Heat Management of UV-LED Panels for Agricultural Applications

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Efficient heat management in high-power UV-LED panels needs urgent attention due to their applications in agriculture industry. LED light source panels are being employed to illuminate plants as UV light has shown potential to speed up germination process for starting seeds. Recent experiments reveal rapid growth of plants in UV environment. One main problem encountered during LED operation is the amount of heat that is generated especially when high power LED panels are used. Both active and passive cooling techniques do exist that include use of circulating cooling liquids, electric fans, and heat sinks. In the proposed work, we are investigating new methods to improve the cooling process by combining various passive cooling techniques. For this purpose, two passive cooling methods that incorporate a heat sink and several heat pipes simultaneously have been tested. Conventional heat transfer equations have been used to develop a theoretical MATLAB based model that gave an optimum heat sink geometry and predicted improved heat transfer rates for this optimum design. Experimental results show that the combined heat sink and heat pipes were able to reduce LED light panel temperature by 56.8%.

I. Introduction

Light Emitting Diodes (LEDs) are the most recent and most energizing innovative progression in the lighting industry. Among many applications, LED are being employed in agriculture sector to illuminate plants as UV light has shown potential to speed up germination process for starting seeds [1]. LEDs are semiconductor devices that produce visible light when an electrical current is passed through them. These lights are a type of Solid State Lighting (SSL), as are organic light emitting diodes (OLEDs), and light emitting polymers (LEPs). These emitting diodes are little, bright lights with great energy efficiency and durability. LEDs work uniquely in contrast to conventional lighting like incandescent or fluorescent light bulbs [2,3]. They also are "directional" light sources, which means they emit light in a specific direction, unlike incandescent and compact fluorescent bulbs, which emit light and heat in all directions. LED technology also offers many other advantages over incandescent, neon and minimal fluorescent lighting gadgets -, for example, extraordinarily longer life expectancy (60,000 hours), lower usage of energy (90% more proficient), higher safety, and reduced maintenance cost [4].

Thermal dissipation is a key variable that constrains the lumen output of a LED light. These bulbs are 80% more energy efficient than traditional incandescent lighting, however the LED parts and the driver hardware still produce a substantial amount of heat. If this heat is not dissipated properly, the LED's quality of light and life expectancy decrease dramatically. For this purpose, both active and passive cooling techniques do exist that include use of circulating cooling liquids, electric fans, and heat sinks. However, active cooling techniques need power and extra maintenance that make them unattractive for agriculture applications where hundreds of LED light panels are employed. Electronic industry has successfully used passive cooling methods where heat sinks are designed and created to properly reduce heat dissipation of electronic equipment. Most heat sinks solve thermal management problems for low-lumen LED lamps. Lighting manufacturers have had little difficulty developing viable 40W-equivalent LED retrofits for larger light panels. In cases where the high lumen counts that thermal management becomes a challenge. An average heat sink alone will not cool a 75W- or 100W-equivalent lamp [5].

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In the proposed work, we are investigating new methods to improve the cooling process by combining various passive cooling techniques. For this purpose, a heat sink along with heat pipes was tested in the lab. Heat pipes were mounted inside the heat sink that was attached to a base plate holding the LED light panel. In addition to experimental work, an attempt has been made to estimate the heat transfer rate for an optimized heat sink suitable for this application. For this purpose, a MATLAB model was developed. Modeling efforts were mostly based on the work that has been cited in literature [6,7,8]. Following section include further details on the equipment, experimental setup, and both experimental and theoretical test results that were obtained in this study.

II. Experimental Set up

The LED panel used in this testing was a LEDLinX (#AB0038) panel. This panel contains fourteen LED's and has an operating temperature of -40 to 90 °C. This LED panel was supplied with 25W, with a range of error of +/- 1W, for all experiments. 14 K-type thermocouples were chosen as the thermal sensors for the experiments, because they are relatively inexpensive, have a large temperature range, and have a suitable range of error, about +/- 2°C, for these experiments. The thermocouples were connected to a Key Sight (Agilent) 34970A Data Acquisition Device (DAQ). This DAQ was used to take readings of all 14 thermocouples every ten seconds and store the data within the internal memory. This DAQ was chosen because it has built in signal conditioning for the thermocouple reading, a large storage of data, and the ability to many channels at once. The DAQ system is paired with the BenchVue software provided by Keysight. This program allowed the data to be directly imported to a user-friendly interface to easily log, analyze, and export data.

The heat sink employed in this work was machined from a billet of 6061 aluminum alloy. It incorporated 13 tapered fins, along with holes in the base into which thermocouples were inserted, Figure 1. This alloy of aluminum was chosen for its thermal properties, cost, and availability. Special holes were drilled to mount seven heat pipes inside the top flat side of the heat sink. Heat pipes used in this study consists of a sealed copper tube with a controlled, low pressure atmosphere, which allows the liquid inside to boil at lower temperatures, Figure 2. When heat enters the heat pipe at its evaporator end, it vaporizes the fluid that exist at a low pressure. The vaporized fluid creates a pressure gradient that forces the vapor towards the condenser end. Vapor travels from the evaporator to condenser through the adiabatic section. Heat exits the heat pipe at the condenser end where the working fluid condenses and releases its latent heat of vaporization. The central section of the heat pipe is filled with porous wick structure. The wick acts as a pump using capillary pressure to return the fluid back to the evaporator [5,6].

III. Theoretical Considerations

In order to keep the LED panel as cool as possible a well-designed heat sink needs to be used to conduct heat from the panel to the fins of the heat sink to then be transferred to the surrounding air via convection. Since the ultimate goal of this project is to find the most efficient passive method of cooling, only natural convection will be used. The heat transfer rate by natural convection from the heat sink can be modeled by Equation 1.

$$\boldsymbol{Q_{dot} = eff_{fin} hnA(T_s - T_{infinity})} \tag{1}$$

Where eff_{fin} is the effectiveness of the fin, h is the heat transfer coefficient, n is the number of fins, $A_t$is the total area of the fins and base spacing, and T_S and T_{inf} are the surface and ambient temperatures. Based on the dimensions of the LED panel, as well as the data gathered from the experiments with the LED panel, the following values have been collected: surface temperature (T_s), ambient temperature (T_{inf}), base width (W), and base length (L). These values, along with the thermal conductivity of Aluminum (k_{Al}), and the fin height (H), kept the same from the previously tested heat sink, are shown in Table 1 below.

The first step in optimizing the heat sink was to determine the fins spacing (S). Given a constraint on the base width of the heat sink, a smaller fin spacing would result in more fins, increasing the surface area for heat transfer, but will decrease the airflow between the fins. The optimum spacing that would maximize the natural convection heat transfer from the heat sink for a given base area was calculated from the following relation [9].

$$\boldsymbol{S_{opt} = 2.714\left(\frac{L}{{Ra_L}^{.25}}\right)} \tag{2}$$

$$h = \frac{1.307k_{air}}{S_{opt}} \quad (3)$$

For these equations, the properties of air at the average temperature, $T_{\text{avg}} = (T_s + T_{\text{inf}})/2 = 55°\text{C}$ were found and shown below in Table 2.

In Equation 2, Ra_L is the Rayleigh number found from Equation 4 below, where g is the acceleration due to gravity.

$$Ra_L = \frac{g\beta(T_s - T_{\text{inf}})L^3}{v^2} \quad (4)$$

Using above equations an optimal spacing of 5.8 mm was found. With this spacing, a resulting heat transfer coefficient of 6.2441 $\text{W/m}^{2\circ}\text{C}$ was found. This value of 5.8 mm should be the optimal spacing for the fins to balance the total surface area of heat transfer and the fluid flow in between the fins.

The next step was to optimize the thickness of the fins. Again, with the constraint on the base width of the heat sink, the larger the thickness, the less fins able to fit within the constraint. However, as the thickness is decreased, the surface area and fin effectiveness will decrease. A Matlab program was used to determine the optimal thickness that balanced both the number of fins, and the surface area and effectiveness of the fin. The program used an array of thickness values from 2 mm to 10 mm. The effectiveness of the fin was first calculated using the following equation given below [9]

$$\varepsilon_{fin} = 1 - \left(\frac{A_f}{A_t} - \eta\frac{A_f}{A_t}\right) \quad (5)$$

With the effectiveness of the fin and the surface area of heat transfer both having an effect on the overall heat transfer rate of the heat sink, the Matlab program returned plots, Figure 3, showing how the fin thickness affected the effectiveness, surface area, number of fins and heat transfer rate.

Simulated results in Figure 3 shows that as the fin thickness increases, the effectiveness and surface area both increase. However, the number of fins is a magnitude of at least ten times higher than the other two. This would suggest that the number of fins has a greater effect on the total heat transfer rate. Figure 3 also shows that a thickness of 0.5 mm would yield the largest heat transfer rate, however machining Aluminum to a thickness of less than 2 mm is not ideal. Therefor, a thickness of 2 mm should be chosen as the optimal thickness of the fins. Once the heat sink was fully dimensioned, the total heat transfer rate could be found. Calculations resulted in a heat transfer rate of 101.82 W. Using the same equation with the geometric dimension of the generic heat sink used for the testing in this work, a heat transfer rate of 75.50 W was calculated. This resulted in a 34.86% improvement in the heat transfer rate that could be achieved from the current design of the heat sink employed in this study.

IV. Experimental Results and Analysis

Experiments were performed using the heat sink shown in Figure 1. As a base line temperature readings were recorded beneath the LED panel. Thermocouples were placed individually under each of the LED light sources on the panel. Figure 4 displays the temperature results for each of the fourteen thermocouples. It seems that within twenty minutes, temperature on the average varied from 90 degree centigrade to about 110 degree centigrade. A base plate was used to mount the LED panel and in this case, all thermocouples collectively reached to an average temperature of about 92.1 degree centigrade as shown in Figure 5. This temperature value was used in the calculations to estimate the heat transfer rates as described in the above section. This temperature value (92.1 C) still exceeds the maximum operating temperature of the LED panel.

At a later stage, the base plate was mounted on the heat sink and thermocouples were used to monitor the temperatures near the top surface of the heat sink. Figure 6 shows temperature results indicating a sharp decrease in temperature from 92.1 degree Centigrade to an average value of about 55.3 degree centigrade giving a 40% reduction in the operating temperature of the LED panel.

As a part of this investigation, seven heat pipes were inserted near the top surface of the heat sink. Due to the presence of heat pipes, temperature under the LED light panel further reduced by 31%. Figure 7 shows the experimental result which clearly indicates a reduction in temperature from 55.3 degree centigrade to 38.11 degree centigrade for almost all fourteen thermocouples.

Currently, effect of environmental humidity and temperature on the heat sink performance are being monitored. For this purpose, Hastest Inc. industrial test chamber (Hastest's industrial chambers (Model HPCH-252XSUH)) are being used. The heat sink with LED panel and all its accessories are kept inside the chamber and remote temperature measurements are recorded while the humidity level and temperature of the chamber is varied. Humidity is varied between 40% to 60% level whereas the ambient temperature inside the chamber is being raised up to 60 C. Results from this study will be presented in future.

V. Conclusions

The purpose of this experiment was to investigate the combined effective of two passive cooling techniques that incorporated a heat sink and seven heat pipes simultaneously. Experimental results clearly indicate 58.6% reduction in temperature values that were recorded using fourteen k-type thermocouples right under the LED panel. It is important to note that future work includes the testing of the optimized heat sink along with heat pipes for the same operating conditions used in the current work. It is anticipated that a reduction of about 60 to 65% can be achieved with new optimized heat sink design. That sharp decrease in the operating temperatures of the LED panel will increase the life time of LED light sources and will optimize its operation for varying environmental conditions especially in agriculture applications.

Acknowledgements

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Table 1. Values used in optimization process

Surface Temp. T_s (°C)	Ambient Temp. T_{inf} (°C)	Base Width W (mm)	Base Length L (mm)	Fin Height H (mm)	Thermal Conductivity of Al 6061 k_{Al} (W/m*K)
92.1	20	165.1	95.3	55.91	180

Table 2. Properties of air at 1atm and 55°C

Prandtl Number Pr	Thermal Conductivity of Air k_{air} (W/m*K)	Kinematic Viscosity υ(m^2/s)	Coefficient of Volume Expansion β(1/K)
.7215	.02772	1.847×10^{5}	3.05×10^{-3}

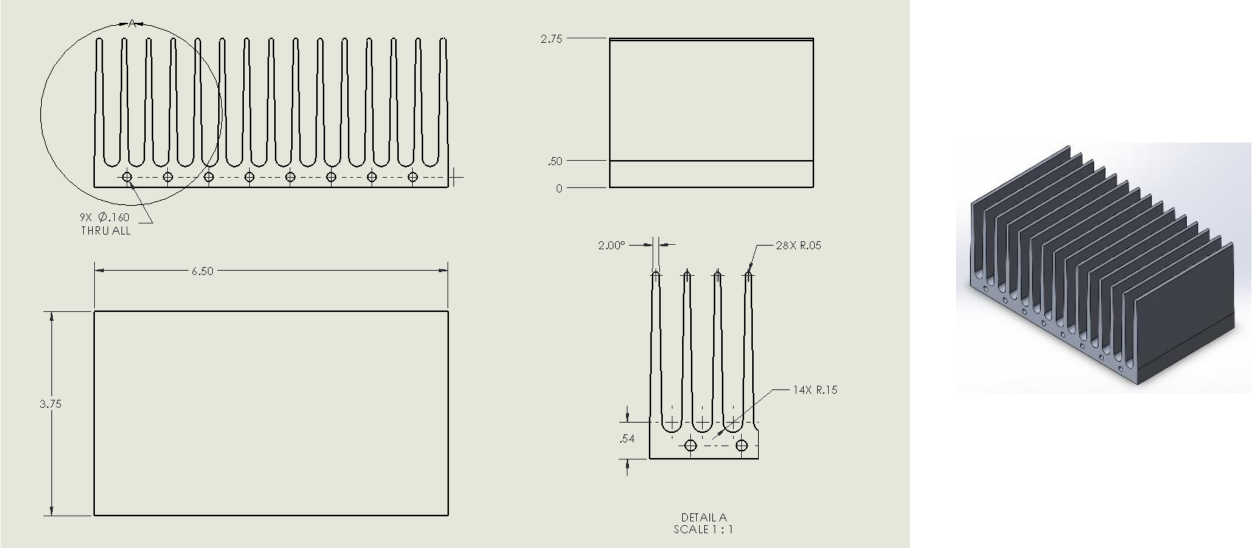

Figure 1. Layout and dimensions of the Heat Sink

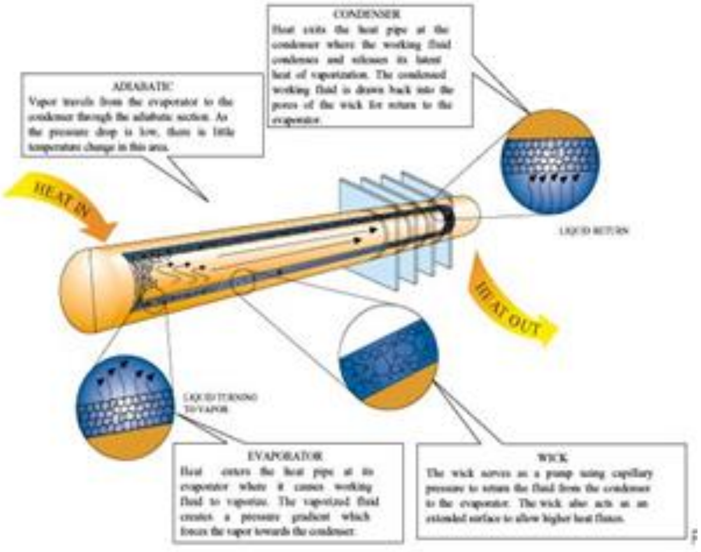

Figure 2. Heat Pipe Concept (http://www.shuttle.eu/_archive/old/en/www.shuttle.eu/index-3002.html)

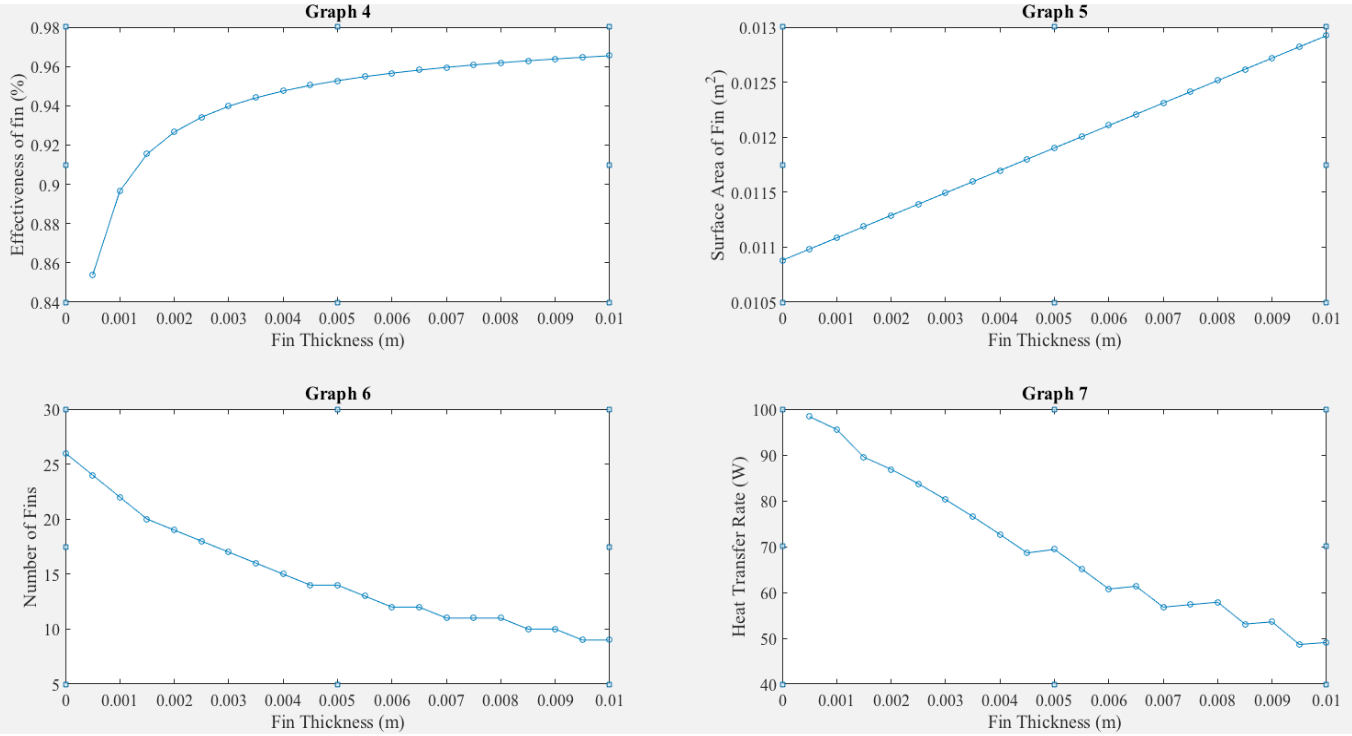

Figure 3. Variation in number of fins, fin effectiveness, surface area of fin, and heat transfer rate as a function of fin thickness.

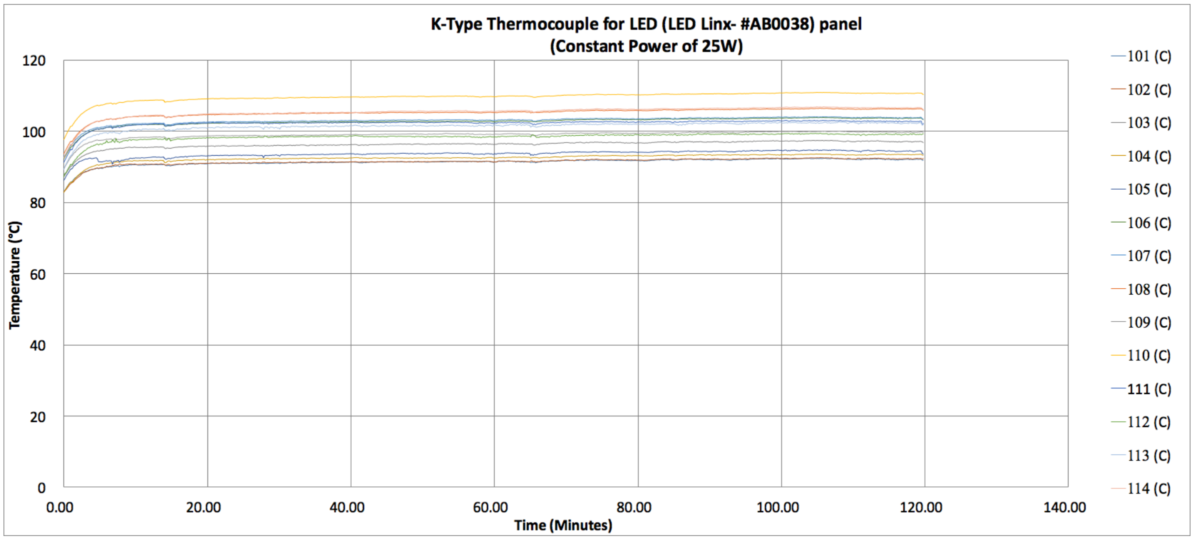

Figure 4. Direct temperature measurements under each LED light source beneath the LED panel

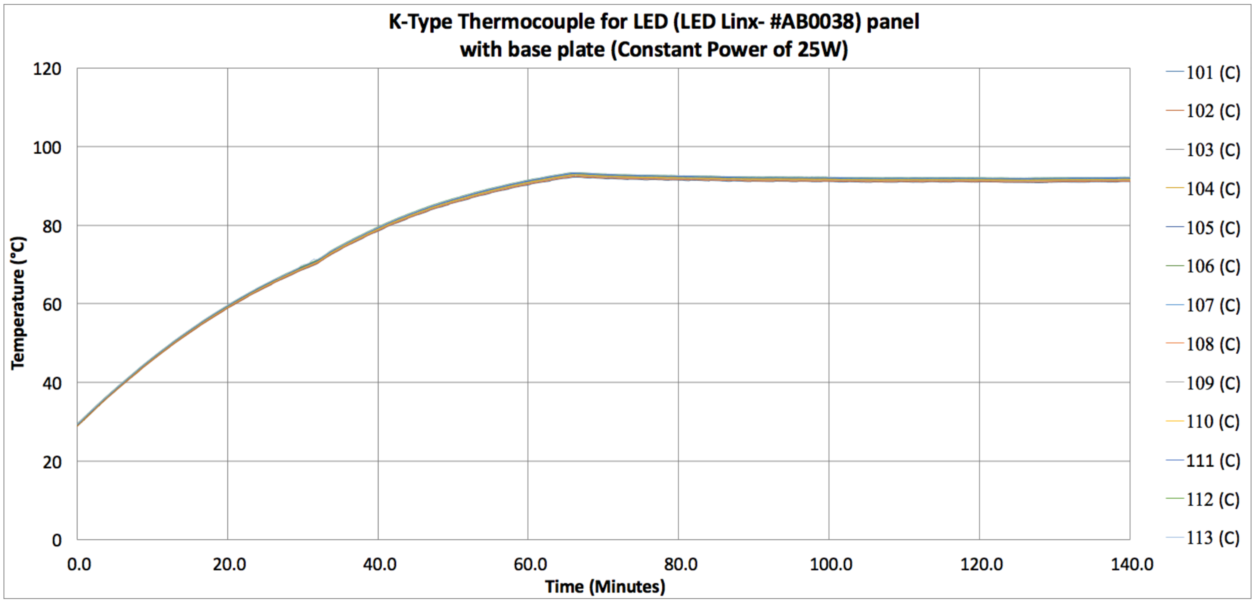

Figure 5. Temperature measurements in the base plate on which LED panel was mounted

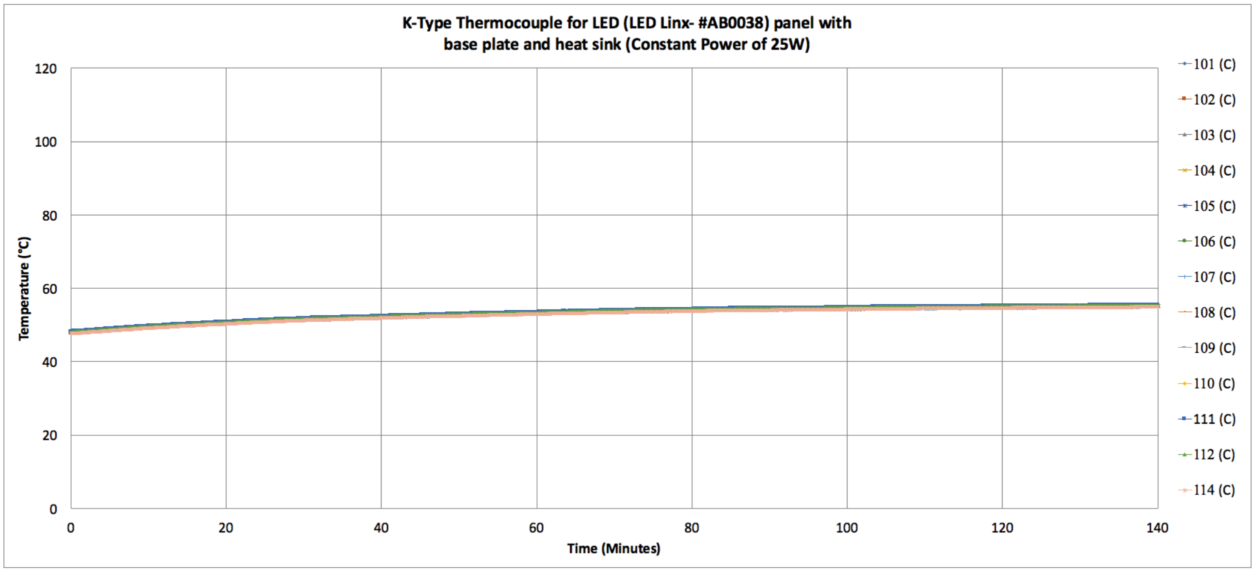

Figure 6. Temperature measurements with the heat sink attached to the LED panel

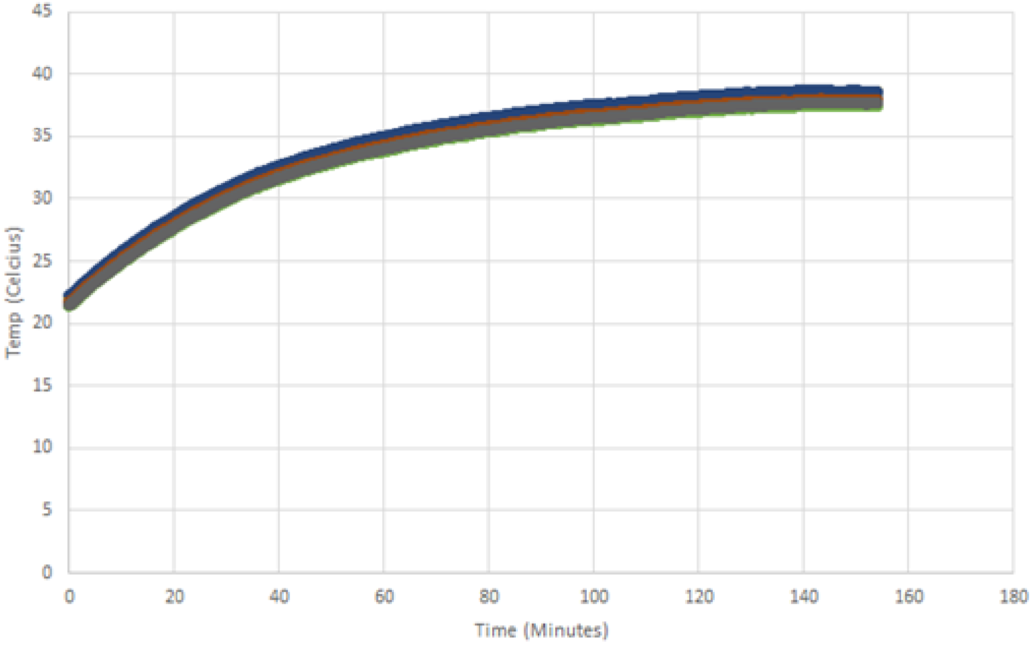

Figure 7. Temperature measurements with heat sink and heat pipes (Combined fourteen thermocouple temperature response)