Advanced Natural Convection Cooling Designs for Light-Emitting Diode Bulb Systems

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The movement to light-emitting diode (LED) lighting systems worldwide is accelerating quickly as energy savings and reduction in hazardous materials increase in importance. Government regulations and rapidly lowering prices help to further this trend. Today's strong drive is to replace light bulbs of common outputs (60 W, 75 W, and 100 W) without resorting to compact fluorescent (CFL) bulbs containing mercury while maintaining the standard industry bulb size and shape referred to as A19. For many bulb designs, this A19 size and shape restriction forces a small heat sink which is barely capable of dissipating heat for 60 W equivalent LED bulbs with natural convection for today's LED efficacies. 75 W and 100 W equivalent bulbs require larger sizes, some method of forced cooling, or some unusual liquid cooling system; generally none of these approaches are desirable for light bulbs from a consumer point of view. Thus, there is interest in developing natural convection cooled A19 light bulb designs for LEDs that cool far more effectively than today's current designs. Current A19 size heat sink designs typically have thermal resistances of 5–7 °C/W. This paper presents designs utilizing the effects of chimney cooling, well developed for other fields that reduce heat sink resistances by significant amounts while meeting all other requirements for bulb system design. Numerical studies and test data show performance of 3–4 °C/W for various orientations including methods for keeping the chimney partially active in horizontal orientations. Significant parameters are also studied with effects upon performance. The simulations are in good agreement with the experimental data. Such chimney-based designs are shown to enable 75 W and 100 W equivalent LED light bulb designs critical for faster penetration of LED systems into general lighting applications. [DOI: 10.1115/1.4028331]

Keywords: LED, cooling, natural convection, heat sink, chimney, electronics cooling, A19, bulb

Introduction

Over the last decade, LED lighting has been growing at a rapid rate, replacing many other traditional light sources in various applications. This trend is expected to continue for the next decade as more applications make use of solid state lighting (SSL) technology. The driving forces toward adoption of SSL technology include energy conservation, reduction of fluorescent technology to reduce hazardous waste (specifically mercury), higher degrees of control over light color temperatures and direction, and government/industry regulations.

One particular area of strong interest is adoption of SSL for regular light bulbs. The large number of bulb sockets in the world currently populated with relatively inefficient incandescent and halogen technology offers a chance for considerable energy savings. Recognizing this, Europe is phasing in regulations to eliminate most incandescent bulb usage and the United States is following with a 1–2 yr delay [1,2]. CFLs have filled some of this gap but concerns remain with long term hazardous waste disposal due to mercury, and some customer acceptance has been limited due to perceived color differences from traditional incandescent sources.

As of late 2012, several LED bulbs have appeared in the market. These have attempted to replicate the light output and distribution of 40 W and 60 W equivalent incandescent lamps and also attempted to conform to the common bulb shape and outline known as A19 in the lighting industry and defined by ANSI [3]. Success in meeting these criteria has been mixed as LEDs are inherently directional light sources and light bulb distribution patterns have been omnidirectional. Cooling requirements for LEDs have further required larger heat sinks as the light output has increased to 60 W equivalent (along with 75 W and 100 W equivalent bulbs now beginning to be sold).

The most desired solution for LED bulbs is to have a bulb that is cooled only by natural convection, fits within the A19 envelope, and is available in the common incandescent equivalent sizes of 60 W, 75 W, and 100 W. Further requirements are soon to be promulgated for light output, lifetime/reliability, lumen maintenance and many other categories from regulatory agencies. The United Statese is in process as of late 2012 with a draft standard for bulbs authored by the government Environmental Protection Agency under a new ENERGY STAR® certification program for all light bulb technologies (includes LED, CFL, and halogen technologies) [4].

The diverse and numerous requirements upon LED light bulbs have created a difficult design issue. On the one hand, the luminous flux (measured in lumens, or lm) required for lamp classes is specified (800 lm for 60 W equivalent, 1100 lm for 75 W, and 1600 lm for 100 W in the U.S.; European values vary slightly) but the current LED and power supply efficacy create power dissipation that is difficult to do within the A19 envelope without specialized cooling (e.g., active) and meet many other bulb requirements. For this reason, bulb designs dependent only upon passive air cooling are not currently found above 60 W for the A19 envelope.

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Note that one bulb shown by SWITCH bulbs does meet the envelope for 75 W and 100 W products but does so with an internal liquid cooling bath; it is not cooled solely by air. A 100 W LED bulb announced by GE uses a synthetic jet to stay within the A19 size. The Philips LED bulbs for 75 W and 100 W are a larger A21 size.

Therefore, the state of the art challenge to SSL bulb design is to find thermal cooling architectures that can effectively work to cool up to 100 W equivalent bulbs and still fit within the industry A19 size.

Current Art

It is worthwhile to examine the current design of LED light bulbs and understand thermal designs and limitations. The bulbs are essentially composed of three regions: a base at the bottom for installing into a luminaire, a central region which is a heat sink and contains space for the electrical driver (power supply), and the upper region which is the optical portion of the bulb where the LEDs reside and some type of optical beam spreading system.

The three main regions' bulb construction is shown Fig. 1 for a typical LED bulb design. The heat sink designs for various bulbs are a variety of styles but incorporate vertical or mostly vertical fins for heat dissipation purposes.

Current System Performance and Effectiveness. To determine the effectiveness of a typical LED bulb, several commercially available bulbs were bought and tested to determine heat sink and system performance. There were two criteria chosen to evaluate them. First, the heat sink itself was evaluated for its convective thermal resistance. This is a straightforward calculation using the average surface temperature, power dissipated, and ambient temperature. The heat sinks showed nearly isothermal conditions under test when examined with an infrared imaging camera (a commercial FLIR SC620 was used in this investigation).

Second, a modified dimensionless parameter was chosen to evaluate the bulb system level performance. In standard dimensionless analysis, the Biot number is defined by

$$\mathrm{Bi} = hL/k \tag{1}$$

This number is often examined to understand the conduction of heat in a solid near its surface compared to the convection off the surface into the surrounding bulk fluid. In this paper, a system level definition will be used, where the thermal resistance to conduction in the solid materials of the lamp is compared to the convection loss at the interface to the air. In reality, the loss at the lamp boundary is a combination of convection (dominant, around 75%) and radiation (around 25%), but the author will lump both effects into a convection term for simplicity (note that for high power lamps in the 100 W class radiation can become 35–40% of the heat transfer). The Biot system equation is similar to the standard definition and is

$$\mathrm{Bi}_{\mathrm{sys}} = \frac{L/kA_{\mathrm{cond}}}{1/hA_{\mathrm{conv}}} = \frac{\text{conductive resistance}}{\text{convective resistance}} \tag{2}$$

where L is the average length from the heat source to the boundary, and the areas A are for the conductive path area in the heat sink and for the convective path the area at the air boundary. In practice, this ratio of resistances can be reduced to a ratio of conductive temperature path drop to the convective temperature path drop when the heat is transferred through the heat sink.

Similar to the standard Biot number, a $\mathrm{Bi}_{\mathrm{sys}}$ number much less than 1 (<0.1) indicates a system where convection resistances are the dominant resistance. On the opposite scale, large $\mathrm{Bi}_{\mathrm{sys}}$ numbers (>10) indicate conduction resistances are the primary resistance factors for heat transfer.

Four typical commercially available lamps similar to those in Fig. 1 were tested, and the system thermal resistances and Biot numbers were evaluated. The lamps were all 40 W class except one 60 W class lamp was tested. The lamp results are shown in Table 1 and provide a baseline performance level of current technology. An average surface temperature for these nearly uniform temperature heat sinks is found using IR images and evaluating an area-weighted average temperature on the heat sink surfaces. Emissivities (found by using thermocouples to correlate to IR images) were between 0.85 and 0.95, depending upon whether the surfaces were anodized aluminum (around 0.85) or a painted surface (0.92–0.95).

Inadequate Performance for Higher Output Bulbs. To understand temperature limits for LED bulb systems, some background information is needed. LEDs are rated at maximum junction temperatures by the manufacturers, but both light output and lifetime are typically reduced at these temperatures. Hence, most designs use a lower design temperature, and this is a compromise based on cost of system components, desired bulb life, and overall light output. The drivers used to convert line voltage to the LED voltage and current also have temperature limitations. As such, the author cannot give an absolute temperature limit for any bulb, but in general, most systems try to keep the LED junction temperature at 100–110 °C or less. In the driver region, electrolytic capacitors (if used) are normally the lowest rated part at 105 °C maximum.

While the performance levels in Table 1 are acceptable for 40 W equivalent bulbs, they reach effective limits with the 60 W class bulb. At the 60 W equivalent lumen output level, the waste heat generated causes the LED packages to be near their maximum thermal limits when operating at the ENERGY STAR® requirement of a 55 °C ambient temperature in Elevated Temperature Life Testing [4]. A different type of thermal solution than currently used is needed for 75 W and 100 W equivalent light bulbs. This is based on the observation that an LED bulb typically uses 70 lm/W (this can range from 50 to 90 lm/W for different products), and for 1600 lm output (the 100 W equivalent) the electrical input power will be near 23 W. Of this input power, 18 W of this will be thermal dissipation assuming the internal driver is an industry standard 85% efficient. A 5.5 °C/W heat sink would have a thermal resistance near 4.5 °C/W at higher power values due to increased radiation transfer. Even with a 4.5 °C/W convective heat sink resistance, convection alone will create an 81 °C rise

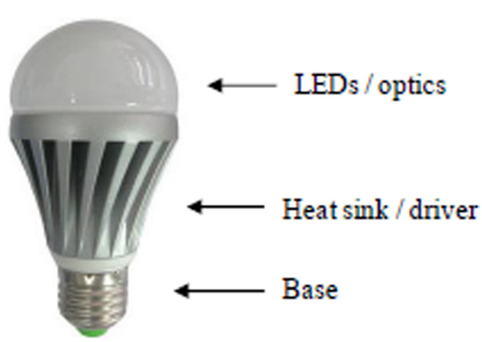

Fig. 1 LED bulb construction

Table 1 LED bulb heat sink performance

Parameter	R_θ, °C/W, range	$\mathrm{Bi}_{\mathrm{sys}}$, range
LED lamps, range	5.5–6.9	0.05–0.10

Table 2 Typical LED bulb heat loads

Bulb equivalent (W)	Luminous flux (lm)	Electrical power (W)	Thermal power (W)
60	800	11.4	9.0
75	1100	15.7	12.4
100	1600	22.8	18.0

over ambient, which is unacceptably high. Typical heat loads are given in Table 2.

Chimney Solutions

One potential solution for this performance shortfall is to consider chimney type designs. Chimneys have existed long before the modern era and been adopted for use in electronics cooling in various applications. Perhaps the earliest example of modern research in this area is that of Ellenbaas with his work examining free convection of parallel plates and vertical tubes with parallel walls in the 1940s [5,6]. In subsequent decades, the research has continued including up to the present time. For example, in the 1970s and 1980s much work was conducted around shrouded heat sink concepts, though the work has continued to the present time. Typical examples can be found in Refs. [7–15], which cover individual component cooling through large systems of cabinet cooling. Basic description of chimney physics has been detailed in discussions of the stack effect [16].

However, a typical LED bulb design lacks one primary element for a chimney design—there is no central core opening. Since many chimneys are of cylindrical shape, this suggests that a design with a cylindrical light guide for the LEDs could be created and allow for a central thermal chimney.

Annular Chimney Design (V3). One proposal for such a solution is shown in Figs. 2 and 3. This bulb assembly still has the same base in the same location, but the LEDs, optical elements and the heat sink all occupy the same general region of the bulb. There is a solid central core about 26 mm in diameter, then an open annular region, which comprises the through chimney, and an outer area with the LEDs, light guide, and other support structure. The entire assembly fits within the A19 envelope defined by the ANSI standard. This particular design was designated prototype V3 (version 3).

As seen in Fig. 2, a prototype bulb was constructed. The main heat sink was made by the lost wax casting process with a standard aluminum casting alloy. Other parts except the base were machined, and different printed circuit boards (PCBs) were created for different testing conditions. Two types of PCBs were created: one with actual LEDs, and a second with surface mount

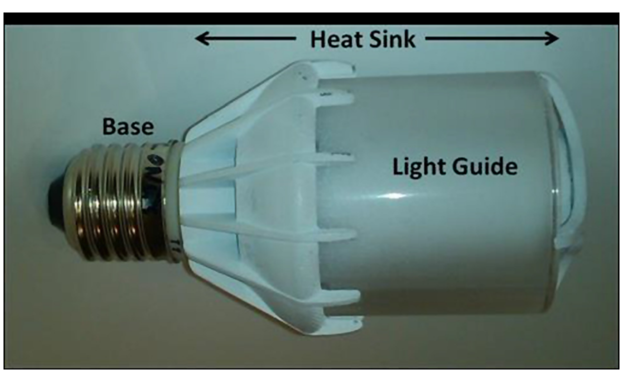

Fig. 2 Prototype V3 assembly

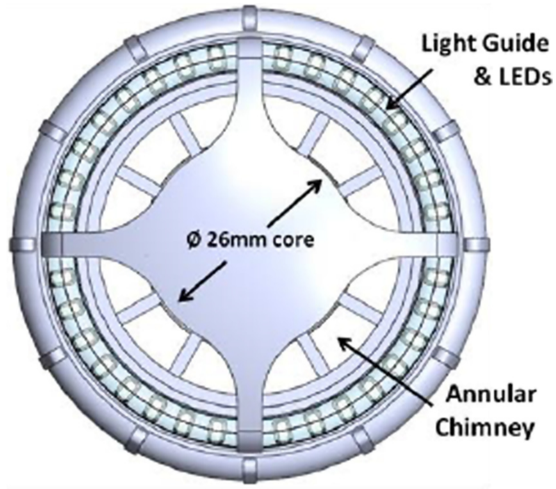

Fig. 3 V3 top view with annular chimney

resistors. The latter design allowed for more accurate measurement of thermal input energy. LEDs can be used but accurately knowing the thermal input energy is difficult; one must accurately measure the radiometric light output energy for an input electrical energy to measure the difference, and light energy can be reabsorbed into the system and become an input load. Further discussion of this issue is found in Ref. [17].

Performance. The V3 design was tested to understand overall system performance and compared to computational fluid dynamics (CFD) numerical solutions using a commercially available CFD code (FloEFD version 10.1 from Mentor Graphics). Key temperature data were obtained using thermocouples and a commercially available infrared imaging camera (FLIR model SC620). Figure 4 shows a typical test setup.

The tests used 11.8 W thermal input energy at the base of the light guide via a PCB. Tests were conducted with the bulb in three orientations—vertical up, vertical down, and horizontal. The bulb was screwed into a standard E26 screw base and held in a vise in open air. Two key locations were instrumented with type T thermocouples (36 ga, or ~0.13 mm diameter)—one on the heat sink external surface, below the PCB area, and a second on the vertical chimney inside the light guide just above the PCB area (these locations correspond to the two upper notations in Fig. 6). IR

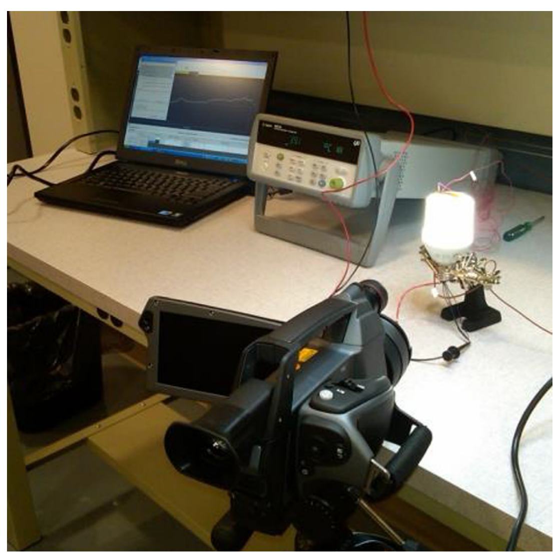

Fig. 4 Test setup (typical)

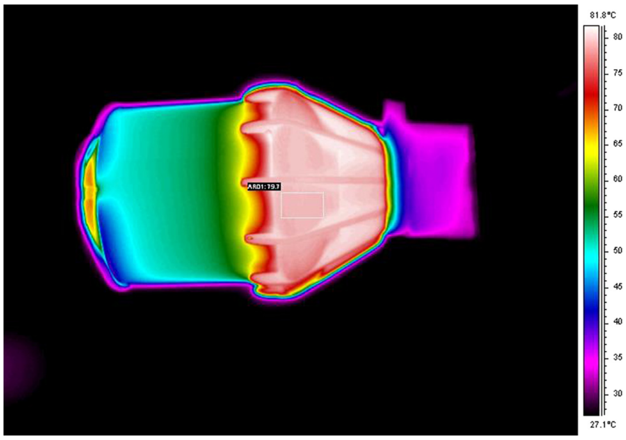

Fig. 5 IR image of V3 in horizontal position

Table 3 Test and analysis correlations (°C)

Test case	Location	*T/C*	IR image	CFD
Vert up	Outer HS	*See text	73.7	73.6
	Chimney	71.6	n/a	72.5
Vert down	Outer HS	74.4	75.2	74.4
	Chimney	71.3	n/a	73.0
Horizontal	Outer HS	78.1	80.7	80.1
	Chimney	75.9	n/a	78.6

Table 4 V3 bulb performance, vertical up

Parameter	R_θ (°C/W)	Bi_{sys}
V3 chimney	4.6	0.04

images were taken of steady state conditions (no more than 0.1 °C change in 15 min, accounting for any ambient temperature changes). Measurement errors for type T class 2 thermocouples are ±1.0 °C in this temperature range, and for the IR camera ±2.0 °C. TC and IR results for experiments were within the 3 °C spread these errors could create. These errors can cause up to a 4% error in the temperature difference calculations from a 50 °C temperature rise above ambient. Measurements were very repeatable by TC or by IR during testing; temperatures measured, again accounting for ambient changes, never varied by more than 0.1 °C (assuming for TCs that proper mounting was used).

Results are shown below. Figure 5 is an IR image of the horizontal test case, and Fig. 6 is the corresponding CFD image (in Fig. 6, all parts are hidden except the heat sink for clarity, though all parts were used in the model). Table 3 gives the correlations between the two methods corrected to ambient of 20 °C ("n/a" indicates not applicable as the IR camera cannot view this location). Table 4 is performance of the heat sink for thermal resistance and the system Bi_{sys} number.

CFD simulations were conducted to correlate to the various tests. In the vertical orientations, about 150,000 cells were used and horizontally about 100,000 cells were used. Sufficient room above and below the lamp (as defined by the gravity vector) is used for proper flow development, and one bulb diameter around the sides was adequate for spacing around the bulb without influencing results unduly. Mesh sensitivity studies were conducted primarily by increasing the number of partial cells the CFD code uses, which refines the mesh around the fluid to solid boundaries (partial cells are part solid, part fluid, and a unique cell used by the commercial CFD code FloEFD, version 10.1, in these simulations). Maximum cell counts of five to six hundred thousand cells were solved and compared to coarser meshes; results were found to be 0.1–0.2 °C different from the coarser meshes so the coarser meshes were deemed acceptable. In all CFD simulations, radiation was selected as part of the solution routine. The heat sinks were painted with a special white paint and the emissivity was measured to be 0.975 on both the heat sinks and a special flat panel painted sample.

A few pertinent observations should be made from these results. First, the thermocouple used to measure the outside heat sink temperature in the Vertical up orientation was not attached properly and is quite sensitive in this orientation (the "*" entry in Table 3). Although this was found later, the test was not repeated though in other orientations the thermocouples gave reliable results. Other temperature differences are within instrument errors. Second, there is a noticeable improvement in the heat sink thermal resistance (nearly 20% better than any of the commercial bulbs tested), but the Bi_{sys} number remained low. Third, horizontal performance is worse—not surprising, since chimneys are meant for vertical operation. As seen in Fig. 7, angles from 45 deg to 0 deg showed significant increases in temperature for the heat sink (0 deg indicates horizontal, and 90 deg is vertical up). This graph is based on

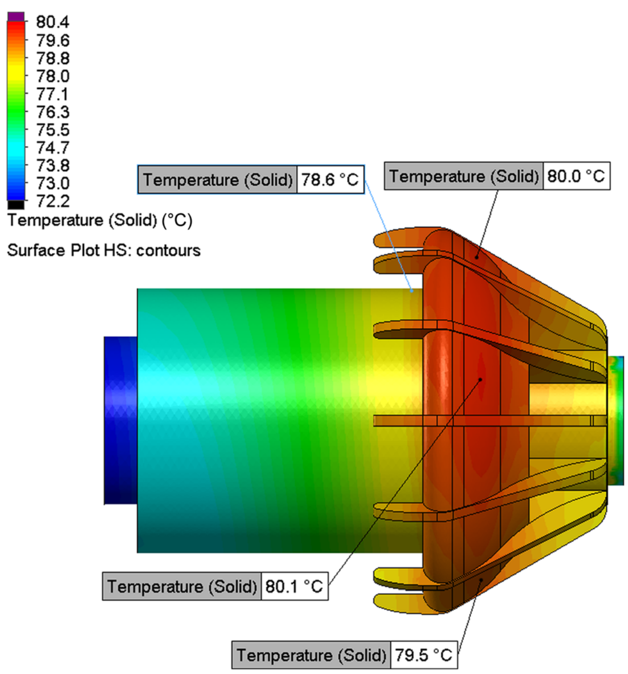

Fig. 6 CFD image of V3 horizontal position (only heat sink part shown for clarity)

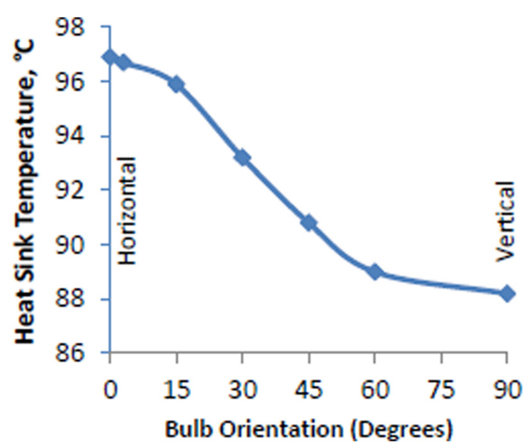

Fig. 7 V3 design orientation sensitivity

Fig. 8 Prototype V6 assembly

the verified CFD model dissipating higher power levels at these angles. Variations between vertical (90 deg) and tipped to 45 deg showed only modest increases in overall thermal resistance.

Other variations of this V3 design were modeled to see how much improvement could be made over this design. They included varying the number of internal fins in the annular chimney and varying the fin height. The best designs were close to this V3 performance and showed that once fully developed flow was reached in the chimney (as seen in the CFD simulation by thermal boundary layers coalescing in the annular ring), little improvement could be made. Further, horizontal performance was poor due lack of flow in the chimney. It is clear this design has limitations.

Chambered Design With Annular Chimney (V6). Given the limitations of the V3 design, another solution was sought. The vertical solutions needed to be better, and some method to improve the horizontal system performance would be needed for bulbs with higher power levels than used for tests in V3.

An advanced chimney system was devised and prototyped, still remaining within the design outline of an A19 bulb. This system involves a unique chamber internal to the chimney yet is open to the lamp bottom, sides and top. The annular chimney is thus split into separate chimneys—in this case, the “Y” shaped chamber creates the three of them—to allow the chamber access to the various sides of the bulb. This heat sink geometry is shown in Figs. 8–11.

What is notable in this design is that the volume for any bulb driver was reduced to only the lower part of the bulb. This allowed the chimney to be built with the chamber in the upper part of the design, only partially blocked with a 15 mm core.

Performance. Similar to the V3 prototype, the V6 prototype was built and tested. A resistor PCB was used to allow higher

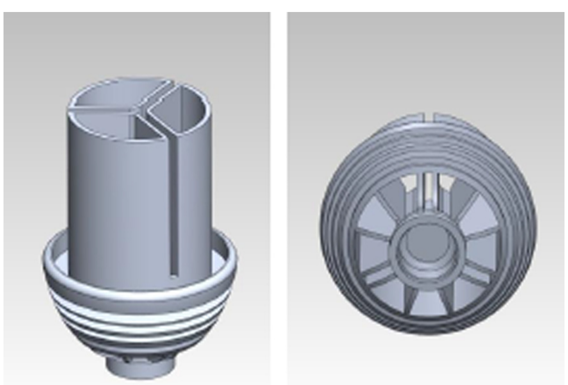

Fig. 9 V6 heat sink detail

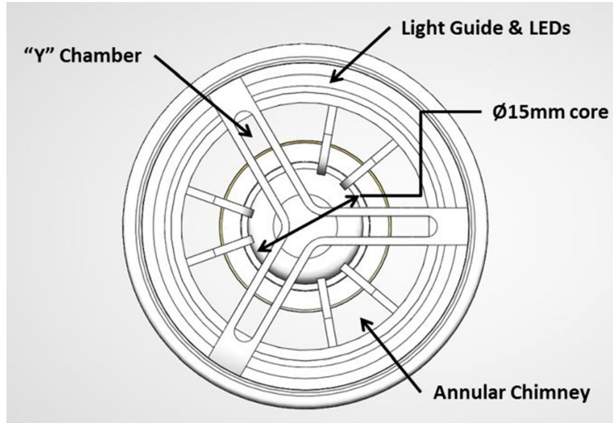

Fig. 10 V6 assembly top view

power levels for testing (up to 21 W). Verification tests (via IR camera and thermocouples) and CFD analyses were again performed, similar to the V3 testing scheme. Agreement between tests and CFD results were as good as or better than V3 (better thermocouple attachments were used).

Test results at 11.9 W of thermal power showed that the V6 heat sink thermal resistance is 4.0 °C/W. This is an improvement of 13% over the V3 performance at a similar power input. It is a fair assessment to state the efficacy of a chamber and chimney type system is a large improvement over the typical LED bulb design (a one third reduction over the typical LED lamp tested). The results for the typical bulbs and the V3 and V6 prototypes are shown in Table 5.

The significant improvement is the improvement in system thermal resistance though the Bi_{sys} number remains similar. There is still room to improve the convective path, and the conductive path is relatively similar to before (the closer the Bi_{sys} number is to zero, the more the heat sink remains isothermal). At this point in the design improvement process, it seemed reasonable to think the Bi_{sys} number could be improved with better convection paths.

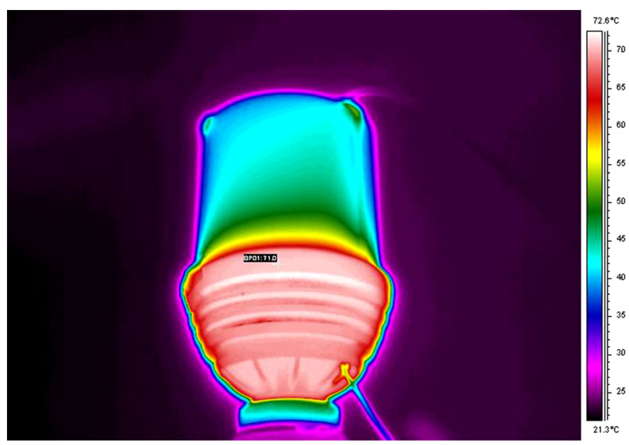

Fig. 11 IR image of V6 assembly

Table 5 Summary of results, vertical up

Parameter	R_θ (°C/W)	Bi_{sys}
LED lamps, range	5.5–6.9	0.05–0.10
V3 chimney	4.6	0.04
V6 chamber	4.0	0.05

Table 6 V3 and V6 heat sink differences by orientation

Parameter	Vert up R_θ (°C/W)	Horiz R_θ (°C/W)
V3 chimney	4.6	5.34
V6 chamber	4.0	4.86

The performance improvements were created by better air flow patterns in the design. Vertically there are strong drafts created in the chimney and chamber sections. Velocities 200 mm above the bulb reached nearly 0.6 m/s in the CFD simulation, indicating a strong draft created by the bulb design, and over 10% higher than the V3 design. Reducing the LED driver core size opens the annular region in V6 to permit greater airflow, along with optimizing the flow paths.

Furthermore, the chamber design has an advantage over the pure chimney design in the horizontal orientations. As noted earlier, one drawback of a pure chimney design is poor horizontal performance. The V3 design horizontal R_θ gains 0.74 °C/W–5.34 °C/W. In the V6 design, there is a slightly larger difference, but the overall system performance is significantly better than V3. The V6 temperature gain is 10.2 °C, and the thermal resistance gain is 0.86 °C/W. Table 6 shows a summary of the performance differences in the orientations.

For horizontal use, the chamber construction is designed for external air to pass through the "Y" shape. From a CFD analysis in one horizontal orientation, one can see the air flow through the chamber as shown in Fig. 12.

As expected, the chamber provides cooling in horizontal orientations that standard chimney designs such as V3 cannot. However, one surprising finding from simulation was that the chamber created the movement of air into the chimney regions when horizontal which provided more cooling. The velocity vector plot of Fig. 13 shows this air flow (seen near the top cap in the right of the figure). Figure 14 shows a similar view with color contours.

This air flow inducing effect of the chamber will be studied in a later paper. It is primarily due to the chamber creating a particular draft that imparts momentum to surrounding air and pushes this external air into the surrounding chimneys.

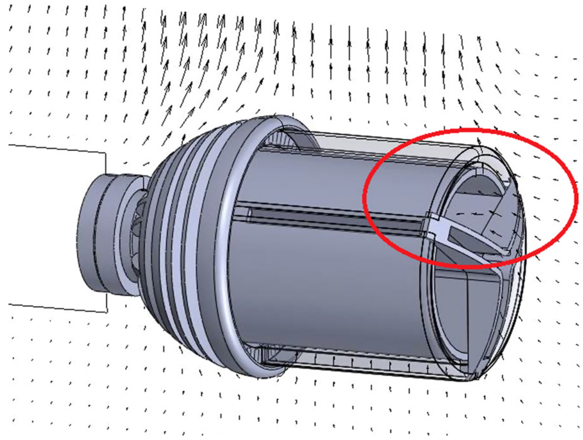

Fig. 13 Air flow into chimney core

Chambered Design With External Fins (V8). While V6 is better than V3, it was clear the thermal performance is not enough for a 100 W equivalent bulb dissipating 18 W. At 4 °C/W and a 55 °C ambient, the boundary condition temperature for the LED PCB would be 127 °C, and the resulting junction temperature would likely be 135 °C or higher (exact temperature would depend on the LED model and drive current applied). To keep the junction temperature below 120 °C (a common maximum), the heat sink should not exceed 110 °C. This 55 °C rise over ambient for 18 W applied means the heat sink resistance should not exceed 3 °C/W as a design goal.

To achieve this, a similar heat sink to V6 was created with a 2 mm larger outer diameter for the annulus outer core, and external fins added outside the light guide section. The lower heat sink section was redesigned for better inlets. Other dimensions were kept the same as V6 and the overall outline was kept within the A19 envelope. Even with the addition of the fins and larger chimney annulus, the heat sink areas are nearly identical between V6 and V8 (39,035 versus 38,705 mm^2). Figures 15–18 show the V8 bulb and heat sink.

Performance. The V8 prototype was tested and simulated at a number of power levels and orientations. A resistor PCB was used with input powers of approximately 6, 9, 13, 17, and 21 W, and simulations were conducted with 9, 13, and 21 W of input power. IR and thermocouple data were within 1.5 °C for all tests. Simulation mesh dependency studies were conducted similar to the process described for V3 to ensure adequate mesh density for the simulations. Figures 19 and 20 show some of the test and simulation results.

The V8 prototype performed significantly better than the V6 design. As seen in Fig. 20 below, the results show reasonable agreement for the tests and the simulations. By performing an examination across a wider range of thermal input powers, a general performance diagram of the heat sink can be generated. Figure 20 shows results for the vertical up orientation. The test data is the

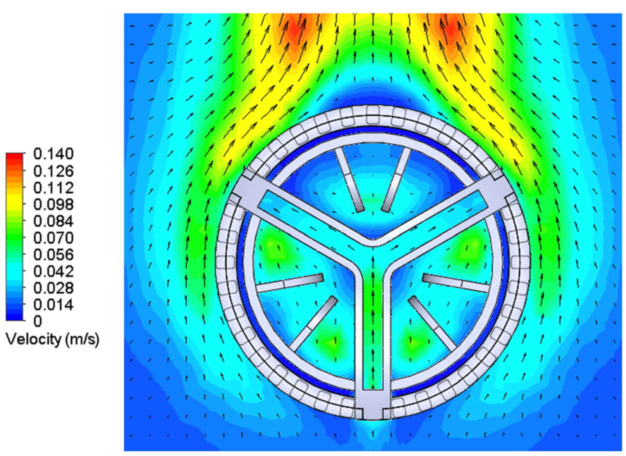

Fig. 12 Air flow in chamber, horizontal orientation

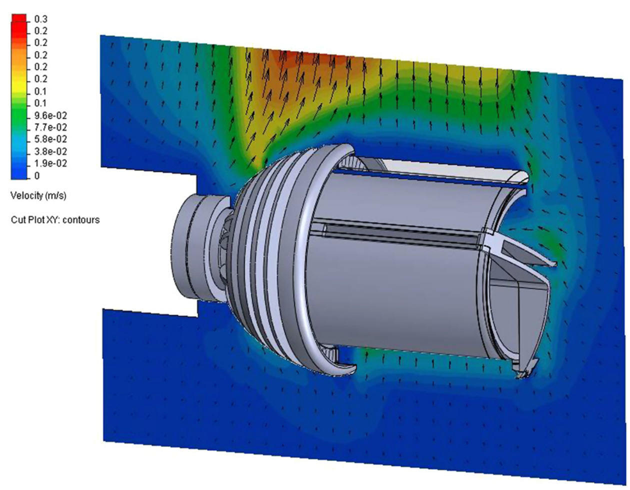

Fig. 14 Velocity vector and contour plot

Fig. 15 Prototype V8 assembly

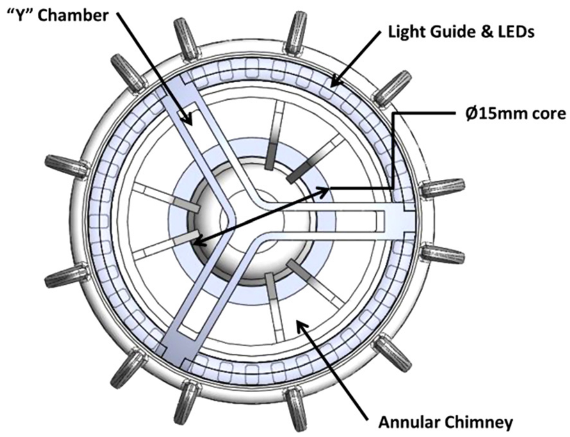

Fig. 16 V8 assembly top view

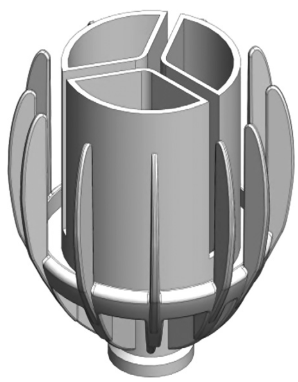

Fig. 17 V8 heat sink detail

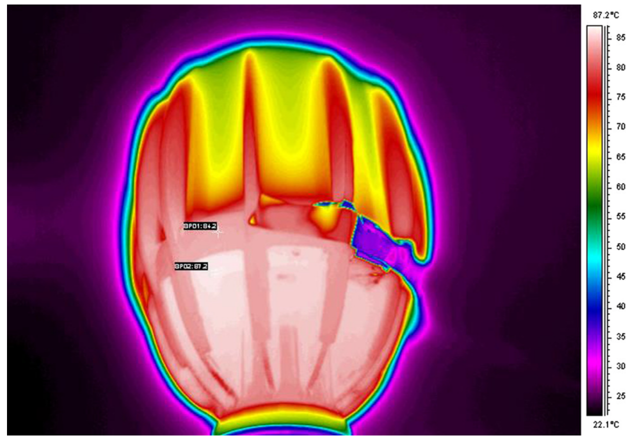

Fig. 18 IR image of V8 vertical up test

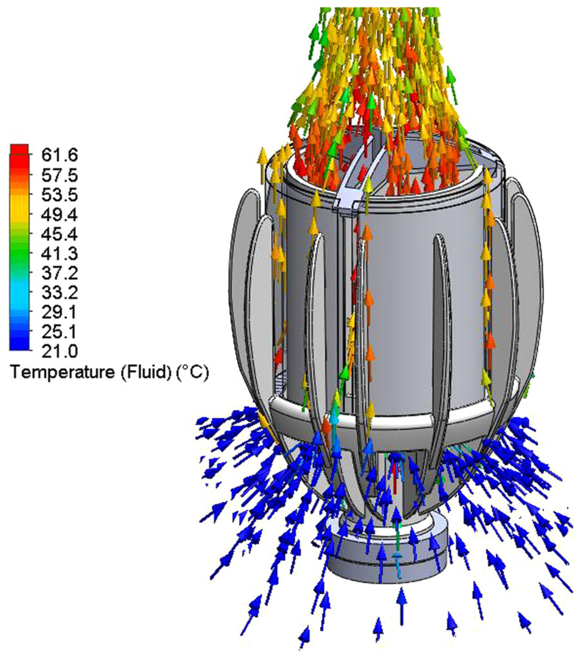

Fig. 19 CFD simulation of V8 vertical up test

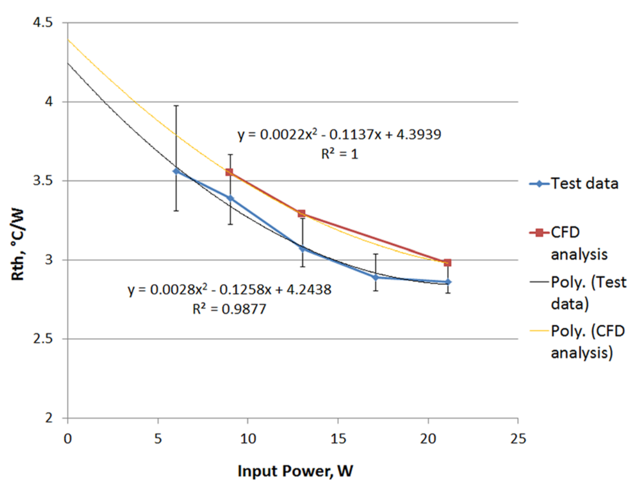

Fig. 20 Performance of V8 in vertical up orientation

average of the thermocouple and the IR image data, and the error estimates based on the possible worst errors for the type of measurement. For each data set, a second-order polynomial curve fit was applied and the equations shown.

A few important observations are found from this performance chart. The simulation solutions are conservative compared to the actual tests though close the upper end of the experimental error band. Small air currents in the lab may account for this as the simulation assumes perfectly still air. The outer fins are very effective at removing heat in the presence of low air currents. Second, the V8 design performance is a large improvement over the V6 design. At 11.9 W of input power, the heat sink resistance is about 3.1 °C/W, almost a 25% improvement in the vertical orientation. At the higher power levels for 100 W equivalent bulbs (18 W input), the heat sink tests just below 3 °C/W, meeting the design target. Third, the two key changes in the V8 design account for the improved performance—the larger annular region (2 mm larger outer diameter) and the fins. Simulations show the outer

Table 7 Summary of results, vertical up

Parameter	R_θ (°C/W)	Bi_{sys}
LED lamps, range	5.5–6.9	0.05–0.10
V3 chimney	4.6	0.04
V6 chamber	4.0	0.05
V8 chamber	3.1	0.08

Table 8 Heat sink differences by orientation (12 W power)

Parameter	Vert up R_θ (°C/W)	Horiz R_θ (°C/W)
V3 chimney	4.6	5.34
V6 chamber	4.0	4.86
V8 chamber	3.1	3.83

fins account for about two-thirds of the improvements, and the larger annular region the other one-third.

Tables 7 and 8 provide a full summary of the V8 prototype performance compared to previously examined bulb designs. The values for these tables use approximately 12 W of input power for V3, V6, and V8 designs. The commercial LED lamps ranged from approximately 7 W (40 W equivalent bulb) to 13 W (60 W equivalent). As seen in Table 7, the Bi_{sys} number improved in the V8 design but still is in the general range of other LED bulbs. Even as the convective resistances of the new designs are an improvement over the commercial units tested, it is still a significantly higher resistance than the conductive resistance in the system. Finally, the horizontal performance of the V8 prototype is also significantly improved over the V6 design. The external fins provide significant new cooling paths in the horizontal orientation.

Conclusions

Several interesting results have been found during this study. First, the current design of LED bulbs performs adequately for the current power dissipations but will not be enough for future 75 and 100 W equivalent bulbs. Second, rather than a standard central LED engine design, a cylindrical LED layout and light guide with a chimney enhances thermal performance and can still fit within the desired A19 design envelope. The metrics of a system thermal resistance and the proposed Bi_{sys} number provide a reasonable way of assessing total performance and where the best enhancements may lie. Last, a novel chimney and chamber design was developed for enhanced performance. Unusual air flows were noted as well in horizontal positions and will be evaluated in future work.

Though the V8 prototype design is far better than current designs, it is a bit marginal of a system that will adequately cool the 100 W equivalent light bulb at 2.9 °C/W. Future work will look at designs beyond the types shown in this paper (and beyond this paper's scope) that again reduce the overall system thermal resistance and lead to higher Bi_{sys} numbers—allowing a 100 W natural convection cooled LED bulb to fit within the A19 envelope.

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Nomenclature

Bi = Biot number, dimensionless
cond = conduction
conv = convection
h = heat transfer coefficient (W/m^2-K)
k = thermal conductivity (W/m-K)
L = characteristic length (m)
R = thermal resistance (°C/W)
sys = system
θ = temperature, thermal resistance

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