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Effectiveness Study of Streamlined Fin with Low-Pressure Loss to Improve Heat Dissipation Performance of Heat Sink

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Abstract— In recent years, circuit boards have been miniaturized to accommodate higher frequencies with the advent of next-generation semiconductors. Miniaturization also means increased heat density. Further miniaturization of switching converters and inverters requires heat sinks that provide excellent heat dissipation performance in a limited volume. The cylindrical fin type has been commonly used as a heat sink structure. However, cylindrical fin type heat sinks have the problem of large pressure loss when refrigerant passes through them. To solve this problem, streamlined fin and elliptical fin have been proposed to replace cylindrical fin type heat sinks. Although many studies have compared the heat dissipation performance of elliptical and cylindrical fin, few studies have compared it with that of streamlined fin. Therefore, in this paper, a heat sink structure with multiple streamlined fin is constructed. The change in pressure loss due to the change in fin shape and the resulting heat dissipation performance was verified by thermo-fluid analysis and actual device verification and compared with the conventional cylindrical fin type. As a result, it was confirmed that the streamlined fin structure slightly increases thermal resistance and significantly reduces pressure loss. A comparison of the thermal resistance-pressure loss characteristics showed that the pressure loss can be sufficiently reduced for the same thermal resistance value, which is excellent for improving the cooling system's efficiency. In the case of forced air cooling, a fan with specific static pressure characteristics improved the streamlined heat dissipation performance.

Keywords— heat sink, thermal design, Zhukovsky wing, streamlined, pressure loss

I. INTRODUCTION

In recent years, the rise of next-generation semiconductors such as SiC-MOSFETs and GaN-HEMTs has led to higher frequency circuit boards, which in turn has led to smaller and lighter main converters for electronic equipment such as computers and inverters for automotive use[1]~[5].

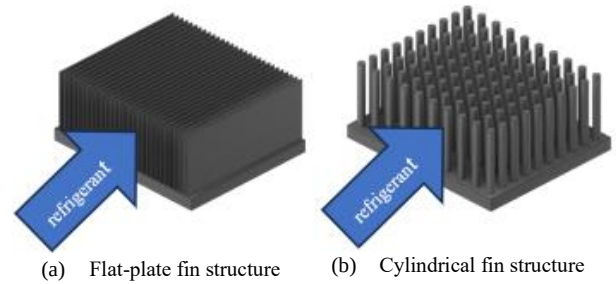


Fig.1. Conventional heat sink

However, as circuits become smaller, the heat dissipation area also becomes smaller, resulting in heat generation in circuits due to power loss. Therefore, heat sinks that enhance heat dissipation efficiency are required to have high performance and be small in size.

Conventional heat sink structures have been of the flat fin type or lattice-arrayed cylindrical fin type as shown in Figs. 1(a) and 1(b)[6][7]. The formula for calculating thermal resistance R_{th} which is an index of heat dissipation performance of a heat sink, is shown in equation(1)[8]. Here h is the heat transfer coefficient by convection and A is the surface area.

$$R_{th} = \frac{1}{h \times A} \quad (1)$$

The flat fin type shown in 1(a) is effective in increasing the contact area A with the refrigerant by attaching it to an element or circuit. According to equation(1), this reduces the thermal resistance and thus enhances the heat dissipation performance.

However, the problem with flat fin heat sinks is that, as shown in a previous study [9], a heat-bearing layer called the temperature boundary layer δ_T is formed from upstream to downstream of the fin surface, along with a velocity boundary layer. Fig.2 shows a top view of air flowing as a refrigerant at 2 m/s through a heated flat-fin heat sink using a thermo-fluid

analysis simulation. The contour is the temperature distribution, and as can be seen from Fig. 2, a layer with heat is formed on the surface of the heat sink. The problem was that this heat-bearing temperature boundary layer reduced the heat dissipation performance. The relationship between the heat transfer coefficient h between the heat sink and the refrigerant and the temperature boundary layer δ_T is shown in equation(2).

$$h \approx \frac{3k}{2\delta_T} \quad (2)$$

As shown in equation(2), the value of the temperature boundary layer δ_T becomes large, resulting in a small heat transfer coefficient h and inefficient heat dissipation performance [9].

Prior studies have compared heat dissipation performance and pressure loss by using methods such as splitting fin to suppress the growth of the temperature boundary layer and fractal shape of the flat fin to destroy the temperature boundary layer [9][10][11]. However, complex heat sink structures such as fractal structures are costly in the manufacturing process due to their complicated geometry.

Unlike flat-fin heat sinks, cylindrical-fin heat sinks have high heat dissipation performance because of their shape, which ensures a large contact area A with the refrigerant by providing a large number of fin and prevents the formation of boundary layers. This shape is also easy to manufacture by extrusion molding [12], one of the most primary manufacturing methods for heat sinks. Furthermore, since the fin body has sufficient cross-sectional area, the generated heat is transferred to the tips of the fin and can efficiently dissipate from the entire fin. Therefore, the entire fin can be efficiently used for cooling.

However, the problem with this geometry is that vortices are generated behind the fin when the boundary layer formed on the fin surface peels off. Fig.3 shows a fluid analysis simulation of air flowing through a cylindrical fin with a diameter of 6 mm at a speed of 0.25 m/s, viewed from the top. The no-slip condition was used as the boundary condition between the object surface and the upper and lower walls of the air region.

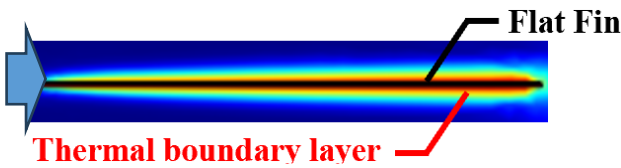


Fig. 2. Temperature boundary layer

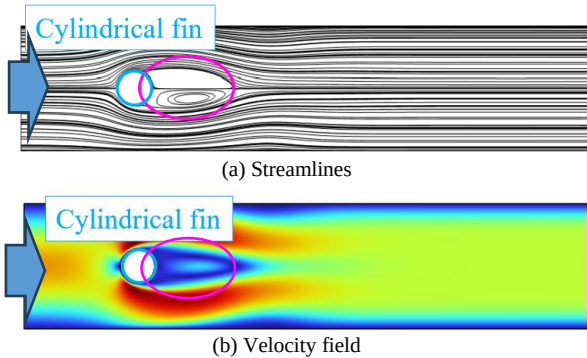


Fig. 3. Behavior of the posterior region of a cylindrical fin

The black line in Fig.3(a) represents the flow line as the air passes through. The streamlines show that vortices are generated in the region behind the fin. Fig.3(b) shows the contour with velocity magnitude. Fig 3(b) shows that the vortex generation impairs the kinetic energy of the refrigerant in the region behind the fin, creating a region where the velocity magnitude is zero. As a result, refrigerant flow becomes difficult and pressure loss occurs.

When considering cooling by forced-air cooling using a fan, even if a constant voltage is applied to the fan, the operating point of the fan changes depending on the size of the pressure loss. As a result, the expected heat dissipation effect is difficult to obtain because the inflow speed of the refrigerant is lower than expected. In the case of water-cooled heat sinks, the load on the pump that circulates the cooling system increases, resulting in higher power consumption during circulation.

In addition, in the case of water-cooled heat sinks, the load on the pump for circulating the cooling system increases, resulting in increased power consumption during circulation.

Therefore, while previous studies [13] have compared heat dissipation performance with elliptical fin while reducing pressure loss due to vortex generation, which is a problem with cylindrical fin, there are only a few studies on heat dissipation performance and pressure loss of heat sink structures with streamlined fin, a shape that reduces vortex more.

In this paper, we examine the effect of pressure loss on the heat dissipation performance of a heat sink structure with a streamlined fin, which is designed to reduce pressure loss by reducing vortex generation, through simulation and experiment. The results are compared with those of the cylindrical fin, and the effectiveness of the streamlined model is discussed.

II. DETAILS OF STREAMLINED FIN

A. Streamlined features

By using a streamlined fin, the kinetic energy of the refrigerant flowing over the fin surface is sustained and the flow proceeds to the rear of the fin because the streamlines of the refrigerant flow more easily along the material surface with less pressure variation due to curvature change than cylindrical fin. As a result, the separation point recedes and the vortex generation area becomes narrower.

Regarding the difference between elliptical and streamlined shapes, which was also mentioned in Section I. Comparing elliptical and streamlined shapes with the same maximum shape

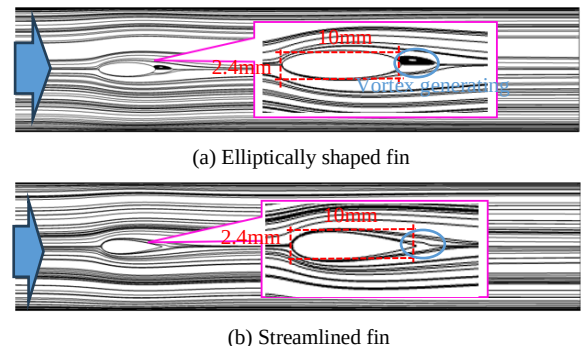


Fig. 4. Vortex generation in the post-fin region

thickness and representative length, the elliptical shape has a steeper shape change as it approaches the trailing edge, which makes it easier to generate a vortex due to delamination.

If a model is created with the maximum shape thickness (2.4 mm) and representative length (10 mm), fluid analysis is performed as in Fig.3, and the behavior of the streamlines is checked using the same number of points, Fig.4 shows the occurrence of vortices due to shape. It has also been reported that the use of streamlines reduces the drag coefficient of the object due to vortex generation compared to elliptical or circular shapes [14].

B. Design of fin

A wide variety of streamlined (airfoil) models exist by varying parameters such as wing thickness and camber. In this study, several streamlines were created using the Zhukovsky transform, which is a method of obtaining a Z-plane streamlined (airfoil) model by transforming a circle of radius 1 in the ξ - η plane into polar coordinates using equations (3) and (4) below.

$$Z = \xi + \frac{c^2}{\xi} \quad (3)$$

$$c = \xi_0 + \sqrt{1 - \eta_0^2} \quad (4)$$

As shown in Fig.5, the shape of the streamlined (airfoil) model can be changed by varying the center coordinates (ξ_0 , η_0) of the circle on the ξ - η plane. When the angle of attack is made concerning the refrigerant inflow direction, the density distribution of the streamlines around the fin differs, and a pressure difference is generated. This pressure difference may generate a blade edge vortex [15].

To exclude the pressure loss due to this vortex, the angle of attack (warp factor) was set to 0 ($\eta_0 = 0$). The fin shape was created by varying the value of ξ_0 , a parameter that changes the blade thickness, between 0.1 and 0.5, which can be identified as streamlined. Details of each parameter are shown in Table 1 and Fig 6.

Table I. Parameter Details

	Surface area	Cross-sectional area
Cylindrical fin	100%	100%
Streamlined fin($\xi_0=0.1$)	100%	24.7%
Streamlined fin($\xi_0=0.2$)	100%	44.9%
Streamlined fin($\xi_0=0.3$)	100%	60.2%
Streamlined fin($\xi_0=0.4$)	100%	72.0%
Streamlined fin($\xi_0=0.5$)	100%	80.1%

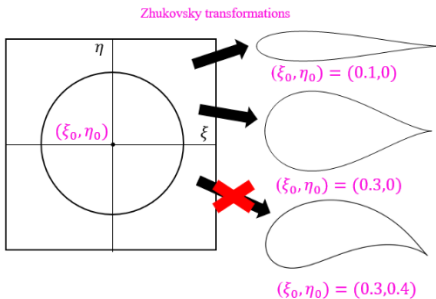


Fig. 5. Example of Zhukovsky wing creation

Here, the model with $\xi_0 = 0.5$ has the largest cross-sectional area of the fin body among the streamlined models. To compare the heat dissipation effects of fin shape for all models in Fig.6(a)~6(f), the fin surface areas were unified by aligning the circumference lengths of the fin bodies.

As for the fin arrangement, a previous study [16] compared the heat dissipation performance of lattice and staggered structures and showed that the heat transfer coefficient of the staggered arrangement is higher. Therefore, in this study, a staggered array structure was adopted instead of a lattice array structure.

III. THERMAL AND FLUID ANALYSIS

A. Simulation conditions

Thermo-fluid simulations using COMSOL Multiphysics 6.2 confirmed the validity of the proposed structure. The fin was made of aluminum, the heat source Q was set to 3000 W/m², and the thermal resistance and pressure loss of each model were evaluated. In the analysis, the refrigerant was air, and the inflow velocity was varied from 0.1 m/s, 0.3 m/s, 0.5 m/s, 1.0 m/s, and 2.0 m/s. The boundary conditions on the sides of the air domain were periodic boundary conditions, non-slip was applied to the upper and lower walls and model surfaces, and L-VEL was used for the turbulence model.

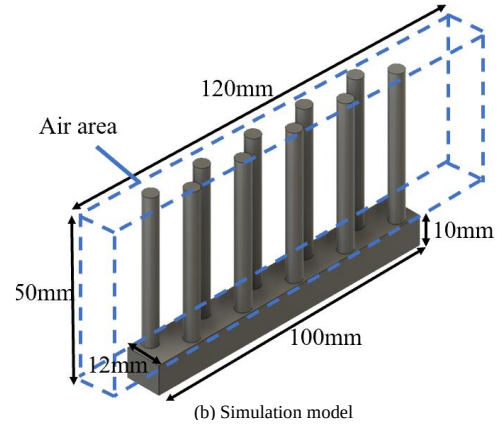
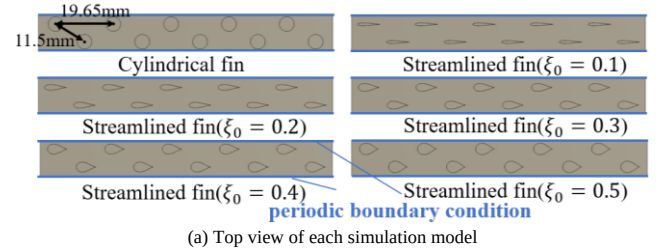


Fig. 6. Model dimensions

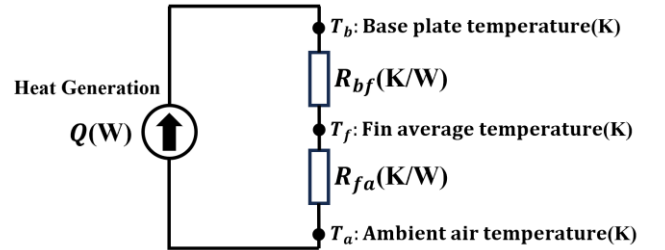


Fig. 7. Thermal equivalent circuit

B. Analysis Results

Thermal resistance R_{th} is expressed by the equation(5)

using the temperature difference ΔT and the heat transfer quantity Q .

$$R_{th} = \frac{\Delta T}{Q} \quad (5)$$

The thermal equivalent circuit of the heat sink is shown in Fig.8. Thermal resistance was calculated from Fig. 7 and equation(5) using the temperature difference ΔT_{ba} between the baseplate temperature and two points of air. The result as well as the pressure difference between the inlet and outlet surfaces, or pressure loss, is shown in Fig .8.

Fig.8(a) shows that the thermal resistance of the conventional cylindrical fin model is the smallest, and that of the streamlined $\xi_0=0.1$ model is the largest. For the streamlined model, the analysis results show that the thermal resistance decreases as the value of ξ_0 is increased. Comparing the thermal resistance of the model with $\xi_0=0.5$, which has the lowest thermal resistance among the streamlined models, with

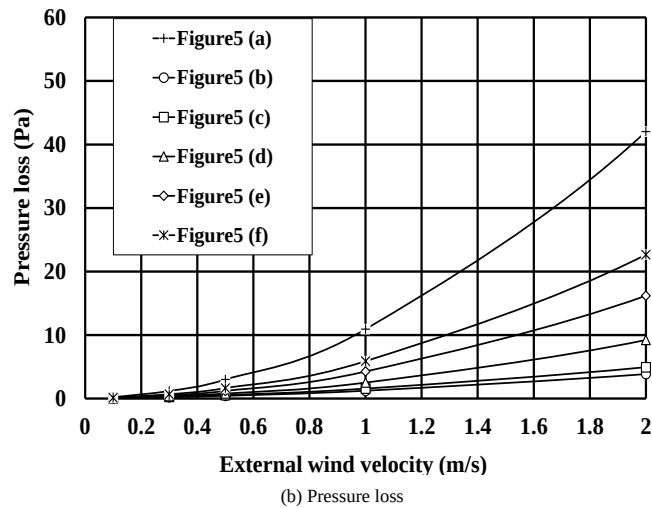
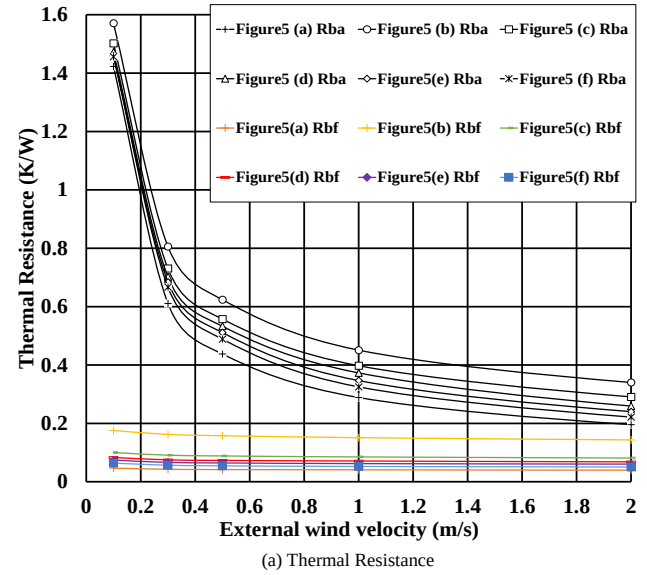


Fig. 8. Simulation results

that of the cylindrical fin, the thermal resistance increased by about 14%.

In addition, since the thermal resistance R_{bf} between base and fin hardly changes with wind velocity, the thermal resistance depends on the R_{fa} of fin-air. In other words, this means that the thermal resistance varies with fin geometry.

Fig 8(b) shows that the pressure loss is largest for the cylindrical fin and smallest for the streamlined model $\xi_0=0.1$. A cross-sectional view of the streamlines including the air region in the simulation is shown in Fig.9 to confirm the behavior of the area behind the fin. The cross-sectional view is at a height of 25 mm above the bottom of the base plate. Fig. 9(a) shows that vortices are generated in the region behind the cylindrical fin. As a result, the pressure loss of the cylindrical fin is increasing. In contrast, Fig.9(b) shows that the vortex generation is completely suppressed in the area behind the fin for the streamlined model $\xi_0=0.1$, where the pressure loss is the smallest. From there, as the blade thickness is increased in Fig.9(c) and 9(d), the formation of vortices due to peeling is encouraged. Comparing the cylindrical fin with the streamlined model $\xi_0=0.5$, which has the largest pressure loss among the proposed models, the pressure loss can be reduced by up to about 45%.

C. Consideration

The data obtained from the simulation is discussed in terms of the cause of the slight increase in thermal resistance of the

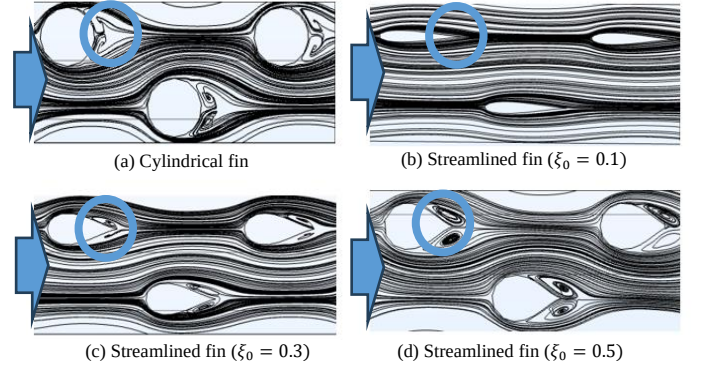


Fig. 9. Vortex generation (Flow velocity 1m/s)

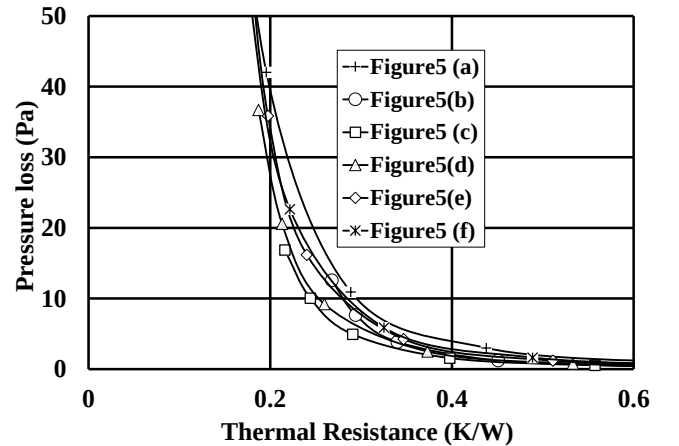


Fig. 10. Thermal resistance-Pressure loss characteristics

streamlined fin and the effect of the decrease in pressure loss on the heat dissipation performance.

A cross-sectional view including the heat sink and air area as in Fig.9 is normalized by the maximum temperature rise, and the temperature distribution is shown as a contour in Fig.11. Fig.11(a) shows the cylindrical fin with the largest pressure loss and the smallest thermal resistance, and Fig.11(b) shows the model with the smallest pressure loss and the largest thermal resistance with the streamlined $\xi_0=0.1$.

Fig.11(a) shows that the heat spreads in the region behind the cylindrical fin and the thermal gradient in the air region is smaller than that of the streamlined $\xi_0=0.1$ shown in Fig.11(b). This is thought to be due to the heat spreading due to the mixing motion of the fluid encouraged by vortex generation, which causes an increase in pressure loss in the rear region of the cylindrical fin. In contrast, the suppression of vortices in the streamlined fin is thought to have prevented efficient heat exchange to the refrigerant and increased thermal resistance compared to the cylindrical fin. However, the experimental results confirmed that the increase in thermal resistance of the streamlined fin was negligible compared to the reduction in pressure loss. From this, the thermal resistance-pressure loss characteristics shown in Fig.10 indicate that all of the streamlined models reduced the pressure loss compared to the cylindrical fin, with the streamlined $\xi_0=0.2$ having the lowest pressure loss.

Consider the effect of this significant reduction in pressure loss for a slight increase in thermal resistance on heat dissipation performance. Assuming that a DC fan CFM-A225BG-123-374-22 (CUI Devices) is used for forced air cooling, the fan operating point obtained from the static pressure characteristics when 12V is applied to CFM-A225BG-123-374-22 and the pressure loss results shown in Fig.9(b), we can conclude that Given the operating point of the fan and the pressure loss results shown in Fig.9(b), the wind speed for the linear model $\xi_0=0.5$ is 1.8 m/s and that for the cylindrical fin is 1.35 m/s. The two models are compared at that wind speed. At that wind speed, the thermal resistance of the streamlined model $\xi_0=0.5$ is about 8% lower than that of the cylindrical fin, confirming its superior heat dissipation performance.

IV. EXPERIMENT

A. Heat sinks used in experiments

As shown in Fig 12, three models were actually created by 3D metal modeling: a conventional cylindrical fin, a streamlined model $\xi_0=0.5$, which was expected to improve cooling performance by the fan in the simulation, and a

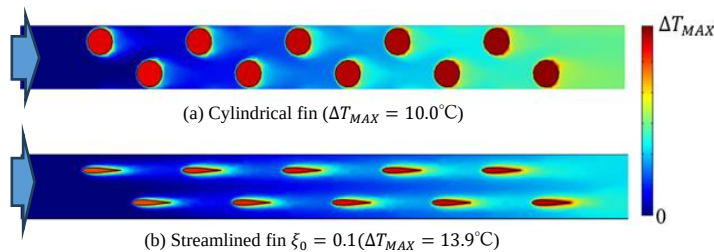


Fig. 11. Temperature distribution in cross section (Flow velocity 1m/s)

streamlined model $\xi_0=0.2$, which reduced pressure loss the most as shown in Fig.10. For fabrication convenience, the base plate was made of *AlSi4130*, and the fin body of *AlSi12* silicon-based aluminum alloy. The dimensions of the experimental heat sink in terms of fin spacing and height were the same as in the simulation model.

B. Experimental Environment and Methods

Dimensions of the wind tunnel used for the experimental

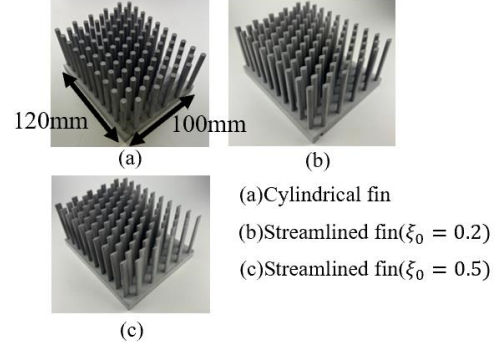


Fig. 12. Actual heat sink

Table II. List of experimental instruments

Instrument	Manufacture	Model Number
DC Regulated Power Supply for fan	GW Instek	GPS-3030DD
PWM signal generator for fan	NF	40.25
Sheet Resistance	Tektronix	AFG2021
DC Regulated Power Supply for sheet resistance	RSPRO	IPSG06D
Date logger	KEYENCE	NR-TH08
DC-fan	CUI Devices	CFM-A225BG-123-374-22
Anemometer	testo	0560 4251
Differential pressure gauge	testo	0560 5126

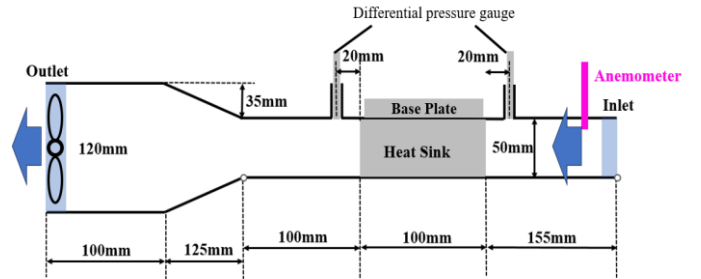


Fig. 13. Detail of wind tunnel

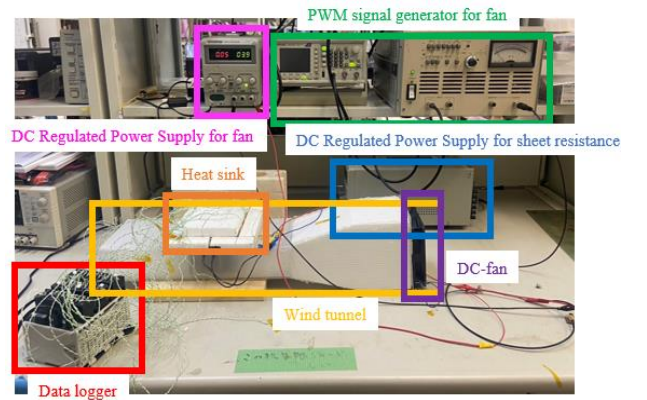


Fig. 14. Experimental environment

environment and photographs of the experimental environment are shown in Fig.13 and 14. Table II shows the equipment used to construct this experimental environment. A heat sink is placed in a wind tunnel 120 mm wide and 50 mm high, and a fan is used to cool the heat sink. A sheet resistor was attached to the back of the base plate as a heat source and heated at 36W. Thermocouples were attached to the fin at the side of the base plate and at three locations upstream, midstream, and downstream from the inlet surface to measure the average base plate temperature and average fin surface temperature after the temperature reached a steady state. At the same time, the average air velocity at the inlet and the pressure difference before and after the heat sink were measured using a differential pressure gauge. Experiments were conducted at several average wind speeds by varying the duty ratio of the applied fan voltage and square wave voltage to investigate the dependence of the thermal resistance and pressure loss on the average wind speed.

C. Experimental Results

The results of thermal resistance R_{ba} against wind velocity obtained from the temperature difference between the average baseplate temperature and room temperature using equation(5)

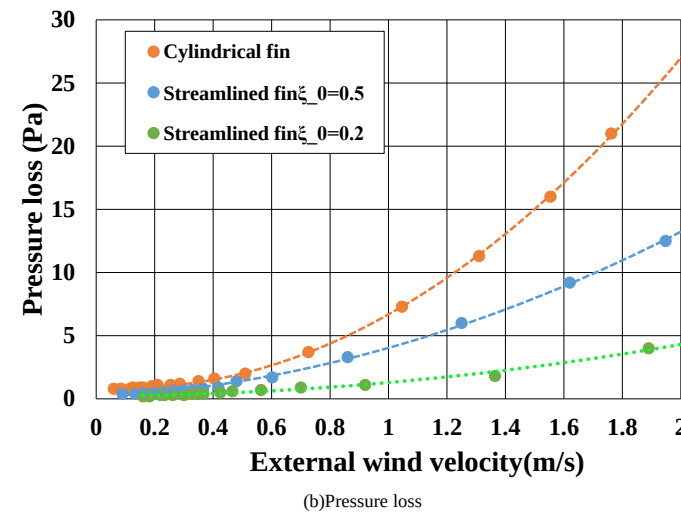
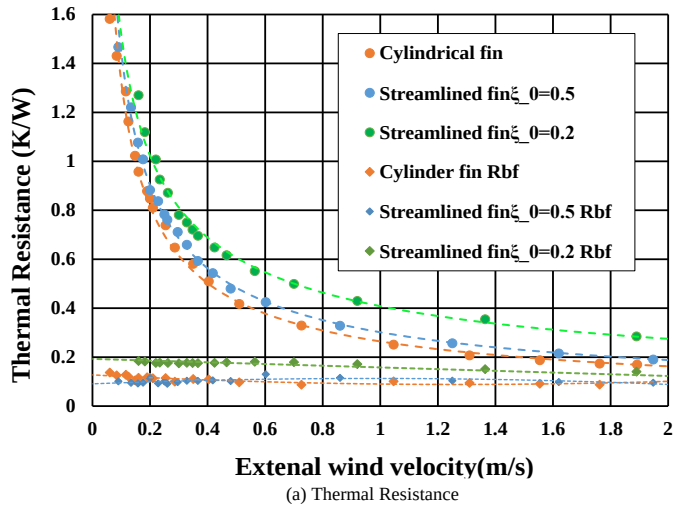


Fig. 15. Experimental Results.

are shown in Fig.15(a), and the results of pressure loss are shown in Fig.15(b).

Fig.15(a) shows that the dependence of thermal resistance on wind speed is similar to the simulation. R_{bf} is almost constant, independent of wind speed, indicating that the thermal resistance depends on R_{fa} . Fig.15(a) shows that the thermal resistance is the smallest for the conventional cylindrical fin. The highest thermal resistance was observed for the streamlined model ($\xi_0=0.2$), and the approximate curve shows that the rate of increase was about 20% at low speeds (below 0.3 m/s) compared to the cylindrical fin. The increase in thermal resistance for the streamlined model ($\xi_0=0.5$) was about 8% at low speeds (below 0.3 m/s) and about 10% at wind speeds of 2.0 m/s compared to the cylindrical fin.

Regarding pressure loss, the largest value was observed for the cylindrical fin, as shown in Fig. 15(b). The pressure loss was the smallest for the streamlined model ($\xi_0=0.2$), with a reduction rate of approximately 50% at low speeds (below 0.3 m/s) compared to the cylindrical fin. It then increased as the wind speed increased, reaching approximately 80% at a wind speed of 1.8 m/s. The pressure loss for the streamlined model ($\xi_0=0.5$) was lower than that for the cylindrical fin, with a reduction rate of approximately 50% at a wind speed of 1.8 m/s. The reduction rate of pressure loss for the streamlined model ($\xi_0=0.5$) was approximately 20% in the low-speed range (0.3 m/s or less) and approximately 50% at a wind speed of 1.8 m/s. Although there were some discrepancies between the measured and simulated pressure loss values, the overall trends showed good agreement.

D. Consideration

From the experimental results, the heat dissipation performance obtained from the relationship between thermal resistance and pressure loss is discussed while mentioning fin efficiency. In this experiment, the aluminum alloy was used for fabrication convenience, which has poor thermal conductivity and deteriorates fin efficiency. Originally, heat sinks are made from materials with high thermal conductivity, so aluminum alloys are not practical. Therefore, this section discusses the data obtained from actual measurements, assuming practical applications, while mentioning fin efficiency. Fin efficiency is an index that expresses the effectiveness of fin surface area for cooling and can be obtained from experimental data using the room temperature T_a , baseplate temperature T_b and fin surface temperature T_f in equation(6).

$$\eta = \frac{T_f - T_a}{T_b - T_a} \quad (6)$$

When fin efficiency is 1, it means that the temperature of the heating element is equal to the temperature of the entire fin surface of the heat sink, indicating that the fin surface area is effectively used for heat dissipation. Thermal resistance can be expressed by equation(7) by including fin efficiency in equation(1).

$$R_{th} = \frac{1}{h \times A \times \eta} \quad (7)$$

From equation (7), it can be seen that the higher the fin efficiency η is, the smaller the value of thermal resistance

becomes. Fig.17 shows the fin efficiency calculated from the measured values using equation(6). Fig.17 shows that the fin efficiency of all models deteriorates to around 50% at a wind speed of 2 m/s. The streamlined model ($\xi_0=0.2$) has the smallest fin efficiency, decreasing by a maximum of about 7% compared to the cylindrical fin. This is because the streamlined model ($\xi_0=0.2$) has the smallest blade thickness and the smallest cross-sectional area, which makes it difficult for the heat from the heating element to be transferred to the fin tip, resulting in the lowest fin efficiency.

The fin temperature distributions for each model at the upstream point at a wind speed of approximately 0.9 m/s are shown in Fig.18.

Three-point thermocouples were fixed to the fin surface to measure the temperature distribution; a gradient was used to interpolate between the three points. The fin temperature distribution shown in Fig.18 also shows that the fin temperature for the streamlined model ($\xi_0=0.2$) in (c) is lower than that in (a) and (b).

Comparing the streamlined model ($\xi_0=0.5$) and the cylindrical fin, there was no significant difference in fin efficiency, with a maximum of about 2%. To eliminate the difference in thermal resistance due to fin efficiency, a heat sink

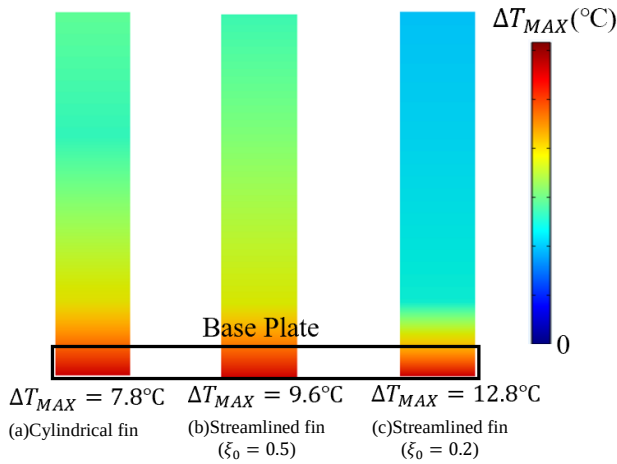


Fig. 17. Fin temperature distribution

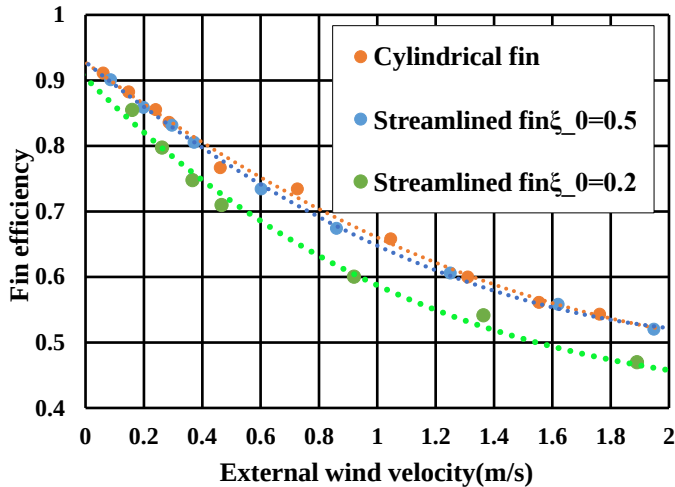


Fig. 16. Fin efficiency

was fabricated using a material with high thermal conductivity, such as copper, and the thermal resistance was corrected using equation(7) when fin efficiency was equal, and the results are shown in Fig.18. Fig.16(b) and Fig.18 shows that the pressure loss is significantly reduced by 52% compared to the 12% increase in thermal resistance of the streamlined model ($\xi_0=0.5$) and the cylindrical fin at a wind velocity of 2 m/s.

From the experimental values, the operating point of the fan due to the fan static pressure characteristic and pressure loss is calculated when forced air cooling is considered as in Chapter 3. As a result, it was found that a fan with a low output airflow and a loose slope of static pressure characteristic, such as CFM-7025B-130-332 (12V), reduces the thermal resistance of the model with a streamlined $\xi_0=0.5$ by 7% and is expected to improve the heat dissipation performance. Thus, the superiority of the streamlined ($\xi_0=0.5$) was confirmed by fan selection, but this advantage was seen only to a limited extent during forced air cooling due to the large rate of increase in thermal resistance compared to the reduction in pressure loss when compared to the cylindrical fin.

However, the streamlined fin used in this experiment shows that both the velocity and flow rate of the refrigerant flowing into the heat sink are higher than those of a conventional cylindrical fin because the pressure loss can be sufficiently reduced at the same thermal resistance compared to the cylindrical fin. Therefore, the pump load required to move the fluid is smaller and energy consumption is lower, which is expected to reduce the operating cost of the cooling circuit and improve the energy efficiency required for circulation.

V. CONCLUSION

In this paper, the heat dissipation performance of a heat sink structure with streamlined fins, which can suppress vortices that cause increased pressure loss, instead of cylindrical fins, was verified through thermo-fluid analysis and experiments. Simulation results showed that the suppression of vortices by streamlined fins reduced pressure loss and slightly increased thermal resistance. For verification on actual equipment, pressure loss, and thermal resistance were measured, and thermal resistance was calculated considering fin efficiency,

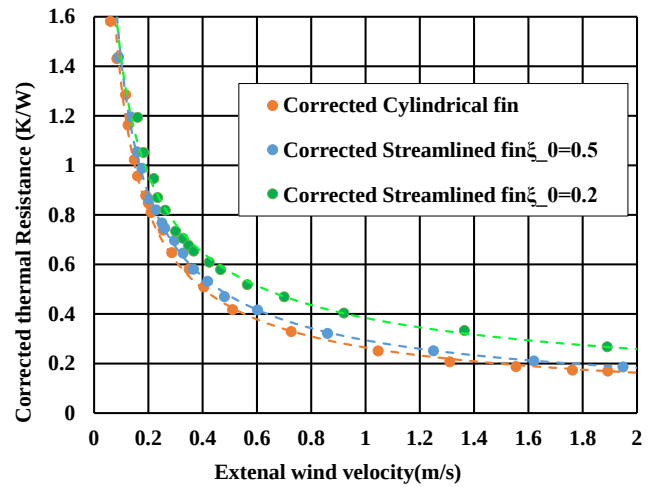


Fig. 18. Corrected Thermal Resistance.

another parameter that determines thermal resistance. As a result, it was confirmed that streamlined fins can significantly reduce pressure drop compared to conventional cylindrical fins, although thermal resistance increases slightly. It was thereby found that for the same thermal resistance, the pressure loss of the streamlined fin is significantly lower than that of the cylindrical fin. The superiority of the heat dissipation performance of the streamlined fins was also confirmed by the selection of the fan during forced air cooling.

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