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A Study on Cooling Characteristics of the LED Lamp Heat Sink for Automobile by Forced Convection

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강제대류에 의한 자동차용 LED 램프 방열판의 냉각 특성에 관한 연구

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ABSTRACT

Automotive headlamps have been continuously developed as one of the most important devices for securing the driver's view, and the LED lamps are getting popular in recent years. However, in case of the LED lamps, because the heat generated by the LED lamps are too high, it shorten the product life and lower the LED efficiency. Therefore, this study was investigated the cooling characteristics of the LED lamp heat sink for automobile by forced convection for LED heat generation control. In order to analyze the cooling characteristics of the heat sink, the temperature distribution results were investigated through the experiment and computational analysis under the increase of the air flow velocity, and the convective heat transfer coefficient was obtained. Also, convective heat transfer coefficient was calculated by the theoretical formula under the same condition and compared with experimental and computational results. From the result of this study, as the air flow velocity around the heat sink fins increased, the convective heat transfer coefficient significantly increased, confirming the improvement in the cooling effect.

Keywords : Forced Convection(강제대류), Cooling Characteristics(냉각특성), Heat Sink(히트싱크), LED Lamp(LED 램프), Computational Analysis(전산해석)

1. Introduction

Automotive headlights secure the view of the drivers and are one of the important external elements that determine the first impression of a vehicle. For this reason, the design of headlights has been changed to suit the emotions and design preferences of consumers using light-emitting diodes (LEDs). LEDs are lamps in which semiconductors are lit. They exhibit low power consumption and have a significantly extended service life compared with

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previous lamps.

The largest benefit of LEDs is that headlights can be produced and utilized in various designs as their volumes decrease[1,2]. Moreover, headlights using LED lamps have been developed and used as countermeasures against regulations on automobile energy consumption, which have become a global issue[3,4]. LEDs, however, use approximately 20% of the power for light emission, and the rest is converted to heat. As the input power increases, the LED junction temperature may increase sharply. When it is more than 70℃, the energy efficiency, light intensity, and service life are reduced[5]. Therefore, the cooling device in an LED lamp is an important element that discharges the heat generated from the LED module into the atmosphere[6].

There are many methods to increase LED efficiency, including the heat sink method, which uses both the cooling fan and the heat sink; the water-cooled cooling plate, which increases heat transfer using a cooling liquid; and heat pipes that use internal circulation caused by phase change[7]. The water-cooled cooling plate[8] and heat pipes exhibit better cooling performance than the heat sink method; however, the use of these heat dissipation systems is limited, because much cost is required for system construction, and there are spatial constraints, because additional devices must be installed. However, the heat sink that uses the cooling fan is subject to less cost and fewer spatial constraint[9]. The heat dissipation performance of a forced-convection-type heat sink that uses a cooling fan is affected by various elements, such as the height and width of the fin. As the area of the fin contacting the air increases, the heat dissipation function becomes better. However, when the spacing between the fins is too narrow, the recycle phenomenon may occur, and the heat transferred to the air can be transferred back to the inside through the heat sink. Therefore, the fin element must be designed appropriately[10-12].

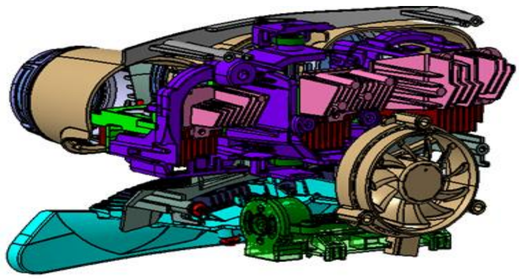


Fig. 1 Images of headlight assemblies used in this study

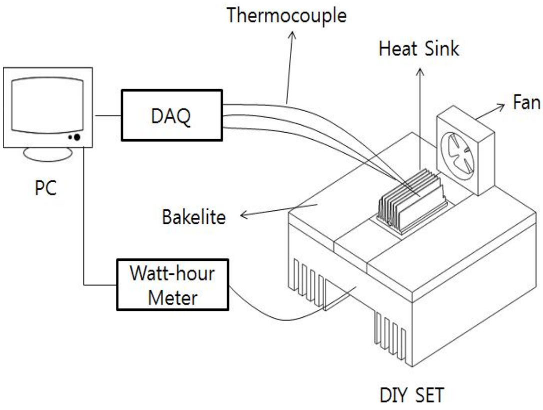
In this study, the heat sink and cooling fan applied to the headlamp of the “E” model from “H” company and installed in a small space, as shown in Fig. 1, were removed, and the forced convection heat transfer characteristics of the components were investigated. In other words, the heat transfer characteristics between the heat sink and the surroundings were investigated by applying an arbitrary heat flux to the heat sink, which had been applied to vehicles in mass production with the determined shape, as well as forced convection with air flow. For this, temperatures were first measured experimentally at the major points of the heat sink with the applied heat flux. The results were compared with those of a numerical analysis, and the forced convection heat transfer coefficient was calculated to examine the heat transfer characteristics.

2. Experimental and Computational Analysis Methods

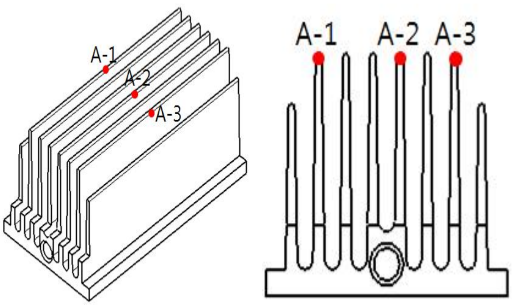
2.1 Experimental set-up and method

Fig. 2(a) shows a schematic of the experimental set-up used in this study. The heat sink could be heated by the thermoelectric element installed on the bottom, and the power consumed for heating was measured using an electricity meter. Based on the measured power consumption, the heat flux applied to the heat sink was determined. Three thermocouples were installed at the locations A-1, A-2, and A-3 on top of the heat sink, as shown in Fig. 2(b), to collect the real-time temperature information. Also, for the heat sink installed on the heated surface of the thermoelectric element, every side was insulated using Bakelite, except for the exposed section.

Moreover, a DC fan controlled using a pulse width modulation (PWM) method was installed 4.5cm in front of the heat sink to investigate the heat transfer characteristics under cooling by forced convection. In addition, as the applied average effective voltage increased from 5.5 to 11.4V, as shown in Fig. 3, the flow velocity created from the fan increased from 2 to 6m/s.



(a) Experimental configuration



(b) Pre-selected locations of thermocouples

Fig. 2 Schematics of experimental set-up

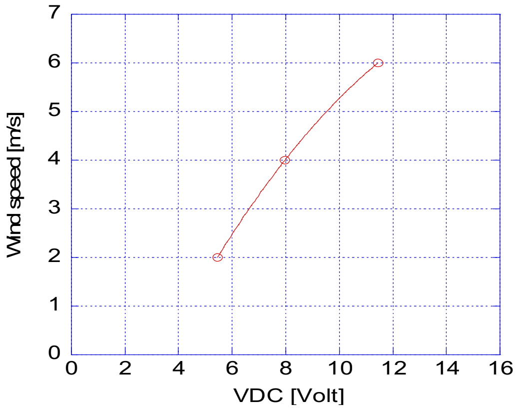
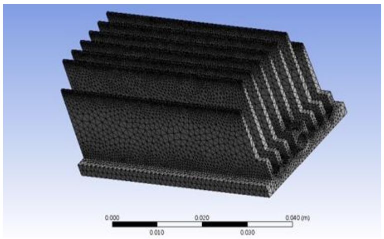


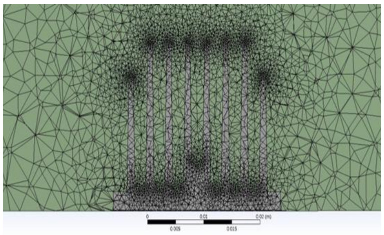
Fig. 3 Air flow control using a PWM

2.2 Computational analysis method and condition

In this study, computational analysis was conducted using Fluent 6.0, a commercial thermo-fluid code. The heat sink was modeled using CATIA, as shown in Fig. 4(a), and the grids were generated through the mesh, as shown in Fig. 4(b).



(a) Heat sink modeling



(b) Meshed heat sink for CFD analysis

Fig. 4 Meshed domain and heat sink for numerical analysis

Table 1 Thermal properties of aluminum heat sink

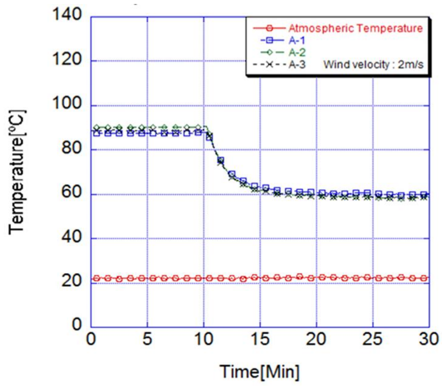
Thermal properties at 300K	
Density	2702 kg/m^3
Specific heat capacity	903 J/kg·K
Thermal conductivity	237 W/m·K

There were 539,975 nodes in the analysis domain and 175,307 nodes in the heat sink. The heat sink used for the analysis was made of aluminum, and its thermal properties are listed in Table 1. To compare the heat transfer characteristics of the heat sink, the thermal load condition of a constant heat flux (27W/m^2) was set on the heat sink bottom surface, and the medium in which conduction occurs in the direction perpendicular to the heat flux of the bottom surface was assumed to be isotropic.

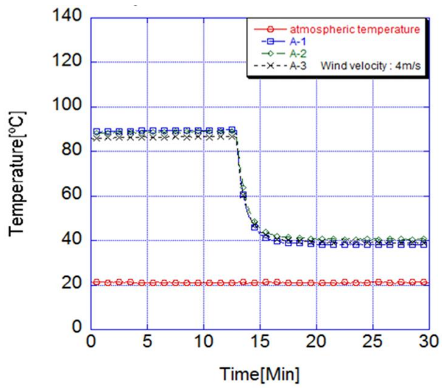
3. Results and Discussion

To investigate the change in the cooling characteristics of the heat sink due to forced convection, the heat sink was heated by applying a 27W/m^2 thermal load using the thermoelectric element, and cooling through the fan was initiated when the temperatures at A-1, A-2, and A-3 shown in Fig. 2(b) were constant.

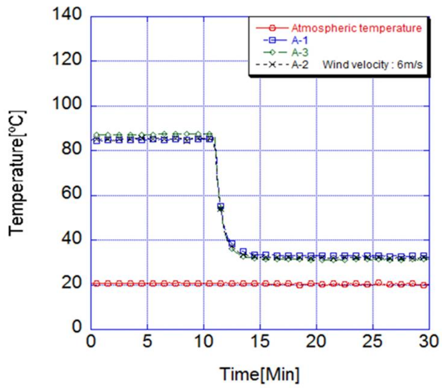
Fig. 5 shows the temperatures measured at A-1, A-2, and A-3 when the heat sink fins were cooled by forced convection. As shown in the Fig. 5, the temperatures of the fins at the measurement points decreased to approximately 59℃ at 2m/s and 32℃ at 6m/s and reached a steady state within 15sec as the flow velocity was applied to the surroundings of the thermoelectric element, which maintained a constant temperature, by the forced convection of the fan. Moreover, after the steady state was reached, the deviations at the three temperature measurement points were found to be less than ±1℃.



(a) Case with air flow velocity of 2m/s



(b) Case with air flow velocity of 4m/s

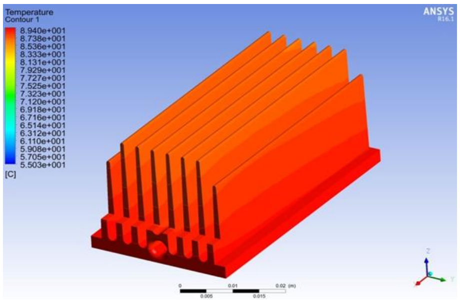


(c) Case with air flow velocity of 6m/s

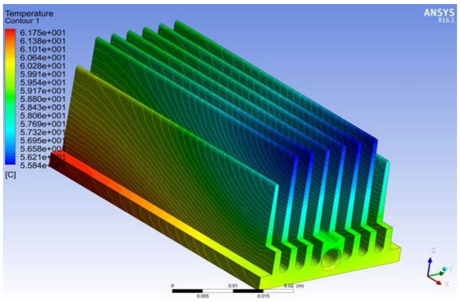
Fig. 5 Measured temperatures at pre-selected location on the heat sink as a function of time

Fig. 6 shows the images of the temperature distribution results of the heat sink derived through computational analysis under the same conditions as the temperature measurement of the heat sink mentioned earlier in the experimental method.

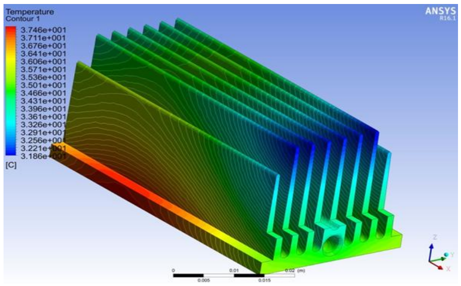
The temperatures of the heat sink fins, which maintained constant values in the steady state after heating, as shown in Fig. 6(a), rapidly decreased and reached a steady state again. It was found that the fin temperatures of the heat sink calculated through computational analysis were also in agreement with the experiment results presented and explained in Fig. 5. Moreover, when the heat sink was cooled by the air flow in the surroundings, it was found that the temperature difference between the bottom surface on the rear side of the heat sink, which is the highest-temperature domain, and the front top surface of the heat sink fins, which is the lowest-temperature domain, was approximately 5℃, regardless of the flow velocity.



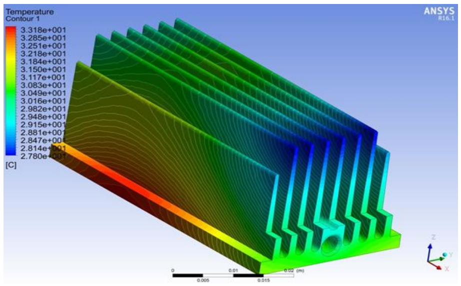
(a) Case without air flow velocity



(b) Case with air flow velocity of 2m/s



(c) Case with air flow velocity of 4m/s



(d) Case with air flow velocity of 6m/s

Fig. 6 Computed temperature distributions for the heat sink without and with air flow applied

When the flow velocity was high, however, the temperature of the heat sink bottom surface decreased as the overall average temperature of the fins decreased. It appears that this can efficiently control the surface temperature of the heating section in contact. As can bee seen from Fig. 6, the average temperature of the heat sink dramatically decreased as the effect of forced convection increased. This is because the convective heat transfer coefficient is related to the Nusselt number(N) and Reynolds number(Re), as shown in Eqs. (1) ~ (4)[13].

Teertstra et al.[14] obtained the Nusselt number according to the heat sink shape using Eqs. (3) ~ (5). Teertstra et al. obtained the Nusselt number(N_{fd}) for the fully developed flow and the Nusselt number (N_{dev}) for the developing flow using the distance between the fins(s_m), the Prantl number(Pr), and the Reynolds number(Re), and obtained the composite Nusselt number(N_u) using the parameter value(n)

depending on the shape. Later, Teertstra et al. obtained the convective heat transfer coefficient using the Nusselt number shown in Eq. (2), fluid conductivity(k_f), and distance between the fins.

Convective heat transfer coefficient

$$h = \frac{N \times k_f}{L} \tag{1}$$

Nusselt number

$$N_L = 0.664 Re_L^{1/2} \times Pr^{1/3} \tag{2}$$

$$N = [(N_{fd})^{-n} + (N_{dev})^{-n}]^{-1/n} \tag{3}$$

$$N_{fd} = \frac{1}{2} \times \frac{s_m}{L} \times Re_{s_m} \times Pr \tag{4}$$

$$N_{dev} = 0.664 \times \left(\frac{s_m}{L} Re_{s_m}\right)^{0.5} \times Pr^{1/3} \times \left(1 + \frac{3.65}{\left(\frac{s_m}{L} \times Re_{s_m}\right)^{0.5}}\right)^{0.5} \tag{5}$$

Fig. 7 shows the convective heat transfer coefficients calculated using Eq. (1) and obtained through computational analysis using the specifications of the heat sink, flow velocity, and Eq. (2) based on the convective heat transfer coefficients measured through an experiment when the air flow velocity around the heat sink fins increased from 2 to 6m/s.

As shown in Fig. 7, the convective heat transfer coefficient calculated through the theoretical formula on the condition of the increase in the air flow velocity increased between approximately 50 and 280W/m^2 · K in a pattern similar to those of the convective heat transfer coefficients obtained through the experiment and computational analysis, despite some differences. In particular, the convective heat transfer coefficients obtained through the experiment and computational analysis exhibited similar results compared with the results of the theoretical formula.

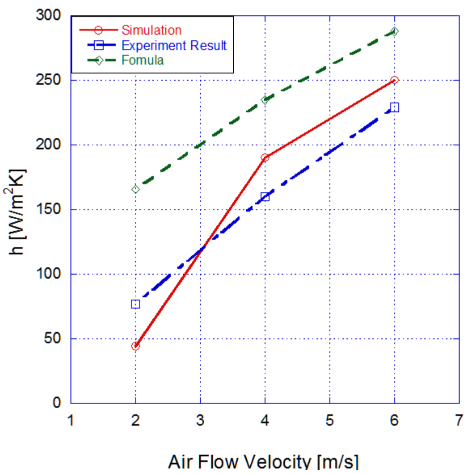


Fig. 7 Estimated convective heat transfer coefficient plotted as a function of air flow velocity

As a result, the convective heat transfer coefficient increased in proportion to the increase in air flow velocity around the heat sink fins. This indicates that, when the air flow velocity increases around the heat sink fins, the performance by forced convection significantly improves under the given conditions.

4. Conclusion

In this study, experiment and computational analyses were conducted to investigate the heat transfer characteristics of the heat sink applied for the cooling of automotive LED lamps. After maintaining the temperature change of the heat sink resulting from the voltage in the steady state, the experiment and computational analysis were conducted while a constant air flow velocity was applied to the surroundings of the heat sink. The following conclusions were derived concerning the heat transfer characteristics.

1. The average temperature of the heat sink decreased as the air flow velocity applied to the heat sink increased, and it was confirmed that the cooling temperature and the locations of the

cooling sections are affected by the air flow velocity.

2. The convective heat transfer coefficient was obtained by the experiment, computational analysis, and theoretical formula while the air flow velocity around the heat sink fins was varied. The obtained convective heat transfer coefficients exhibited similar patterns, despite some differences, and they increased between approximately 50 and 280W/m^2 · K as the flow velocity increased. The experiment results and computational analysis results exhibited similar convective heat transfer coefficients, confirming the objective reliability of the computational analysis results.
3. As the air flow velocity around the heat sink fins increased, the convective heat transfer coefficient significantly increased, confirming the improvement in the cooling effect.

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