thereby dissipating more power than an individual LED die and concentrating waste heat within a small region. Consequently, it would be reasonable to expect some form of dedicated thermal management to be employed. However, in this case the COB array appeared to have been chosen for its form factor rather than output. The luminaire’s luminous flux is relatively small, so there is less power and waste heat to manage. The construction also employed thick sections of highly conductive material (aluminium) to conduct heat from the small source throughout the luminaire. Again, this study did not evaluate how such a strategy provides sufficient thermal management. It should also be noted that the COB array, along with mid-power LED categories includes only a small number of samples, allowing atypical designs and data errors (such as mistakes made during capture or inaccurate source material) to significantly skew the results.

4.3 2015 Survey results

The full database of sampled products can be found in appendix A, Table A-2. Once again, the luminous flux of each luminaire, sorted by flux and differentiated by LED component type employed, was plotted as shown in Fig. 4-4.

![Fig. 4-4: 2015 survey results for luminous flux emitted by luminaire differentiated by LED component category](image-url)
The range of luminous flux from the surveyed luminaires (250 – 38,000 lm) was higher than in the previous survey. The predominant luminous flux reflects literature comments that an ideal luminous flux ‘sweet spot’ of around 3000 lm (Christensen and Graham, 2008) exists. Compared to the previous survey there were fewer samples in the low luminous flux (< 300 lm) range.

In comparison to the 2013 survey, by 2015 there was far more data available for luminaires employing low, mid and COB type LED components. COB LED arrays were the largest category of light source represented in this survey. These components tend to be more compact, offering improved aesthetics and permitting smaller luminaire designs. Luminaires designed around mid-power LED components were also far more numerous in this survey, suggesting that they offer some commercial benefit which has driven their increase. There were two luminaires identified based upon low-power LED components, which represents an increase from the previous survey, but is of limited significance. The greater quantity of lower power LED die required to deliver equivalent luminous flux to the higher power alternatives may provide beneficial photometric characteristics or lower cost. However, the small number of luminaires employing these components suggest they offer few commercial advantages over the alternatives and so were restricted to niche applications. This category appears, therefore, to present a negligible potential influence on the general lighting industry.

The occurrence of each thermal management technique employed by the luminaires, sorted by LED component category, was plotted as shown in Fig. 4-5.

![Fig. 4-5: 2015 survey results for distribution of thermal management methods employed by each LED component category](image-url)
The majority of high power and COB array based luminaires still required some form of heatsink to establish suitable operating conditions. However, compared to the 2013 survey, there was a significantly greater proportion of luminaires which implemented no dedicated thermal management.

Examples taken from the data indicate the thermal management strategy expressed less association with luminaire output than in the previous survey. The one actively cooled luminaire emitted just 2880 lm while many of the highest output luminaires incorporated no dedicated thermal management. These observations indicate that the thermal management strategy was primarily dictated by the LED component category and form of the luminaire rather than the magnitude of heat transfer.

The heatsink materials, manufacturing methods and surface finishes were similar to those seen in the previous study. The majority of luminaires designed around mid-power components, and all those designed around low-power components, employed no dedicated thermal management. With respect to luminaires based on mid-power LED components, the sample size was large enough to ensure an accurate representation of industry practice. Again, the findings do not prove specific thermal management measures were unnecessary. However, this appeared to be an increasingly common configuration and so must be presumed to be commercially viable.

### 4.4 Evaluation of results

#### 4.4.1 Limitations

There were several limitations of this analysis which need to be highlighted before evaluating the findings.

- The categorisation of LED types was based on an extremely simple definition which left some ambiguity. Some of the reported high-power LED components may have employed multiple chips within a single package, essentially making them COB arrays, whereas industry tends to limit the term COB array to a particular style of module (see Fig. 4-1). Consequently, the results may under report the proportion of luminaires employing COB arrays. This is not believed to significantly affect the investigation’s findings. If anything, it lessened the
apparent growth in the use of COB arrays. However it was clear that COB arrays still became much more common in recent products.

- It was assumed that the surveys provide a fair representation of the industry, but many products, across multiple manufacturers’ ranges, had to be excluded because of incomplete data. This was an industry wide issue, so was not believed to have unfairly influenced the findings. Its effect was believed to be consistent for both surveys so should not have had any influence when drawing comparisons.

- The initial survey drew from each manufacturer’s entire product catalogue, while the later survey was limited to products released in the interim period. This means the samples of the initial survey do not necessarily correspond to that particular time period, distorting the observed pace of any development and preventing historical trends from being incorporated into the work. The sources of information rarely provide clear product history that could have been used to avoid these issues. Consequently, it was not possible to accurately evaluate the rate at which the industry is evolving.

- The data presented by suppliers was predominantly contained within their marketing material. This would tend to emphasise certain features over others. For instance, COB arrays are a distinctive type of light source which may be seen as being more desirable, so are commonly highlighted, whereas low and mid-power LED packages are more rarely identified. This would have influenced the relative proportions of sample products found under each category. There was no reason to believe the emphasis of published data changed between surveys, so the general growth / contraction of each category can be established, but any comparison between the sizes of each category would be invalid.

- The manufacturer’s published data tended to lack document references, independent corroboration or consistency. This introduces considerable uncertainty, makes results difficult to reproduce and hinders verification. Every effort was made to manage the quality of the data and any errors have to be assumed to be reasonably consistent (although the effect on smaller sample groups may have created some anomalous results). With these issues in mind, the findings of this work can only be treated as a very general assessment.

- This review, in accordance with the focus of this research, was specifically concerned with luminaires targeted at general lighting applications because this represents the largest sector of the lighting industry (McKinsey and Company, 2012) and, therefore, has the widest relevance. However, there are a number of other applications which impose different constraints and demands that may result in different trends. For instance, it might be
expected that automotive applications are less constrained by commercial factors but
demand for extremely high output and geometric constraints make active thermal
management technologies a necessity. Unfortunately reviews of these other sectors could
not practically be included alongside this review.

4.4.2 Discussion

Comparison of the results of the two surveys shows that luminaires employing high-power LED
components are being superseded by systems designed around alternative LED types, particularly
mid-power components and COB arrays. At the present time high-power LED components lag
behind mid-power packages in terms of efficacy and also behind the compact, high luminous flux
of COB arrays. A high-power LED component\(^1\), for instance, can emit 525 lm at a luminous efficacy
of 112 lm.W\(^{-1}\) (Samsung, 2016 a). A mid-power component\(^2\) on the other hand can produce 90
lm at a far greater efficacy of 169 lm.W\(^{-1}\) (Samsung, 2016 b) while a COB array\(^3\) can produce
18,550 lm at a luminous efficacy of 95 lm.W\(^{-1}\) (Tridonic, 2016). It is believed these advantages
have driven the increase in the proportion of luminaires employing mid-power components and
COB arrays, and that the impact on luminaire design trends can be expected to continue.

Awareness of luminaire lifecycle performance and environmental impact appears to be a growing
market influence, illustrated by a major manufacturer taking steps to provide environmental
performance declaration (EPD) certificates for all of its products (Zumtobel Lighting, n.d.).
According to various lifecycle analyses (e.g. United States Department of Energy, 2012),
increasing efficacy has the greatest potential to reduce the luminaire’s environmental impact.
Reducing material consumption also plays a significant role. Mid-power LED components
appeared to impose less demanding thermal management requirements (i.e. less heatsink
material content) alongside superior efficacy. As LED technology matures and the market
saturates, the ability of manufacturers to differentiate their products from those of competitors
will become increasingly challenging. Highlighting environmental and lifecycle performance
offers the means to address that challenge, which will further support the adoption of mid-power
LED packages.

\(^1\) Samsung LH351B.

\(^2\) Samsung LM561B Plus CRI 80.

\(^3\) Tridonic TALEXmodule STARK FLE GEN1.
The literature review agrees that reducing the LED chip’s operating temperature is beneficial to its performance. However, luminaires based around low and mid-power LED components often omit dedicated features to enhance heat transfer from the luminaire. The associated costs and complexity of integrating these enhancements presumably outweigh the performance benefits. This also hints that LED packages may be becoming more tolerant to high temperatures, thereby circumventing the need for dedicated thermal management. This appears to be an incorrect strategy for achieving the greatest lifecycle performance, but further investigation is required (see ‘Chapter 9: System optimisation’). However, the increasing abundance of products omitting dedicated thermal management indicates that this approach is commercially feasible and advantageous. The potential cost reductions, simplified system design and reduction in material content that this permits are clearly valuable factors and would support the future growth of this strategy.

The later survey captured very few examples of luminaires that emit low luminous flux. This may be because there was limited justification to develop new products which satisfy a relatively small segment of the market, and for which a number of luminaires already exist (as identified in the earlier survey). It appears the focus of commercial product development shifted to higher output systems. By superimposing the two sets of survey data (see Fig. 4-6) it is clear there has been an overall upward shift in luminaire output occurring alongside developments in thermal management strategy. The observed trend for increased output seems to have no influence on thermal management strategy; instead luminaire design seems to be evolving to accommodate preferred thermal management strategies independently of luminaire output.

![Fig. 4-6: Comparison of luminaire luminous flux survey data](image-url)
Low-power LED components appeared to have very little bearing on the general lighting industry. There was limited evidence to suggest the growth of this category should be expected and no obvious advantages to drive any change.

The data showed limited demand for enhanced heat transfer from low and mid-power LED components. As luminaires developed around these components seem to be displacing systems employing high-power LEDs, the incentive to develop superior thermal management strategies to facilitate high-power components is expected to diminish. On the other hand, luminaires based on COB arrays appeared to be a growing category. The aesthetic and physical constraints which are believed to promote the use of this type of light source are not expected to disappear, and cannot be achieved by the alternative low- and mid-power components which require multiple packages to deliver equivalent output. Their ability to produce high luminous flux also makes them well suited to the ongoing drive for increased luminaire output, although there are practical limits to this factor. Luminaires designed around COB modules are expected, therefore, to continue being developed for the foreseeable future. Dedicated thermal management of these light sources remained common, and apparently more challenging (demonstrated by the occurrence of actively cooled luminaires which are known to offer superior heat transfer). However, the 2015 survey data also shows a greater quantity of these luminaires avoiding dedicated thermal management, possibly enabled by improving robustness of LED components. The demand for high performance thermal management techniques does not appear to be essential, but the majority of systems still required a passive heatsink. Enhancing the performance of these systems or reducing their material content and cost still holds considerable commercial value. It may allow active cooling technologies and their drawbacks to be completely avoided, permit more compact luminaire designs, reduce costly and environmentally damaging material content, or improve lifecycle performance. Development of enhanced thermal management techniques for luminaire’s based around this type of light source can be justified and this is therefore the focus of the research presented in the following chapters.
Robust techniques to evaluate the thermal behaviour of a luminaire are essential for this research. With them, the properties of a system can be quantified and their suitability confirmed. They also allow the thermal management performance of different systems to be compared, so that those offering superior commercial or operational benefits can be identified. This chapter discusses appropriate definitions of thermal management performance, methods of capturing the data necessary for its evaluation and how the techniques can be implemented regarding the subject of this research. This was undertaken with reference to commercial practice where simple and rapid techniques are valued. The methods discussed in this chapter are subsequently used to; capture benchmark data for the validation of simulation boundary conditions in Chapter 6; measure the evolving thermal behaviour of luminaires exposed to typical operating environments in Chapter 7; and quantify the thermal management performance of different heatsink designs in Chapters 8, 9 and 10.

5.1 Physical measurements

There are a variety of techniques which can be employed to acquire thermal data from a physical component, each with associated advantages and limitations (Pryde, 2012 a). For the purposes of this investigation the following methods were employed. They were selected for their ability to provide sufficient data, consistent accuracy and practicality.

5.1.1 Luminaire preparation

Handling and surface contaminants such as dust and residue from production processes are to be expected under normal circumstances. It was assumed that these would have a negligible effect on the luminaire sample’s operating behaviour, so no attempt to clean them was made. Exceptions were made where special preparation is required (i.e. drilling holes for applying
temperature sensors). Any resulting contamination or debris was removed from the affected area.

Where required, mechanical assembly was conducted in accordance with the manufacturer’s or suppliers instructions. For consistency all unspecified threaded fasteners were tightened to a torque of 10 N.m using a torque wrench\(^1\). The same torque wrench was used throughout this investigation.

**5.1.2 Controlled test environment**

To ensure repeatability, physical analysis was conducted in a controlled environment (see Fig. 5-1). On account of its practicality, suitability and availability, an 825 mm tall, closed chamber with a floor area measuring 600 x 600 mm was used in this investigation. The chamber walls were formed from 15 mm thick double layered corrugated cardboard which created a thermally insulating barrier against the surrounding environment and blocked disruptive airflow, ensuring consistent and reproducible test conditions. A matt black surface coating was applied to the internal walls of the chamber. This enhanced radiation absorption, thereby minimising reflected energy which may affect the test piece. The luminaire was positioned in the centre of the chamber floor. To minimise heat lost by conduction to the chamber base, it was supported above the floor on a thin-walled, 50 mm tall cardboard tube. It was assumed this had a negligible effect on the studied item’s thermal behaviour owing to its minimal contact area and poor thermal conductivity. This setup was believed to represent a similar situation to many typical installations where a luminaire sits within a ceiling void or mounted to a surface. The base of the test chamber provides a comparable obstruction airflow below the product whilst also allowing unimpeded airflow from the sides and above the luminaire. The enclosed chamber does cause heated air to be recirculated across the test piece. However, the parts being tested in this investigation were far smaller than the chamber (typically occupying a region smaller than 0.5 % of the chamber’s volume), and dissipated very little power (less than 20 W). Their impact on the test environment, and likewise the test chamber’s impact on the luminaire, was, therefore, assumed to be negligible. In practice, the ambient environment temperature never showed more than a 2 K increase during testing which would support this assumption. All temperatures were measured

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\(^1\) X-Tools XTLSTRQWRNCH2TO24NM.
with reference to the ambient environment temperature negating the effect of any change in ambient conditions during testing.

Measurements were conducted without disturbing the test environment. Power supply and sensor cables were routed across the chamber base and under the edge of the chamber walls. An overlapping edge arrangement prevented significant gaps which would allow the internal environment to be disturbed during tests. The ambient environment air temperature \( (T_{amb}) \) was measured at the point indicated in the image. The sensor was positioned 25 mm above the floor of the chamber to exclude boundary wall influences.

The luminaire was allowed to reach a stable thermal equilibrium within this test environment before its properties were measured. To ensure this, a minimum of 1 hour was allowed after
imposing a change to the sample’s operating conditions (i.e. connecting a power supply), but measurements did not occur until a $\leq 0.2$ K variation at each probe location was observed over a 15 minute period\(^1\). Preliminary tests showed this was sufficient to reach a stable operating condition, and therefore produce repeatable results. A tighter limit on temperature variation was not practical owing to inherent measurement fluctuation. Statistical methods to suppress the effects of fluctuating readings (for example using an average of three consecutive measurements) should be considered for future analyses. They were not applied here because of the additional complexity which opposes the objective of defining simple, rapid and commercially practical methods.

5.1.3 Electrical properties and thermal power

The electrical parameters of the luminaire are used to derive a number of thermal properties and monitor the condition of the heat source. During tests, electrical power was supplied to the LED components of the luminaire by the accompanying driver. The driver is a sub-system of the luminaire which converts electrical power from a source to an appropriate voltage and current for the luminaire’s LED components. The luminaires considered in this investigation all employed a discrete driver module. As these do not interact directly with the luminaire, except to supply electrical power to the LED, their behaviour was excluded from consideration by locating them outside the test chamber. A 500 mm long piece of 6242YH cable was used to connect the driver to the LED package within the luminaire. This cross-sectional areas of the cable’s conductive cores were both 1 mm\(^2\).

In this investigation electrical properties were measured using a Digital-Multimeter (DMM)\(^2\). The quoted accuracy is summarised in Table 5-1. The DMM used was purchased specifically for this research. It was factory calibrated and all testing was concluded within 24 months of purchase. Due to the restricted use, it was assumed the DMM remained within the quoted accuracy throughout the period of this research.

\(^1\) With regard to luminaire temperatures, this applied to the change in temperature, i.e. temperature rise above that of the ambient environment, rather than the absolute value.
\(^2\) Voltcraft Digital-Multimeter VC270.
Table 5-1: DMM resolution and accuracy

<table>
<thead>
<tr>
<th>Measurement type</th>
<th>Measurement range</th>
<th>Resolution</th>
<th>Rated accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC voltage</td>
<td>400 mV</td>
<td>0.1 mV</td>
<td>± 0.8 %, + 1 mV</td>
</tr>
<tr>
<td></td>
<td>4 V</td>
<td>1 mV</td>
<td>± 0.8 %, + 10 mV</td>
</tr>
<tr>
<td></td>
<td>40 V</td>
<td>0.01 V</td>
<td>± 0.8 %, + 0.1 V</td>
</tr>
<tr>
<td></td>
<td>400 V</td>
<td>0.1 V</td>
<td>± 0.8 %, + 1 V</td>
</tr>
<tr>
<td>DC current</td>
<td>40 mA</td>
<td>0.01 mA</td>
<td>± 1.6 %, + 0.04 mA</td>
</tr>
<tr>
<td></td>
<td>400 mA</td>
<td>0.1 mA</td>
<td>± 1.6 %, + 0.4 mA</td>
</tr>
<tr>
<td></td>
<td>4 A</td>
<td>0.001 A</td>
<td>± 2 %, + 0.01 A</td>
</tr>
</tbody>
</table>

The total electrical resistance of both conductors supplying power to the LED package within the luminaire, when unpowered and at an ambient temperature of 295.35 K, was measured to be < 0.1 Ω (± 1.5 %, + 1 Ω). This was considered to be too small to cause any significant voltage drop between the driver and the LED package, and so all power supplied to the luminaire by the driver was assumed to be dissipated within the LED package. Consequently, the LED component’s electrical parameters could be measured outside of the test chamber via the cable used to supply power to the device. Although it is preferable to measure electrical parameters near to the site of consumption, for the sake of simplicity and to eliminate any need to disturb the test environments point this was considered to be an acceptable compromise. Connection of the DMM was made via the accompanying leads\(^1\). Errors arising from the use of these leads will have been consistent for all measurements. For voltage measurements the DMM was connected in parallel fashion to the LED component. Current measurements were performed by connecting the DMM in series fashion with the LED component. Electrical power consumed, \(P_e\), with units of watts, can be calculated from the measured voltage and current characteristics by the equation:

\[
P_e = I \times V
\]

Equation 5-1

\(^1\) Voltcraft Safety test lead (Banana jack 4 mm – Test probe) MS-1A.
Where, I is current (measured in amperes) and V is voltage (measured in volts). Unless otherwise specified, in this investigation it was assumed that 75% of electrical power supplied to an LED package was dissipated as waste heat in accordance with the findings of a report by the United States Department of Energy (2009). The remainder of the electrical power was assumed to be emitted as useful light with no other significant losses to consider. It was assumed energy emitted as light had a negligible impact on the behaviour of the luminaire or test environment.

5.1.4 Thermal measurement
Surface temperatures were measured using thermocouples. Pollock (1991) offers a solid foundation for thermocouple theory and practice on which the following methods were based. Alternative techniques of measuring temperatures were feasible but less practical and/or require equipment that is unlikely to be available in most commercial environments. To ensure the thermocouple effectively reports the test part’s temperature, rather than that of the general environment around the target, their attachment requires special attention. With suitable calibration and control, accuracies of ± 0.1 K are achievable (Kinzie, 1973). IEC 60584 tolerance class 1, K-type thermocouples were used throughout this work. This grade of thermocouple has a measurement tolerance of ± 1.5 K in the temperature range relevant to this research. Connection between the thermocouple and instrumentation was made via a 1 m length of thermocouple cable (wire made from the same materials as the thermocouple). Ageing and other factors which can affect the behaviour of the thermocouple were assumed to be negligible.

The thermocouple was bonded to the test part’s surface using a polymer with enhanced thermal conductivity\(^1\). Preliminary tests showed bonding the thermocouple to the test specimen provided the most consistent contact whilst also providing a degree of shielding from environmental influences. Approximately 0.0625 (1/16\(^{th}\)) ml of adhesive was used to attach each thermocouple. Accurately dispensing this amount of material with readily available equipment was not practical. Instead, it was estimated using the head of a match as a comparable guide to maintain some basic control. To ensure the thermocouple maintained contact with the target surface while the adhesive cured, the free thermocouple cable was fixed to the sample using self-adhesive fibreglass tape at the edge of the adhesive bead. The thermocouple cable’s spring qualities exerted a small force at the thermocouple against the test piece to ensure effective thermal

\(^1\) Loctite 315 OUTPUT.
contact. The adhesive was allowed 72 hours at room temperature to cure before the self-adhesive tape was removed and testing could begin.

A variation on the preferred thermocouple attachment discussed above occurred when the sensor needed to be embedded within the body of a luminaire component. In these cases they were installed inside a 2 mm diameter drilled hole. Thermal adhesive was added to the bottom of the hole and the thermocouple inserted. To ensure the thermocouple was correctly placed at the base of the hole, a piece of stiff copper wire was used to help feed it into position. This piece of wire was then slowly withdrawn to avoid disturbing the thermocouple.

The resulting thermocouple electrical potential and derivation of temperature was performed using a digital thermometer\(^1\). The measurement accuracy of this device was ± 0.3 %, plus an additional 1 K. An additional uncertainty of ± 0.01 %, +0.03 K per degree over 301.2 K or under 291.2 K must also be applied to all temperature measurements within the range of 273.2 - 323.2 K. The thermometer was calibrated prior to each test using an ice-bath and the built-in adjustment facility. To stabilise the readout during measurement, the thermometer’s “Max” feature was used to display only peak temperatures. The thermometer display was temporarily frozen at the required measurement time using the device’s “hold” feature.

The commonly-used reference point when assessing the LED chip’s temperature dependent behaviour is the semiconductor junction. However, the packaging of the LED component (as with many electrical components) prevents direct junction temperature measurement and a thermocouple must be electrically isolated. For these reasons they are not appropriate for directly measuring junction temperatures. Thermal imaging methods require additional calculation using component properties that are not normally available or of questionable accuracy. It also requires more expensive equipment, is limited by calibration accuracy and has a restricted spatial resolution (Wang et al., 2011). EIA / JESD51-1:1995 (EIA / JEDEC, 1995) and a later update relating specifically to LED components (EIA / JESD51-51:2012 (EIA / JEDEC, 2012)) outline a method to use voltage drop to determine junction temperature. The benefit of this is that the junction temperature can be assessed without the need for direct contact or observation of the LED die. However, the data supplied by manufacturers, which is required to derive the junction temperature, is often difficult to interpret and verify (Siegal, 1992). The method is further complicated by the presence of multiple die and the need to make extremely accurate

\(^1\) Voltcraft Digital Handthermometer K102.
voltage measurements (Lasance, 2008). To implement this technique requires equipment capable of holding the device at specific temperatures, sensitive measurement equipment, extremely accurate pulsed power supplies and suitable software to interpret data. While the data produced is useful to have, the specialist equipment and techniques to acquire it present an obstacle to its implementation in many commercial settings. In addition, where a simple estimate of system performance is needed, and in most cases all that can reasonably be achieved considering limitations of the associated data (see ‘2.4 Reliability’), this method could be considered excessive. For general commercial requirements, thermocouple measurements of case temperature present a practical option. While they cannot be used to directly measure the temperature of an LED die (and consequently extrapolate lifetime, optical or other performance characteristics of the luminaire), they do meet the requirements of this investigation, i.e. allow the evaluation of the luminaire’s thermal management performance. If necessary, the LED junction temperature and associated performance characteristics can often be extrapolated from thermocouple measurements taken from specified reference points. Therefore, their use was considered more appropriate considering commercial demands and not unduly restrictive.

5.2 Virtual simulation

Simulation provides a powerful technique to evaluate the behaviour of a system. It can be applied early in the product development cycle when the potential to guide designs is greatest; it allows hypothetical scenarios to be explored; and it enables properties and performance characteristics that would be difficult to evaluate by other methods to be analysed.

To support this investigation, the computational fluid dynamics (CFD) package Solidworks Flow Simulation\(^1\) was chosen. It was employed alongside the parametric modelling package Solidworks\(^2\). Models of the luminaire were created using these packages’ inbuilt tools. Assigning and benchmarking of simulation boundary conditions is addressed in Chapter 6.

Star-CCM+\(^3\), an alternative CFD package, was evaluated for its potential to perform the required simulation studies. The package is very sophisticated and offers a comprehensive suite of analysis capabilities. However, it was judged to be poorly suited to the demands of this research.

\(^1\) Solidworks Flow Simulation 2014 SP4.0 Build 2765.
\(^2\) Solidworks Professional 2014 x64 Edition SP4.0.
\(^3\) Star-CCM+ 8.02.0111.
Development of effective systems prioritises strong parametric design capabilities that enable rapid evaluation of multiple scenarios. Star-CCM’s integrated geometry modelling capabilities were very limited and the process of importing and interpreting data from other sources would severely hinder the development process. Although the sophisticated physics models and simulation controls were useful, they were judged to be excessive for this research and in a commercial setting. The additional time and expertise required to implement this programme effectively, in addition to the expense of the package, further opposed its use. CFD simulation tools embedded within Computer Aided Design (CAD) software suites often lack some of the more sophisticated controls offered by dedicated simulation packages. However, the applied engineering focus and integrated nature of these software packages promote their use in commercial product development practice. While they do have drawbacks, they can be very accurate and effective if correctly employed (Mikjaniec et al., 2013). Consequently, the simpler Solidworks suite was selected for this investigation. It offered the ability to model and evaluate the luminaire within the same environment, allowing multiple configurations of complex geometry to be rapidly defined and analysed.

### 5.3 Evaluation of luminaire thermal management performance

Measuring a system’s physical properties alone does not provide a sufficient evaluation of its thermal management performance. Methods of interpreting this information and relating it to the objectives of the system are also needed. To establish some appropriate metrics, the thermal management performance of a typical LED luminaire was analysed and the results evaluated.

#### 5.3.1 Application of test methods

Fig. 5-2 shows an exploded view of the luminaire selected for evaluation\(^1\). This luminaire had a thermal feedback system to reduce its power consumption at elevated temperatures. To ensure the operating characteristics of the luminaire remained consistent throughout testing, the feedback system was disabled. The heat source employed was a COB type LED array\(^2\). The original

---

\(^1\) Tamlite Lighting TD20DL19L.

unidentifiable thermal interface material, used to enhance contact between this LED array and the luminaire heatsink was replaced with a known graphite-filled sheet\(^1\).

![Fig. 5-2: Exploded view of a typical LED luminaire](image)

The main elements of this luminaire are:

1. Heatsink
2. COB LED array
3. Thermal interface material
4. Power cable
5. Bracket
6. Mechanical fixings
7. Retaining spring
8. LED array holder
9. Reflector attachment clip
10. Reflector
11. Bezel

Tests were conducted in the specified test environment. Electrical and thermal parameters were recorded in accordance with the methods previously described (‘5.1 Physical measurements’). For the purposes of evaluation thermal measurements were taken at two locations on the heatsink. One, indicative of the heatsink’s upper temperature \((T_{\text{high}})\), was 2 mm behind the centre of the LED component. The second, capturing the heatsink’s lower temperature \((T_{\text{low}})\), was at the

\(^1\) HALA Contec TFO-S250-CB.
outermost tip of the fin farthest from the heat source (highlighted in Fig. 5-3). Note these are not referred to as maximum and minimum as the exact location of these temperatures was not known. Subsequent simulation (see ‘6.3.2 Analysis results’) revealed the temperature difference between the upper temperature probe location \( (T_{\text{high}}) \) and maximum heatsink temperature to be less than 0.3 K. The difference between the lower probe location temperature \( (T_{\text{low}}) \) and heatsink minimum temperature was less than 0.8 K. As the upper temperature measurement \( (T_{\text{high}}) \) represents a reasonable approximation of the heatsink’s peak temperature, and the lower temperature measurement \( (T_{\text{low}}) \) is an arbitrary reference to assess the conductive behaviour of the heatsink (which, as will be, discussed is relatively insignificant), these differences were considered acceptable.

A virtual model of the luminaire was created using the parametric modelling software package.

Fig. 5-3: Sample luminaire and thermocouple position for heatsink minimum temperature reference point
5.3.2 Luminaire properties

The properties of the luminaire are presented in Table 5-2. The heatsink material volume and surface area were calculated from the virtual model. All other properties were obtained from measurement of the physical component. Heatsink temperatures are expressed as the rise with respect to the ambient environment temperature \(T_{\text{amb}}\), which was measured as 294.5 K ± 2.6 K. This was measured in a similar manner to the heatsink temperatures and was of a comparable magnitude. It was, therefore, subject to similar uncertainty. It was not known what proportion of this uncertainty could be attributed to systematic errors and so the potential uncertainty was considered to be cumulative.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC voltage during operation ((V))</td>
<td>25.65</td>
<td>± 0.31</td>
</tr>
<tr>
<td>Supplied current ((A))</td>
<td>0.697</td>
<td>± 0.024</td>
</tr>
<tr>
<td>Electrical power consumption ((W))</td>
<td>17.88</td>
<td>± 0.83</td>
</tr>
<tr>
<td>Heatsink material volume ((m^3, x10^{-6}))</td>
<td>96.50</td>
<td>Unknown uncertainty transcribing from physical model</td>
</tr>
<tr>
<td>Heatsink surface area ((m^2, x10^{-3}))</td>
<td>76.90</td>
<td>Unknown uncertainty transcribing from physical model</td>
</tr>
<tr>
<td>Heatsink upper temperature relative to ambient environment (T_{\text{high}} - T_{\text{amb}}) ((K))</td>
<td>+ 30.9</td>
<td>± 2.7 + Ambient temperature measurement uncertainty</td>
</tr>
<tr>
<td>Heatsink lower temperature relative to ambient environment (T_{\text{low}} - T_{\text{amb}}) ((K))</td>
<td>+ 29.3</td>
<td>± 2.7 + Ambient temperature measurement uncertainty</td>
</tr>
</tbody>
</table>

5.3.3 Thermal management performance parameters

The first thermal management performance parameter of interest was the heatsink’s resistance to heat transfer. This must be sufficiently low to maintain suitable temperatures for all components in the system. It can also be used to calculate the temperature of a particular
location (such as the LED junction) from its operating conditions. The absolute thermal resistance, $R_\theta$, between two points is described by the equation:

$$R_\theta = \frac{(\Delta T)}{P_\theta}$$

Equation 5-2

Where, $P_\theta$ is the power transferred as heat (measured in watts, W), and $\Delta T$ is the difference in temperature (measured in kelvin, K), between the two points. Unless otherwise stated, in this research thermal resistance was calculated between the locations of the peak heatsink temperature and far field quiescent ambient air ($T_{amb}$). Absolute thermal resistance, with units of kelvin per watt (K.W$^{-1}$), captures the combined effects of all heat transfer modes and associated thermal resistances, and so provides a useful assessment of overall thermal management capability. However, it is also limited in that it relates to the specific conditions for which it is determined. This is because it is based on parameters that develop from the way the system’s various physical properties interact. Altering any of these properties changes the nature of their interaction and so thermal resistance is also modified. Methods of overcoming this limitation have drawbacks. For example, models to equate different load conditions such as those developed by Sadeghi et al. (2010), offer the necessary tools to compensate for different configurations but tend to be complex. Standardised thermal load definitions (such as those proposed by Poppe et al., 2014) are of limited relevance when the application does not adhere to the same configuration, e.g. a luminaire with a unique arrangement of LED components. It should be noted that transient thermal testing is an invaluable tool for evaluating and validating thermal resistance, but relies on availability of both a physical specimen and specialist equipment (Farkas and Poppe, 2013). It can, therefore, often be more practical to assess the thermal resistance of each load case separately.

A heatsink employing high thermal conductivity materials (aluminium in this instance), a relatively large surface area to volume ratio, and cooled by passive heat transfer modes (convection and radiation) will often be described by a small Biot number. This means heat transfer is primarily governed by the body’s interaction with the environment rather than internal conduction (Holman, 2010). For the example luminaire, thermal resistances was calculated through the heatsink body to the surrounding environment (between $T_{high}$ and $T_{amb}$). The specification of the analysed luminaire’s LED component states it emits up to 12.6 W of heat
under the operating conditions of the test (Philips, 2013). Based on the measured temperature differential, the thermal resistance through heatsink body and to the surrounding environment (between \( T_{\text{high}} \) and \( T_{\text{amb}} \)) was 2.452 K.W\(^{-1}\). The temperature difference through the heatsink body (between \( T_{\text{high}} \) and \( T_{\text{low}} \)) was measured to be 1.6 K while the difference between the heatsink and the surrounding environment (between \( T_{\text{high}} \) and \( T_{\text{amb}} \)) was 30.9 K. This indicates resistance against heat transfer to the environment via radiation and convection did dominate the heatsink’s thermal management performance and that Biot number could be considered to be small. There is more potential, therefore, to enhance the heatsink’s interaction with the surrounding environment and so it would be the more appropriate focus for development.

Thermal resistance can be achieved by various heatsink designs, some of which may exploit heat transfer mechanisms to better effect or be more commercially attractive to produce. These considerations are not captured by the heatsink’s thermal resistance properties and so a secondary measure of how effectively the heatsink’s design develops its behaviour is required. As heat transfer to the environment is facilitated by the heatsink’s surface, achieving the same thermal resistance using less surface area (\( A \)) represents better utilisation of the heat transfer interface. A simple thermal management performance metric can therefore be calculated by the equation:

\[
\text{Performance} = A \times R_\theta
\]

Equation 5-3

This expression, describing the specific thermal resistance of a heatsink, carries units of squared metre kelvin per watt (m\(^2\).K.W\(^{-1}\)). Smaller values of specific thermal resistance represent better thermal management performance. Where the Biot number of the heatsink is small the inverse of specific thermal resistance would translate to an average heat transfer coefficient, \( h \). This carries the units watts per squared metre kelvin (W.m\(^2\).K\(^{-1}\)). Thermal management performance can alternatively be evaluated, therefore, using the equation:

\[
h \approx \frac{1}{(A \times R_\theta)}
\]

Equation 5-4
Hereafter, average heat transfer coefficient refers to this derivation. From the example heatsink’s surface area and thermal resistance, this average heat transfer coefficient was calculated to be 5.30 W.m\(^{-2}\).K\(^{-1}\). This is a reasonable, but relatively low, magnitude considering the typical range of convective heat transfer by natural convection of a gas (Cengel, 2003).

The definition of effectiveness does not reference how fully the potential heat transfer from the part’s surface was exploited. Integrating an additional parameter akin to the Number of Transfer Units (NTU) property would be a valuable extension to this thermal management performance criterion. However, the familiar and simple average heat transfer coefficient provides an adequate basis for comparison.

From this evaluation it was clear the greatest obstruction to heat transfer was the rejection of heat to the environment. The resulting average heat transfer coefficient from the heatsink was relatively low. The heatsink’s interaction with the environment, therefore, presents a significant opportunity for development.

### 5.4 Evaluation of methods

This chapter discussed the methods used to measure and evaluate the thermal behaviour of a luminaire. Although they were fairly basic, they were also within the means of this investigation’s resources. This simplicity and practicality also makes them ideal for use in a commercial environment, where similar limitations are to be expected. While the various simplifications introduced considerable uncertainty into the analysis, it was believed that they were adequately managed to maintain their consistency. Therefore, the techniques discussed enable the robust comparison of different system’s thermal management performance. More accurate test methods may allow for greater confidence, but were considered unnecessary for the requirements of this research.
Chapter 6: Validation of simulation methodology

Computational simulation methods are well established as useful tools for evaluating the performance of many types of system. Performing the associated calculations can be a complex task which is well suited to computer processing. Consequently, a number of commercial software packages exist for such purposes. However, while these simulation packages are generally very robust and capable of accurately evaluating an input model, they must all be supplied with appropriate data to arrive at a valid solution. This leaves computational analysis prone to considerable uncertainty (Lasance, 2002). Anecdotal evidence appears to suggest diminishing processing constraints, increasing accessibility and the general inexperience of the lighting industry promote the use of simulation in a fairly simplistic fashion (e.g. modelling the entire system to an unnecessarily fine level of geometric detail). This also introduces numerous potential sources of error in terms of its definition. Establishing appropriate simulation parameters, assumptions and simplifications to enhance their application is, therefore, required to support this industry’s adoption of thermal management design tools.

The objective of this chapter was to define and validate some appropriate simulation methodologies which accurately and efficiently reproduce the thermal behaviour of some typical luminaire components. A series of representative case studies were used to determine appropriate properties. The cases considered here were: an extruded aluminium heatsink with black surfaces; a die-cast aluminium heatsink with black and reflective surfaces; an extruded aluminium heatsink with reflective surfaces; and an extruded aluminium heatsink coupled with multiple heat sources on a circuit board. Examples of each of these were found by the market survey presented in Chapter 4 to be common in industry. They also have distinct thermal management properties. It is valuable, therefore, to focus on these cases. Benchmark data acquired from corresponding physical samples was then used to validate the results.

6.1 Case study 1: Black, extruded aluminium heatsink

The market survey presented in Chapter 4 highlighted extruded aluminium with a black surface finish to be commonly used for luminaire heatsinks. This was, therefore, selected for study.
6.1.1 Model definition

The case study was based on a stock extruded aluminium heatsink, comparable to those used in LED luminaires, measuring 125 mm in diameter and 25 mm in depth (Fig. 6-1). It was from an unknown source and so a detailed specification was unavailable. It was known to be formed from an aluminium alloy and the surface was treated with a black coating believed to be anodisation. The heat source employed was a COB type LED array. It was attached to the centre of the bottom face of the heatsink by two threaded fasteners tightened to 10 N.m torque in accordance with ‘5.1.1 Luminaire preparation’. A graphite-filled thermal interface sheet material was sandwiched between the LED array and heatsink contact surfaces to enhance heat transfer.

Thermal and electrical measurements were performed on a physical specimen according to the procedures outlined in ‘5.1 Physical measurements’. The heatsink’s upper \( T_{\text{high}} \) and lower \( T_{\text{low}} \) temperatures were measured at the two points indicated in Fig. 6-1. The simulation model temperatures were monitored at the same locations as the thermocouples attached to the physical sample.

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Fig. 6-1: Diagram of case study 1; extruded heatsink with LED package

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2 HALA Contec TFO-S250-CB.
The case study was modelled in the simulation package with all small (< 2 mm) holes and fillets excluded. These geometrical simplifications were assumed to have a negligible effect on the model’s behaviour. Geometric data was acquired by measurement of the physical sample. The presence of mechanical fixings was assumed to have negligible impact on the thermal behaviour of the luminaire and so were also excluded from the simulation.

Fig. 6-2 shows the computational domain (represented by the shaded region). It was based on the dimensions of the controlled experimental test chamber used to acquire benchmark experimental measurements (see ‘5.1.2 Controlled test environment’), i.e. extending 0.3 m in the x and z directions, and - 0.07 m to + 0.755 m in the y direction with gravity acting in – y direction. The system’s co-ordinate origin was positioned at the centre of the heatsink base. Symmetry conditions (a frictionless boundary with no fluid flow and no heat transfer across it) were applied to the xy and zy planes. The symmetric region is represented by the hatched outline.

![Fig. 6-2: Heatsink computation domain](image)

For a practical compromise between predictive accuracy and processing resources the target computational mesh employed 150,000 to 200,000 cells per quarter domain. Mesh refinement
was focused in a spherical region of 75 mm radius positioned around the centre of the heatsink. Outside this region cell density was uniform. The vertical to horizontal aspect ratio of all mesh cells was approximately 1:1. At least five cells spanned each inter-fin space. The computational mesh, shown for a horizontal cross-section through the heatsink region in Fig. 6-3, is a non-conformal structured Cartesian grid. The maximum and minimum heatsink temperatures were used as convergence goals. Convergence criteria were determined automatically by the CFD package, but only came into effect following one simulation travel (the propagation of each boundary condition’s influence throughout the entire computational domain). This occurred after approximately 100 iteration steps.

Fig. 6-3: Local mesh refinement applied around heatsink
For the purposes of simulation, it was assumed any interaction between the luminaire and the controlled test chamber was negligible. This was based on the chamber being large with respect to the luminaire and the internal surfaces being treated to minimise any effect on the subject’s behaviour. Because of this, the simulation domain was defined as though the luminaire existed in an open environment and not bounded by solid surface. Walls of the computational simulation domain, excluding those described by symmetry conditions, were allowed to act as both an inlet and outlet using the software’s default free stream boundary condition. This does not recreate the effect of recirculating air within the test environment. However, as interaction between the luminaire and test environment was assumed to be negligible, aided by the sample being raised above the chamber floor to minimise boundary wall effects, this was believed to be an acceptable simplification. In some of the following analyses, to ensure such a simplification did not adversely affect the accuracy of the simulation results, a simple representation of the luminaire’s surroundings was included where it was believed to have a significant influence on the system’s behaviour (see Chapter 9 and Chapter 10). The fluid behaviour was simulated using the program’s inbuilt model for air under standard atmospheric conditions and an initial temperature of 298.15 K. A combined laminar and turbulent fluid flow model was employed in order to accommodate the full range of potential flow phenomena. As noted in literature review, radiative heat transfer can contribute up to 40 % to total heat transfer from the system (Sparrow and Vemuri, 1985). It was therefore considered necessary to include its effects. Radiative heat transfer was modelled using a wall-wall interaction model that employs a ray-tracing procedure to determine the proportion of energy transferred in each direction. The radiation view factor resolution (the number of rays traced for each view factor calculation) was set at 5. The environment to which radiative heat transfer from the luminaire was emitted was specified as 298.15 K. Because the material specification for the heatsink component was not available it was estimated from similar examples. It was assigned custom material properties based on a 6000 series aluminium alloy with a thermal conductivity of 171 W.m⁻¹.K⁻¹ (MATWEB, n.d. a). The surface of the heatsink was semi-reflective and black. This was estimated to have an emissivity value of 0.7 (estimated based on values supplied by Fluke Corporation (2007)). In this case it was the behaviour of the heatsink which was of interest, from which the temperature of the LED component can be derived using its already well characterised properties. A 3.15 W thermal load was assigned to the heatsink based on the LED component’s specification (Philips, 2013) and the symmetry conditions applied to the simulation. It was assumed the heat
source was accurately modelled by being uniformly distributed across the entire contact region between the heatsink and LED component bodies (i.e. a 27.3 mm by 38 mm area). Previous studies have treated heat transfer from the exposed surfaces of the LED component body to its environment to be negligible (Christensen and Graham, 2008). It is unclear if that condition applies to the COB array employed in this case. As the exposed surface area of the LED package represents just 2% of the entire model’s surface area, excluding its behaviour from the simulation was still believed to be an acceptable simplification. Consequently, both as a simplification and to eliminate several potential sources of error, the LED component body was removed from the simulation model. This also made it possible to omit the thermal behaviour of the interface between the LED component and heatsink body. Radiative heat transfer was disabled where the surface of the heatsink was obscured by the LED component body. Detailed thermal models of the LED component and its interface behaviour are rarely available. Consequently, these simplifications were considered to be appropriate and representative of common industrial practice.

All unassigned radiating surfaces were left at the default setting of non-radiating surface behaviour. All undefined material interfaces were assumed to be perfect. It has long been established that surface roughness can have a significant effect on heat transfer (Dipprey and Sabersky, 1962). Accordingly, to accurately reproduce the behaviour of the component all surfaces were assigned a surface roughness of 1.6 µm based on a typical value for the manufacturing processes employed (Booker et al., 2001). Based on measurement of the physical part’s steady state operating temperature, the heatsink body was assigned an initial uniform temperature of 320 K to accelerate simulation convergence.

### 6.1.2 Analysis results

A typical simulation result is shown in Fig. 6-4, with fluid flow velocities, heatsink surface temperature and specific reference temperatures plotted. The graph in Fig. 6-5 compares the measured and simulated temperatures at the heatsink’s upper ($T_{\text{high}}$) and lower ($T_{\text{low}}$) reference points with respect to the ambient environment temperature. Cumulative uncertainties in the physical measurements are indicated. Thermal resistance was calculated from the recorded temperatures and is reported in Table 6-1.
Fig. 6-4: Case study 1, simulated fluid flow and heatsink temperature profile

Fig. 6-5: Case study 1, comparison of simulated and measured temperatures
The measured temperatures of the heatsink component were accurately reproduced using the assigned simulation parameters. The slight deviations were well within the bounds of thermocouple measurement uncertainty. The thermal resistance likewise showed an accurate agreement between simulation and measurement (for this research defined as the greater of ± 0.1 K.W$^{-1}$ or ± 10 % difference). The small difference between the heatsink’s upper ($T_{\text{high}}$) and lower ($T_{\text{low}}$) reference points, compared to the large difference between the heatsink’s lower ($T_{\text{low}}$) reference point and the ambient environment, demonstrates that the system’s interaction with the environment had the greatest influence on its overall thermal management performance. The accuracy of this simulation implies the assigned boundary conditions, assumptions and simplifications were valid. For comparison, a second analysis was performed in which the effects of radiative heat transfer were suppressed. The resultant upper and lower temperature reference points ($T_{\text{high}}$ and $T_{\text{low}}$) of the heatsink were in turn 28.1 K and 26.0 K above that of the ambient environment. Although these simulated temperatures were within the bounds of measurement uncertainty, they were approaching the extreme limit of the range and so were considered less likely to be accurate. The corresponding thermal resistance based on these temperatures was 2.229 K.W$^{-1}$ (representing an error from the measured value of 19 %). This was outside the range defined as acceptable. As an approximate means of evaluating radiative heat transfer’s contribution to total thermal power dissipated, for a thermally simple system where conductive heat transfer can largely be ignored, radiative and convective heat transfer can be considered the two defining parallel modes of heat transfer from the system. The resistance to heat transfer by these modes can then be evaluated using established principles of parallel resistance described by the relationship:

$$\frac{1}{R_{\theta}} = \frac{1}{R_{r}} + \frac{1}{R_{c}}$$

Equation 6-1
Where, $R_\Theta$ is the total thermal resistance of the heatsink, $R_r$ is the thermal resistance opposing radiative heat transfer and $R_c$ is the thermal resistance opposing convective heat transfer (all of which are measured in K.W$^{-1}$). If the total combined thermal resistance of the heatsink, $R_\Theta$, is taken to be 1.813 K.W$^{-1}$ as per the simulated value combining both heat transfer modes, and that predicted with radiative heat transfer suppressed (2.229 K.W$^{-1}$) represents the thermal resistance opposing convective heat transfer alone, $R_c$, it is possible to calculate the thermal resistance opposing convective heat transfer, $R_r$, and its fractional contribution to total power dissipated by the heatsink. In this case radiative heat transfer appeared to account for about 19% of total power dissipated. This reveals that in these circumstances radiative heat transfer can indeed be considered significant and to exclude its effect would not be a valid simplification.

The results show that the location of the heatsink’s lower temperature probe ($T_{low}$) did not correspond with the location of the lowest temperature (see labels in Fig. 6-4). The difference between the two was 0.07 K. As the lower temperature reference is an arbitrary point used to validate the conductive behaviour of the heatsink body it can still be used for comparison. However, it does highlight that predicting the behaviour of a system is not straightforward and simulation can be a valuable tool to help define a robust test procedure. Similarly, the simulated heatsink’s peak temperature was 321.11 K, 0.11 K higher than the upper temperature reference ($T_{high}$). Again, because the upper temperature reference ($T_{high}$) was measured at the same location on the simulated and physical models it can be used to make a valid comparison between the two.

### 6.2 Case study 2: Black and reflective, die-cast heatsink

In addition to the extruded heatsink types considered in Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’), the market survey presented in Chapter 4 also highlighted die-cast aluminium heatsinks as common thermal management devices. This was, therefore, selected for study.

#### 6.2.1 Model definition

The die-cast heatsink selected for study (Fig. 6-6) measured 130 x 102 x 38 mm. Its upper surfaces were treated with a black powder coating, while the lower faces were bare. The heatsink was
from a known source (Tamlite Lighting) and formed from ADC12 aluminium alloy\(^1\). It employed the same COB type LED array, thermal interface material and fixing method as the previous case study. The LED component’s position was dictated by existing fixing holes on the physical component.

The heatsink’s properties were evaluated in the same fashion as the previous case study. The heatsink’s upper \((T_{\text{high}})\) and lower \((T_{\text{low}})\) temperatures were measured at the same two points as indicated in Fig. 6-6. The simulation model temperatures were monitored at the same locations as the thermocouples attached to the physical sample.

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The case study was modelled in the simulation package using the same simplifications as for the previous case study i.e. small features removed and mechanical fixings omitted. Identical environment parameters, calculation settings and mesh conditions were also applied. The heat source was treated in an identical manner to the first case. However, the surfaces of this heatsink had been treated differently. The upper surfaces were treated with a matt black finish powder coating process while the lower faces (see note in Fig. 6-6) were left bare, resulting in a semi-reflective finish. Therefore, in the simulation model the black surfaces were assigned an emissivity of 0.8 and the bare surfaces were assigned an emissivity of 0.1 (estimated based on values supplied by Fluke Corporation (2007)).

The material used to form the heatsink carries a Japanese designation. It is comparable to the United States of America’s 300 series alloy designations (ed. CVERNA, 2001), for which material properties are more readily available. Accordingly, the heatsink body was assigned a thermal conductivity of 92 W.m\(^{-1}\).K\(^{-1}\) based on aluminium alloy 384.0-F (MATWEB, n.d. b). Properties were assigned using a custom material definition.

All undefined surfaces were assigned a default surface roughness of 3.2 µm based on a cautiously chosen value attainable by the manufacturing process employed (Booker et al., 2001). Based on measurement of the physical part’s steady state operating temperature, the heatsink body was assigned an initial uniform temperature of 335 K to accelerate simulation convergence. Aside from the specific conditions detailed here, the simulation parameters were the same as for Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’).

### 6.2.2 Analysis results

A typical simulation result is shown in Fig. 6-7, with fluid flow velocities, heatsink surface temperature distribution and specific reference temperatures plotted. The graph in Fig. 6-8 compares the measured and simulated temperatures at the heatsink’s upper \(T_{\text{high}}\) and lower \(T_{\text{low}}\) reference points with respect to the ambient environment temperature. Cumulative uncertainties in the physical measurements are indicated. Thermal resistance was calculated from the recorded temperatures and is reported in Table 6-2.
Fig. 6-7: Case study 2, simulated fluid flow and heatsink temperature profile

Fig. 6-8: Case study 2, comparison of simulated and measured temperatures
Table 6-2: Case study 2, calculated thermal resistances

<table>
<thead>
<tr>
<th>Evaluation boundaries</th>
<th>Thermal resistance (K.W⁻¹)</th>
<th>Difference*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Measured</td>
<td>Simulated</td>
</tr>
<tr>
<td>Heatsink upper temperature reference ((T_{\text{high}})) and ambient environment ((T_{\text{amb}}))</td>
<td>3.310</td>
<td>3.418</td>
</tr>
</tbody>
</table>

*Difference between simulated and measured value, expressed as a percentage of measured value

This system established higher temperatures and larger thermal gradients than the previous case. Again, the results of this study show excellent correlation between simulated and measured temperatures. The differences were within the range of thermocouple measurement uncertainty, but were of a greater magnitude than the previous case. Observations indicate this was in part the result of the definition of the thermal load. The COB type array employed in this case was composed of multiple LED chips (the heat sources) clustered about its centre. Approximating the thermal load as being uniformly distributed across the entire area of the COB array’s footprint was believed to have a greater effect in this case. In addition, the heatsink was formed from a material with significantly lower thermal conductivity and the thickness of the wall to which it was attached was much thinner than in the previous case. These conditions amplify the effects of thermal spreading resistance (the resistance to conductive heat transfer from the small localised heat source through the heatsink) hence result in the erroneously low simulated heatsink thermal resistance. Restricting the simulated thermal load to a smaller region improved the accuracy of the results, but to apply this modified definition accurately and consistently would have significantly increased the model’s complexity. Because the results were within acceptable limits, changing the heat source’s definition was considered unnecessary. However, any cases where the effects of thermal spreading resistance are more pronounced would be likely to need an improved definition of the thermal load to maintain the simulation’s accuracy. As with the previous case study, a second analysis was performed in which the effects of radiative heat transfer were suppressed. In this case, the resultant upper and lower temperature reference points \((T_{\text{high}} \text{ and } T_{\text{low}})\) of the heatsink were in turn 59.2 K and 51.4 K above that of the ambient environment, both of which were outside the bounds of measurement uncertainty. The predicted temperatures would represent a total system thermal resistance of 4.700 K.W⁻¹ (a 42 % error compared to the measured value). It can confidently be concluded, therefore, that excluding the effect of radiative heat transfer in this case was unacceptable. Here radiative heat transfer accounted for approximately 27 % of all thermal power dissipated. This is greater than the
previous case and is likely to be a consequence of the different heatsink geometry and surface emissivity. However, further investigation would be required to understand exactly how these differences contribute to the change of behaviour.

Once again, the heatsink’s lower temperature reference point and the minimum reported temperature location did not correspond. In this case the difference between two these temperatures (1.66 K) was larger than the previous case (0.07 K). Using the actual minimum temperature would, in this case, have improved the calculation of internal thermal resistance by providing a larger thermal gradient over which it was evaluated, thereby reducing the significance of any uncertainty. However, the accuracy was within reasonable limits and so repeating the analysis of the physical component to measure the heatsink’s actual minimum temperature offered little value. The heatsink’s upper temperature reference point ($T_{\text{high}}$) and peak temperature also differed. In this case the simulated peak heatsink temperature was 0.16 K higher than the upper temperature measurement ($T_{\text{high}}$). As with the previous case, because the upper temperature reference ($T_{\text{high}}$) was measured at the same point on the simulated and physical component it provided an acceptable means of comparison.

**6.3 Case study 3: Reflective, extruded aluminium and copper heatsink**

Reflective surface finishes are employed by some commercially available LED luminaires (e.g. Tamlite Lighting TD20DL19L). These have significantly different radiative heat transfer properties to the black surfaces considered in the previous case studies. To ensure the behaviour of luminaires incorporating such surfaces could accurately and efficiently be reproduced, a separate study was conducted.

**6.3.1 Model definition**

The same luminaire employed in ‘5.3 Evaluation of luminaire thermal management performance’ formed the basis of this case study. The heatsink measured 80 mm in diameter and 55 mm in depth. Detailed specifications of the materials and processes used for its production could not be obtained but it was apparent a copper core had been sandwiched between two extruded aluminium sections (Fig. 6-9). The heatsink’s exposed surfaces were all highly reflective. It employed the same COB type LED array, thermal interface material and fixing method as the first case study.
The heatsink’s properties were evaluated in the same fashion as Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’). The heatsink’s upper ($T_{\text{high}}$) and lower ($T_{\text{low}}$) temperatures were measured at the same two points as indicated in Fig. 5-3. The simulation model temperatures were monitored at the same locations as the thermocouples attached to the physical sample.

The case study was modelled in the simulation package using the same simplifications as for Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’) i.e. small features removed and mechanical fixings omitted. Identical environment parameters, calculation settings and mesh conditions were also applied. The heat source was treated in an identical manner to the first case.

Fig. 6-9: Diagram of case study 3; heatsink with reflective surface and LED package
Because the specifications for the materials used in the heatsink were unavailable, their properties had to be estimated. The copper core was assigned the material properties of pure copper taken from the software package’s internal database, i.e. thermal conductivity of 401 W.m\(^{-1}\).K\(^{-1}\) at 300 K. The 6000 series aluminium alloy material property applied to the extruded heatsink simulated in the first case study was used again for the extruded aluminium sections of this heatsink. Thermal contact between the three heatsink components was assumed to be perfect. The presence of some type of thermal grease and the high contact pressure in the physical sample minimise any error arising from this assumption. The exposed surfaces of the heatsink structure were all assigned a custom emissivity value of 0.05 (estimated based on values supplied by Fluke Corporation (2007)).

Based on measurement of the physical part’s steady state operating temperature, the heatsink bodies were assigned an initial uniform temperature of 325 K to accelerate simulation convergence. Aside from the specific conditions detailed here, the simulation parameters were the same as for Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’).

**6.3.2 Analysis results**

A typical simulation result is shown in Fig. 6-10 with fluid flow velocities, heatsink surface temperature distribution and specific reference temperatures plotted. The graph in Fig. 6-11 compares the measured and simulated temperatures at the heatsink’s upper \((T_{\text{high}})\) and lower \((T_{\text{low}})\) reference points with respect to the ambient environment temperature Cumulative uncertainties in the physical measurements are indicated. Thermal resistance was calculated from the recorded temperatures and is reported in Table 6-3.
Fig. 6-10: Case study 3, simulated fluid flow and heatsink temperature profile

Fig. 6-11: Case study 3, comparison of simulated and measured temperatures
The simulated temperatures match those of the physical sample extremely accurately and were well within the range of measurement uncertainty. This translates to an excellent prediction of the thermal characteristics. System thermal resistance ($T_{high}$ to $T_{amb}$) was almost perfectly reproduced by the simulation, indicating its definition was valid. The multiple components of the heatsink assembly had no noticeable effect on the results and so the associated assumptions were considered appropriate. Again, a separate analysis was performed with the effects of radiative heat transfer suppressed. In this case, the resultant upper and lower temperature reference points ($T_{high}$ and $T_{low}$) of the heatsink were in turn 33.8 K and 32.0 K above that of the ambient environment. These were within the bounds of measurement errors and extremely close to the measured value. The corresponding thermal resistance under these conditions would be 2.683 K.W$^{-1}$ (representing an error of 2 % compared to the measured value). The fractional contribution of radiative heat transfer in this case was just 4 %, demonstrating it had relatively little impact on the system’s thermal management performance compared to the previous cases. Again, it is believed the reason for the difference in the observed impact of radiative heat transfer is a consequence of the heatsink’s geometry and surface emissivity. However, without further investigation to determine when radiative heat transfer is significant it is believed to be more appropriate to always include its effects in the simulation definition.

Once again, the heatsink’s lower temperature reference point ($T_{low}$) and the minimum temperature locations did not correspond. The minimum heatsink temperature was actually 328.19 K, 0.75 K lower than $T_{low}$. The heatsink’s upper temperature reference point ($T_{high}$) and peak temperature also differed. The peak heatsink temperature was simulated to be 331.01 K, 0.28 K higher than the upper temperature reference ($T_{high}$). Despite the relatively large differences, the temperatures were measured consistently on the simulated and physical component so they can still be used to make a valid comparison.
6.4 Case study 4: Multiple heat sources on a circuit board

The previous case studies have all employed a single COB type LED array as the heat source. The market survey results of Chapter 4 identified multiple discrete LED packages as a common, albeit diminishing, alternative. It is worthwhile identifying simulation parameters to allow accurate and efficient reproduction of the behaviour of this configuration. However, there are numerous variations which must be considered and so in addition to accurately reproducing the behaviour of a single case study the behaviour of the model for a variety of different typical conditions was also explored.

6.4.1 Model definition

This study was based on a commercially available LED array\(^1\). This was chosen so any subsequent experimental analysis could employ a readily available component. The module comprised 4 discrete LED packages attached to a circuit board measuring 60 mm by 65 mm. The LED array was supplied with an attached lens. To avoid any influence this may have had on the system’s thermal behaviour it was removed. The LED array was mechanically fixed to the underside of the same heatsink studied in the Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’) by four threaded fasteners tightened to 10 N.m torque in accordance with ‘5.1.1 Luminaire preparation’. The same graphite-filled thermal interface sheet material\(^2\) used in the previous studies was used to enhance thermal contact between the two bodies (Fig. 6-12).

\(^1\) Vossloh Schwabe WU-M-444/B-NW 553939.
\(^2\) HALA Contec TFO-S250-CB.
The heatsink’s properties were evaluated in a similar fashion to Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’). The heatsink’s upper \(T_{\text{high}}\) and lower \(T_{\text{low}}\) temperatures were measured at the same two points as indicated in Fig. 6-12. A thermocouple was also attached to the face of the LED array at the manufacturer’s nominated test site \(T_{\text{ref}}\). The simulation model temperatures were monitored at the same locations as the thermocouples placed on the physical sample.

The previous case studies all employed a single thermal load applied directly to the heatsink across the heat source body’s contact region. This provided acceptable simulation results and so additional model refinements were considered unnecessary. However, the heat sources in this case were widely separated and attached to a metal (aluminium alloy) core circuit board. It was unclear if the previously successful modelling approximations were still appropriate, considering this complex circuit board structure and unknown level of interaction occurring between the distinct heat sources. To evaluate how this configuration could be accurately represented, a series of comparative analyses were undertaken. These configurations were referred to as “detailed”, “no-tracks”, “no-layers” and “distributed” (Fig. 6-13). The aim of the “detailed” simulation was to capture the impact of the circuit board’s notable features, and therefore accurately reproduce the thermal load applied to the attached heatsink. The model included a
representation of the circuit board’s internal layers (copper film, dielectric isolation and aluminium base). In this model the copper layer also included a representation of the conductive tracks and solder pads around each LED package (Fig. 6-14). Models of the circuit board with a simpler continuous copper layer (“no-tracks”) and a further simplified model containing only the aluminium substrate without copper or dielectric layers (“no-layers”) were modelled. A final model in which the discrete LED packages, dielectric isolation and copper layer were omitted, with the thermal load applied uniformly throughout the aluminium substrate was also simulated (“distributed”). Although this configuration was not much simpler, and less representative, than the other models considered here, being able to assign a single distributed thermal load may be beneficial when attempting to reproduce a large quantity of discrete heat sources. Its validity was, therefore, tested.

Fig. 6-13: Case study 4, circuit board simulation configurations shown as exploded views
Alongside an analysis of various heat source simplifications, the influence, if any, that the other aspects of the system have on the validity of the model was explored. Parametric studies were employed to modify the system’s properties by varying the heatsink depth between 1 and 50 mm. This established a range of thermal resistances. Parallel studies were undertaken using the same sized circuit board but with the heat sources separated by 30 mm (as per the physical part), 15 mm and 45 mm. In these studies the heat source was represented using the “no-layers” definition. For comparison a further study was conducted using the “distributed” heat source definition.

The case study was modelled in the simulation package using the same simplifications as for Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’) i.e. small features removed and mechanical fixings omitted. Identical environment parameters, calculation settings and mesh conditions were also applied. As shown in case study 1, the impact of radiative heat transfer from this heatsink can be significant so was included in the simulation definition.

Each individual heat source was treated in a similar manner to the first case (i.e. component body omitted, thermal load applied uniformly across the footprint area, non-radiating footprint surface etc.). The thermal power dissipated by each LED package was not publicly available. As discussed in ‘5.1.3 Electrical properties and thermal power’, a reasonable estimate of electrical power supplied to the array converted to heat is 75 %. Joule heating of the component interconnections was assumed to be negligible. Power was assumed to be equally distributed between the 4 LED
packages present. From measurement of the physical sample the thermal power dissipated by each LED package was estimated to be 1.46 W.

The circuit board measured 60 x 65 mm and was 1.6 mm thick. Because of its small features an exception had to be made to exclude it from geometric simplifications. The manufacturer did not provide details of this component’s composition. Its properties were instead approximated from known alternatives. There were three distinct components of the circuit board. The substrate was formed from an unknown aluminium alloy so was assigned custom properties based on a typical sheet material (MATWEB, n.d. c). This had a thermal conductivity of 138 W.m\(^{-1}\).K\(^{-1}\). The top layer of the circuit board was composed of copper and formed the electrical interconnections between the LED packages. This was assigned material properties using the simulation software’s existing pure copper model. The copper was assumed to be 35 µm thick as per the electronics industry standard. Separating the copper interconnections and aluminium substrate was a layer of dielectric material. The thermal properties of this were based on a similar material which had an in-situ thermal conductivity of 1.3 W.m\(^{-1}\).K\(^{-1}\) (The Bergquist Company, n.d. b). The material thickness was modelled as 76 µm in accordance with the reference material’s specification.

The surfaces of the LED array not bonded to the PCB were exposed to the surrounding environment, thereby permitting the rejection of heat from the system. This was consistent across all models. The surface to which the LED components were attached had a white, matt coating. This was recreated by assigning the corresponding faces of the simulation model a radiation emissivity value of 0.8 to approximate the component’s surface finish. The other faces had a semi-reflective finish and so were assigned an emissivity of 0.1 (estimated based on values supplied by Fluke Corporation (2007)).

Because the body of the LED array was included in this simulation, it was also necessary to model the thermal impedance of the contact between it and the heatsink. The datasheet of the employed interface material specifies it establishes a thermal impedance of 154.8 \(10^{-6}\) K.m\(^2\).W\(^{-1}\) under 68.95 kPa of pressure (HALA Contec, 2014). As the pressure applied to the thermal interface material could not be accurately determined, these worst-case properties were employed. Thermal impedance was assumed to be uniform across the entire contact area. All other material interfaces were assumed to be perfect.

Based on measurement of the physical part’s steady state operating temperature, the heatsink body was assigned an initial uniform temperature of 310 K to accelerate simulation convergence.
Aside from the specific conditions detailed here, the simulation models were setup the same as for Case study 1 (‘6.1 Case study 1: Black, extruded aluminium heatsink’).

6.4.2 Analysis results
Some typical simulation results are shown in Fig. 6-15, with fluid flow velocities, heatsink surface temperature distribution and specific reference temperatures plotted. The graph in Fig. 6-16 compares the measured and simulated temperatures of the heatsink’s upper ($T_{\text{high}}$) and lower ($T_{\text{low}}$) reference points with respect to the ambient environment temperature. The temperature of the LED array’s reference point ($T_{\text{ref}}$) is also included. Cumulative uncertainties in the physical measurements are indicated. Thermal resistances were calculated from the recorded temperatures and are reported in Table 6-4.

Fig. 6-15: Case study 4, simulated fluid flow and heatsink temperature profiles for “detailed” (top left), “no-tracks” (top right), “no-layers” (bottom left) and “distributed” (bottom left) simulation models
As Fig. 6-16 shows, the definition of the heat source within the model only had a minor influence on the temperature rise. Even defining the heat source as a thermal load distributed throughout the body of the LED array produced a reasonably accurate assessment of the heatsink’s thermal resistance. Once again the simulated results were well within the bounds of measurement uncertainty. However, the measured temperature rises were smaller than those of the previous cases, so the relative uncertainty is larger. As Table 6-4 illustrates, each simulation configuration resulted in an acceptable 5 % difference from the measured system thermal resistance (\( T_{\text{high}} \) to \( T_{\text{amb}} \)). The thermal power applied in the simulation was based on an approximation that would
have critically, but consistently, affected the accuracy of the simulation. This was believed to be responsible for the higher temperatures reported by each model. Even so, the results show sufficient accuracy to validate the approximations and assumptions made.

The thermal resistances predicted by the parametric studies for each configuration modelled are presented in Fig. 6-17. The average heat transfer coefficient for each of these simulations, based on the thermal resistances and total surface area, is plotted in Fig. 6-18. These properties are evaluated according to the methods outlined in Chapter 5. The profile of the curve in Fig. 6-17 demonstrated an initially non-linear relationship between heatsink depth and system thermal resistance. Above approximately 20 mm, increasing the heatsink depth had far less impact on thermal resistance. It was believed the most likely cause for this is that as heatsink depth increased, resistance to heat transfer through the body also increased, limiting any improvement in thermal resistance gained by the additional surface area of the heatsink. This is supported by the results presented in Fig. 6-18. They reveal that increasing the depth of the heatsink initially produces some small improvement in average heat transfer coefficient (most likely because conductive heat transfer is enhanced by the presence of additional material), but further increases are unable to exploit the heatsink surface area as effectively and so average heat transfer coefficient reduces. Further study would be required to characterise all the potential effects which produce this response. The consequence of this behaviour is a trade-off exists between the heatsink depth (and therefore volume of material) and system thermal resistance. Eventually, thermal resistance and heatsink depth become nearly independent, as too do heatsink depth and average heat transfer coefficient. Increasing heatsink depth to improve thermal management performance, therefore, has limits.
The results for the different heat source configurations showed very little variation. When compared to the model of discrete heat sources, applying the thermal load uniformly across the entire circuit board resulted in only a slight underestimation of the system’s thermal resistance. The greatest divergence between simulation configurations occurred when the heat sources
were separated by the smallest distance and coupled with a high thermal resistance heatsink. These LED packages were patterned in a 2 x 2 arrangement at 15 mm centres. Taking the effective region covered by this configuration to be 30 x 30 mm, the heat source array accounted for only 23 % of the entire circuit board area. For such a confined region, and when the system thermal resistance was greater than 3 K.W\(^{-1}\), modelling the heat source as a single load noticeably diverged from the result predicted by the presumably more accurate model employing discrete heat sources. It would therefore be an unacceptable simplification to apply.

### 6.5 Conclusions

The simulation definitions employed here, for the typical cases studied, have been shown to provide reasonably accurate results. They can, therefore, be used to guide commercial heatsink development without unnecessary complexity. Interaction between the system and its environment appears to impose the greatest barrier to heat transfer. The contribution of radiative heat transfer to total heat transfer can be significant (up to 27 % in the cases studied here). It is, therefore, necessary to include its effects in the simulation definition. The systems analysed here can be considered thermally simple. Consequently, simulation models could be significantly streamlined. Excluding small features from the simulation model appeared to allow for substantial simplification without any loss of accuracy. Indications from the case studies suggest the simulated thermal behaviour was reasonably insensitive to minor definition errors (demonstrated by the accurate results obtained from the numerous estimated properties). For the low thermal resistance systems studied here, distributing the thermal load across a large area produced acceptable results. However, the findings also suggest wherever thermal spreading resistance imposes a greater influence on the system’s behaviour (i.e. when lower conductivity materials are employed and / or component wall thicknesses are smaller), the thermal load should be applied in the vicinity of the heat source for the greatest accuracy. The resulting simulated thermal resistances of the studied systems were within 0.1 K.W\(^{-1}\) or 10 % of the measured values. This was judged to be sufficient for the temperature of the LED component to be quantified and predictions made about its performance. Consequently, the simulation parameters (including all associated assumptions, assigned material properties, surface behaviours environment interactions and heat source definition) were considered to be valid for the requirements of this research.

It should be noted that it was possible to achieve greater simulation accuracy for any individual case studied here. Practicality imposed the need to apply a degree of judgement and estimation
to the definition of each model. To reduce the impact of any subjective reasoning and judgement, the range of case studies was carefully selected. These covered a variety of properties and conditions relevant to this research. Treating each consistently and achieving reasonably accurate results across the entire series ensured the simulation parameters were truly representative of the physical component rather than a coincidentally accurate solution to a specific case.

The physical measurement uncertainties present in these analyses were relatively large. The actual uncertainty was believed to be much smaller than the potential margin, but this could not be verified. As well as using more precise equipment, applying these methods to cases which experience a greater temperature change would be recommended for future work. This would reduce the relative magnitude of measurement uncertainty and improve confidence in the results. It should also be noted that the heatsink’s actual maximum and minimum temperatures capture a larger thermal gradient across the heatsink body and so could be used to minimise the effect of any uncertainty when calculating thermal resistance. The heatsink’s upper and lower temperature references ($T_{\text{high}}$ and $T_{\text{low}}$) used in these case studies provided a close approximation of the actual maximum and minimum temperatures, and so did not have had a critical effect on the results. However, future evaluation would be advised to consider performing some preliminary tests in order to identify the most appropriate reference points to measure.

The studies conducted here unsurprisingly revealed a trade-off between heatsink depth and thermal resistance, which reaches a limit when further improvement becomes impractical or insignificant. The average heat transfer coefficient results also show that increasing heatsink depth, after some small initial improvement, makes increasingly ineffective use of the available surface area. The heatsink depth that offers the lowest thermal resistance did not achieve the highest average heat transfer coefficient. The reviewed literature offers very little exploration of this trade-off. The influence on thermal management performance of various system parameters, as well as the appropriate weighting to define an optimum configuration, are valuable properties to understand. Using the insight this provides to enhance thermal management performance requires further attention and was used to guide the following research.

These case studies have been used to determine valid simulation parameters and simplifications that acceptably reproduce the behaviour of some common systems. These can enhance the speed and accuracy of future simulations with valuable consequences for commercial product development, as the following chapters will demonstrate.
Chapter 7: Characterisation of operating environment effects

Chapters 5 and 6 have established effective methods to analyse the thermal management performance of a system. These provide a set of tools to aid in the design of an effective luminaire. However, the literature review also highlighted that the thermal properties of a system can evolve during service. This is expected to be especially critical when considering the anticipated extended lifetime of a typical LED luminaire. The thermal resistance of the LED package, and of the interface materials in particular, were shown to alter significantly during service. The outcome of this was an increase in the system’s total thermal resistance, which would negatively impact on its performance and potentially increase the LED junction temperature beyond its maximum rating. To ensure the maintenance of suitable LED die operating conditions throughout the lifetime of the system it must be designed to accommodate such potential changes. One topic absent in the reviewed literature was the impact of the operating environment on the system’s behaviour. It is reasonable to expect that exposure to typical operating environments will introduce effects such as surface fouling. This could interfere with heat transfer mechanisms, thus altering the thermal resistance between the luminaire and its surroundings. This chapter sets out to evaluate what impact, if any, the operating environment has on a typical luminaire’s thermal behaviour. The environments chosen for study were exterior and interior ceiling voids. These are two of the most common settings for a luminaire, and so offer the greatest analysis value.

7.1 Test methodology

Two identical luminaires were subjected to two typical application environments. The focus of this investigation was on the impact of these environments on the thermal resistance of the system. As discussed in Chapter 2 and ‘3.4 Thermal interface material enhancements’, the behaviour of thermal interfaces and LED components have already been explored in the
literature. Further analysis of these aspects was not an objective of this study, so it instead focused on how the thermal resistance of the luminaire’s heatsink alters with set time.

The luminaire used in this analysis was the same as used in ‘5.3 Evaluation of luminaire thermal management performance’. The focus of this study was the evolution of the heatsink’s thermal resistance. As similar heatsinks are employed in commercially available luminaires designed for indoor\(^1\) and outdoor\(^2\) environments, it was considered appropriate to use this component for both tests. However, owing to the unsuitability of some of the luminaire components for the studied environments (labelled 7 - 10 in Fig. 5-2), and because they were believed to be of little relevance to the system’s thermal management, they were removed from the luminaire. The luminaire bracket and bezel were retained in order to support the heatsink in its intended orientation during testing. The behaviour of the LED component and thermal interface was not included in this analysis. It was assumed the thermal load applied to the heatsink would remain constant throughout the product life. To minimise any degradation of these components due to the test environment they were removed between each series of measurements. Between measurements, the LED component was stored according to the manufacturer’s guidelines. The thermal interface was reinstated using the same stock material for each series of measurements.

Measurements were conducted on each luminaire at 6 month intervals over a total period of 18 months (between the 1st August, 2014 and 1st February, 2016). This was done to allow 4 measurements within the available time and consequent identification of any trends in the luminaire’s behaviour. Unless stated otherwise, the luminaire thermal management performance was measured using the procedures outlined in ‘5.1 Physical measurements’. A thermocouple was applied to the same upper temperature reference point (\(T_{\text{high}}\)) used in ‘5.3.1 Application of test methods’. The thermal power dissipated by the LED component was taken to be 12.6 W, as per the manufacturer’s specification (Philips, 2013). Electrical parameters were measured at each interval to monitor for any unexpected changes in the LED component’s behaviour.

Care was taken to avoid affecting the luminaire during handling. To minimise transportation of the sample all physical measurements were performed near to the assessed environment


locations. When absolutely necessary the luminaire was handled only by the bezel component as this was believed to have the least risk of impacting on the samples thermal behaviour.

Owing to the nature of each test, some specific handling precautions and preparations were required for the luminaires (discussed below).

7.2 Interior ceiling void
A luminaire fitted in an interior ceiling is a common situation. Often the body of the luminaire is recessed into the void behind the ceiling for improved aesthetics from below. However, this space is not normally cleaned, so could contain significant amounts of dirt and dust that may accumulate on the luminaire, affecting its thermal resistance.

7.2.1 Test location
The location was chosen with the objective of being as representative of a typical example as possible. It was not in the vicinity of any ventilation, power or other building systems. Neither was it sited directly above any windows, doors or high traffic walkways. For this study, the ceiling void above a store room was used. The ceiling void was not disturbed between measurements of the luminaire’s properties. The ceiling had been installed and free from major disturbances for approximately 5 years prior to the test.

7.2.2 Sample preparation
The luminaire was not fitted through the ceiling barrier for analysis as this was believed to play no role in its interaction with the environment. Instead, the entire luminaire was placed in the ceiling void (Fig. 7-1). Thermocouples remained bonded to the luminaire throughout the study. It was assumed their thermal contact properties did not alter. Placement and retrieval of the luminaire was done with great care to minimise disturbance of the surrounding environment.
7.2.3 Results

The thermal resistance through the heatsink to the surrounding test environment (between $T_{\text{high}}$ and $T_{\text{amb}}$) was calculated after each test. The heatsink’s thermal resistance was initially measured to be 2.214 K.W\(^{-1}\). This value was slightly lower than the equivalent luminaire tested in ‘5.3 Evaluation of luminaire thermal management performance’ but the discrepancy was well within the bounds of thermocouple measurement uncertainty. The change in magnitude of thermal resistance, with reference to the initial value, is plotted in Fig. 7-2. Error bars to represent the cumulative uncertainty of both ambient environment and upper heatsink reference point ($T_{\text{high}}$) temperatures, divided by the thermal power dissipated by LED component as stated by the manufacturer (12.6 W), have been included for reference. The overall trend of the results suggests thermal resistance increased with the duration of exposure. However, the measured changes in thermal resistance were still extremely small relative to the total magnitude. These changes were well within the bounds of measurement uncertainty and so no conclusive trend can be identified.
After 18 months of exposure, the luminaire displayed very few changes. There was no observed change in its appearance. There was a very fine layer of dust present on the sample’s surfaces, which appeared to accumulate as the test progressed. However, the quantity was not measured so this outcome could not be evaluated.

7.2.4 Discussion

The initial temperatures of the sample were consistent with the equivalent luminaire used to demonstrate the test methods (‘5.3 Evaluation of luminaire thermal management performance’). This validates the assumption that removing several components of the system had no significant effect on its thermal management performance.

The results of the analysis suggest exposure to a ceiling void environment caused a gradual increase in the system’s combined thermal resistance. From observation, it appeared a fine layer of dust did accumulate on the surface of the luminaire, which could account for this change. However, the changes in thermal behaviour were small and well within the bounds of measurement uncertainty so it was not possible to state definitively the impact of subjecting the luminaire to this environment.

![Graph](image.png)

**Fig. 7-2: Evolution of system thermal resistance after being subjected to a ceiling void environment**
If the results are accepted as accurate, there appeared to be a correlation between thermal resistance and duration of exposure. It is believed accumulation of dust on the luminaire’s surfaces was responsible for the change in thermal resistance, as no other changes were observed. There also appeared to be no mechanism to limiting the accumulation of dust so longer exposure could be expected to have a greater impact on the system’s thermal resistance. Judging from the gradient of the trend line, the change in system thermal resistance after 18 months would equate to a 0.88 K increase in the LED junction temperature. Making a linear extrapolation of this trend to a typical LED luminaire’s design lifetime predicts a significant and potentially critical change in LED junction temperature.

7.3 Exterior environment

An alternative environment commonly encountered by luminaires is exterior. The nature of this environment varies greatly depending on geographical location and specific installation conditions. In the UK an exterior environment is generally mild and not representative of extreme conditions that can occur in other markets. However, it does provide a wide variety of conditions for assessment. Conditions can range from warm to freezing and wet to dry. Direct sunlight is also common. These were expected to impose a range of damaging effects on a luminaire. As with the ceiling void, being subjected to this environment was expected to affect its properties.

7.3.1 Test location

The test location was in the West Midlands region of the UK. The position was partially shielded by a wall 2 m away on the west side. The location was not disturbed during the study other than to retrieve the sample for testing. The location was reasonably typical of an exterior environment. To ensure the sample was not erroneously affected by standing water, it was placed on a raised bed of gravel.

7.3.2 Sample preparation

The thermocouples used to evaluate the luminaire’s behaviour were not expected to be able to tolerate long term exposure to the test environment. Consequently, they had to be re-applied for each set of measurements and removed before returning the luminaire to the test environment.
Because the use of adhesive would have prevented this the thermocouples were instead attached using a small piece (5 x 5 mm) of removable adhesive tape. The thermocouple contact with the luminaire was enhanced using a small amount of thermally conductive grease$^1$ in place of adhesive. Care was taken to avoid handling the heatsink while applying the thermocouples. The measurements were used to monitor the evolution of the sample’s thermal resistance rather than compare its properties to another. Any inconsistencies introduced by this modified thermocouple attachment were, therefore, considered to be irrelevant. The thermocouples were attached using a consistent process throughout the test and so conclusions drawn about the changes occurring were assumed to be valid.

### 7.3.3 Results

After 18 months of exposure, the luminaire showed some minor changes in its surface appearance (see Fig. 7-3 in comparison to Fig. 5-3). The mechanical fixings had corroded, but the integrity of the assembly did not seem to be affected. Some dirt had been deposited on its surfaces after 6 months but this did not appear to increase further over the remainder of the test. It is believed that the action of wind and rain was also responsible for removing dirt from the luminaire surfaces, thereby limiting its accumulation. However, the quantity of any contamination was not measured, so its evolution could not be assessed. The reflectivity of the heatsink surfaces also appeared to reduce following exposure to the environment, but again this was not measured.

As with the indoor study (‘7.2 Interior ceiling void’), the total thermal resistance through the heatsink to the surrounding environment was calculated to be 2.230 K.W$^{-1}$. Once again this value was slightly lower than the equivalent luminaire tested in ‘5.3 Evaluation of luminaire thermal management performance’ but the discrepancy was well within the bounds of thermocouple measurement uncertainty. The change in magnitude of thermal resistance, with reference to the initial value, is plotted in Fig. 7-4. Again, error bars to represent the cumulative uncertainty of both ambient environment and upper heatsink reference point ($T_{high}$) temperatures, divided by the thermal power dissipated by LED component as stated by the manufacturer (12.6 W), have been included for reference.

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$^1$ Arctic Cooling ARCTIC MX-4.
The thermal resistance through the heatsink to the surrounding environment appeared to fluctuate around the initial value. It initially demonstrated an increase before steadily falling below its original value. This suggests the environment has two counteracting effects. A possible
explanation for this would be the thermal resistance initially increased rapidly in response to dirt accumulating on the heatsink’s surfaces and hindering heat transfer to the environment. The reflectivity of the surfaces also reduced throughout the test and so radiation emissivity can be assumed to have increased, consequently enhancing heat transfer and reducing total thermal resistance. The counteracting effects result in very little overall change in total thermal resistance after 18 months of exposure to the test environment. It is also possible changing weather conditions introduced inconsistencies into the results. If the build-up of dirt and debris on the surface of the heatsink was indeed responsible for the observed change in thermal resistance, and action of wind and rain helped limit its accumulation, then the elapsed time between measuring the heatsink’s properties and any significant wind and rain is also likely to be a factor in the resulting behaviour. As this was not controlled (by imposing a minimum period of dry, calm weather before recording any measurements, for example), weather conditions may have had a varying influence on the luminaire’s thermal behaviour. Further analysis is necessary to understand if the linear trend plotted in Fig. 7-4 is truly representative of the environments long term impact. However, all these changes were well within the bounds of measurement uncertainty so once again no effect can be conclusively identified.

7.3.4 Discussion

Despite the sample luminaire not being intended for external use, the heatsink proved to be resilient enough to survive the conditions of the test. However, the test methodology had to be adapted for this analysis. This potentially introduced inconsistencies that would need to be thoroughly evaluated in order to reliably compare the results to the indoor study (‘7.2 Interior ceiling void’).

The trend in the results suggest the environment had very little overall impact on the luminaire within the timeframe of this test. However, the linear trend applied in this analysis may not be appropriate. There may also be longer term impacts that were not captured within the timeframe of this test. As with the previous study, the margin for uncertainty and small changes occurring means it was not possible to conclusively determine what impact the environment had.
7.4 Evaluation of findings

The results from this investigation are in line with those established in the literature (for example, Skuriat et al., 2013), i.e. there are indications that the thermal resistance of the system alters during service. From the general trend in the data, it was estimated that after 18 months of the luminaire’s exposure to an interior ceiling environment the LED junction temperature would have increased by 0.88 K. The counteracting changes in thermal resistances hint that the effects of an exterior environment are actually quite significant, and that the response of the system is complex. However, ultimately there was no significant result from this investigation. To draw meaningful conclusions on the long-term impact with confidence would require further study. It is safest to presume exposure to the environment results in some increase in the system’s thermal resistance, which should be anticipated during development. However, both environments demonstrated very little overall impact, and these could reasonably be safeguarded against by incorporating a small safety margin in the system’s thermal management capability. Based on the observed trends, allowing for a 1 K increase in LED junction temperature for each year of service demanded from the luminaire appeared to be sufficient, but further investigation would be necessary to achieve a reasonable degree of confidence in this value.

Assessing a wider array of conditions and components would be a worthwhile expansion of the findings of this analysis. The behaviour of these samples appeared to undergo some change, so alternative materials and surface treatments are expected to show varying sensitivity to the operating environment, which should also be evaluated. Extending the duration of the study, or employing a suitable accelerated test method, would help to ascertain whether a linear extrapolation of the observed behaviour is appropriate and if the trends are accurately described. Using more precise methods and repeating measurements to reduce the uncertainty in the results would also be valuable. The investigation measured the resulting change in component temperatures but did not measure the changes in physical properties as a result of subjecting the sample to each environment. Therefore, the apparent changes in thermal resistance may have several causes which can only be hypothesised about here. Capturing more data from the samples to overcome this would be useful but extremely time consuming. Unfortunately, nothing more was possible within the scope of this research.

Aligning the recommended analysis improvements with commercial practice would be extremely challenging. The resources to overcome the technical limitations are unlikely to be available and commercial pressures demand rapid results which cannot be achieved when long periods of
testing are required. Deriving a method to predict the potential change in thermal resistance for a particular system design and without the need for testing would be extremely valuable, and overcome many of the commercial constraints. Changes could then be anticipated during the product design phase to ensure the system remains within a tolerable specification throughout the design life. To achieve this would require further study of the environment’s impact, how it relates to the system design, and how it can be translated to alternative cases. However, this could not be pursued here.

This chapter provided some quantitative evaluation of the initial changes in a luminaire’s thermal behaviour after being subjected to two common operating environments. It suggested some change in the thermal behaviour of a luminaire does occur during service, and this could have potential significance over the course of a typical LED luminaire’s design life. It is therefore advisable to make some allowance for the evolution of these properties as part of the design of a luminaire’s thermal management in order to ensure it remains effective and sufficient throughout the anticipated design life. However, the results were ultimately too insignificant to confidently draw any conclusions. Further investigation is required to definitively state what changes, if any, occur.
Chapter 8:
Analysis of heatsink concepts

This thesis has focused so far on characterising the thermal properties of a luminaire and the constraints imposed on it during service. This provides the means to evaluate thermal management performance and to ensure it meets operating constraints. The focus now moves onto exploring the potential to enhance thermal management performance within relevant constraints and with reference to an application (LED based general lighting and equivalent systems). The previous work has highlighted that passive heatsinks are particularly well suited to this task, but also indicated that their heat transfer into the surrounding environment tends to impose the limit on thermal management performance. This chapter describes how CFD modelling was used to first evaluate and then compare various heatsink geometry concepts to determine how thermal management performance can be maximised within given size constraints. The impact of augmenting the best performing heatsink with a chimney was then assessed. A range of chimney geometries, again within given size constraints, were studied to understand their influence on heatsink thermal management performance.

8.1 Evaluation of heatsink geometry

The literature, technical and market reviews presented in Chapters 2, 3 and 4 identified a number of common heatsink forms but no evidence to indicate they provide optimum thermal management. It is also unclear what thermal management performance other possibilities may offer. An analysis of different heatsink forms was, therefore, undertaken to compare their relative thermal management performance and to identify features that could potentially improve the thermal management capabilities of heatsinks for LEDs within practical constraints. Manufacturability considerations were put aside to avoid restricting the proposed concepts to conventional forms. Proposals for potential forming methods have been offered to demonstrate the concepts considered here are realisable. However, the focus of this analysis was to evaluate the thermal management performance of different heatsink forms on the basis the associated manufacturing and commercial challenges could be addressed should worthwhile benefits be discovered.
8.1.1 Concept geometry definition

A range of designs including both conventional and novel concepts to provide an extended heatsink surface were selected as summarised in Table 8-1. Each heatsink body incorporated a 5 mm thick base plate. This minimised effects particular to localised heat sources ensuring the widest relevance of the study. All heatsink fins were 3 mm thick, although some small variation resulted from the way the parametric models were defined. This thickness was found to offer a reasonable compromise between minimum resistance to conductive heat transfer and material content. In each case one parameter (indicated in Table 8-1) was modified to vary the surface area available for heat transfer whilst also adjusting spacing for the flow of cooling fluid between fins. For any heatsink, a compromise between these parameters is known to exist, and has been thoroughly studied for conventional geometry (e.g. Kraus and Bar-Cohen, 1995). Translating these principles to the diverse selection of heatsink forms considered here was not feasible using methods from the reviewed literature, so multiple simulations were performed to identify the geometries offering superior thermal management performance.

Table 8-1: Summary of analysed heatsink forms (full details provided in appendix B)

<table>
<thead>
<tr>
<th>Case</th>
<th>Description</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Parallel plate fins (conventional geometry). Suitable for production by extrusion processes</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Radial plate fins (conventional geometry). Suitable for production by die-casting processes</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Spiral plate fins (conventional geometry). Suitable for production by die-casting processes</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Diagonal plate fins (conventional geometry). Suitable for production by die-casting processes</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
5. Staggered pin fins (conventional geometry). Suitable for production by die-casting processes

6. Staggered pin fins with open centre (conventional geometry). Suitable for production by die-casting processes

7. Stepped staggered pin fins (conventional geometry). Suitable for production by die-casting processes

8. Capped radial plate fins (novel geometry). Suitable for production by lost-wax casting processes

9. Mesh (novel geometry). Suitable for production by lost-wax casting processes

10. Vertical tube (novel geometry). Suitable for production by lost-wax casting processes

11. Helical plate fins (novel geometry). Suitable for production by lost-wax casting processes

For practicality the models considered were governed by some generic size constraints (see Fig. 8-1). The heatsink bounds were restricted to 60 x 65 x 65 mm (width, breadth and height). The
width and breadth were defined by the extents of a commercially available LED array\textsuperscript{1}. This was chosen so that any subsequent experimental analysis could employ a readily available component. Heatsink height was an arbitrary value in keeping with the base dimensions of the component, an approach used in the established literature (e.g. Bar-Cohen et al., 2006). The LED array was modelled as being as attached to the base of the heatsink in the same way as for ‘6.4 Case study 4: Multiple heat sources on a circuit board’.

![Heatsink geometry constraints shown using exploded view of Case 1 for reference](image)

Simulation parameters were defined in accordance with Chapter 6. The interface between the heatsink and heat source module was assigned a thermal impedance of $154.8 \times 10^{-6}$ K.m$^2$.W$^{-1}$, as per the chosen thermal interface material’s specified properties (HALA Contec, 2014). The heatsink body was assigned the material properties of 6000 series aluminium alloy (MATWEB, n.d. a), considered to be a suitable candidate for its production. The heatsink surface was assigned an emissivity of 0.1 to represent a reflective finish. The low emissivity value represents an approximate “machined” finish on the basis that this would be the simplest to reproduce for

\textsuperscript{1} Vossloh Schwabe WU-M-444/B-NW 553939.
future experimental validation. As shown in ‘6.3 Case study 3: Reflective, extruded aluminium and copper heatsink’, assigning a low emissivity also appears to restrict radiative heat transfer thereby minimising a potential source of simulation error as well as emphasising the relationship between heatsink geometry and convective heat transfer. The heatsink was evaluated with the finned surfaces in a vertical and inverted orientation (with acceleration due to gravity, \( g \), aligned as shown in Fig. 8-1) to assess the different fluid flow behaviour developed. The heat sources and LED module were modelled as per the simplified “no-layers” definition employed in ‘6.4 Case study 4: Multiple heat sources on a circuit board’. For simplicity a thermal load of 1.5 W was applied at each LED component (i.e. 6 W in total for the whole LED array).

8.1.2 Case 1: Parallel plate heatsink results

This basic form was analysed to provide a baseline for comparison with subsequent designs. Within the specified geometric constraints the inter-fin spacing was varied to create heatsinks with 3, 5, 7, 9 and 11 evenly spaced fins (see appendix B, Table B-1 for details). The thermal resistance was calculated along with the corresponding average heat transfer coefficient as per Equation 5-2 and Equation 5-4 respectively (see ‘5.3.3 Thermal management performance parameters’). These results were plotted against the wetted (exposed) surface area of the heatsink available for heat transfer offered by each variation of the form simulated, and in each orientation (Fig. 8-2, data points, from left to right, correspond to the 3, 5, 7, 9 & 11 fin heatsink models in that order). The simulation model that offered the lowest thermal resistance is displayed in Fig. 8-3.

The average heat transfer coefficient for each heatsink variation was calculated. The maximum value was achieved when the number of fins was smallest (and therefore spacing was greatest). The trend indicates that this reduction in average heat transfer coefficient continues beyond the range studied here. The optimum thermal resistance was achieved by the 7 finned heatsink with 95.10 x10^{-6} m^3 of material and a surface area of approximately 0.058 m^2. This demonstrates the trade-off between maximising heatsink surface for heat transfer and open space for the unhindered flow of cooling fluid and maximum average heat transfer coefficient. The optimum thermal resistance of 3.35 K.W^{-1} was achieved at an average heat transfer coefficient of 5.14 W.m^{-2}.K^{-1}. There appears to be some benefit to orienting the fins vertically unless using very few, widely spaced fins. Fig. 8-3 shows fluid entering the inter-fin spaces from the open sides before rising vertically. The flow profile was very smooth and peak velocity was small, demonstrating turbulence parameters are of little relevance. The peak fluid flow velocity of 0.456 m.s^{-1} was
achieved about 0.7 m above the centre of the top of the heatsink. The geometry obstructs cooling fluid entering from the sides, which is expected to have a more limiting effect on thermal management performance if the fin length was greater. This is highlighted in Fig. 8-4, which shows a cross-section through the heatsink taken 0.1 mm above the base of the fins (note the plot has been mirrored to recreate the entire heatsink). It can be seen that the heat flux passing through the fins was greatest in the vicinity of the LED package and the outer fins. Heat flux in centre of heatsink was approximately 50 % of the maximum value and the lowest heat flux was approximately 30 %, revealing the significant variation in the performance of each fin as a result of geometric constraints. The thermal profile of the heatsink displayed some graduation, but this was small compared to the overall component temperature rise above the ambient environment. This indicates conductive transfer imposes a relatively minor restriction on thermal dissipation through the heatsink. Following the procedure described in ‘6.1.2 Analysis results’, radiative heat transfer’s fractional contribution to total power dissipated by the heatsink was calculated to be approximately 11 %. This reveals radiative heat transfer was significant but convective heat transfer dominates thermal management performance under the conditions analysed here. These findings were in line with expectations.

Fig. 8-2: Predicted thermal behaviour of parallel plate heatsink
Fig. 8-3: Predicted temperature distribution and fluid flow profile for the vertically mounted, parallel plate heatsink offering the lowest thermal resistance.

Fig. 8-4: Predicted heat flux passing through a plane 0.1 mm above base each fin for the vertically mounted, parallel plate heatsink offering the lowest thermal resistance.
8.1.3 Case 2: Radial plate heatsink results

Removing the obstruction to airflow entering from the sides (one of the limitations of Case 1) can be achieved by orienting the fins towards the heatsink centre in a radial arrangement. Several variations of this concept were simulated by varying the quantity of fins between 8 and 20 (see appendix B, Table B-2 for details). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-5.

![Predicted temperature distribution and fluid flow profile for the vertically mounted, radial plate heatsink offering the lowest thermal resistance](image)

As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated. The results were found to follow a similar relationship to the previous case (see appendix B, Fig. B-1). The average heat transfer coefficient improved with increased fin spacing, but the optimum thermal resistance was achieved by a compromise between spacing and surface area. The fin configuration generally provides less surface area than the parallel plate design and consequently the part’s thermal resistance was higher, but because fluid flow was less obstructed the average heat transfer coefficient for a corresponding thermal resistance was greater. It is believed the negative consequences of increasing fin length would be less significant with this arrangement, but the reduced fin spacing towards the heatsink centre imposes a new obstruction to airflow. The optimum thermal
resistance of 3.44 K.W⁻¹ was achieved at an average heat transfer coefficient of 5.74 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.051 m² and a material volume of approximately 79.86 x10⁻⁶ m³.

8.1.4 Case 3: Spiral plate heatsink results

A slight evolution of the radial plate heatsink form is to employ a curved fin. This provides similar benefits to the radial heatsink fin design (Case 2) but creates a slightly greater surface area for a given number of fins. The quantity of fins were varied from 8 to 20 to evaluate several variations of this concept (see appendix B, Table B-3 for details), the results of which were comparable to the radial plate heatsink (see appendix B, Fig. B-3). Although this spiral fin design can offer a slight increase in surface area compared to the radial plate design (up to 7.4 % for given number of fins), its thermal management performance benefit was marginal. It also suffered from the same restrictions as the radial plate heatsink. The optimum thermal resistance of 3.40 K.W⁻¹ was achieved at an average heat transfer coefficient of 5.50 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.053 m² and a material volume of approximately 83.37 x10⁻⁶ m³.

8.1.5 Case 4: Diagonal plate heatsink results

One observation made about the radial plate heatsink (Case 2) was that the inter-fin channel narrows towards the heatsink’s centre, restricting airflow. Aligning the fins in a parallel fashion, but still oriented towards the heatsink centre, was expected to provide consistent and optimised spacing for the flow of cooling fluid. Variations of this concept employing 13 to 29 fins and different inter-fin spacing were simulated (see appendix B, Table B-4 for details). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-6.

As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated. The results were found to follow a similar relationship to Case 1 (see appendix B, Fig. B-4). However, in this case the geometry overcame many of the limitations of the previous cases to achieve greater thermal management performance. The optimum thermal resistance of 3.21 K.W⁻¹ was achieved at an average heat transfer coefficient of 6.25 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.050 m² and a material volume of approximately 77.37 x10⁻⁶ m³. It should be
noted that regardless of fin quantity and spacing, there was always a fin positioned directly behind the heat source (as illustrated by Fig. 8-7). This would enhance conductive heat transfer away from its source, reducing heatsink thermal resistance and increasing average heat transfer coefficient. Situations with differently positioned heat sources may not benefit from this same enhancement and so thermal management performance would be lower. It was not tested but this enhancement is believed to be small considering the part’s relatively high thermal conductivity and low rates of passive heat transfer to the surrounding environment.
8.1.6 Case 5: Staggered pin heatsink results

Pin fins in a staggered hexagonal grid array have been seen in commercial practice (Cooliance, n.d.) and promise high thermal management performance. Within the geometric constraints of the study, pin spacing was varied to create heatsinks with 27 to 163 evenly spaced pins for simulation (see appendix B, Table B-5). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-8. As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated but in this case the results showed a different pattern, as plotted in Fig. 8-9.

In this case the thermal resistance appeared to plateau as pin fin spacing decreased (Fig. 8-9). It is believed that this can be attributed to the definition of the heatsink’s geometry. The design permits a degree of flexibility regarding pin fin spacing without a corresponding effect on heatsink surface area (demonstrated by Fig. 8-10).
Fig. 8-8: Predicted temperature distribution and fluid flow profile for the vertically mounted, staggered pin heatsink offering the lowest thermal resistance

Fig. 8-9: Predicted thermal behaviour of staggered pin heatsink
Because the surface area does not correlate so strongly with the fin spacing, it allows the corresponding thermal performance trend to be distorted. However, it may be the case that the analysed models do not reveal the full extent of any trend and that the optimum thermal performance lies outside the studied range of parameters. This is considered unlikely, but would require further analysis to confirm. The lowest thermal resistance of the models studied here was 3.53 K.W\(^{-1}\) at an average heat transfer coefficient of 5.83 W.m\(^{-2}\).K\(^{-1}\). The corresponding heatsink had an exposed surface area of approximately 0.049 m\(^2\) and a material volume of approximately 52.16 x10\(^{-6}\) m\(^3\).

The central region of the heatsink was all found to be a similar temperature (Fig. 8-8). A plot of the temperature profile distribution over a cross-section through the model (Fig. 8-11) reveals that thermal equilibrium had been established between the inner heatsink pin fins and fluid. With little to no thermal gradient between the heatsink and fluid, there would be no heat transfer between the two. Fig. 8-12 demonstrates this point very clearly. There were some extremely high heat fluxes passing through many of the pin fins, but the heat flux through the central pins was only 2.5 \% of the peak value. Therefore, as the heatsink pin fins in this region were almost completely redundant, removing them has the potential to improve the heatsink’s effectiveness (represented by the average heat transfer coefficient), to reduce unnecessary material content and to minimise cost. While convention dictates that more surface area should be advantageous to thermal management performance, much of this is based on conditions where the cooling fluid can be driven to maintain a thermal gradient across the entire heatsink surface area (for example, actively cooled by an electromagnetic fan). For the conditions considered here the relatively low
temperatures establish small buoyancy forces to drive fluid convection and consequently provide insufficient airflow to maintain a thermal gradient across all of the heatsink’s surface area. In designing an effective heatsink for these conditions it is, therefore, necessary to manage fluid flow as well as avoid adding heat transfer surfaces where they are ineffectively cooled. The staggered pin heatsink model with the lowest thermal resistance identified here was adapted to explore this concept in Case 6 (‘8.1.7 Case 6: Staggered pin with open centre heatsink results’).

![Fig. 8-11: Predicted thermal profile through staggered pin heatsink cross-section](image_url)
8.1.7 Case 6: Staggered pin with open centre heatsink results

The previous case study found that the pins in the centre of the heatsink were relatively redundant. The staggered pin fin heatsink geometry from Case 5 that achieved the lowest thermal resistance was used to evaluate the potential thermal management performance enhancement available if these pins are removed. Several variations of this concept, with between 32 and 71 pins, were simulated (see appendix B, Table B-6). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-13. As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated, as plotted in Fig. 8-14.
These results were noticeably different to those for the previous case studies. The optimum thermal resistance was achieved by different geometric configurations depending on the model.
orientation. The orientation also had a much larger effect on thermal resistance. The stagnant fluid region amongst the pins shown in Fig. 8-15 suggests that the open centre of the heatsink fins captures rising heated air when in an inverted orientation. This results in greater thermal resistance and is the most probable explanation for the observed differences. Removing redundant pins from the centre of the heatsink offered a significant improvement in average heat transfer coefficient whilst reducing material content by 18.7% compared to the optimum staggered pin fin heatsink (Case 5) used as the basis of this analysis. The optimum thermal resistance of 3.24 K.W⁻¹ was achieved at an average heat transfer coefficient of 8.65 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.036 m² and a material volume of approximately 42.40 x10⁻⁶ m³.

8.1.8 Case 7: Stepped staggered pin heatsink results
The staggered pin heatsink (Case 5) simulations revealed the coolest regions occurred at the top of the outermost fins. To understand the effect of removing these regions another variation of the staggered pin heatsink model offering the lowest thermal resistance in Case 5 was simulated. Several versions of this concept were modelled by changing the step down angle between 55°
and 15° (see appendix B, Table B-7 for details). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-16. As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated, as plotted in Fig. 8-17.

The lowest thermal resistance of 3.66 K.W⁻¹ was achieved at an average heat transfer coefficient of 6.03 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.045 m² and a material volume of approximately 49.59 x10⁻⁶ m³. With regard to the optimum staggered pin heatsink (Case 5) used as the basis of this analysis, the smaller surface area results in a 3.7 % poorer thermal resistance. However, the reduced surface area was realised by removing less effective regions, providing a 3.4 % improvement in average heat transfer coefficient whilst reducing material content by 5 %.

![Fig. 8-16: Predicted temperature distribution and fluid flow profile for the vertically mounted, stepped staggered pin heatsink offering the lowest thermal resistance](image-url)
8.1.9 Case 8: Capped radial plate heatsink results

There is some value in exploring how the cooling fluid can be constrained to direct it more effectively across the heatsink surfaces and, therefore, enhance heat transfer. To assess the feasibility of this, the radial plate heatsink offering the lowest thermal resistance in Case 2 was modified with a simple cap to constrain fluid flow. In this case the cap was modelled as an extension of the heatsink body and so also adds to its surface area. An aperture was added to the centre of the cap feature to allow rising air to exit after passing across the heatsink fins. The aperture size was varied between 55 mm and 15 mm in diameter whilst all other heatsink dimensions were fixed (see appendix B, Table B-8). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-18, with fluid flow around the heatsink cap detailed in Fig. 8-19. As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated, as plotted in Fig. 8-20.
Fig. 8-18: Predicted temperature distribution and fluid flow profile for the vertically mounted, capped radial plate heatsink offering the lowest thermal resistance

Fig. 8-19: Predicted fluid flow around cap feature
As Fig. 8-19 shows, the cap did obstruct fluid flow, forcing the cooling fluid to flow around it and so help direct it towards the hotter regions of the heatsink surface (highlighted in the red circle). However, the graph (Fig. 8-20) shows that the constraining cap feature, with any size aperture, only hinders heat transfer. As surface area increased (i.e. hole diameter decreased), thermal resistance increased and average heat transfer coefficient decreased. The lowest thermal resistance achieved by the heatsink configurations studied in this case was 3.39 K.W⁻¹ at an average heat transfer coefficient of 5.52 W.m⁻².K⁻¹. The corresponding heatsink had an exposed surface area of approximately 0.053 m² and a material volume of approximately 83.46 x10⁻⁶ m³. This represents a 1.5 % improvement in thermal resistance but also a 3.8 % reduction in average heat transfer coefficient and approximately 4.9 % more material compared to the optimum radial plate heatsink with no cap feature (Case 2) used as the basis of this analysis. Note the results indicate the heatsink offering the optimum thermal resistance lies outside the range of geometries studied here (with a larger diameter aperture). The additional surface area provided by the cap seemed to enhance the thermal resistance, but the negative effect on average heat transfer coefficient reveals it to be an ineffective utilisation of the additional surface area and material.
8.1.10 Case 9: Mesh heatsink results

A concept to maximise heatsink surface area and open space for fluid flow was developed around a mesh structure. Several variations of this concept were created for simulation by varying the width (and hence quantity) of interlinking channels running through the heatsink bounding region (see appendix B, Table B-9). Channel width was varied between 12.5 mm and 4.5 mm. The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-21.

As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated. The results were found to follow a similar relationship to Case 1 (see appendix B, Fig. B-9). The optimum thermal resistance of 3.44 K.W\(^{-1}\) was achieved at an average heat transfer coefficient of 5.65 W.m\(^{-2}\).K\(^{-1}\). The corresponding heatsink had an exposed surface area of approximately 0.051 m\(^2\) and a material volume of approximately 66.75 x10\(^{-6}\) m\(^3\). The optimum thermal resistance was higher than achieved by the optimum parallel plate heatsink design (Case 1). Fluid flow in the vicinity of the heatsink was also found to be at lower velocities than that achieved by the preceding concepts. It appears that the mesh
structure obstructs smooth, and consequently fast, fluid flow, which in turn hinders heat transfer. The thermal gradient through the heatsink was also relatively large, suggesting conductive transfer imposed a greater restriction on thermal management performance than the previous cases. These effects limit the concept’s potential to provide effective thermal management. The geometry of the heatsink is also likely to present a number of manufacturing challenges, further opposing its commercial adoption.

8.1.1.1 Case 10: Vertical tube heatsink results

To minimise obstruction to rising fluid flow, whilst still maximising surface area, a concept was devised around a series of vertical tubes. Several versions of this concept with between 4 and 36 tubes were simulated (see appendix B, Table B-1). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-22.

As with Case 1, the thermal resistance and average heat transfer coefficient with respect to the surface area of each heatsink was evaluated. The results were found to follow a similar
relationship to Case 1 (see appendix B, Fig. B-10). It can be seen from the fluid velocities in Fig. 8-22 that the channels had a noticeable constraining effect on fluid flow. The narrow aperture where air enters the vertical tube section established a region of fast moving cool air over the heatsink base. However, as with the mesh heatsink model, the overall reduction in flow velocities and poor conductive heat transfer through the heatsink body seem to limit its thermal management performance. The optimum thermal resistance of 3.70 K.W$^{-1}$ was achieved at an average heat transfer coefficient of 4.86 W.m$^{-2}$.K$^{-1}$. The corresponding heatsink had an exposed surface area of approximately 0.056 m$^2$ and a material volume of approximately 94.80 x10$^{-6}$ m$^3$. This concept seems more straightforward to manufacture than the mesh heatsink so could be worth further attention to determine if its limitations can be overcome.

The literature shows that for forced convection situations, micro-channels have the potential to offer extremely high heat transfer (Shao et al., 2007), but in this study the smallest channels offered the worst thermal management performance. This can probably be attributed to the different driving forces behind the fluid flow. Forced convection can overcome the large friction effects in small channels, whereas passive buoyancy-driven (natural) convection, as exploited here, generates a very small pressure gradient to drive fluid flow. This is relatively easily obstructed and so micro scale channels become ineffective in this situation. However, there are a number of other considerations such as operating power, temperature and heatsink orientation that would need to be explored before this conclusion can confidently be translated to all passively cooled heatsinks.

**8.1.12 Case 11: Helical plate heatsink results**

Curving radial plate fins into a spiral form was shown in Case 3 to offer a small thermal management performance benefit. The same principle can be applied along the height direction of the fin in a helical arrangement. By using the radial plate fin heatsink geometry from Case 2 that produced the optimum thermal resistance, but adding an angle of twist of between 40° and 108° to the fins upwards projection (helical sweep angle), several variations of this concept were modelled for simulation (see appendix B, Table B-11). Owing to the asymmetric nature of this concept, the full model was simulated (no symmetry conditions). The behaviour of the configuration that offered the lowest thermal resistance is displayed in Fig. 8-23. As with Case 1, the thermal resistance and average heat transfer coefficient with respect to heatsink surface area was evaluated, as plotted in Fig. 8-24.
The results plotted in Fig. 8-24, from left to right, correspond to increasing sweep angles of twist. The behaviour of this helical fin heatsink showed there was an optimum sweep angle of twist to minimise the heatsink’s thermal resistance but the benefit was extremely small (just 0.02 K.W\(^{-1}\))
between a vertically oriented heatsink with 40° of sweep angle and the optimum 56° sweep angle). Increasing the sweep angle only reduced average heat transfer coefficient. The optimum thermal resistance of 3.45 K.W⁻¹ was achieved at an average heat transfer coefficient of 5.43 W.m⁻².K⁻¹. For both criteria, the performance was lower than the radial plate heatsink (Case 2) on which it was based. It is interesting to note the baseline heatsink’s thermal resistance, essentially the helical plate heatsink with 0° of sweep angle, does not fit the trend demonstrated by the vertically oriented helical plate heatsinks. When the baseline model was re-run without symmetry conditions (i.e. with identical boundary conditions to the helical plate heatsink simulation) the same result was obtained so it appears to be accurate and is unlikely to be a consequence of simulation boundary condition errors. An explanation for the radial plate heatsink’s observed deviation could be that, without any helical sweep angle, radiation view factor from the surface of each fin is less obstructed. This allows greater heat transfer to the environment than the helical plate heatsink and so it does not follow the same trend. Therefore it could be concluded that even a small twist angle is detrimental to thermal management performance. The heatsink offering the lowest thermal resistance had an exposed surface area of approximately 0.053 m² and a material volume of approximately 79.42 x10⁻⁶ m³. Owing to the way the model was created, the fin thickness was below the nominal 3 mm used in the other models. This divergence from the nominal value increased with larger sweep angle. The geometry also means the length of the thermal path from the heatsink base to the top of the fin increased with greater sweep angle of twist. These features impact the effectiveness of the fins and would explain the observed reduction in thermal management performance. It would be relatively straightforward to compensate for this by modifying the fin geometry (e.g. increase fin thickness or reduce heatsink height). However, the results suggest this concept offers limited potential for enhancing heatsink thermal management. Manufacturing challenges would also oppose its adoption. There is, therefore, very little commercial incentive to pursue this concept further.

8.1.13 Comparison of results
The properties of the heatsink models offering the lowest thermal resistance are summarised in Table 8-2.
Table 8-2: Predicted optimum thermal resistance, associated average heat transfer coefficient and material volume of each heatsink concept

<table>
<thead>
<tr>
<th>Heatsink design</th>
<th>Isometric view</th>
<th>Thermal resistance ((K.W^{-1}))</th>
<th>Thermal resistance relative to Case 1</th>
<th>Average heat transfer coefficient ((W.m^2.K^{-1}))</th>
<th>Average heat transfer coefficient relative to Case 1</th>
<th>Heatsink material volume ((m^3, x10^{-6}))</th>
<th>Heatsink material volume relative to Case 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1:</td>
<td></td>
<td>3.35</td>
<td>-</td>
<td>5.14</td>
<td>-</td>
<td>95.10</td>
<td>-</td>
</tr>
<tr>
<td>Parallel plate fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 2:</td>
<td></td>
<td>3.44</td>
<td>+ 2.4 %</td>
<td>5.74</td>
<td>+ 11.8 %</td>
<td>79.86</td>
<td>- 16.0 %</td>
</tr>
<tr>
<td>Radial plate fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 3:</td>
<td></td>
<td>3.40</td>
<td>+ 1.4 %</td>
<td>5.50</td>
<td>+ 7.1 %</td>
<td>83.37</td>
<td>- 12.3 %</td>
</tr>
<tr>
<td>Spiral plate fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 4:</td>
<td></td>
<td>3.21</td>
<td>- 4.2 %</td>
<td>6.25</td>
<td>+ 21.6 %</td>
<td>77.37</td>
<td>- 18.6 %</td>
</tr>
<tr>
<td>Diagonal plate fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 5:</td>
<td></td>
<td>3.53</td>
<td>+ 5.1 %</td>
<td>5.83</td>
<td>+ 13.5 %</td>
<td>52.16</td>
<td>- 45.2 %</td>
</tr>
<tr>
<td>Staggered pin fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 6:</td>
<td></td>
<td>3.24</td>
<td>- 3.4 %</td>
<td>8.65</td>
<td>+ 68.5 %</td>
<td>42.40</td>
<td>- 55.4 %</td>
</tr>
<tr>
<td>Staggered pin fins with open centre</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 7:</td>
<td></td>
<td>3.66</td>
<td>+ 9.2 %</td>
<td>6.03</td>
<td>+ 17.4 %</td>
<td>49.59</td>
<td>- 47.9 %</td>
</tr>
<tr>
<td>Stepped staggered pin fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 8:</td>
<td></td>
<td>3.39</td>
<td>+ 1.0 %</td>
<td>5.52</td>
<td>+ 7.5 %</td>
<td>83.46</td>
<td>- 12.2 %</td>
</tr>
<tr>
<td>Capped radial plate fins</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>
These results were consistent with expectation. The lowest thermal resistances of the models evaluated here ranged between 3.21 K.W⁻¹ (Case 4) to 3.70 K.W⁻¹ (Case 10). Average heat transfer coefficient of the corresponding heatsinks spanned 8.65 W.m⁻².K⁻¹ (Case 6) to 4.86 W.m⁻².K⁻¹ (Case 10) and their material volume ranged from 42.40 x10⁻⁶ m³ (Case 6) to 95.10 x10⁻⁶ m³ (Case 1). A conventional parallel plate fin heatsink (Case 1) was neither the most effective nor offered the lowest thermal resistance yet had the largest material volume of all the heatsink designs tested. By employing a superior design, it was possible to reduce thermal resistance by 0.14 K.W⁻¹ (4.2 %), improve average heat transfer coefficient by 3.51 W.m⁻².K⁻¹ (69 %) or reduce heatsink material volume by 52.70 x10⁻³ m³ (55.4 %). Some key points to summarise are:

- There was no apparent correlation between thermal resistance and effectiveness in terms of average heat transfer coefficient or heatsink material volume. No criteria in isolation provides a full assessment of a heatsink’s thermal management performance.
- Of the models analysed, the basic parallel plate heatsink (Case 1) makes efficient use of the available heatsink bounding region to create a large surface area. This achieved one of the lowest thermal resistances (3.35 K.W⁻¹) but the fin alignment also constricts fluid flow and so resulted in one of the lowest average heat transfer coefficients (5.14 W.m⁻².K⁻¹).
- The radial plate heatsink (Case 2) imposed 0.09 K.W⁻¹ (2.7 %) higher thermal resistance but also a 0.6 W.m⁻².K⁻¹ (11.7 %) higher average heat transfer coefficient and required 15.24 x10⁻⁶ m³ (16.0 %) less material volume than the best performing parallel plate heatsink design (Case 1). This appeared to be because the arrangement of fins did not
obstruct airflow from any direction but provided an overall smaller surface area which restricts heat transfer.

- The spiral plate fin arrangement (Case 3) can provide up to 7.4 % more surface area for a given number of fins than the radial plate heatsink (Case 2). This resulted in 0.04 K.W⁻¹ (1.2 %) lower thermal resistance but poor utilisation of the additional surface area led to a 0.24 W.m⁻².K⁻¹ (4.2 %) decrease in average heat transfer coefficient and a 3.51 x10⁻⁶ m³ (4.4 %) increase in material volume.

- The diagonal plate heatsink (Case 4) offered the lowest thermal resistance. A number of factors appeared to contribute to this. The parallel fin arrangement maintained a consistent spacing for optimal fluid flow, the fin positions enhanced heat conduction away from the source, and the geometry did not obstruct fluid flow from any direction.

- Excluding pins from the centre of the staggered pin heatsink (Case 6) appeared to enhance fluid flow whilst removing ineffective heat transfer surfaces. Consequently, it provided one of the lowest thermal resistances (3.24 K.W⁻¹) along with the highest average heat transfer coefficient and lowest material volume.

- As demonstrated by the staggered pin (Case 5) and stepped staggered pin (Case 7) design, removing the coolest portions of the heatsink showed limited benefit. Between the two models there was just a 0.2 W.m⁻².K⁻¹ (3.4 %) improvement in average heat transfer coefficient and a 2.57 x10⁻⁶ m³ (4.9 %) reduction in material volume but an accompanying 0.13 K.W⁻¹ (3.7 %) increase in thermal resistance.

- Adding a horizontal cap to the top of the radial plate heatsink (Case 8) reduced thermal resistance by 0.05 K.W⁻¹ (1.5 %) but also reduced the average heat transfer coefficient by 0.22 W.m⁻².K⁻¹ (3.8 %) and increased heatsink material volume by 3.6 x10⁻⁶ m³ (4.9 %) compared to the basic radial plate (Case 2), demonstrating the additional surface area was poorly utilised. However, the results do show that the cap feature did help direct fluid between the heatsink fins. This effect could potentially be exploited to direct cooling fluid towards higher temperature regions and maximise heat transfer. The possibility of developing this effect, and minimising the negative impact on heatsink thermal management performance by utilising a separate component is considered in more detail in ‘8.2 Evaluation of chimney structure augmentation’ and Chapter 10.

- Restricted heat transfer through the structure and poor fluid flow hinders the thermal management performance of the mesh (Case 9) and vertical tube (Case 10) heatsinks. However, the fluid flow patterns through these heatsinks again demonstrate that it is
possible to direct flow through a complex structure (i.e. through internal channels towards specific sites) using passive, buoyancy-driven convection.

- Case 10 offered the worst thermal resistance and average heat transfer coefficient, and only a marginal reduction of heatsink material volume, so would be unattractive for further consideration.

- The average heat transfer coefficient of the helical plate heatsink (Case 11) was 0.31 W.m\(^{-2}\).K\(^{-1}\) (5.4 %,) less than for straight radial fins (Case 2). This appeared to be because the helical form obstructed fluid flow and radiative heat transfer. As a result of the way it was defined the helical fin geometry also suffered from reduced fin thickness. As a consequence, fin efficiency and thermal management performance was reduced.

In this study 11 heatsink designs, subjected to a narrow range of conditions, were analysed. A thorough and definitive comparison of thermal management performance would demand a more detailed assessment. For more complex models, incorporating multi-objective optimisation, the number of conditions to consider would quickly become impractical. In an effort to manage this, a number of constraints were applied. A broader exploration of the potential of various concepts without the arbitrary physical constraints imposed here would be valuable. It was believed that the results still provided an acceptable indication of thermal management performance and adequately demonstrated the ability to determine which concepts offered the most potential for development. Some of the more complex models also showed the potential to manipulate fluid flow. To exploit the full potential of these findings requires further development.

It is important to recognise that some features of the geometries studied gave rise to some inconsistent results. For example, in Case 5 the staggered pin fin heatsink surface area, and hence average heat transfer coefficient, did not directly relate to the modified pin spacing parameter and so the results showed a weaker correlation. In some models the relationship between fin position and heat source altered as the geometry was modified (e.g. Case 2) and so the thermal behaviour would have been affected. It was shown in Chapter 6 that the simulation results tended to be reasonably insensitive to small geometric discrepancies. It is believed, therefore, that the results offer a reasonable approximation of the true thermal management performance and the conclusions are accurate.
8.2 Evaluation of chimney structure augmentation

Managing downstream fluid flow to improve its interaction with the heatsink is an interesting concept to explore. The preceding simulations suggest integrating this facility with the heatsink compromised its thermal management performance, but augmenting the heatsink with a secondary component was not considered. The literature review highlighted the chimney effect as one method to enhance fluid flow (Fisher and Torrance, 1998) (Park et al., 2016) and so some studies were conducted to explore how this could be effectively integrated.

8.2.1 Concept geometry definition

For this study, the simulation model was adapted from the lowest thermal resistance staggered pin with open centre heatsink developed previously (‘8.1.7 Case 6: Staggered pin with open centre heatsink results’). A square profile tube section was defined above the heatsink to create the chimney body (see Fig. 8-25). To limit this assessment to the chimney structure’s impact on fluid flow, and therefore heat transfer, its physical properties were defined to minimise all other effects. For this reason it was assigned the material properties of an ideal insulator material (0 W.m\(^{-1}\).K\(^{-1}\)); it did not contact the heatsink body; and its surfaces were assigned non-radiating behaviour.

![Fig. 8-25: Model of heatsink with chimney structure](image)
The simulation software’s parametric modelling capabilities were employed to adjust the chimney length (x) and wall angle (α) during analysis. The effect of the chimney was assessed for lengths of 15 mm, 30 mm, 45 mm and 60 mm (an arbitrary selection to briefly evaluate a range of practical values in keeping with the heatsink’s dimensions). At each length the wall angle was modelled when at 0°, 45°, 90°, 135° and 180° (again, an arbitrary selection to cover a summary range of values). It was hypothesised that increasing cross-sectional area along its length may affect a change on the pressure of the column of air rising through it. Following the Bernoulli principle, and assuming incompressible fluid behaviour for simplicity, the highest fluid flow velocity would then be established at the narrow opening of the chimney in close proximity to the heatsink (Holman, 2010). Consequently, there was some reason to believe this may enhance heat transfer from the heatsink.

The simulation software’s parametric study tool was used to perform a full factorial evaluation of the two specified parameters. Considering multiple parameters creates large numbers of configurations which demand substantial analysis time and computational resources. To minimise this, a simplified simulation definition was employed. The computational domain was split into fewer, larger calculation cells (approximately 8 times the volume of the cell size verified as accurate in Chapter 6). This had a corresponding effect on the time taken to solve each simulation but effectively reduced its resolution. Re-running the baseline model at this definition calculated its peak heatsink temperature rise to be 0.22 K smaller, corresponding to a 0.04 K.W⁻¹ lower thermal resistance and a 0.10 W.m⁻².K⁻² higher average heat transfer coefficient. This is an error of approximately 3 %. Sacrificing this small amount of accuracy to accelerate simulation processing was considered a necessary compromise to allow a solution within an acceptable time. It was assumed that this sparse mesh introduced a generally consistent error, therefore the relative behaviour of each model can still be compared and overall trends in results identified. However, the results were not generated on an equivalent basis to the underlying heatsink concept (‘8.1.7 Case 6: Staggered pin with open centre heatsink results’) and so cannot reliably be directly compared.

8.2.2 Results of modifying chimney length

The results of the parametric study are plotted in Fig. 8-26. As discussed above, the properties of the non-augmented heatsink taken from ‘8.1.7 Case 6: Staggered pin with open centre heatsink results’
results’ are included for guidance only as they were obtained using a different simulation definition.

These results clearly showed that the heat transfer from the heatsink can be improved with the addition of a chimney structure. The greatest improvement, a 9.61% increase in average heat transfer coefficient compared to the reference model, was attained with the tallest chimney structure (i.e., 60 mm long, at a 0° wall angle). The literature suggests that increasing chimney height improves heatsink thermal management performance, but the beneficial effect gradually diminishes as height increases (Park et al., 2016). Within the constraints of this study, the impact on heatsink thermal management performance appeared to follow a linear relationship with chimney height, so it is reasonable to conclude that chimney height should be maximised. Increasing chimney height beyond the range considered here is potentially advantageous but would require further study to determine the limit of its effectiveness.

8.2.3 Results of modifying chimney wall angle
The results plotted in Fig. 8-26 indicate the vertical wall chimney provided the greatest enhancement. Contrary to the initial hypothesis there was no observable benefit from adding an
outward taper angle to the chimney walls. However, the interval between the assessed wall angles was too large to conclude reliably that there was no benefit at smaller angles. A second parametric study was performed to assess wall angle (α) between -45° to 45° in 5° increments for each chimney length (x). The results from this analysis are presented in Fig. 8-27.

![Graph](image-url)

**Fig. 8-27: Predicted average heat transfer coefficient achieved using different chimney wall angles**

From Fig. 8-27 it is apparent that the effect of chimney wall angle (α) on thermal behaviour was more pronounced the longer the chimney’s length (x). An optimum wall angle appears to exist, but contrary to expectation this occurred with a small inward taper of the chimney (approximately -5° to -10°). It can also be seen from the data that this improvement quickly disappeared as the wall angle decreased further.

The optimum chimney geometry identified by these simulations (when x = 60 mm and α = -5°) was assessed using the benchmarked mesh density defined in Chapter 6. This calculated the augmented heatsink achieved an average heat transfer coefficient 9.55 W.m⁻².K⁻¹, a 10.4 % increase over the performance of the heatsink alone.

The fluid flow profiles developed by the optimum, the - 45°, the + 45° and the vertical walled (0°) chimneys are shown in Fig. 8-28. The heatsink and surrounding fluid temperatures for each of
these configurations are plotted in Fig. 8-29. For comparison the behaviour of the non-augmented heatsink is presented in Fig. 8-30.
8.2.4 Discussion

Apart from the tallest, most inward tapering chimney, each configuration shown in Fig. 8-28 can be seen to create a barrier against entrained air joining the rising plume of heated air. This forced cool air to be drawn downwards, under the chimney and across the heatsink (indicated in Fig. 8-28). Compared to the non-augmented heatsink (Fig. 8-30), adding the chimney can produce a slight increase in fluid flow velocity in the vicinity of the heatsink (demonstrated by the appearance of a new contour region indicated in Fig. 8-28 and the region of peak flow velocity moving closer to the heatsink). As a result, the fluid temperature in the vicinity of the heatsink can be reduced (see bottom right image of Fig. 8-29 compared to right had image of Fig. 8-30) and thermal management performance improved. This appears to be a result of constraining the rising column of heated air. Consequently, some of the cool air displacing it, which drives fluid convection, is forced across the heatsink rather than entrained into the downstream flow. In comparison to the vertical (0°) walled chimney, the reduction in cross-sectional area imposed by the 5° inward taper appears to help further contain and direct the fluid flow across the heatsink. This resulted in an additional improvement of thermal management performance (0.2 W.m⁻².K⁻¹ increase in average heat transfer coefficient compared to the vertical walled configuration).

However, for long chimney lengths and large inward taper angles it is clear the reduced chimney cross-section becomes too much of an obstruction. Compared to the non-augmented heatsink shown in Fig. 8-30, fluid flow across the heatsink is clearly reduced (see top centre image of Fig. 8-28) and consequently temperatures are higher (as shown in top centre image of Fig. 8-29), resulting in a drop in performance. The heatsink's behaviour became nearly totally independent of chimney wall angle as the outward taper increased. It can be seen in the bottom centre image of Fig. 8-28 that fluid was actually flowing down into the top of the chimney to join the rising...
column of heated air. Consequently, less air was drawn across the heatsink and so the thermal management performance enhancement was reduced. There appears, therefore, to be little benefit to employing large outward tapering chimney wall angles. Incorporating a chimney clearly has an impact on heatsink thermal management performance and so, as per Chapter 10, is worth further consideration. However, fully understanding the complex interaction between heatsink and chimney, and how the chimney’s geometry impacts this, cannot be achieved within the scope of this research so instead must be recommended for a future investigation.

The principles of fluid mechanics show that turbulent fluid flow would be desirable for maximum heat transfer from the heatsink (Holman, 2010). However, the observed flow conditions give no indication that this can be achieved within the constraints of these designs. As described by Holman (2010), the Reynolds number (Re) in a tube can be calculated according to the equation:

\[
Re = \frac{\rho vD_h}{\mu}
\]

Equation 8-1

Where \( \rho \) is the fluid density (measured in kilograms per cubic metre), \( \mu \) is dynamic viscosity of the fluid (expressed in kilograms per metre second), \( v \) is fluid velocity (in metres per second) and \( D_h \) is hydraulic diameter of the tube, expressed in metres and calculated by the equation:

\[
D_h = \frac{4A}{W}
\]

Equation 8-2

With \( A \) being the tube’s cross-sectional area (given in square metres) and \( W \) being the wetted perimeter (measured in metres). If the chimney is treated as a tube, according to these equations, the properties of air (appendix C, Table C-1), and the velocity of the fluid passing through the chimney, Reynolds number can be estimated. Accordingly, Reynolds number would be less than 1000. Empirical data shows fluid flow through a tube typically becomes turbulent for Reynolds numbers greater than 2000 (Holman, 2010). Although there are a number of factors that mean this threshold cannot be considered definitive, the flow regime appears to be well within laminar bounds.
8.3 Conclusions

This chapter explored the thermal management potential of several concepts under conditions relevant to the research topic. The conventional heatsink simulation models (i.e. parallel plate and radial finned heatsinks) showed reasonable thermal management performance but also a number of limitations. Minimising the opposition to airflow passing across the heatsink and removing redundant features was shown to significantly enhance thermal management performance. Enclosing the heatsink demonstrated a direct effect on the fluid flow, but this was detrimental to heat transfer. It is unclear from the results whether this could be refined to provide a positive effect. Table 8-3 summarises the performance of the most promising concepts, including their improvement relative to the conventional parallel plate heatsink design offering the optimum thermal resistance. Heatsink material volume is included to indicate the potential commercial and environmental benefits provided. It is clear the concepts modelled here can offer considerable thermal management performance benefits over the conventional heatsink designs.

Table 8-3: Summary of simulation models predicted performance

<table>
<thead>
<tr>
<th>Simulation model</th>
<th>Isometric view</th>
<th>Thermal resistance (K.W⁻¹)</th>
<th>Thermal resistance relative to Case 1</th>
<th>Average heat transfer coefficient (W.m⁻².K⁻¹)</th>
<th>Average heat transfer coefficient relative to Case 1</th>
<th>Heatsink material volume (m³, x10⁶)</th>
<th>Heatsink material volume relative to Case 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td></td>
<td>3.35</td>
<td>-</td>
<td>5.14</td>
<td>-</td>
<td>95.10</td>
<td>-</td>
</tr>
<tr>
<td>Case 4</td>
<td></td>
<td>3.21</td>
<td>- 4.2 %</td>
<td>6.25</td>
<td>+ 21.6 %</td>
<td>77.37</td>
<td>- 18.6 %</td>
</tr>
<tr>
<td>Case 6</td>
<td></td>
<td>3.24</td>
<td>- 3.4 %</td>
<td>8.65</td>
<td>+ 68.5 %</td>
<td>42.40</td>
<td>- 55.4 %</td>
</tr>
<tr>
<td>Case 6 with most effective chimney augmentation</td>
<td></td>
<td>2.93</td>
<td>- 12.5 %</td>
<td>9.55*</td>
<td>+ 86.0 %</td>
<td>42.40*</td>
<td>- 55.4 %</td>
</tr>
</tbody>
</table>

*chimney body excluded from calculation
The pin fin heatsink concept, with no fins in the centre region, provided the highest average heat transfer coefficient and required the least heatsink material, while a parallel, diagonally-oriented plate fin heatsink offered the lowest thermal resistance. Average heat transfer coefficient was enhanced by incorporating a chimney structure above the system. The results suggest the height of the chimney should be maximised. The wall angle can also be optimised to provide an additional benefit, but the system’s thermal management performance can be very sensitive to changes so this would need to be carefully controlled.

The studied concepts represented a small selection of ideas, which underwent limited optimisation, and were evaluated under a narrow set of conditions. The limitations also created some uncertainty when extending these findings to alternative situations. The work presented in Chapter 6 suggests that the boundary conditions being used provide reasonably accurate results and the findings can confidently be extended to similar cases. It is believed that this is sufficient to guide development towards concepts with the greatest potential, but does not provide a definitive assessment of the maximum thermal management performance enhancement available. Practicality and limited resources prevented a more thorough analysis being conducted here, but there is an opportunity for further investigation.

This chapter employed CFD simulation to evaluate and compare the thermal management performance of various heatsink concepts. It revealed that different heatsink forms offer significantly different levels of performance. It was also shown that fluid flow can be manipulated to enhance heatsink thermal management performance. It demonstrates how simulation can be used to: identify systems that offer superior thermal management performance; evaluate the benefits of different configurations; and identify features that facilitate effective heat transfer. It also has implications for commercial product development. By employing these same techniques, LED luminaire design can be directed towards forms that offer more effective thermal management and so extract similar performance and material consumption improvements.
Chapter 9: System optimisation (Constrained)

The analyses presented in Chapter 8 were based on an extensively simplified model of a luminaire. This was useful for comparing the relative thermal management performance of different concepts but it is difficult to extend the conclusions to alternative operating conditions or to state definitively what benefit would be attained by the novel concepts in a practical situation. To enable this it would be necessary to evaluate their behaviour under a much wider range of conditions, following extensive optimisation to exploit their full potential. Unfortunately, such an exhaustive analysis would be impractical. This chapter, therefore, evaluates the concepts considered previously in the context of a typical commercial application and relevant constraints. This provides a meaningful basis for comparison, allowing findings to be extended to similar situations with greater confidence. It also allows the impact on performance to be quantified in terms of heatsink material volume, embodied energy and cost in addition to the established thermal management criteria.

The objective of this study was to demonstrate that the findings of Chapter 8 could be implemented to provide tangible thermal management performance benefits, yet still allow the system to operate within the same constraints. The chosen constraints were to maintain an equivalent heatsink maximum temperature (within a tolerance of 0.5 K) and fit within the same bounding region. For simplicity this study was restricted to heatsink geometries without an additional chimney structure. This was considered an independent augmentation and is evaluated separately in Chapter 10.

9.1 Simulation definition

The heatsink already simulated in ‘6.2 Case study 2: Black and reflective, die-cast heatsink’, was selected as the basis of this evaluation (see Fig. 9-1). This model included a series of features which are typical of a generic luminaire. It incorporates a chamber around the LED package, a common feature for accommodating some means of controlling the luminaire’s photometric behaviour (i.e. housing a reflector or lens). It is well suited to being simulated as a body attached to a surface (i.e. Fig. 9-2), recreating a similar configuration to a recessed luminaire protruding


into a ceiling void. Finally, it employs die-cast aluminium, which introduces material and manufacturing constraints consistent with commercial practice.

Fig. 9-1: Baseline model

Fig. 9-2: Cross-section view of baseline heatsink model attached to representation of a ceiling, as per simulation configuration
Simulation boundary conditions were defined as per ‘Case study 2: Black and reflective die-cast heatsink’. As with the previous simulation, the model was simplified by excluding unnecessary components and small features. Several bosses and artefacts of the manufacturing process that were present in the original case study model were also removed. These features were considered unnecessary as this study did not need to reproduce accurately the behaviour of an existing component. This study was based on the relative performance of different heatsink models. These simplifications were considered to be acceptable as they were applied consistently so the results can be used for the purposes of comparison and any unwanted influence on the system’s behaviour is eliminated.

The shaded faces shown in the lower left view of Fig. 9-1 were treated as fixed geometric constraints that define the heatsink’s bounding region. Along with a limit on the overall heatsink height, these represent the component’s dimensional constraints. Within these restrictions, and those of manufacturability such as incorporating no hollow chambers, the heatsink was free to be modified. Imposing these restrictions also helped limit the analysis’s complexity.

It was assumed that the established simulation parameters remained suitable for this model. However, there were some elements that needed to be modified or added in order to facilitate this analysis:

- Two mesh densities were employed for this study: sparse and fine. The sparse mesh definition, as discussed in ‘8.2.1 Concept geometry definition’, sacrifices accuracy to reduce simulation time by approximately 85%. Again, it was assumed this imposed similar limitations on the results, but that they were suitable for parametric optimisation studies. The fine mesh was defined through a series of preliminary analyses of the optimised heatsink model. The total cell count, and therefore mesh density, was increased in steps of approximately 25,000 until the resulting peak heatsink temperature changed by less than 0.1 K with a further increase in cell quantity. This fine mesh definition was used to quantify the thermal management performance of the optimised heatsink models.
- Software limitations prevent the allocation of radiative properties to new surfaces created during parametric driven optimisation studies. Therefore, during these studies, no radiative properties were assigned to the heatsink. Instead, the default radiative surface property (non-radiating) applied to the entire component. While removing a potentially significant heat transfer mechanism may compromise the accuracy of the results, out of necessity the effect was assumed to be consistent across all models. The various
geometries created would mean this simplification would have had a differing effect on each model. However, the constraints on the general form of the heatsink mean the visible surface presented to the surrounding environment, from which radiative heat transfer occurs, remained relatively consistent. Consequently, radiative heat transfer was believed to be relatively insensitive to heatsink geometry and so this was considered to be an acceptable simplification. Specific radiative surface properties were applied during the final evaluation studies. These were assigned on an equivalent basis to the benchmark case.

- The body representing a ceiling surface was assigned a thermal conductivity of 0.28 W.m\(^{-1}\).K\(^{-1}\) based on the properties of an appropriate material (wood lath and plaster, Holman, 2010). This poor conductivity minimised its impact on the thermal behaviour of the luminaire whilst avoiding excessive artificial temperature increases resulting from an idealised insulating material property.

- The default surface roughness was defined as 3.2 μm (as per the benchmarked die-cast surface property) while the ceiling surface (the only other surface within the computational domain) was specifically assigned a surface roughness of 1.6 μm (as per the benchmarked default wall roughness). Assigning surface roughness properties in this way ensured all heatsink surfaces were assigned the relevant property during the parametric optimisation study. Owing to the wide variety of possibilities, a typical surface roughness for the ceiling cannot be defined. Rather than revert to an idealised smooth surface that may have produced misleading or unexpected results, assigning the ceiling a surface roughness value of 1.6 μm was considered more appropriate. The significance of this condition was tested by re-running the baseline simulation with a surface roughness of 1000 μm assigned to the ceiling component. This had no impact on the heatsink’s peak temperature and so the ceiling’s surface roughness, therefore, can be considered inconsequential in this situation.

- The thermal contact resistance between the heatsink and ceiling surface was specified as 0.05 K.m\(^2\).W\(^{-1}\) based on a poor contact condition established under low compression (estimated from Fletcher, 1972). This allowed some limited heat transfer which was considered more accurate than the default perfect contact properties.

- The heatsink body was assigned the material properties of a die-cast aluminium alloy (92 W.m\(^{-1}\).K\(^{-1}\), based on Aluminium 384.0-F (MATWEB, n.d. b)), diverging from Chapter 8 which employed a higher conductivity material property more typical of extruded
material (‘8.1.1 Concept geometry definition’). For components characterised by a small Biot number (i.e. those employing highly conductive materials, large surface area to volume ratio, and passively cooled, such as the component considered here) interaction between the heatsink and environment tends to constrain heat transfer (Holman, 2010). As thermal conductivity is not a bottleneck the exact value of conductivity is not critical, but the use of a die-cast material property was considered more appropriate to represent likely commercial realisation of the concept.

These new parameters introduced multiple simultaneous changes to the verified boundary conditions, which obscure the effect of any individual change on the results. However, the effect of these changes was not of interest here. Chapter 6 indicated that the accuracy of the boundary conditions tends not to be too critical considering the requirements of this evaluation, so these changes were applied with ample justification and reasonable confidence. Regardless of this, the results still need to be treated with caution and validation against a physical sample would be a valuable exercise.

With the available resources it was not possible to conduct a comprehensive, concurrent optimisation of every heatsink parameter. This analysis explored select features of the heatsink geometry. The simulation software’s parametric optimisation study tool was used to evaluate these conditions individually across a range of values. Plots of heatsink properties’ sensitivity to changes in these parameters were then used to identify approximately which combination of values would provide the greatest thermal management performance. Statistical techniques (e.g. factorial design of experiments and response surface methods) were considered for analysing the interaction between these parameters and identifying the optimum performing heatsink designs. These are well established for this purpose (see Montgomery, 2001). However, the value in this research lies in defining an approach to thermal management design that can be readily adopted in commercial practice. Rather than employ complex statistical methods that require a high level of expertise, it was deemed to be more valuable to explore the potential thermal management performance that can be realised directly from simulation results following a one factor at a time approach. This was believed to be a better representation of typical commercial practice. Exploring how these results compare to a well-designed statistical approach, and defining simple tools to allow such an approach to be implemented in commercial practice, would be extremely valuable and a recommendation for further study.
9.2 Initial model analysis

Owing to the changes to the verified boundary conditions and simulation model, it was not appropriate to use the results from the original case study (‘6.2.2 Analysis results’) as a basis for comparison. Therefore, a preliminary analysis of the basic model (Fig. 9-1) was conducted. This was performed with both simplified and refined simulation definitions. The simplified definition allows the relative effects of the subsequent parametric optimisation studies to be evaluated. The refined definition provides a comparison for the accurate quantification of results. The simplified definition employed the sparse mesh arrangement and applied no specific radiative surface properties to the heatsink. The refined definition employed the fine mesh settings and specified heatsink radiative surface properties. The results of these preliminary analyses are summarised in Table 9-1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Simplified simulation definition</th>
<th>Refined simulation definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heatsink material volume (m$^3$, x10$^{-6}$)</td>
<td>83.82</td>
<td>83.82</td>
</tr>
<tr>
<td>Exposed surface area (m$^2$, x10$^{-3}$)*</td>
<td>42.43</td>
<td>42.43</td>
</tr>
<tr>
<td>Peak heatsink temperature relative to ambient environment (K)</td>
<td>+ 56.0</td>
<td>+ 40.7</td>
</tr>
<tr>
<td>Thermal resistance (K.W$^{-1}$)</td>
<td>4.44</td>
<td>3.23</td>
</tr>
<tr>
<td>Average heat transfer coefficient (W.m$^2$.K$^{-1}$)</td>
<td>5.30</td>
<td>7.29</td>
</tr>
</tbody>
</table>

*The total surface area of the heatsink body, excluding faces which do not contact the surrounding fluid or contact an enclosed internal fluid region. It was believed this best represents the surfaces responsible for transfer of heat to the surrounding environment and therefore the most appropriate basis for deriving the part’s average heat transfer coefficient.

Under the refined simulation definition, but without radiative heat transfer behaviour enabled, the peak heatsink temperature rise above that of the ambient environment was found to be 55.2 K (compared to 40.7 K when radiative heat transfer was enabled). Following the procedure described in ‘6.1.2 Analysis results’, radiative heat transfer’s fractional contribution to total power dissipated by the heatsink was calculated to be approximately 26 %. It can be seen then that radiative heat transfer is significant and accounts for the majority of the difference between
the results of the two simulation definitions. It also represents a much higher fractional contribution to total heat transfer than the models considered in Chapter 8 (approximately 11% for the parallel plate heatsink considered in ‘8.1.2 Case 1: Parallel plate heatsink results’). By comparison, this would amplify the effects of radiative heat transfer and suppress the impact of any changes to convective heat transfer. Obviously it would have been preferable to include the effects of radiative heat transfer in both simulation definitions. However, as already discussed this could not be applied consistently to all the heatsink models created during the parametric driven optimisation studies. It was, therefore deemed better to exclude it from the simplified simulation definition used for these studies on the assumption that its impact would be reasonably consistent and predictable.

9.3 Concept selection and definition of optimisation parameters

The baseline case study in ‘9.2 Initial model analysis’ was based on a simple parallel plate fin heatsink. Chapter 8 identified a number of concepts that have the potential to improve upon the thermal management performance of this configuration. Of these, Case 6 offered the greatest average heat transfer coefficient (‘8.1.7 Case 6: Staggered pin with open centre heatsink results’), while Case 4 had the lowest absolute thermal resistance (‘8.1.5 Case 4: Diagonal plate heatsink results’). The heatsink that provides a basis for this evaluation was formed by die-casting parallel plate fins. As seen in Chapter 4, this appears to be a common heatsink form and manufacturing method suggesting it offers some commercial advantages. The relevance of this research is reinforced by focusing on commercially feasible concepts. It was more appropriate, therefore, to restrict this study to concepts which share similar manufacturability characteristics. Case 4, the diagonal plate design, has many of the same features and so was selected for development. One of the features that appeared to enable Case 6 to perform so highly was the removal of heatsink material and surface area from regions that were ineffective at transferring heat to the surrounding environment. To explore if this strategy could offer similar benefits when applied to the diagonal plate fin heatsink arrangement similar features were incorporated. On this basis, the heatsink was redesigned with the form shown in Fig. 9-3 (note fixed dimensions have also been labelled).

A series of parametric optimisation studies were setup to modify the features labelled in Fig. 9-3 using the values provided in Table 9-2. Each parameter was modified in isolation while the other parameters defaulted to the values given in bold and enclosed in brackets. So the impact of
removing material from the central section of the heatsink could be evaluated, the parametric optimisation study also assessed the thermal management performance of the heatsink without the central cut-out (i.e. heatsink fins present and no material removed). This was modelled by suppressing the modelling operations that generated the central-cut-out geometry. This was effectively the same as a cut-out offset equal to fin height and so results from this configuration were plotted at the corresponding position in Fig. 9-6. Note that in this configuration the cut-out diameter and taper angle would be immaterial.

Fig. 9-3: Modified heatsink concept and optimisation parameters
Table 9.2: Parametric study parameters

<table>
<thead>
<tr>
<th>Heatsink parameter</th>
<th>Assessed value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>0.5, 1, 1.5, 2, (2.5), 3, 3.5, 4, 4.5, 5</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>1, 2, 3, 4, (5), 6, 7, 8, 9, 10</td>
</tr>
<tr>
<td>Central cut-out offset (mm)</td>
<td>0, 2.5, 5, (7.5), 10, 12.5, 15, Suppressed</td>
</tr>
<tr>
<td>Central cut-out diameter (mm)</td>
<td>5, 10, 15, (20), 25, 30, 35</td>
</tr>
<tr>
<td>Central cut-out taper angle (degrees)</td>
<td>0, 10, 20, (30), 40, 50, 60</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>1, 3, 5, 7, 9, 11, 13, 15, (17)</td>
</tr>
<tr>
<td>Base thickness (mm)</td>
<td>1, 2, 3, 4, (5), 6, 7, 8, 9, 10</td>
</tr>
</tbody>
</table>

9.4 Preliminary results

The results of each parametric study were used to produce a simple plot of the resulting heatsink properties (plotted in Fig. 9-4 to Fig. 9-10 and tabulated in appendix D, Table D-1 to Table D-7). The average heat transfer coefficient was derived from each heatsink’s exposed surface area and thermal resistance according to Equation 5-4, while the thermal resistance was calculated from the difference between the ambient environment and peak heatsink temperatures, divided by power of 12.6 W, as per Equation 5-2.

Fig. 9-4: Predicted heatsink thermal properties with varying fin thickness
Fig. 9-5: Predicted heatsink thermal properties with varying fin spacing

Fig. 9-6: Predicted heatsink thermal properties with varying central cut-out offset

Fig. 9-7: Predicted heatsink thermal properties with varying central cut-out diameter

Fig. 9-8: Predicted heatsink thermal properties with varying central cut-out taper angle
These plots reveal a number of points. The results of varying cut-out offset (Fig. 9-6), cut-out diameter (Fig. 9-7) and taper angle (Fig. 9-8) demonstrated that, for the studied conditions, these parameters had very little effect on the heatsink’s thermal resistance or average heat transfer coefficient. However, the dimension of the central cut-out appeared to have more influence on the heatsink’s average heat transfer coefficient than the thermal resistance. The fin spacing (Fig. 9-5) had a large impact on the part’s average heat transfer coefficient and thermal resistance, which could be exploited to offset increases arising from other changes. Modifying base thickness (Fig. 9-10) had a similar, but smaller effect on heatsink performance. Finally, minimising fin height (Fig. 9-9) has significant potential to maximise the heatsink’s average heat transfer coefficient. However, to exploit this to its full extent would be extremely detrimental to the LED junction temperature owing to the similarly large associated increase in heatsink thermal resistance. Maximising fin thickness (Fig. 9-4) demonstrated a similar relationship but to a much lesser extent, suggesting manipulating it optimise either parameter would be less effective but also carry less penalty.

Only fin spacing demonstrated a clear optimum value to minimise heatsink thermal resistance within the range studied here. The other parameters demonstrated non-linear, but progressive, impacts on thermal resistance and average heat transfer coefficient. This limits the range across
which modifying the parameter value significantly benefits heatsink thermal management performance. However, as Fig. 9-9 shows, there was clearly a strong association between fin height and thermal resistance that continues beyond the parameter range imposed by the constraints of this analysis. Significant improvement could, therefore, still be realised before this limit is reached. Neither property appears to be fully optimised using the initial parameters. Thermal resistance can be reduced further and there are significant improvements to average heat transfer coefficient that could potentially be realised. While in most instances the most effective (highest average heat transfer coefficient) heatsink geometry was produced by one of either the maximum or minimum parameter values, to rely on this criteria alone to define the most effective combination of parameters would be flawed as each result was obtained while the other parameters were set to their default value (Table 9-2). To truly optimise the heatsink would require each of these parameters to be assessed against every permutation of all others. Even within the focused scope of this assessment, this would require 3,528,000 simulations, clearly an impractical approach. It would also be difficult to evaluate how any individual parameter influences thermal management performance. Treating them separately in this manner allowed the relative impact of each to be assessed, providing guidance to the identification of the optimum combination of parameters.

As defined in ‘5.3.3 Thermal management performance parameters’, a higher average heat transfer coefficient represents a better performing heatsink design (i.e. more effective at rejecting heat to the environment from a smaller surface area), so is desirable to maximise providing thermal resistance constraints are not exceeded. Excluding fin thickness and base thickness, as each heatsink parameter varied the resulting heatsink material volume and exposed surface area show a strong positive correlation (Fig. 9-11). Because average heat transfer coefficient is derived from the heatsink surface area, to some extent material volume can also be considered optimised by proxy. Should material content be the primary optimisation objective, similar techniques to interrogate the effect of changing heatsink parameters as presented here, and application of this knowledge in a manner similar to that described in the following sections, could be used to optimise it directly. However, the limitations of this research restricted what could be accomplished. Therefore, only heatsink thermal resistance and average heat transfer coefficient were considered in this investigation.
9.5 Average heat transfer coefficient optimisation

The findings of the parametric optimisation studies showed that the default parameters produced a reasonably effective heatsink, but there was still some potential for further improvement. Exploiting this whilst remaining within the constraints of this study (to maintain LED junction temperature within 0.5 K of the baseline model and fit within the same bounding region) requires further analysis.

Using the preliminary simulation results (‘9.4 Preliminary results’) and their insight onto their respective impact, different combinations of parameters were trialled with the aim of maximising the heatsink’s average heat transfer coefficient. To determine the optimum combination of parameters, the respective impacts of each were first considered. The aim was to maximise the heatsink’s average heat transfer coefficient without any increase in thermal resistance. The plots shown in Fig. 9-4 to Fig. 9-10 were studied to see where average heat transfer coefficient improvements could be realised and where any associated increase in thermal resistance could be compensated for by modifying other more sensitive parameters (i.e. realising average heat transfer coefficient gains by increasing central cut-out taper angle, and compensating for the associated drop in thermal resistance by modifying fin spacing, which had a larger effect). The individual responses of each parameter were then used to identify the upper limit where the impact of modifying each parameter diminished (i.e. the gradient of the trend line through the data became almost 0) to restrict the scope of the parameter’s range under consideration. Because of the interaction between the heatsink’s parameters, several different combinations...
were tested to narrow in on the optimum performing configuration. These trials used the refined simulation definition (using the fine mesh settings and radiative heat transfer behaviour). The greatest average heat transfer coefficient was found using the parameters listed in Table 9-3, the results of which are shown in Fig. 9-12. Table 9-4 summarises the properties of the optimised design with reference to the baseline model (‘9.2 Initial model analysis’). These results also include an estimation of the associated impact on material cost and embodied energy to help relate the findings to market factors.

Table 9-3: Predicted parameter values for optimum heatsink average heat transfer coefficient

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Minimum assessed value</th>
<th>Maximum assessed value</th>
<th>Default assessed value</th>
<th>Independently optimised value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>0.5</td>
<td>5</td>
<td>2.5</td>
<td>2.5</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>1</td>
<td>10</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Central cut-out offset (mm)</td>
<td>0</td>
<td>Suppressed</td>
<td>7.5</td>
<td>7.5</td>
</tr>
<tr>
<td>Central cut-out diameter (mm)</td>
<td>5</td>
<td>35</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Central cut-out taper angle (degrees)</td>
<td>0</td>
<td>60</td>
<td>30</td>
<td>50</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>1</td>
<td>17</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Base thickness (mm)</td>
<td>1</td>
<td>10</td>
<td>5</td>
<td>5</td>
</tr>
</tbody>
</table>

Fig. 9-12: Predicted temperature distribution and fluid flow profile of heatsink optimised for maximum average heat transfer coefficient
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Baseline model</th>
<th>Optimised model</th>
<th>Difference between baseline and optimised models</th>
<th>Properties of optimised model relative to baseline model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heatsink material volume (m$^3$, x10^{-6})</td>
<td>83.82</td>
<td>75.48</td>
<td>8.34</td>
<td>- 9.9 %</td>
</tr>
<tr>
<td>Exposed surface area (m$^2$, x10^{-3})</td>
<td>42.43</td>
<td>35.09</td>
<td>7.34</td>
<td>- 17.3 %</td>
</tr>
<tr>
<td>Mass* (kg)</td>
<td>0.237</td>
<td>0.213</td>
<td>0.024</td>
<td>- 9.9 %</td>
</tr>
<tr>
<td>Material cost** (GBP)</td>
<td>£0.2868</td>
<td>£0.2583</td>
<td>£0.0285</td>
<td>- 9.9 %</td>
</tr>
<tr>
<td>Embodied energy*** (MJ)</td>
<td>47.32</td>
<td>42.62</td>
<td>4.71</td>
<td>- 9.9 %</td>
</tr>
<tr>
<td>Peak heatsink temperature relative to ambient environment (K)</td>
<td>+ 40.7</td>
<td>+ 41.2</td>
<td>0.4</td>
<td>+ 1.1 %</td>
</tr>
<tr>
<td>Thermal resistance (K.W^{-1})</td>
<td>3.23</td>
<td>3.27</td>
<td>0.04</td>
<td>+ 1.1 %</td>
</tr>
<tr>
<td>Average heat transfer coefficient (W.m^{-2}.K^{-1})</td>
<td>7.29</td>
<td>8.72</td>
<td>1.43</td>
<td>+ 19.7 %</td>
</tr>
</tbody>
</table>

*Mass derived from heatsink material volume and density of 2823 kg.m$^{-3}$ (MATWEB, n.d. b)

** Price correct at 12.00 am, 29th July 2016. Price calculated from Aluminium cash buyer value of $1597 per tonne (London Metal Exchange, n.d.) and currency exchange rate of £0.7590 per USD (Exchange rates UK, n.d.)

*** Based on Aluminium embodying 200 MJ.kg$^{-1}$ (Bar-Cohen et al., 2006)

Although the thermal resistance was worse than the baseline model, the peak heatsink temperature rise was within the 0.5 K tolerance applied to these studies. The optimum fin thickness was found to be 2.5 mm but it was the thinnest fins, which create the most heatsink surface area, that were initially expected to produce the best thermal management performance. It appeared that the thinnest fins restricted conductive heat transfer to, and consequently hindered heat transfer from, the surface of the fins. This could well be offset by increasing surface area. However, to maximise how effectively the heatsink utilises the available surface area, the parameters of the heatsink have to balance surface area against transporting heat through the structure. Hence the optimum fin thickness did not correspond to the maximum surface area.
These results also demonstrate that a series of benefits can be readily achieved. In absolute magnitude the improvement was small, but the relative effect was considerable. The heatsink’s average heat transfer coefficient increased by almost 20% and its surface area was reduced by 17%. Concurrently its total mass was reduced by 10%. This reduction in mass offers direct savings in material cost and embodied energy as well as less tangible benefits to the production and transportation of the component. The relative improvements were smaller than displayed by the initial concepts explored in Chapter 8. This indicates that the constraints of the design and the different conditions restricted the impact of refining the heatsink geometry. A potential explanation for this is that radiative heat transfer represented a much higher fractional contribution to total heat transfer from this heatsink than the models considered in Chapter 8. This would have magnified the effects of radiative heat transfer whilst also reducing the significance of any improvement in convective heat transfer. The geometry optimisation, which predominantly impacts the heatsink’s interaction with the cooling fluid (i.e. its convective heat transfer behaviour), would have, therefore, had a correspondingly smaller impact on its thermal management performance. The introduction of a surrounding surface and a different heat source configuration are also likely to be factors that led to this outcome. As multiple properties were modified the exact cause is unclear, but it is believed to result from a combination of the imposed changes.

To verify the assumption that radiative heat transfer was significant, but also relatively insensitive to the changes made to the heatsink through its optimisation, the simulation was repeated with radiative heat transfer behaviour suppressed. Following the same procedure described in ‘6.1.2 Analysis results’, radiative heat transfer’s fractional contribution to total thermal power dissipated by the heatsink was calculated to be approximately 28%. This was almost identical to the 26% fractional contribution it made to the initial model and so the assumption was considered valid.

9.6 Thermal resistance optimisation

Optimising the heatsink’s average heat transfer coefficient provided a direct benefit to the required material content and consequently to its embodied energy. Embodied energy provides an approximate measure of the part’s environmental impact, so, provided other factors such as manufacturability are not compromised, reducing it is likely to be beneficial. The literature indicates that energy consumption during the system’s use phase is far greater (United States
Department of Energy, 2012). Therefore, enhancing energy consumption during use potentially offers greater benefits to the system’s total energy demands.

To evaluate which avenue offers the greatest energy saving a second optimisation study was performed. Enhancing the operating characteristics of the system imposes different objectives to the previous optimisation analysis. As discussed in the literature review, enhancing extraction of waste heat from the system improves the operating performance of the LED light source. This translates to greater operating efficiency and, therefore, a reduction in energy consumption, from which the impact on the product lifecycle can be estimated. Therefore, the lowest possible peak heatsink temperature (i.e. the lowest thermal resistance configuration) was sought. For the purposes of comparison, the same geometric constraints and boundary conditions were imposed. Following a similar procedure to that discussed in ‘9.5 Average heat transfer coefficient optimisation’, but with thermal resistance now the main objective, the impact of each heatsink parameter was again evaluated and an optimum configuration identified. After a series of trials, the optimum thermal resistance was found to be achieved using the parameters listed in Table 9-5, the results of which are plotted in Fig. 9-13. The properties of the heatsink using these parameters, with reference to the baseline model (‘9.2 Initial model analysis’), are summarised in Table 9-6.

### Table 9-5: Predicted parameter values for optimum heatsink thermal resistance

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Minimum assessed value</th>
<th>Maximum assessed value</th>
<th>Default assessed value</th>
<th>Independently optimised value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>0.5</td>
<td>5</td>
<td>2.5</td>
<td>0.5</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>1</td>
<td>10</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Central cut-out offset (mm)</td>
<td>0</td>
<td>Suppressed</td>
<td>7.5</td>
<td>Suppressed</td>
</tr>
<tr>
<td>Central cut-out diameter (mm)</td>
<td>5</td>
<td>35</td>
<td>20</td>
<td>-</td>
</tr>
<tr>
<td>Central cut-out taper angle (degrees)</td>
<td>0</td>
<td>60</td>
<td>30</td>
<td>-</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>1</td>
<td>17</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Base thickness (mm)</td>
<td>1</td>
<td>10</td>
<td>5</td>
<td>10</td>
</tr>
</tbody>
</table>
Fig. 9-13: Predicted temperature distribution and fluid flow profile of heatsink optimised for minimum thermal resistance

### Table 9-6: Summary of predicted thermal resistance optimised heatsink properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Baseline model</th>
<th>Optimised model</th>
<th>Difference between baseline and optimised models</th>
<th>Properties of optimised model relative to baseline model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heatsink material volume (m$^3$, x10$^{-6}$)</td>
<td>83.82</td>
<td>90.96</td>
<td>7.14</td>
<td>+ 8.5 %</td>
</tr>
<tr>
<td>Exposed surface area (m$^2$, x10$^{-3}$)</td>
<td>42.43</td>
<td>46.12</td>
<td>3.69</td>
<td>+ 8.7 %</td>
</tr>
<tr>
<td>Mass* (kg)</td>
<td>0.237</td>
<td>0.257</td>
<td>0.020</td>
<td>+ 8.5 %</td>
</tr>
<tr>
<td>Material cost** (GBP)</td>
<td>£0.2868</td>
<td>£0.3112</td>
<td>£0.0244</td>
<td>+ 8.5 %</td>
</tr>
<tr>
<td>Embodied energy*** (MJ)</td>
<td>47.32</td>
<td>51.36</td>
<td>4.03</td>
<td>+ 8.5 %</td>
</tr>
<tr>
<td>Peak heatsink temperature relative to ambient environment (K)</td>
<td>+ 40.7</td>
<td>+ 38.6</td>
<td>2.2</td>
<td>- 5.4 %</td>
</tr>
<tr>
<td>Thermal resistance (K.W$^{-1}$)</td>
<td>3.23</td>
<td>3.06</td>
<td>0.17</td>
<td>- 5.4 %</td>
</tr>
<tr>
<td>Average heat transfer coefficient (W.m$^2$.K$^{-1}$)</td>
<td>7.29</td>
<td>7.09</td>
<td>0.20</td>
<td>- 2.8 %</td>
</tr>
</tbody>
</table>

*Mass derived from heatsink material volume and density of 2823 kg.m$^{-3}$ (MATWEB, n.d. b)

** Price correct at 12.00 am, 29$^{th}$ July 2016. Price calculated from Aluminium cash buyer value of $1597 per tonne (London Metal Exchange, n.d.) and currency exchange rate of £0.7590 per USD (Exchange rates UK, n.d.)

*** Based on Aluminium embodying 200 MJ.kg$^{-1}$ (Bar-Cohen et al., 2006)
These results show the improvement in thermal resistance (of approximately 5%) comes at the expense of all other criteria. The geometry of this heatsink may have also compromised its manufacturability. An analysis of a model using a fin thickness comparable to the original component demonstrated far less benefit to the heatsink’s thermal resistance and, as a consequence, its energy consumption during use. Its feasibility would, therefore, need to be reassessed.

The optimisation achieved a 2.190 K decrease in peak heatsink temperature rise. From the datasheet for the current generation of LED component around which the simulation is based, an estimate of the luminous efficacy increase resulting from this reduction in operating temperature would be 0.5 % (Philips, 2015). This could be exploited to increase the luminous flux emitted by the luminaire. Alternatively, at this improved luminous efficacy the same luminous flux, and thereby the luminaire specification, could be maintained with less input power, thus reducing the total lifecycle energy consumption. From the literature review, a typical LED luminaire’s lifecycle energy consumption during use accounts for approximately 3500 MJ (United States Department of Energy, 2012. Part 1). The reduction in luminaire lifecycle energy consumption achieved by maintaining the same luminous output, but accounting for a 0.5 % greater operating efficacy, can then be calculated as follows.

\[
\text{Original luminous output as a percentage of improved system's output} = \frac{1}{1.005}
\]

Equation 9-1

Therefore:

\[
\text{Percentage reduction of improved system's output to match original specification} = 1 - \left( \frac{1}{1.005} \right)
\]

Equation 9-2

And so, assuming a linear relationship between luminous output and power consumed:

\[
\text{Lifecycle energy consumption saving (MJ)} = 3500 \times \left( 1 - \left( \frac{1}{1.005} \right) \right)
\]

Equation 9-3
According to this calculation, the total reduction in consumed energy through the luminaire’s lifetime would be 17.41 MJ. If the additional material content and embodied energy of the thermal resistance optimised heatsink compared to the average heat transfer coefficient optimised heatsink is taken into account this saving is reduced to 13.38 MJ (equivalent to 3.72 kWh). These figures were derived from extrapolated data. Even so, it can confidently be stated that the saving is negligible compared to the total lifecycle energy consumption.

9.7 Conclusions and evaluation

The thermal management performance of the heatsink can be optimised in different ways, depending on the most important criteria. Applying relatively small modifications to the heatsink was shown to increase its average heat transfer coefficient by up to 20% whilst reducing the material content, embodied energy and heatsink cost by up to 10% with little accompanying increase in thermal resistance (and hence LED junction temperature). Alternatively, the system thermal resistance can be reduced by up to 5%, which was extrapolated to a lower LED junction temperature and consequently lower system lifecycle energy consumption. However, in comparison to the concepts considered in Chapter 8, the constraints, optimisation methods applied and boundary conditions of the application apparently restricted the thermal management performance gains available were reduced.

The usage phase represents the most significant portion of the product’s lifecycle energy consumption. This was reduced by optimising the heatsink’s thermal resistance, but the benefits were relatively insignificant and the same outcome can be realised by other means; for example, increasing the heatsink surface area. The chosen heatsink concept offered an inherently low thermal resistance, but uncertainty regarding the lifecycle benefit, the consequences for other properties such as manufacturability, and commercial demands limit the value of further optimising this property. On the other hand, maximising effectiveness provides direct tangible benefits (material savings, lower cost) that cannot be realised by other means. For this reason, it is considered to be the more appropriate target for optimisation.

Although this study builds on the concepts of Chapter 8, there is still uncertainty regarding how these findings translate to different situations, in particular to differently sized heatsinks and to higher power applications. It is believed that, in these situations, the benefits of optimisation could be amplified. For instance, higher rates of heat transfer to the environment could establish
higher rates of buoyancy driven, convective fluid flow, optimising fluid flow across the heatsink would, therefore, have a greater impact on its thermal management performance. This study simply demonstrated the proof of concept in a typical application. Further investigation is needed to explore the response under different conditions. The simulation boundary conditions have changed from the benchmarked definition established earlier. For confidence in the conclusions, these conditions should be benchmarked against further physical samples but it is strongly believed the conclusions are accurate. Finally, the outcomes of these optimisation studies are believed to represent the best performing heatsink designs for the studied conditions, but without evaluating every combination of heatsink parameters this cannot be guaranteed. A comprehensive optimisation study, evaluating every permutation of the model parameters, would have been impractical and so the process had to be simplified. This was realised by evaluating the impact of each heatsink parameter in isolation (one factor at a time), and then assessing the results, along with some trial and error, to identify an improved combination of parameters. Statistical methods for analysing the effects of different factors are available, but implementing them is not straightforward. The techniques employed here are believed to be more appropriate to achieve widespread commercial adoption. Refining the optimisation process with statistical methods appropriate for use in a commercial setting could improve the results and reduce uncertainty so would be a potentially valuable avenue for further study.

This chapter demonstrated how the concepts discussed in Chapter 8 can be applied to a commercially available component. The results predicted significant improvements in thermal resistance, average heat transfer coefficient and heatsink mass (and hence cost) can all be achieved with relatively minor modifications. It was also shown that a reduction in lifecycle energy consumption can be achieved through reducing the heatsink’s thermal resistance (although the benefit was relatively small). This outcome showed that thermal management could be achieved more effectively than under current practices (i.e. requiring less material, at lower cost, reducing environmental impact, consuming less energy, minimising susceptibility to environment affects) simply by adopting a thorough development process.
Chapter 9 demonstrated the potential benefits of optimising conventional heatsink geometry, but was restricted by several constraints and simplifications. The objective of this chapter is to evaluate the potential thermal management performance that can be achieved if those restrictions are relaxed. This includes an assessment of what impact incorporating a chimney structure and enclosing the heatsink channels might have on thermal management performance. These were concepts identified in Chapter 8 for their ability to influence the behaviour of the system and potentially open up new opportunities to enhance thermal management performance. Incorporating an enclosing body may also provide a secondary benefit as a shroud to guard against the burn hazard presented by the hot heatsink surfaces. Based on the conclusions of Chapter 9, thermal management performance was evaluated solely on average heat transfer coefficient (effectiveness).

10.1 Design objectives
Maintaining the system’s equivalence to an existing product potentially restricts its thermal management performance but ensures a degree of commercial relevance. The case studied in Chapter 9 was tightly constrained for this reason. To evaluate fully the potential for thermal management performance enhancement, it is necessary to remove as many constraints as possible. However, within this objective the system’s primary function should not be lost. Therefore, this study was undertaken around the same LED component which imposed the same thermal load and temperature boundary conditions applied previously. The design was also required to provide the same physical arrangement for the emitted light (i.e. same sized aperture and depth of light source chamber).

The geometric constraints previously applied to the model (part height and fixed faces) were not employed in this analysis. High Density Die-Casting (HDDC) now allows high volume production of thinner, taller, closer spaced fins with superior thermal properties to traditional die-casting (Sce and Caporale, 2014) and additive manufacturing processes such as Selective Laser Sintering (SLS) are rapidly evolving to offer a variety of materials, with improved properties, in almost any
form imaginable (Singh et al., 2016). These, and similar advances, overcome production constraints so the relevance of manufacturability is expected to diminish. It can be argued that this would influence the heatsink concept selection (see ‘9.1 Simulation definition’) and thus a staggered pin-fin with open centre (i.e. the concept evaluated in ‘8.1.7 Case 6: Staggered pin with open centre heatsink results’) design may be more appropriate. However, the timescales of manufacturing process improvements are unclear, creating uncertainty as to when concepts predicated on these advances will become commercially viable. Restricting development to the heatsink form studied previously minimises the manufacturing advances necessary for its realisation. Consequently, improvements can be exploited sooner, providing greater commercial incentive for further development. The decision was made to retain the same heatsink concept and explore the potential thermal management performance it could offer. This also meant the previous analysis could be used to provide a comparable benchmark for evaluation and guide the identification of the optimum heatsink parameters. Further analysis to compare the pin-fin heatsink concept would be valuable, but owing to the limited time and resources, was not possible here.

With relatively open design objective it is worth reiterating the principles of heatsink design which are being employed. To summarise: heatsink fins need to be oriented to minimise obstruction to airflow, fins should as far as possible be parallel to each other, a thermal gradient between each heatsink surface and the contacting fluid must be maintained, and redundant surfaces should be removed. The constraints on the LED component position and optical chamber created the starting point of a heatsink concept to which these principles were applied. To this model, an enclosing body was added above the heatsink fins. The enclosing body also extended upwards to create a chimney arrangement in order to enhance fluid flow across the heatsink. This enclosing component was modelled as a distinct body, to represent a two piece assembly, on the basis that this would be easier to manufacture by traditional methods. This design would call for some means of fixing the two parts together but for the purposes of evaluation were omitted for simplicity. For analysis, mechanical fixings were envisaged for connecting the two components, which could potentially result in a small but significant air gap and poor thermal contact between them. As the enclosing structure was not intended to act as a heat exchanging surface, only constrain airflow, conduction of heat between the two components was not considered to be important. This was, therefore, believed to be a reasonable representation of how this concept would potentially be realised. The design is shown in Fig. 10-1 (note fixed dimensions and parameters of interest have been labelled).
Fig. 10-1: Enclosed heatsink concept
10.2 Simulation definition

A full factorial parametric optimisation study was created to refine the heatsink design. For simplicity, only those parameters which previously (‘9.4 Preliminary results’) expressed a strong association with heatsink thermal management performance (labelled in Fig. 10-1) were modified. A small range around the optimum values found in the previous study (‘9.5 Average heat transfer coefficient optimisation’) were selected for study (see Table 10-1). All other model parameters were assigned a fixed value. As already discussed (‘9.4 Preliminary results’), a more extensive optimisation study which incorporated these fixed parameters, as well as a wider range of values, would have been very time consuming. This was not achievable within the scope of this analysis and so judgement was employed to restrict its size. The findings of the previous investigation were used to make an informed judgement, so the outcome was believed to represent a reasonably accurate approximation of the optimum design.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assessed value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>2, 2.5, 3</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>4, 6, 8, 10</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>5, 10, 15, 20</td>
</tr>
<tr>
<td>Chimney height (mm)</td>
<td>25, 50, 75</td>
</tr>
</tbody>
</table>

Most of the assigned simulation conditions were identical to those used in Chapter 9. Once again, it was simulated attached to a representation of a ceiling (Fig. 10-2) and two simulation definitions were employed in its analysis: simplified and refined. The simplified simulation employed a sparse mesh density and was used for the parametric optimisation study. The refined simulation employed a fine mesh density and was used to quantify accurately the thermal management performance of particular models. The mesh was defined as previously discussed.
The enclosing chimney component was designed to constrain fluid flow between the heatsink fins, not transfer heat to the environment. For this reason it was assigned a custom thermal conductivity of 0.2 W.m\(^{-1}\).K\(^{-1}\). This was comparable to many polymers (a potential candidate material for this component) and minimised any artificial effects that would have been present if it defaulted to an idealised insulating material.

During the parametric optimisation study, the interface between the heatsink fins and chimney bodies defaulted to idealised contact behaviour. Software limitations prevented assignment of an accurate thermal resistance to the contact surfaces (owing to the creation of new faces during the study). Applying a low thermal conductivity material property to the chimney body limited the impact of this ideal condition and so was accepted for the sake of simplicity. In the subsequent evaluation of individual models under a refined simulation definition, the thermal contact resistance was 0.05 K.m\(^2\).W\(^{-1}\) (based on an estimation of a poor contact condition generated under low compression taken from the work of Fletcher, 1972). This allows some limited heat transfer between the two and so is considered more accurate than an idealised or an adiabatic boundary condition. However, it was not benchmarked against a physical specimen so cannot be relied upon as accurate where the property has a significant impact on system behaviour. For this initial study, the chimney acts to constrain fluid flow rather than facilitate heat exchange with the environment and so the poor contact condition was considered acceptable.

As noted in the previous chapter, the parametric optimisation process prevents radiative surface properties being assigned to the entire model. Instead, the default surface properties (non-radiating) were once again accepted and the error assumed to have a consistent impact on all
model configurations. While this was useful in assessing the influence of the chimney structure on conductive and convective heat transfer alone, the effects of radiative heat transfer are known to be significant (approximately 26% fractional contribution to total heat transfer from the initial heatsink model considered in the previous chapter) and so cannot be ignored when attempting to accurately quantify the heatsink’s thermal management performance. This was especially critical here because of the presence of the enclosing chimney structure, which would interfere with radiative heat transfer from the heatsink. The previous chapter determined the peak heatsink temperature rise above the ambient environment temperature would be 56.0 K when analysed using the simplified simulation definition (‘9.2 Initial model analysis’). Only model configurations which demonstrated an equal or lower temperature rise under equivalent conditions were considered able to meet the design constraints. The radiative surface properties of the refined simulation model are as described in ‘6.2.1 Model definition’.

The parametric optimisation study was conducted both on the heatsink model alone and with the chimney structure present to assess its effect on thermal management performance. The heatsink models offering optimum thermal management performance, in terms of average heat transfer coefficient, were subsequently evaluated using a refined simulation definition. This employed the fine simulation mesh and validated radiative surface properties discussed, as well as a defined thermal contact resistance between the heatsink and chimney. For the refined simulation the chimney structure was assigned an emissivity of 0.8, to represent a matt, flat coloured surface (estimated based on values supplied by Fluke Corporation (2007)).

10.3 Analysis of results

The full factorial parametric optimisation study found no single design configuration could concurrently optimise all properties of interest (thermal resistance, average heat transfer coefficient and heatsink material volume). After assessing each combination of heatsink parameters, those that failed to meet the peak heatsink temperature constraint of 56.0 K (as defined by the baseline model evaluated using the same simplified simulation definition) were rejected. Of those that remained the greatest average heat transfer coefficients from the heatsink, when the chimney structure was included and excluded, were found using the values summarised in Table 10-2. The properties of each of these configurations were evaluated using the refined simulation conditions and are reported in Table 10-3 along with an assessment of the relative effect of incorporating the chimney structure.
Table 10-2: Predicted parameter values for optimum heatsink average heat transfer coefficient

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Minimum assessed value</th>
<th>Maximum assessed value</th>
<th>Optimised value without chimney</th>
<th>Optimised value with chimney</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>4</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>5</td>
<td>20</td>
<td>15</td>
<td>10</td>
</tr>
<tr>
<td>Chimney height (mm)</td>
<td>25</td>
<td>75</td>
<td>-</td>
<td>75</td>
</tr>
</tbody>
</table>

Table 10-3: Summary of predicted average heat transfer coefficient optimised heatsink properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Without chimney</th>
<th>With chimney</th>
<th>Predicted difference with addition of chimney augmentation</th>
<th>Predicted change in properties with addition of chimney augmentation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak heatsink temperature relative to ambient environment (K)</td>
<td>+ 39.5</td>
<td>+ 46.7</td>
<td>+ 7.1</td>
<td>+ 18.1 %</td>
</tr>
<tr>
<td>Thermal resistance (K.W⁻¹)</td>
<td>3.14</td>
<td>3.70</td>
<td>0.56</td>
<td>+ 17.8 %</td>
</tr>
<tr>
<td>Average heat transfer coefficient (W.m⁻².K⁻¹)</td>
<td>7.82</td>
<td>8.33</td>
<td>0.51</td>
<td>+ 16.5 %</td>
</tr>
<tr>
<td>Heatsink material volume (m³, x10⁶)</td>
<td>73.91</td>
<td>62.77</td>
<td>- 11.14</td>
<td>- 15.1 %</td>
</tr>
</tbody>
</table>

These results appear to show that integrating the chimney structure had a big impact on the system’s behaviour, but not necessarily a beneficial one. As seen in ‘9.5 Average heat transfer coefficient optimisation’, the highest average heat transfer coefficient was not produced by the thinnest fins, supporting the findings of the previous chapter. It can be seen that while average heat transfer coefficient and material volume and were improved (by 15.1 % and 16.5 % respectively), adding the chimney also increased the peak temperature and thermal resistance of the heatsink (by 18.1 % and 17.8 % respectively). However, it must be noted that when the chimney structure was present the peak heatsink temperature rise exceeded thermal constraints (defined as 40.7 K as per ‘9.2 Initial model analysis’). It is believed this was a consequence of the simulation definition and selection process used to identify the ‘optimum’ configuration. The initial parametric optimisation study was conducted using a simplified simulation definition which
applied the default non-radiating surface property to all model surfaces. The baseline model used to define the thermal constraints was also analysed using this same simplified definition. It was assumed that the difference between the simplified and refined simulation models was consistent. Therefore, any model analysed using the simplified simulation that meets the thermal constraints (as defined by the baseline model analysed under the same conditions) would also meet the thermal constraints when a refined simulation definition was used. This did lead to the selection of a suitable design when the chimney body was excluded. However, when the chimney was present its influence on radiative heat transfer from the heatsink meant the thermal management performance of models under the simplified and refined simulation conditions were no longer comparable, leading to the selection of an unsuitable chimney configuration. When a default emissivity of 0.8 was employed in the simplified simulation definition, the results from the heatsink models incorporating a chimney showed a much more consistent correlation to the refined simulation results. They could, therefore, be used to determine the optimum configuration within the assessed parameter space and design constraints. To determine the relevant peak heatsink temperature constraint in order to eliminate unsuitable configurations the baseline model was reassessed using a default emissivity of 0.8 in place of non-radiating behaviour. As before, the heatsink and chimney configurations that failed to meet this constraint were then eliminated from consideration and the peak performing design identified. Table 10-4 is an updated version of Table 10-2, which defines the models found to be the closest match. The behaviours of these models are shown in Fig. 10-3 and Fig. 10-4 while their properties are summarised in Table 10-5.

Table 10-4: Parameter values for predicted optimum average heat transfer coefficient, within thermal constraints

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Minimum assessed value</th>
<th>Maximum assessed value</th>
<th>Optimised value without chimney</th>
<th>Optimised value with chimney</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness (mm)</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Fin spacing (mm)</td>
<td>4</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Fin height (mm)</td>
<td>5</td>
<td>20</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Chimney height (mm)</td>
<td>25</td>
<td>75</td>
<td>-</td>
<td>75</td>
</tr>
</tbody>
</table>
Fig. 10-3: Temperature distribution and fluid flow profile of unconstrained heatsink optimised for maximum average heat transfer coefficient, without chimney augmentation

Fig. 10-4: Temperature distribution and fluid flow profile of unconstrained heatsink optimised for maximum average heat transfer coefficient, with chimney augmentation (not shown)
Table 10-5: Summary of predicted average heat transfer coefficient optimised heatsink properties, within thermal constraints

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Without chimney</th>
<th>With chimney</th>
<th>Difference with addition of chimney augmentation</th>
<th>Change in properties with addition of chimney augmentation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak heatsink temperature relative to ambient environment (K)</td>
<td>+39.5</td>
<td>+38.8</td>
<td>-0.7</td>
<td>-1.8 %</td>
</tr>
<tr>
<td>Thermal resistance (K.W⁻¹)</td>
<td>3.14</td>
<td>3.08</td>
<td>-0.06</td>
<td>-1.9 %</td>
</tr>
<tr>
<td>Average heat transfer coefficient (W.m⁻².K⁻¹)</td>
<td>7.82</td>
<td>8.01</td>
<td>0.19</td>
<td>+2.4 %</td>
</tr>
<tr>
<td>Heatsink material volume (m³, x10⁻⁶)</td>
<td>73.91</td>
<td>73.91</td>
<td>0.00</td>
<td>0 %</td>
</tr>
</tbody>
</table>

The configurations detailed in Table 10-4 were within the thermal constraints, but as Table 10-5 reveals, the potential impact of the chimney was much smaller than initially thought. While there appears to be some benefit to incorporating the chimney structure, it is likely to be too small to be considered worthwhile. This implies that efforts to improve the thermal management performance of the heatsink through modifying fin geometry can be more productive than adding the enclosing chimney structure proposed here. The performance of these concepts is related to the baseline reference model (‘9.2 Initial model analysis’) in Table 10-6.

Table 10-6: Predicted benefits of heatsink optimisation

<table>
<thead>
<tr>
<th>Model</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Peak heatsink temperature relative to ambient environment compared to baseline model</th>
<th>Thermal resistance (K.W⁻¹)</th>
<th>Thermal resistance relative to baseline model</th>
<th>Average heat transfer coefficient (W.m⁻².K⁻¹)</th>
<th>Average heat transfer coefficient relative to baseline model</th>
<th>Heatsink material volume (m³, x10⁶)</th>
<th>Heatsink material volume relative to baseline model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline model</td>
<td>+40.7</td>
<td>-</td>
<td>3.23</td>
<td>-</td>
<td>7.29</td>
<td>-</td>
<td>83.82</td>
<td>-</td>
</tr>
<tr>
<td>Without chimney</td>
<td>+39.5</td>
<td>-3.0 %</td>
<td>3.14</td>
<td>-2.8 %</td>
<td>7.82</td>
<td>+7.3 %</td>
<td>73.91</td>
<td>-11.8 %</td>
</tr>
<tr>
<td>With chimney</td>
<td>+38.8</td>
<td>-4.7 %</td>
<td>3.08</td>
<td>-4.6 %</td>
<td>8.01*</td>
<td>+9.9 %</td>
<td>73.91*</td>
<td>-11.8 %</td>
</tr>
</tbody>
</table>

*chimney body excluded from calculation
The improvement in average heat transfer coefficient achieved by adding the chimney was smaller than that offered by the optimised designs developed earlier (‘9.5 Average heat transfer coefficient optimisation’). The evidence suggests that the heatsink design considered here, with or without the addition of the chimney structure, offers no significant thermal management performance advantage over the optimised designs identified in Chapter 9. Consequently, pursuing this chimney concept further appears to be of limited commercial value. This appears to contradict the findings of Chapter 8 but there are some probable causes. As discussed in the previous chapter where similar surface emissivity properties were employed, the impact of radiative heat transfer is believed to have changed. Following the procedure described in ‘6.1.2 Analysis results’, radiative heat transfer’s fractional contribution to total power dissipated by the optimised heatsink with chimney augmentation was calculated to be approximately 14 %. In comparison to the model considered in Chapter 9, where radiative heat transfer accounted for approximately 26 % of total heat dissipated, the addition of the chimney appears to create a significant obstruction to radiative heat transfer. At the same time, compared to the 11 % fractional contribution to total heat transfer made by radiation from the parallel plate heatsink considered in ‘8.1.2 Case 1: Parallel plate heatsink results’, the greater contribution would diminish the significance of convective heat transfer and the benefit offered by the chimney augmentation. There have also been some significant design changes imposed on the heatsink’s design. Rather than the chimney rising straight up from the sides of the heatsink (as in Chapter 8), it instead sits across the top of the heatsink fins with a narrow vertical section in the centre. The heatsink’s fins have also been moved away from the heat source and, as with the previous chapter, the effects of a surrounding surface below the heatsink have been introduced. It is not known what impact these changes had on the heatsink’s thermal management performance or whether the concept truly represents an optimum design. It is clear that design and optimisation has to build on a proven foundation to arrive at the best performing system as trying to base it on predictions leads to uncertainty. As such, further analysis is necessary to overcome the uncertainty associated with these results and evaluate the true potential of augmenting the heatsink with a chimney structure.

Some design configurations demonstrated improvements of the heatsink’s thermal resistance are possible. The greatest thermal resistance improvements were achieved at the expense of average heat transfer coefficient, but there were configurations which offered a compromise allowing both properties to be enhanced simultaneously.
As shown in Chapter 8, the data once again indicated greater chimney height equated to a larger benefit to thermal management performance. There were a number of cases from the parametric optimisation studies that demonstrated even the smallest chimneys could have a positive impact on heatsink thermal management performance. However, when a refined simulation definition was used to evaluate these configurations accurately, the chimney was actually revealed to have a negative impact. This discrepancy is likely to be a consequence of the simulation boundary conditions employed. The simplified simulation definition (used in the parametric optimisation studies) employed a coarse mesh which, in this situation, produced a misleading result. There is, therefore, some uncertainty surrounding the optimum design configurations identified through the parametric studies. While a better optimisation procedure would be desirable, because the simplified simulation definition was applied consistently, and the more accurate refined simulation definition was used to quantify the thermal management performance, the uncertainty was considered to be relatively inconsequential. What these results also reveal is that the enclosing chimney restricts heat transfer from the heatsink. However, once the chimney height reaches a certain threshold, any negative impact appears to be offset by the enhanced convective heat transfer and so overall thermal management performance is improved. Exploring how this behaviour translates to different conditions may reveal situations where the chimney’s impact is more beneficial.

The model described in Table 10-4 integrated the tallest chimney. The fluid flow velocities through that chimney, which had a cross-sectional area of \(2.66 \times 10^{-3} \text{ m}^2\) and wetted perimeter of 0.19 m, were in the region of 0.25 m.s\(^{-1}\). According to Equation 8-1 and Equation 8-2, this would equate to Reynolds numbers below 1000, so once again well within the bounds of laminar behaviour (Holman, 2010).

There is a strong possibility that the material properties assigned to the chimney body limited its impact on system thermal management performance. To validate the significance of the low chimney thermal conductivity a separate analysis was required for comparison, the results of which are presented in Table 10-7. The same simulation definition and optimum design parameters were used but a high thermal conductivity material was substituted for the chimney body. For simplicity, and because it was expected to be a likely candidate material, die-cast aluminium properties were employed for the role. However, it was still considered to be a distinct body, on the basis it would potentially need to be manufactured as such, and so the poor contact condition with the heatsink was maintained. The intention behind assigning a high thermal
conductivity material property to the chimney structure was it would enhance the transport of heat through the structure, therefore allowing its surfaces to be utilised more effectively to reject heat to the surrounding environment. On this basis, it may have been more appropriate to consider an alternative interface condition between the heatsink and chimney with lower thermal contact resistance (e.g. bonded). However, to minimise the number of changes imposed on the simulation the poor contact condition was accepted as an assessment of a worst case scenario. Despite the poor contact condition between them, because the chimney plays some role in facilitating heat exchange with the surrounding environment it was appropriate to consider it an extension of the heatsink body. As such, it was included in the calculation of the heatsink’s material volume and surface area, which were subsequently used to derive the part’s average heat transfer coefficient. The consequence of this was a huge decrease in average heat transfer coefficient and a large increase in material volume. The additional material content represents adverse cost and production implications as well as being detrimental to the system’s environmental impact. The highly conductive chimney structure did reduce the system’s thermal resistance, but the additional material required to achieve this was poorly utilised, as revealed by the systems low average heat transfer coefficient.

| Table 10-7: Comparison of predicted performance with alternative chimney material properties |
|-----------------------------------------------|---------------------------------|-------------------------------|-------------------|-------------------|----------------------|-------------------------|-----------------------|
| Model                                       | Peak heatsink temperature relative to ambient environment (K) | Peak heatsink temperature relative to ambient environment compared to baseline model | Thermal resistance (K.W⁻¹) | Thermal resistance relative to baseline model | Average heat transfer coefficient (W.m².K⁻¹) | Average heat transfer coefficient relative to baseline model | Heatsink material volume (m³, x10⁶) | Heatsink material volume relative to baseline model |
| Baseline model                              | + 40.7                          | -                             | 3.23                           | -                              | 7.29                              | -                                     | 83.82                          | -                          |
| Low thermal conductivity chimney             | + 38.8                          | - 4.7 %                       | 3.08                           | - 4.6 %                       | 8.01*                             | + 9.9 %                              | 73.91*                           | - 11.8 %                   |
| High thermal conductivity chimney            | + 37.1                          | - 9.0 %                       | 2.94                           | - 9.0 %                       | 3.90                              | - 46.5 %                             | 119.24                          | + 42.3 %                   |

*chimney body excluded from calculation
10.4 Conclusions

With respect to the baseline model of Chapter 9, the system developed here offered superior thermal management performance. Although it was still not possible to concurrently optimise the thermal resistance and average heat transfer coefficient of the system, there were several model configurations that did offer simultaneous improvement of both properties. The heatsink design which met the objectives and constraints of this study offered the potential to increase the average heat transfer coefficient by up to 10 % whilst reducing the heatsink’s total material content by up to 12 % and thermal resistance by up to 5 %. However, this average heat transfer coefficient improvement was less than that offered by the optimum design developed in the previous chapter (‘9.5 Average heat transfer coefficient optimisation’), indicating the design was less effective. The impact of integrating a chimney structure was also less than expected based on the results of Chapter 8, suggesting the different radiative properties and significant heatsink design modifications compromised its thermal management performance. It is clear that design should build on proven principles rather than rely on prediction. Consequently, the results of this analysis may not represent the optimum thermal management performance that could be achieved. Further analysis is, therefore, required to evaluate the true potential of augmenting the heatsink with a chimney structure.

The benefits of incorporating a chimney structure and enclosing the heatsink channels in the manner considered in this study appear to be very limited. The greatest impact was achieved using the tallest (75 mm) chimney height. In cases where the chimney height was small, its impact on the system’s thermal management performance was detrimental. This suggests that enclosing the heatsink restricts heat transfer. However, it can have a beneficial effect once the chimney height reaches a threshold value, where the additional convective heat transfer generated exceeds the restriction imposed. With reference to the non-augmented heatsink, the thermal management performance enhancement provided by the chimney structure was marginal (1 – 3 %). The results also failed to support any of the secondary benefits that enclosing the heatsink might offer. Exposure to the environment was shown in an earlier investigation to have very little effect on the part’s thermal management performance. In a typical ambient environment of 298.15 K, the peak temperature rise in the heatsink would be 336.96 K. As defined by BS EN ISO 13732-1:2008 (British Standards Institute, 2008), human skin’s burn threshold starts at 342.15 K for 1 second of exposure to a coated metal body. On this basis the heatsink poses very little risk so a protective shroud is superfluous. Even with the addition of the chimney structure the fluid flow displayed no turbulent flow characteristics and the associated enhancement of convective...
heat transfer. The practicality and commercial considerations of this concept would further oppose its adoption. On the basis of these results there is, therefore, very little justification to pursue this concept. Greater thermal management performance enhancement was achieved by modifying the heatsink and so, for maximum impact, that presents a more attractive focus of future development. However, the implementation of the chimney augmentation needs further study before it is possible to definitively state it offers no thermal management performance benefit.

Assigning the chimney structure a high thermal conductivity did achieve a small (0.29 K.W$^{-1}$, 9%) reduction of the system’s thermal resistance. However, because the enclosing chimney structure was then considered an extension of the heatsink, the average heat transfer coefficient and material volume were severely compromised. This reveals the concept’s ineffective utilisation of material and surface area for heat transfer. It should be noted these results were calculated with unverified contact conditions between the heatsink and chimney bodies. Assigning a high conductivity material property to the chimney meant this condition had a greater influence on the part’s heat transfer behaviour. While this introduces a potentially significant error, the negative thermal management performance impact was great enough to dismiss this concept with a reasonable degree of confidence.

For practical purposes, this analysis was performed using simplified simulation boundary conditions; it considered a small range of operating circumstances and optimised a focused selection of parameters. As a result, a degree of subjective judgement was required during the optimisation process. In addressing these limitations, further thermal management performance enhancements may be discovered which could overturn the conclusions drawn here. However, the findings are believed to present a reasonably accurate assessment of the systems behaviour and any major contradiction is unlikely. There is very little evidence, therefore, to justify pursuing the thermal management strategies discussed here any further.

This chapter explored the potential thermal management performance benefits of enclosing a heatsink. It was shown that in comparison to a typical commercially available component, thermal resistance and heat transfer can be enhanced. However, the improvement was no greater than could be achieved without enclosing the heatsink (see ‘Chapter 9: System optimisation (Constrained)’). There also appears to be no obvious advantage provided by protecting the heatsink or manipulating fluid flow. There is no evidence, therefore, to suggest that this strategy has any commercial value in the circumstances considered here. However, further analysis is required before it can be dismissed entirely.
Chapter 11:

Discussion of results and opportunities for further study

The work presented in this thesis represents a collection of advances towards effective thermal management for LED luminaires. This chapter summarises its main findings and puts them into context in order to clarify their potential value and commercial impact. The present limits and possible avenues for progression of this work are also discussed.

11.1 Thesis summary

LED technology offers a number of benefits over traditional light sources, particularly efficiency and reliability. As a result it has rapidly evolved to meet market demands and revolutionise lighting. However, it also imposes a number of unique demands on the luminaire: their long design life demands extreme reliability; their luminous efficacy promotes low energy consumption; and the physical limits of the semiconductor device requires effective removal of waste heat. Thermal management of electronics is a well-established field, but to date there has been relatively little assessment of how this should be applied to LED technology considering its particular demands or the relative inexperience of the lighting industry. The aim of this research was clear: to bring together relevant topics and then define the most effective strategies to manage the temperatures of LED components. It was conducted with a commercial focus to ensure the potential for implementation of its findings and the available thermal management performance improvements to be realised.

This work began with a review of the current state of the art and potential developments that may influence LED thermal management constraints. A review of thermal management technologies was conducted in order to identify opportunities to enhance their performance or exploit new concepts in commercial practice that had not yet been considered. From this it was possible to identify the key technologies for the focus of this research, along with a clearer definition of the constraints and challenges that need to be addressed. This led to an analysis of commercial practice, conducted over a two-year period, to identify any new trends and preferred
thermal management strategies. The middle phase of the research was concerned with acquiring and defining any missing information needed to develop effective thermal management strategies. This encompassed the definition of appropriate test methods, an analysis of two common operating environments’ impacts, and the benchmarking of suitable simulation parameters. The final part of this work was concerned with implementing computational models to evaluate the potential improvement offered by a more optimised design. Multiple concepts were compared to show how development can be directed towards better performing options. Optimisation analyses were applied to refine these concepts. The findings of these studies were then applied to a commercially available component to evaluate the potential benefits they offer.

11.2 Main conclusions

- Minimising the temperature of the LED component is known to enhance its light output and reliability. This research showed that despite technological advances, thermal management still remains an essential element of luminaire design. However, its role did appear to be becoming less critical. A combination of factors such as improving efficacy (which reduces waste heat) and reducing component cost (which makes it commercially viable to use more components at lower power density) have facilitated new strategies that appear to bypass dedicated thermal management. At the same time there is still a significant market segment employing high power density COB type LED arrays. This calls for high performance thermal management strategies. There is, therefore, still value in pursuing improvements to these systems.

- Passive heatsinks are inherently well suited to the demands of LED luminaire systems owing to their intrinsic reliability, low energy consumption and lack of noise so were chosen as the focus for this research. There are some active technologies that promise low energy consumption and enhanced convective heat transfer. These may potentially offer superior thermal management performance and should be explored in more detail. However, for the most immediate and widespread impact, improvements to passive heatsinks are believed to represent the most promising technology to develop.

- The long-term effects of some common operating environments on luminaire thermal management performance was assessed as part of this research. These initial results suggest that some change in the thermal properties does occur, but they were not definitive and longer term tests in a wider range of environments are needed.
• Commercially available passive heatsinks currently seem to be limited to conventional forms, probably as a result of the constraints of traditional manufacturing techniques but may also be due to a lack of knowledge regarding more advanced designs. These constraints are slowly becoming less appropriate as new processes such as additive manufacturing develop, presenting opportunities to explore new concepts. Computational simulation offers the freedom to explore these possibilities and extract more thermal management performance. To support this the industry requires clearly defined thermal management performance metrics, for which this investigation identified thermal resistance, heatsink material volume and average heat transfer coefficient. By using these criteria, a range of concepts were evaluated to demonstrate how superior designs can be developed.

• After applying a sequence of optimisation studies, the potential improvement of an existing commercially available component was predicted. This revealed a 20 % increase in average heat transfer coefficient and a 10 % lower heatsink mass or a 5 % lower thermal resistance could all be achieved with minimal development. These benefits were achieved when radiative heat transfer accounted for a relatively high proportion of the thermal power dissipated by the heatsink. As the thermal management performance enhancement was primarily a result of geometric changes that influence how the heatsink interacts with the cooling fluid of the surrounding environment (i.e. convective heat transfer), where radiative heat transfer makes a smaller contribution to performance (for example where reflective surface treatments are employed) the potential benefit could be greater. Enclosing the heatsink with a chimney structure offered a small (1 – 3 %) improvement in thermal management performance over an equivalent heatsink without the enclosing structure. If the enclosing structure was included in the evaluation of the heatsink’s thermal management performance the total material volume and average heat transfer coefficient were found to be severely compromised for the scenario considered. Its role as a protective shroud is also questionable owing to the low temperatures of the heatsink which mean there is very little risk of damage or injury arising from physical contact. Based on the results of this investigation there appears, therefore, to be very little value in pursuing this concept further. However, it is unclear how effective the chimney design was and so its benefit may not be indicative of that offered by a truly optimised design.
11.3 Impact of findings

The results of this work show there is potential to enhance established LED luminaire thermal management strategies. This offers a number of potential commercial and physical benefits. Firstly, for a given input power, an enhanced luminaire could dissipate more heat within the same physical constraints to reduce the junction temperature of the LED light source. As has been thoroughly reported in the literature, this would improve the LED component’s reliability and output. By utilising the improved thermal management to reduce the LED chip’s temperature its luminous efficacy can also be increased. This can significantly reduce the total energy consumed by the luminaire during its lifetime and the savings can outweigh any additional energy embodied by an increase in the (aluminium) heatsink material content. This means there are circumstances when it is environmentally beneficial to invest more resources during manufacturing to improve overall heatsink performance. Alternatively, for a specific LED junction temperature, increasing the luminaire’s capacity to dissipate waste heat can permit an increase in input power with an accompanying increase in light emission. This is desirable to meet ongoing commercial demands for higher output without the need for additional resources, i.e. more luminous flux from a luminaire for equal system capital cost. Finally, for the same operating conditions (i.e. same LED junction temperature and input power), enhanced thermal management could allow the luminaire’s size and material content to be reduced. This offers commercial and physical advantages, such as cost savings or improved aesthetics. As LED packages become cheaper, the cost of the associated thermal management represents a growing proportion of the total system cost, thus making reductions here increasingly valuable.

Thermal resistance, average heat transfer coefficient and heatsink material volume have been used throughout this work as simple but broad metrics to evaluate a system’s thermal management performance. In combination, these criteria provide an analysis of the system’s physical suitability, effectiveness, environmental impact and commercial value. This allows the evaluation and direct comparison of different concepts. From these it becomes clear which strategies show the most promise for development and where resources should be focussed. By implementing the design and optimisation techniques discussed, it was shown that the thermal management performance of an existing component can be significantly improved. This work represents the collection of knowledge and tools needed to reach this result. The reviewed literature did not provide any sources that combine the broad range of topics needed to develop an effective thermal management strategy in a commercial setting i.e. market demands, performance definition, test methods, environmental impacts, simulation parameters,
optimisation procedures and concept evaluation. A lack of understanding of these factors within commercial practice could lead to a failure to realise the full potential of a design. A single point of reference, combining all these relevant fields along with a demonstration of the commercial value, is needed to combat that failure. This work represents a significant contribution towards that need.

11.4 Limitations of this work

This research was intentionally kept as generic as possible, to have the most wide reaching impact. However, it was still necessary to restrict its scope in places to maintain a practical scale. Unfortunately this also limits the broader applicability of its findings. For instance, the analysis of the effects of exposure to a typical environment only evaluated a reflective, extruded aluminium heatsink. Alternative materials, manufactured by different processes and with different surface finishes may not react the same. Similarly, only a small number of operating conditions were simulated. The system’s behaviour under different circumstances may be considerably different. Based on the small effects and insensitivity observed in many of the simulation studies conducted here it is believed these results are representative of most typical cases. However, this cannot be assumed with confidence and so would need to be tested. The range of parameters where these conclusions remain valid is also unclear. For instance, adding heatsink material to improve thermal management was shown to have a beneficial impact on lifecycle energy consumption, but there is almost certainly a threshold when additional material is no longer utilised effectively. The findings cannot, therefore, reliably be extended to alternative situations.

A significant restriction was the narrow range and simplified treatment of heatsink parameters considered during the optimisation studies. In particular, the optimisation conducted in Chapter 9 considered the studied heatsink parameters in isolation. These were then integrated by evaluating their independent impact on heatsink thermal management performance, assessing where improvements could potentially be realised and negative consequences offset for an overall positive effect, and trialling different configurations until an improved solution was obtained. Multi-parameter optimisation studies were conducted in the final (Chapter 10) but considered a reduced range of variables based on the findings of the previous optimisation studies. This means it is possible the outcomes do not represent the maximum potential thermal management performance improvements available.
In this study the thermal performance of various systems was assessed and improvements identified under reasonably typical conditions. One result of trying to keep the work as generic as possible is that there are potential improvements that could be realised with more specific constraints. Under explicitly defined circumstances (i.e. a particular component, thermal load, maximum tolerable temperature and environment) there would be no need to accommodate any approximations or uncertainty. Safety margins could then be reduced to enable greater performance to be realised for that specific scenario.

11.5 Suggestions for further study

- The obvious next step for this research would be verification of the findings. This would involve the production of physical samples of the optimised designs, measurement of component temperatures under equivalent operating conditions, and the comparison of these results to the simulations reported here. The fluid flow behaviour displayed low velocity and few features, so its measurement for comparison would not be advised.

- Following verification, and presuming the conclusions of this work are confirmed, the logical progression would be to address the limitations discussed. Expanding on this work with a wider range of operating conditions, thermal loads, physical properties and heatsink geometries is needed to evaluate the potential for heatsink optimisation to deliver significant benefits under different circumstances. Evaluating larger thermal loads should be a priority owing to the greater necessity of effective thermal management under those conditions.

- Thermally conductive polymers present an interesting alternative to the assessed heatsink material. Their thermal conductivity is lower than conventional materials (Weber et al., 2003), but they offer a number of commercial benefits and the material’s physical limits can be overcome through design (Hussain et al., 2017).

- Many of the active thermal management technologies considered in Chapter 2 had limitations that made them poorly suited to the subject of this work. However, there were some that promised to enhance convective heat transfer without compromising the advantages of the LED luminaire. These were initially dismissed on the grounds of their supposed cost, complexity and the difficulty of integrating them into a luminaire. Exploring the potential benefits of these technologies for luminaire design relative to established techniques would be worthwhile.
Two common operating environments’ impact on a heatsink’s thermal properties were studied in this work but the mechanisms of any changes occurring were not identified. It may be that the heatsink’s surface area is best minimised to reduce the impact of corrosion and surface fouling, which potentially restrict heat dissipation during service. Similarly, specific surface finishes may perform more consistently throughout the luminaire’s design life in particular situations. In order to limit any undesirable evolution of thermal management performance, defining the nature of any relationships between the operating environment and specific features of a heatsink would be useful.

This study attempted to cover a broad range of possibilities, but in doing so it failed to conclusively optimise a system for any specific property. To extract the maximum thermal management performance requires any generalisations or approximations to be eliminated. However, restricting its scope risks it becoming too narrow to be useful. Instead, defining the uncertainty associated with different conditions would be recommended. Consequently, reasoned judgements could be made to address any uncertainty whenever critical or where additional effort may be beneficial.

The optimisation procedures applied in this investigation were extensively simplified. As a result, the outcomes may not fully realise the potential thermal management performance enhancements that could be achieved. Evaluating heatsink material content as an optimisation objective in its own right would be useful to understand if average heat transfer coefficient does actually represent an acceptable proxy. Multi-parameter and multi-objective optimisation would be desirable but its implementation should also be practical. Evaluating the potential improvement offered by refining the optimisation procedures with appropriate statistical methods (e.g. with factorial design of experiments and response surface methods) would be recommended to determine if the additional complexity can be justified.

An integral aspect of this investigation was its commercial relevance. It was concerned with practical techniques with the aim of maximising its impact. To enhance this, the development methodology should be refined and integrated with commercial product development practices. A simple tool to optimise a luminaire, without the need to understand the variables and principles involved, would open the potential improvements on offer to a far wider inexpert audience and, it might be hoped, accelerate the evolution of thermal management strategies.

The potential to exploit transient properties for thermal management would be an interesting subject for further study and one that had not been explored in the reviewed literature. By
defining luminaire operating regimes and evaluating the requirement for, or feasibility of, integrating thermal storage it would be possible to determine if this strategy has any practical value.

- There was some evidence in Chapter 8 to suggest augmenting a heatsink with a chimney structure had the potential to improve its thermal management performance. However, the implementation of this considered in Chapter 10 failed to realise any significant benefit. It was not clear if the studied chimney represented an optimum design and so it is difficult to dismiss the concept based solely on the findings of this investigation. The chimney’s behaviour needs to be understood in more detail to evaluate if its implementation can be improved. Exploring the impact of changing the chimney’s geometry and its relationship with the heatsink is, therefore, recommended.

- As a final comment, this work began from a suspicion that fluid flow could potentially be passively driven through a series of enclosed channels to achieve better heat transfer. The findings here suggest that any attempt to enclose the fluid flow compromises the system’s thermal management performance and so the concept shows very little potential. It might be that, under different conditions, a different conclusion could be reached. In addressing the limitations of this study, the potential opportunities to develop this concept should become clear.
References


http://www.ncbi.nlm.nih.gov/pmc/articles/PMC3215219/.


Appendices
Appendix A: Luminaire survey results
2013 luminaire survey results
Table A-1: 2013 luminaire survey results
Entry
Manufacturer / Product name
number Supplier

Manufacturer / Supplier reference

LED power Total
Power dissipation
category Luminous method
flux from
lamp (lm)

Thermal
managing
structure
material

Thermal
managing
structure
forming
process

General
release
date

1
2
3
4
5
6

Solow LED
Canolux
LEDBay
Olympus basic
iPlan
Prolix

SWL 15582
CX 14643L
LY 14904L
OPBSMW/MC2-1000-840/NF/AND
ME72+LED
PR0368F/444XSC028/036/0350

High
High
High
High
High
High

20600
11150
8070
5490
5147
4500

Passive heatsink
Passive heatsink
Passive heatsink
Body redistribution
Passive heatsink
Passive heatsink

Aluminium
Aluminium
Aluminium
Unspecified
Aluminium
Aluminium

Extruded
Extruded
Extruded
Unspecified
Extruded
Extruded

2013
n.d.
n.d.
2013
2013
n.d.

Olympus
XL-20
Magna

OP-BS-MW/MC1-810-80-40/22
XL 15389
MA0167F/4000SC028/018/0650

High
High
High

4446
3935
3156

Body redistribution
Body redistribution
Passive heatsink

Unspecified
Aluminium
Aluminium

Unspecified 2012
Circuit board 2012
Die-cast
n.d.

Nemesis
Frame
G4
Linealuce
Flute
G3
Illuceo 260

NS-LG-MW/JT1-3350-80-40/60
MF97+LED
GF 15104
BB83+LED
SH 15157L
GT 15071
IL04402/4000WF070/016/0700

High
High
High
High
High
High
High

2909
2542
2340
2209
2000
1996
1970

Passive heatsink
Active fan
Passive heatsink
Body redistribution
Passive heatsink
Passive heatsink
Passive heatsink

Aluminium

Die-cast

Aluminium
Aluminium
Aluminium
Aluminium
Aluminium

Die-cast
Extruded
Extruded
Extruded
Die-cast

2012
2013
n.d.
2013
n.d.
n.d.
n.d.

DOT 14794
VNSMMS/JT2-2000-840/VW/AND
VNSMMW/JT1-1950-80-40/110
MF49+LED
MF25+LED
MJ78+LED

High
High
High
High
High
High

1810
1808
1697
1558
1558
1540

Body redistribution
Body redistribution
Body redistribution
Passive heatsink
Passive heatsink
Passive heatsink

Aluminium
Unspecified
Unspecified
Aluminium
Aluminium
Aluminium

Sheet metal
Unspecified
Unspecified
Die-cast
Die-cast
Die-cast

n.d.
2013
2012
2013
2013
2013

LRF1MW/VN2-1600-840/S/AND
MMXOMB/JT2-2000-840/VW/AND
M650+LED
MMX0MB/JT1-2150-80-40/110
CYADMW/VN2-1600-840/NF/AND
M977+LED
VTMXMW/MC2-1000-840/S/AND
LR-A1-MW/VS1-1000-80-40/22
PT-F1-MW/VS1-1000-80-40/22
VTMDMW/MC2-1000-840/S/AND
LTA1MW/MC2-1000-840/NF/AND
CSEDMW/MC2-1000-840/NF/AND
CDR1/MW/MC221000/840/NF/AND
SSASMW/MC2-1000-840/NF/AND
CY-AD-MW/VS1-1000-80-40/22
TA-AD-MW/VS1-1000-80-40/22
VSU1MW/MC2-1000-840/S/AND
SRLGMW/AR2-1000-840/WF/AND
VT-MD-MW/AR1-875-80-40/50
VT-MX-MW/AR1-875-80-40/50
CHS3MW/MC2-1000-840/NF/AND
SR-LG-MW/AR1-875-80-40/50
SS-AS-MW/MC-810-80-40/22
LT-A1-MW/MC1-810-40/22
CS-ED-MW/MC1-810-80-40/22
CD-R1-MW/MC1-810-80-40
VS-U1-MW/MC1-810-80-40/22
CHS3MW/MC1-810-80-40/32
GU 15407
VTMOMW/PT2-420-840/NF/AND

High
High
High
High
High
High
High
High
High
High
High
High
High

1389
1335
1334.8
1286
1270
1074
926
925
925
922
915
915
915

Passive heatsink
Body redistribution
Body redistribution
Body redistribution
Body redistribution
Passive heatsink
Body redistribution
Passive heatsink
Passive heatsink
Body redistribution
Passive heatsink
Passive heatsink
Body redistribution

Aluminium
Unspecified
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Aluminium
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Die-cast
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High
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High
High

915
881
881
860
842
800
800
782
763
741
741
741
741
741
706
644
473

Body redistribution
Passive heatsink
Passive heatsink
Body redistribution
Body redistribution
Passive heatsink
Passive heatsink
Body redistribution
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Thorlux
Thorlux
Thorlux
Photonstar
iGuzzini
Integrated
System
Technologies
Photonstar
Thorlux
Integrated
System
Technologies
Photonstar
iGuzzini
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iGuzzini
Thorlux
Thorlux
Integrated
System
Technologies
Thorlux
Photonstar
Photonstar
iGuzzini
iGuzzini
iGuzzini

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Photonstar
Photonstar
iGuzzini
Photonstar
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Photonstar
Photonstar
Photonstar
Photonstar

Dot
Votan
Votan
Frame
Minimal
Palco medium
body
lorem
Maxi Muro
Primopiano
Maxi Muro
Cryos
Deep laser
Venturi maxi
Laser
Phaeton
Venturi
Lightslot
Ceilingstar ED
Cordus 1

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Photonstar
Photonstar
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Photonstar
Photonstar
Photonstar
Photonstar
Thorlux
Photonstar

Aero
Cryos
Tesla
Vespertine
Simetra
Venturi
Venturi Maxi
Chime
Simetra
Aero
Lightslot
Ceilingstar ED
Cordus 1
Vespertine
Chime
G2
venturi micro

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## 2015 luminaire survey results

### Table A-2: 2015 luminaire survey results

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<thead>
<tr>
<th>Entry number</th>
<th>Manufacturer / Supplier</th>
<th>Product name</th>
<th>Manufacturer / Supplier reference</th>
<th>LED power category</th>
<th>Total Luminous flux from lamp (lm)</th>
<th>Power dissipation method</th>
<th>Thermal managing structure material</th>
<th>Thermal managing structure forming process</th>
<th>General release date</th>
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<td>Aluminium</td>
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<td>HB1-250W-NW</td>
<td>High</td>
<td>25000</td>
<td>Passive heatsink</td>
<td>Aluminium</td>
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<td>LXCAC012E43WH5</td>
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<td>Laser Blade high contrast</td>
<td>P193</td>
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<td>Sheet metal</td>
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<td>Centro LED PRO 120</td>
<td>5400004</td>
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<td>EKIP1630/WB</td>
<td>Medium</td>
<td>16300</td>
<td>Body redistribution</td>
<td>Steel</td>
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<td>Corine Trunking 140X</td>
<td>140X</td>
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<td>BY740P</td>
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<td>FS484F</td>
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<td>MLED5950/NW/WC</td>
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<td>9500</td>
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<td>Extrusion</td>
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<td>Pacific LED</td>
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<td>iNH 60</td>
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<td>Arano LED</td>
<td>BC6440</td>
<td>Medium</td>
<td>4250</td>
<td>Body redistribution</td>
<td>Aluminium</td>
<td>Extrusion</td>
<td>2015</td>
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</table>
Appendix B: Heatsink geometries and simulation results

Case 1: Parallel plate heatsink

Table B-1: Parallel plate heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter value (m, x10^{-3})</th>
<th>Exposed heatsink surface area (m^2, x10^{-3})</th>
<th>Heatsink material volume (m^3, x10^{-6})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parallel plate fins (conventional geometry). Suitable for production by extrusion processes</td>
<td>Inter-fin spacing (fin quantity) 18.66 (3 fins)</td>
<td>27.83</td>
<td>51.90</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inter-fin spacing (fin quantity) 10 (5 fins)</td>
<td>42.95</td>
<td>73.50</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inter-fin spacing (fin quantity) 6.29 (7 fins)</td>
<td>58.07</td>
<td>95.10</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inter-fin spacing (fin quantity) 4.22 (9 fins)</td>
<td>73.19</td>
<td>116.7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inter-fin spacing (fin quantity) 2.91 (11 fins)</td>
<td>88.31</td>
<td>138.30</td>
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Fig. B-1: Thermal behaviour of parallel plate heatsink models
Case 2: Radial plate heatsink

Table B-2: Radial plate heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (Degrees)</th>
<th>Exposed heatsink surface area ($m^2 \times 10^{-3}$)</th>
<th>Heatsink material volume ($m^3 \times 10^{-6}$)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial plate fins (conventional geometry).</td>
<td>Angular fin spacing (fin quantity)</td>
<td>45.00° (8 fins)</td>
<td>32.89</td>
<td>56.38</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Suitable for production by die-casting processes</td>
<td></td>
<td>30.00° (8 fins)</td>
<td>43.13</td>
<td>69.66</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>25.73° (14 fins)</td>
<td>50.73</td>
<td>79.86</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>22.50° (16 fins)</td>
<td>66.06</td>
<td>88.85</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>18.00° (20 fins)</td>
<td>69.39</td>
<td>104.44</td>
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Fig. B-2: Thermal behaviour of radial plate heatsink models
**Case 3: Spiral plate heatsink**

Table B-3: Spiral plate heatsink models

<table>
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<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (Degrees)</th>
<th>Exposed heatsink surface area ($m^2 \times 10^3$)</th>
<th>Heatsink material volume ($m^3 \times 10^6$)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spiral plate fins (conventional geometry). Suitable for production by die-casting processes</td>
<td>Angular fin spacing (fin quantity)</td>
<td>45.00° (8 fins)</td>
<td>32.10</td>
<td>55.22</td>
<td>![Image]</td>
<td>![Image]</td>
<td>![Image]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>30.00° (12 fins)</td>
<td>45.56</td>
<td>73.30</td>
<td>![Image]</td>
<td>![Image]</td>
<td>![Image]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>25.73° (14 fins)</td>
<td>53.47</td>
<td>83.37</td>
<td>![Image]</td>
<td>![Image]</td>
<td>![Image]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>22.50° (16 fins)</td>
<td>60.42</td>
<td>93.13</td>
<td>![Image]</td>
<td>![Image]</td>
<td>![Image]</td>
</tr>
<tr>
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<td></td>
<td>18.00° (20 fins)</td>
<td>74.50</td>
<td>110.70</td>
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<td>![Image]</td>
<td>![Image]</td>
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Fig. B-3: Thermal behaviour of spiral plate heatsink models
**Case 4: Diagonal plate heatsink**

**Table B-4: Diagonal plate heatsink models**

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<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (m, x10^3)</th>
<th>Exposed heatsink surface area (m^2, x10^3)</th>
<th>Heatsink material volume (m^3, x10^6)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
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<td>Diagonal plate fins (conventional geometry). Suitable for production by die-casting processes</td>
<td>Inter-fin spacing (fin quantity)</td>
<td>9.00 (13 fins)</td>
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<td>68.47</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>7.50 (13 fins)</td>
<td>49.84</td>
<td>77.37</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.00 (17 fins)</td>
<td>57.14</td>
<td>86.61</td>
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<td></td>
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<tr>
<td></td>
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<td>4.50 (21 fins)</td>
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<td>103.40</td>
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<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.00 (29 fins)</td>
<td>90.53</td>
<td>127.15</td>
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Fig. B-4: Thermal behaviour of diagonal plate heatsink models
## Case 5: Staggered pin heatsink

Table B-5: Staggered pin heatsink models

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<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (m, x10^{-3})</th>
<th>Exposed heatsink surface area (m^2, x10^{-3})</th>
<th>Heatsink material volume (m^3, x10^{-6})</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Staggered pin fins (conventional geometry). Suitable for production by die-casting processes</td>
<td>Centre-centre pin spacing (pin quantity)</td>
<td>11.00 (27 pins)</td>
<td>20.42</td>
<td>30.95</td>
<td>[Image]</td>
<td>[Image]</td>
<td>[Image]</td>
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<td>9.00 (45 pins)</td>
<td>30.60</td>
<td>38.59</td>
<td>[Image]</td>
<td>[Image]</td>
<td>[Image]</td>
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<td>7.00 (77 pins)</td>
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<td>52.16</td>
<td>[Image]</td>
<td>[Image]</td>
<td>[Image]</td>
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<td>5.50 (127 pins)</td>
<td>76.97</td>
<td>73.36</td>
<td>[Image]</td>
<td>[Image]</td>
<td>[Image]</td>
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<td>5.00 (163 pins)</td>
<td>97.32</td>
<td>88.63</td>
<td>[Image]</td>
<td>[Image]</td>
<td>[Image]</td>
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![Fig. B-5: Thermal behaviour of staggered pin heatsink models](image)

- Thermal resistance (Vertical heatsink orientation)
- Thermal resistance (Inverted heatsink orientation)
- Average heat transfer coefficient (Vertical heatsink orientation)
- Average heat transfer coefficient (Inverted heatsink orientation)
Case 6: Staggered pin with open centre plate heatsink

Table B-6: Staggered pin with open centre heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value</th>
<th>Exposed heatsink surface area (m², x10⁻²)</th>
<th>Heatsink material volume (m³, x10⁻⁶)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
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<tbody>
<tr>
<td>Staggered pin fins with open centre (conventional geometry). Suitable for production by die-casting processes</td>
<td>Pins removed from centre of array (pin quantity)</td>
<td>45 pins removed (32 pins remaining)</td>
<td>23.25</td>
<td>33.07</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>31 pins removed (46 pins remaining)</td>
<td>31.16</td>
<td>39.01</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>23 pins removed (54 pins remaining)</td>
<td>35.69</td>
<td>42.40</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>13 pins removed (64 pins remaining)</td>
<td>41.34</td>
<td>46.64</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7 pins removed (70 pins remaining)</td>
<td>44.73</td>
<td>49.19</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. B-6: Thermal behaviour of staggered pin with open centre heatsink models
Case 7: Stepped staggered pin heatsink

Table B-7: Stepped staggered pin heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (Degrees)</th>
<th>Exposed heatsink surface area (\text{m}^2, x10^{-3})</th>
<th>Heatsink material volume (\text{m}^3, x10^{-6})</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stepped staggered pin fins (conventional geometry). Suitable for production by die-casting processes</td>
<td>Step down angle</td>
<td>55°</td>
<td>30.86</td>
<td>38.49</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>45°</td>
<td>36.15</td>
<td>42.59</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>35°</td>
<td>39.87</td>
<td>45.46</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>25°</td>
<td>42.80</td>
<td>47.69</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>15°</td>
<td>45.29</td>
<td>49.59</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. B-7: Thermal behaviour of stepped staggered pin heatsink models
**Case 8: Capped radial plate heatsink**

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (m, $x10^{-3}$)</th>
<th>Exposed heatsink surface area ($m^2, x10^{-3}$)</th>
<th>Heatsink material volume ($m^3, x10^{-6}$)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capped radial plate fins (novel geometry). Suitable for production by lost-wax casting processes</td>
<td>Aperture diameter</td>
<td>55.00</td>
<td>53.48</td>
<td>83.46</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>45.00</td>
<td>54.11</td>
<td>85.18</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>35.00</td>
<td>54.43</td>
<td>86.44</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>25.00</td>
<td>54.65</td>
<td>87.38</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>15.00</td>
<td>54.77</td>
<td>88.01</td>
<td></td>
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<td></td>
</tr>
</tbody>
</table>

**Table B-8: Capped radial plate heatsink models**

**Fig. B-8: Thermal behaviour of capped radial plate heatsink models**
**Case 9: Mesh heatsink**

**Table B-9: Mesh heatsink models**

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (m, x10⁻³)</th>
<th>Exposed heatsink surface area (m², x10⁻³)</th>
<th>Heatsink material volume (m³, x10⁻⁶)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mesh (novel geometry). Suitable for production by lost-wax casting processes</td>
<td>Channel width</td>
<td>12.50</td>
<td>34.31</td>
<td>49.20</td>
<td><img src="top_view.png" alt="Top view" /></td>
<td><img src="side_view.png" alt="Side view" /></td>
<td><img src="isometric_view.png" alt="Isometric view" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>10.00</td>
<td>41.39</td>
<td>53.70</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>7.50</td>
<td>51.41</td>
<td>66.75</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.00</td>
<td>60.01</td>
<td>75.00</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.50</td>
<td>77.61</td>
<td>87.00</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. B-9: Thermal behaviour of mesh heatsink models
**Case 10: Vertical tube heatsink**

Table B-10: Vertical tube heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter value (m², x10⁻⁶)</th>
<th>Varied parameter value (m², x10⁻³)</th>
<th>Exposed heatsink material volume (m³, x10⁻⁶)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical tube</td>
<td>Tube cross-sectional area (tube quantity)</td>
<td>714.00 (4 tubes)</td>
<td>4.91</td>
<td><img src="image1" alt="Cross-sectional Area" /></td>
<td><img src="image2" alt="Side View" /></td>
<td><img src="image3" alt="Isometric View" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>282.72 (9 tubes)</td>
<td>55.61</td>
<td><img src="image4" alt="Cross-sectional Area" /></td>
<td><img src="image5" alt="Side View" /></td>
<td><img src="image6" alt="Isometric View" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.63 (16 tubes)</td>
<td>65.05</td>
<td><img src="image7" alt="Cross-sectional Area" /></td>
<td><img src="image8" alt="Side View" /></td>
<td><img src="image9" alt="Isometric View" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>78.96 (25 tubes)</td>
<td>73.21</td>
<td><img src="image10" alt="Cross-sectional Area" /></td>
<td><img src="image11" alt="Side View" /></td>
<td><img src="image12" alt="Isometric View" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>47.65 (36 tubes)</td>
<td>80.11</td>
<td><img src="image13" alt="Cross-sectional Area" /></td>
<td><img src="image14" alt="Side View" /></td>
<td><img src="image15" alt="Isometric View" /></td>
</tr>
</tbody>
</table>

**Fig. B-10: Thermal behaviour of vertical tube heatsink models**
### Case 11: Helical plate heatsink

#### Table B-11: Helical plate heatsink models

<table>
<thead>
<tr>
<th>Description</th>
<th>Varied parameter</th>
<th>Varied parameter value (Degrees)</th>
<th>Exposed heatsink surface area ($m^2$, x10^-3)</th>
<th>Heatsink material volume ($m^3$, x10^-6)</th>
<th>Top view</th>
<th>Side view</th>
<th>Isometric view</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helical plate fins (novel geometry)</td>
<td>Fin sweep angle</td>
<td>40°</td>
<td>51.92</td>
<td>79.36</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>56°</td>
<td>53.43</td>
<td>79.42</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>72°</td>
<td>55.19</td>
<td>79.28</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>90°</td>
<td>57.65</td>
<td>79.41</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>108°</td>
<td>60.49</td>
<td>79.40</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Suitable for production by lost-wax casting processes.

#### Fig. B-11: Thermal behaviour of helical plate heatsink models

- Thermal resistance (Vertical heatsink orientation)
- Thermal resistance (Inverted heatsink orientation)
- Thermal resistance (Case 2 baseline model)
- Average heat transfer coefficient (Vertical heatsink orientation)
- Average heat transfer coefficient (Inverted heatsink orientation)
- Average heat transfer coefficient (Case 2 baseline model)
Appendix C: Air properties

Table C-1: Properties of dry air at standard atmospheric pressure and a temperature of 300 K (Holman, 2010)

<table>
<thead>
<tr>
<th>Density (kg.m(^{-3}))</th>
<th>Specific heat capacity (J.kg(^{-1}).K(^{-1}))</th>
<th>Dynamic viscosity (kg.m(^{-1}).s(^{-1}).x10(^{-5}))</th>
<th>Thermal conductivity (W.m(^{-1}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1774</td>
<td>1005.7</td>
<td>1.8462</td>
<td>0.02624</td>
</tr>
</tbody>
</table>

Appendix D: Preliminary parametric optimisation results

Table D-1: Predicted heatsink thermal properties with varying fin thickness

<table>
<thead>
<tr>
<th>Fin thickness (m, x10(^{-3}))</th>
<th>Exposed heatsink surface area (m(^2), x10(^{-3}))</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance (K.W(^{-1}))</th>
<th>Average heat transfer coefficient (W.m(^{-2}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.50</td>
<td>40.00</td>
<td>54.402</td>
<td>4.32</td>
<td>5.79</td>
</tr>
<tr>
<td>1.00</td>
<td>38.94</td>
<td>54.562</td>
<td>4.33</td>
<td>5.93</td>
</tr>
<tr>
<td>1.50</td>
<td>37.99</td>
<td>54.963</td>
<td>4.36</td>
<td>6.03</td>
</tr>
<tr>
<td>2.00</td>
<td>37.29</td>
<td>55.616</td>
<td>4.41</td>
<td>6.08</td>
</tr>
<tr>
<td>2.50</td>
<td>36.75</td>
<td>56.209</td>
<td>4.46</td>
<td>6.10</td>
</tr>
<tr>
<td>3.00</td>
<td>36.19</td>
<td>56.491</td>
<td>4.48</td>
<td>6.16</td>
</tr>
<tr>
<td>3.50</td>
<td>35.58</td>
<td>56.807</td>
<td>4.51</td>
<td>6.23</td>
</tr>
<tr>
<td>4.00</td>
<td>35.00</td>
<td>57.178</td>
<td>4.54</td>
<td>6.31</td>
</tr>
<tr>
<td>4.50</td>
<td>34.20</td>
<td>57.589</td>
<td>4.57</td>
<td>6.40</td>
</tr>
<tr>
<td>5.00</td>
<td>33.64</td>
<td>57.978</td>
<td>4.60</td>
<td>6.46</td>
</tr>
</tbody>
</table>

Table D-2: Predicted heatsink thermal properties with varying fin spacing

<table>
<thead>
<tr>
<th>Fin spacing (m, x10(^{-3}))</th>
<th>Exposed heatsink surface area (m(^2), x10(^{-3}))</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance (K.W(^{-1}))</th>
<th>Average heat transfer coefficient (W.m(^{-2}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>69.07</td>
<td>70.263</td>
<td>5.58</td>
<td>2.60</td>
</tr>
<tr>
<td>2.00</td>
<td>56.69</td>
<td>66.692</td>
<td>5.29</td>
<td>3.33</td>
</tr>
<tr>
<td>3.00</td>
<td>48.86</td>
<td>61.034</td>
<td>4.84</td>
<td>4.23</td>
</tr>
<tr>
<td>4.00</td>
<td>43.49</td>
<td>57.380</td>
<td>4.55</td>
<td>5.05</td>
</tr>
<tr>
<td>5.00</td>
<td>39.48</td>
<td>56.195</td>
<td>4.46</td>
<td>5.68</td>
</tr>
<tr>
<td>6.00</td>
<td>36.75</td>
<td>56.244</td>
<td>4.46</td>
<td>6.10</td>
</tr>
<tr>
<td>7.00</td>
<td>34.24</td>
<td>57.357</td>
<td>4.55</td>
<td>6.42</td>
</tr>
<tr>
<td>8.00</td>
<td>31.88</td>
<td>58.770</td>
<td>4.66</td>
<td>6.73</td>
</tr>
<tr>
<td>9.00</td>
<td>30.65</td>
<td>59.720</td>
<td>4.74</td>
<td>6.88</td>
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<td>10.00</td>
<td>29.50</td>
<td>60.328</td>
<td>4.79</td>
<td>7.08</td>
</tr>
</tbody>
</table>

Table D-3: Predicted heatsink thermal properties with varying central cut-out offset

<table>
<thead>
<tr>
<th>Central cut-out offset (m, x10(^{-3}))</th>
<th>Exposed heatsink surface area (m(^2), x10(^{-3}))</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance (K.W(^{-1}))</th>
<th>Average heat transfer coefficient (W.m(^{-2}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>35.40</td>
<td>56.668</td>
<td>4.50</td>
<td>6.28</td>
</tr>
<tr>
<td>2.50</td>
<td>35.92</td>
<td>56.413</td>
<td>4.48</td>
<td>6.22</td>
</tr>
<tr>
<td>5.00</td>
<td>36.37</td>
<td>56.411</td>
<td>4.48</td>
<td>6.14</td>
</tr>
<tr>
<td>7.50</td>
<td>36.75</td>
<td>56.276</td>
<td>4.47</td>
<td>6.09</td>
</tr>
<tr>
<td>10.00</td>
<td>37.04</td>
<td>56.143</td>
<td>4.46</td>
<td>6.06</td>
</tr>
<tr>
<td>12.50</td>
<td>37.27</td>
<td>56.240</td>
<td>4.46</td>
<td>6.01</td>
</tr>
<tr>
<td>15.00</td>
<td>37.47</td>
<td>56.161</td>
<td>4.46</td>
<td>5.99</td>
</tr>
<tr>
<td>Suppressed</td>
<td>37.60</td>
<td>56.172</td>
<td>4.46</td>
<td>5.97</td>
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</tbody>
</table>

Table D-4: Predicted heatsink thermal properties with varying central cut-out diameter

<table>
<thead>
<tr>
<th>Central cut-out diameter (m, x10(^{-3}))</th>
<th>Exposed heatsink surface area (m(^2), x10(^{-3}))</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance (K.W(^{-1}))</th>
<th>Average heat transfer coefficient (W.m(^{-2}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.00</td>
<td>37.37</td>
<td>56.125</td>
<td>4.45</td>
<td>6.09</td>
</tr>
<tr>
<td>10.00</td>
<td>37.17</td>
<td>56.126</td>
<td>4.45</td>
<td>6.04</td>
</tr>
<tr>
<td>15.00</td>
<td>36.98</td>
<td>56.167</td>
<td>4.46</td>
<td>6.07</td>
</tr>
<tr>
<td>20.00</td>
<td>36.75</td>
<td>56.196</td>
<td>4.46</td>
<td>6.10</td>
</tr>
<tr>
<td>25.00</td>
<td>36.37</td>
<td>56.303</td>
<td>4.47</td>
<td>6.15</td>
</tr>
<tr>
<td>30.00</td>
<td>35.84</td>
<td>56.404</td>
<td>4.48</td>
<td>6.23</td>
</tr>
<tr>
<td>35.00</td>
<td>35.26</td>
<td>56.635</td>
<td>4.49</td>
<td>6.31</td>
</tr>
</tbody>
</table>
Table D-5: Predicted heatsink thermal properties with varying central cut-out taper angle

<table>
<thead>
<tr>
<th>Central cut-out taper angle (Degrees)</th>
<th>Exposed heatsink surface area ( m^2 \times 10^{-3} )</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance ( K.W^{-1} )</th>
<th>Average heat transfer coefficient ( W.m^{-2}.K^{-1} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>37.04</td>
<td>56.225</td>
<td>4.46</td>
<td>6.05</td>
</tr>
<tr>
<td>10°</td>
<td>36.96</td>
<td>56.162</td>
<td>4.46</td>
<td>6.07</td>
</tr>
<tr>
<td>20°</td>
<td>36.88</td>
<td>56.229</td>
<td>4.46</td>
<td>6.08</td>
</tr>
<tr>
<td>30°</td>
<td>36.75</td>
<td>56.143</td>
<td>4.46</td>
<td>6.11</td>
</tr>
<tr>
<td>40°</td>
<td>36.52</td>
<td>56.347</td>
<td>4.47</td>
<td>6.12</td>
</tr>
<tr>
<td>50°</td>
<td>36.16</td>
<td>56.344</td>
<td>4.47</td>
<td>6.18</td>
</tr>
<tr>
<td>60°</td>
<td>35.56</td>
<td>56.707</td>
<td>4.50</td>
<td>6.25</td>
</tr>
</tbody>
</table>

Table D-6: Predicted heatsink thermal properties with varying fin height

<table>
<thead>
<tr>
<th>Fin height ((m, \times 10^{-3}))</th>
<th>Exposed heatsink surface area ( m^2 \times 10^{-3} )</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance ( K.W^{-1} )</th>
<th>Average heat transfer coefficient ( W.m^{-2}.K^{-1} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>19.03</td>
<td>83.647</td>
<td>6.64</td>
<td>7.91</td>
</tr>
<tr>
<td>3.00</td>
<td>21.55</td>
<td>81.252</td>
<td>6.45</td>
<td>7.20</td>
</tr>
<tr>
<td>5.00</td>
<td>24.01</td>
<td>79.179</td>
<td>6.28</td>
<td>6.63</td>
</tr>
<tr>
<td>7.00</td>
<td>26.42</td>
<td>75.556</td>
<td>6.00</td>
<td>6.31</td>
</tr>
<tr>
<td>9.00</td>
<td>28.66</td>
<td>70.005</td>
<td>5.56</td>
<td>6.28</td>
</tr>
<tr>
<td>11.00</td>
<td>30.81</td>
<td>66.163</td>
<td>5.25</td>
<td>6.18</td>
</tr>
<tr>
<td>13.00</td>
<td>32.88</td>
<td>62.318</td>
<td>4.95</td>
<td>6.15</td>
</tr>
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Table D-7: Predicted heatsink thermal properties with varying base thickness

<table>
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<tr>
<th>Base thickness ((m, \times 10^{-3}))</th>
<th>Exposed heatsink surface area ( m^2 \times 10^{-3} )</th>
<th>Peak heatsink temperature relative to ambient environment (K)</th>
<th>Thermal resistance ( K.W^{-1} )</th>
<th>Average heat transfer coefficient ( W.m^{-2}.K^{-1} )</th>
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