Optimum Fin Spacing of Rectangular Fins on Aluminum Heat Sinks Plates

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OPTIMUM FIN SPACING OF RECTANGULAR FINS ON ALUMINUM HEAT SINKS PLATES

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Abstract. Aluminum heat sink cooling fin which are especially used for cooling electronic circuits, are directly related to their heat transfer capability and design. In heat transfer, the most important factors are the surface area where the heat is taken, the fin length, the distance between the fins, the airflow rate, and the thermal conductivity. In this study, the effects of the width between the fins on the heat transfer were investigated. In the study, the dimensions of the rectangular finned cooling plate were 200 mm in width, 200 mm in length, 25 mm in fin height, fin spacing between 4 and 2.5 mm, fin thickness of 0.5 mm and the number of fins between 100 and 66 pieces. The plate surface dimensions were kept constant, the fin spacing was narrowed, and the number of fins was increased. It was measured that the total heat wool transferred from the fins on the plate surface increased. The airflow rate was increased from the side surface of the plate and it was observed that the total heat load ejected from the rectangular fins increased.

Key words: heat transfer, fin heat sinks, finned surfaces.

Nomenclature

Symbol	Nomenclature	SI Units
Q	Total heat load	W
H	Wing height	m
W	Wing width	m
L	Wing length	m
U	The mean velocity	m/s
D	Wing pitch	–
h	Heat transfer coefficient	$W/m^2 K$
k	Heat transfer coefficient of air	W/m K
Δp	Pressure	Pa
ρ	Density	kg/m^3
Pr	Prandtl number	–

Symbol	Nomenclature	SI Units
Re	Reynolds number	–
T	Temperature	°C
V	Airflow rate	m/s
$\dot{m}$	Mass flow	kg/s
ν	Kinematic viscosity	m^2/s
μ	Viscosity	kg/s m
α	Thermal diffusivity	m^2/s
c_p	Specific capacity	J/kg K
n	Number of fins	–
τ	Surface shear stress	N/m^2
Π	Pressure difference number	–

Subscripts: face = surface, 0 = environment; opt = optimum.

1. INTRODUCTION

The heat generated by the operation of the devices will prevent the efficient and safe operation of the devices. Moreover, it will exceed the operating temperature limits and will deteriorate or burn. For this reason, cooling of the parts used in electronic circuits is of great importance. The process of removing the heat from the working devices and throwing it to the lower temperature environment is either left to the natural flow or convection is applied with the help of a fan. In some cases, it may not be enough to remove this heat on the device due to the small volume. In this case, the surface needs to be enlarged, so finned plates are placed on the device. The study of expanding the surfaces of heat exchangers used in cooling and heating systems with the addition of fins is the most emphasized research. Generally, finned plates are selected from materials with high thermal conductivity coefficient. The fin spacing is very important in rectangular fin coolers. Too narrow fin spacing increases the flow pressure. Very wide wing spacing can both create volume width and perform the required performance. Knowing the optimum fin spacing of the fins used for cooling will also provide gain in volume. In the study, it is seen that as the heat convection value on the surfaces increases, the heat load released to the environment increases. However, as the number of blades increases on the surface, the gap between the blades narrows and the heat load decreases. In addition, as the air flow rate on the surfaces increases, the heat load value thrown to the environment increases.

2. LITERATURE STUDIES

Heat transfer by heat dissipation method from expanded surfaces has been used for a century. It is particularly interested in the use of expanded surfaces in the fields of engineering, in demands for various applications such as electrical and electronic equipment, air conditioning, cooling, and process heat. By adding fins to the surfaces and expanding the surface, the cooling performance has been increased significantly by increasing the heat dissipation [1].

Wang mentions that work is being done on the use of finned coolers to solve the overheating problem of PV modules. In his experimental observations, he observed that PV modules cause low efficiency in electricity production due to overheating from the sun [2].

Ahmad et al. investigated the thermal performance of discrete multilevel finned heat sink (MLFHS) profiles for naturally convection PV cooling. The stagnation against the flow created by the fins were numerically examined. They observed that the heat transfer performance of the discrete multilevel fin heat sink is better than that of the rectangular plate fin heat sink due to the improved surface geometry. The discrete multi-level fin heat sink design offered approximately 6.13% lower average temperature than the previous design. They stated that they have developed promising alternatives for the development of a passive cooling method in the cooling of photovoltaic systems [3].

They investigated whether the pressure drop causes a negative effect in rectangular profile finned heat sinks with natural convection [4, 5, 6, 7]. In their study, Mousavi et al. applied natural convection to the surface of a vertical finned cooling system and discussed numerical values. The current 3D simulation has been compared with the available experimental data in the literature for the continuous fin heat sink. In the study, ten different forms of discontinuous, stepped and capped finned heat sink were used to find the appropriate configuration. Natural convection and heat transfer have been calculated separately in cooling systems. As a result, they observed that less than 3 mm of fin clearance in stepped fin models does not improve the cooling process [8].

Shen *et al*. used rectangular fin heat sinks to cool the light emitting diodes (LEDs) due to overheating in their study. Experimental and numerical studies were carried out to determine the ability of rectangular fin heat sinks on fluid flow and heat transfer under natural convection conditions. For this, they used 8 different rectangular fin coolers and made performance evaluations. They determined that heat transfer in natural convection flows is an important factor in rectangular fin coolers [9].

Ray *et al*. used numerical simulation in their study to analyze the performance of branched and discontinuous fins to investigate the possibility heat sink configurations, discontinuous and branched fins. They changed the temperature difference (ΔT) between the fin base and the environment in the range of 10 °C to 60 °C ($1.1\times10^5 \leq \mathrm{Ra} \leq 6.1\times10^5$). As a result, it has been observed that all heat sink designs provide better performance by branching the fins, taking into account the change in heat transfer rate (Q), specific heat transfer rate (Q/m), efficiency and Nusselt number (Nu) [10].

Shahzadi *et al*., using the response surface methodology (RSM), optimized the heat conduction rate in a moving permeable fin in the natural convection and radiation environment. Also, the sensitivity of the heat transfer rate was evaluated using RSM. They investigated that for a finite fin, the heat along the fin length decreases as the value of the dimensionless ratio of the ambient temperature T_∞ to the base temperature T_b and the difference between the ambient temperature and the radiation parameter. They also observed that the heat transfer rate at the fin tip decreases as the Peclet number increases. As a result of the sensitivity analysis, they measured that the expansion in the pore parameter increased the heat conduction rate and the highest value was observed at the (–1) level [11].

There are many studies on heat conduction of fins used for cooling. Razelos *et al*. [12] studied convection-emitting rectangular fins with ideal properties. Kiwan [13] investigated the radiation effect on the convection displacement of heat from the fin along an isothermal surface from the base of the wing to the tip.

Rao *et al*. [14] performed a mathematical calculation of the heat exchange from the flat fin assembly with regular convection and radiation.

Choudhary et al. discussed the experimental investigation of heat transfer and airflow behavior of finned and finless cooler under forced convection.

Fin pitch ratio (S/D) and fin size ratio (Lw/D) were investigated in the Reynolds range of 6800–15,100 for the sequential and stepped arrangement of pin fins. By reducing the (S/D) and (Lw/D) ratios, the heat transfer rate and friction losses have been increased. They calculated that the optimum cooling performance is in pin fin dimensions with a (S/D) ratio of 2 and a (Lw/D) ratio of 0.2 [15].

In Kundu's study, thermal analysis and optimization of pin fins subjected to completely wet, partially wet and completely dry surface conditions was analyzed analytically. For the same thermo-geometric and psychometric parameters, a longitudinal fin was calculated to provide higher efficiency than a pin fin. From the optimization results, it was determined that the optimum design of both the longitudinal and pin fin under the partially wet surface condition is possible only for a narrow range of relative humidity, while for a full wet surface it is within a wide range [16].

In addition [17, 18, 19, 20, 21, 22, 23], scientists have presented extensive research on design optimizations, applications, and mounting patterns of fins in many different forms.

Yazicioğlu *et al*., in the study on rectangular fins of different sizes, considered a fin height of 5 to 25 mm, a fin length of 250 to 340 mm, and a spacing of 5.75 to 85.5 mm. Fin thickness was kept fixed at 3 mm. They observed that all fin configurations provide heat input ranging from 25 to 125W. They showed that the optimum fin spacing for the fin spacing was between 6.1 and 11.9 mm, based on the base temperature of the plate and the ambient temperature [24].

In the study of Tu et al., the use of pin fins to provide direct cooling of a computer chip mounted on a printed circuit board (PCB) has been experimentally investigated. It shows that placing a pin fin between a chip and a PCB can provide a significant improvement in direct heat transfer. Numeric values are provided for pin heights of 4 mm, 7 mm and 10 mm [25].

3. HEATSINK DESIGN

As seen in the figure, the surface temperature of a finned heat sink was kept constant at 80 °C and the heat load on the environment was calculated by changing the distance between the fins. In addition, the number of fins and the heat transfer coefficient were changed by keeping the surface width constant in the calculations. The airflow is laminar. Rectangular finned cooler plate dimensions $H = 25$ mm, $L = 200$ mm, $W = 200$ mm, ambient temperature 25 °C. Aluminum heat transfer coefficient is $k = 200$ W/m °C.

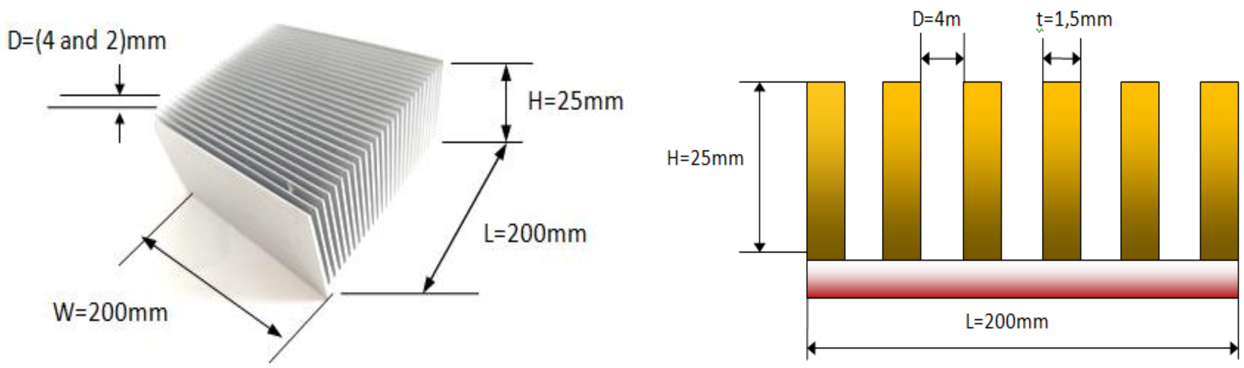

Fig. 1 – Geometric model of the rectangular finned cooler examined in this study.

4. FUNDAMENTAL EQUATIONS AND BOUNDARY CONDITIONS

Heat transfer by convection is achieved at the molecular level. Thermal energy transfer is also provided by the mass or macroscopic movement of the liquid. In the case of a temperature gradient between the surfaces and the media flow, heat transfer from the surface to the flow is achieved. If there is a flow at a given velocity U over a fixed hot plate, a hydrodynamic boundary layer or velocity boundary layer is formed between the surfaces due to the fluid. If $T_{face} > T_0$, convection heat transfer takes place between the surface and the external flow [26].

4.1. Small fin spacing

Too narrow wing openings will reduce the airflow rate. In this case, sufficient heat transfer will not be ensured. As a result of the decrease in heat transfer, the temperature on the cooling plate will increase and reach the maximum. The heat transfer from the cooling plate surface in a certain volume to the environment is [27]

$$\dot{Q} = \dot{m}\, c_p\, (T_{max} - T_0) \tag{1}$$

Equation (1) is also calculated. The airflow mass is $\dot{m} = \rho HWU$, where U is the airflow velocity between the fins

$$U = \frac{D^2}{12\mu}\frac{\Delta p}{L} \tag{2}$$

According to the (2) equation, (1) equation can be rewritten as

$$\dot{Q} = \rho HW \frac{D^2}{12\mu}\frac{\Delta p}{L} c_p \left(T_{max} - T_o\right) \tag{3}$$

The total heat load equation valid for small D is obtained.

2. Large fin spacing

In cases where the distance between the fins designed horizontally to the cooling plate surface is large, a certain U velocity of the airflow from the thermal boundary layer formed on the surface must be known. Since the pressure drop Δp is constant in such a flow, the flow force must be balanced for the control volume $(H \times L \times W)$. If this expression is formulated [27]

$$\Delta p\, HW = \tau (2n) LW \tag{4}$$

where τ is the shear stress on the fin surface. Under the conditions of $\mathrm{Re} \leq 0.5 \times 10^5$, in laminar flow along the length L is

$$\tau = 0.664\, \rho\, U_\infty^2\, \mathrm{Re}_L^{-1/2} \tag{5}$$

If equation (4) is arranged, the speed of the airflow U_∞ is defined by

$$U_\infty = \left(\frac{\Delta pH}{1.328\; n\; L^{1/2} \rho v^{1/2}}\right)^{2/3} \tag{6}$$

$$\dot{Q} = h\, LW\, (T_{max} - T_0) \tag{7}$$

where h means heat transfer coefficient over L of the fin is the equation when $\mathrm{Pr} \geq 0.5$

$$\frac{hL}{k_{air}} = 0.664\, \mathrm{Pr}^{1/3}\, \mathrm{Re}_L^{1/2} \tag{8}$$

provided with. If equations (6) and (8) are arranged, the heat load passing through a single finned surface according to equation (7) becomes like equation (9)

$$\dot{Q}_1 = 1.21\, k_{air}\, HW\, (T_{\max} - T_0)\, \left(\frac{\Pr L\, \Delta p}{\rho\, v^2 D^2}\right)^{1/3} \tag{9}$$

The total heat transfer of the entire heat sink is determined by $\dot{Q} = 2n\dot{Q}_1$ depending on the number of fins. As the distance D' between the fins on the plate increases, the amount of heat transfer also changes. The most suitable fin D' spacing (10) is determined according to the cooling plate that transfers the most heat [28]

$$\frac{D_{opt}}{L} = 2.7\left(\frac{\Delta p L^2}{\mu\alpha}\right)^{-1/4} \tag{10}$$

Pressure drop occurs especially in the plates used to cool the electronic circuits. This pressure drop is defined as dimensionless [29]

$$\Pi_L = \frac{\Delta p L^2}{\mu\alpha} \tag{11}$$

and is determined by equation (11). Total heat transfer is obtained [27] by arranging equation (3) and (9) by considering the total heat load discharged to the environment from the rectangular finned cooler plate in the pressure drop

$$\dot{Q}_{max} = 0.6\, k\left(T_{\max} - T_0\right)\frac{HW}{L}\left(\Pi_L\right)^{1/2} \tag{12}$$

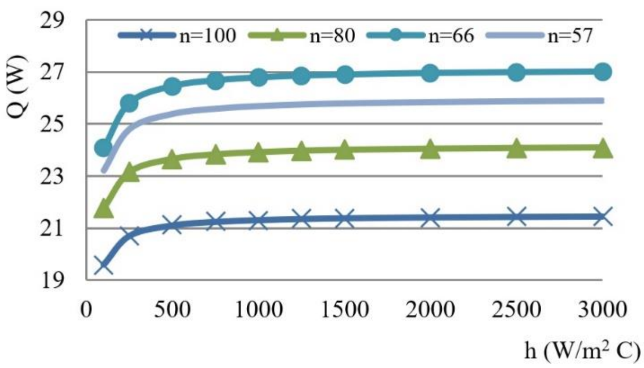

Fig. 2 – Heat transfer coefficient and total heat load change at different fin numbers on the plate surface.

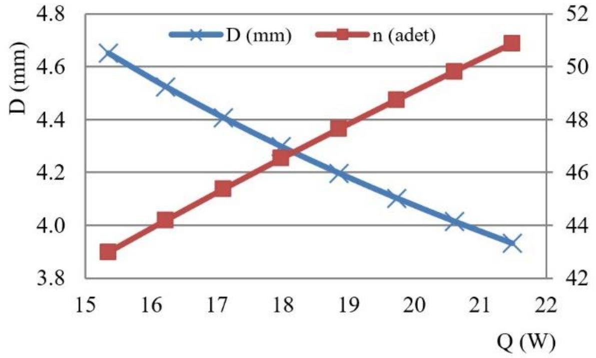

Fig. 3 – Total heat transfer according to fin number and fin spacing change.

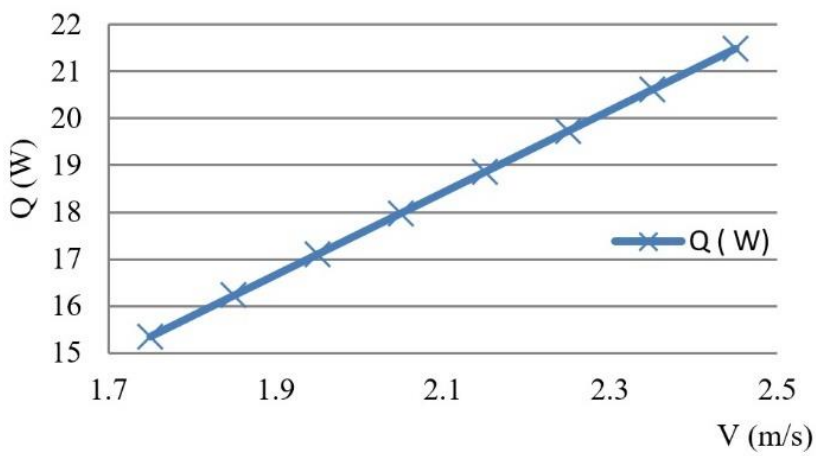

Fig. 4 – Heat transfer according to airflow rate.

5. CONCLUSIONS

The design of rectangular finned cooling plates used for cooling heated systems is very important in heat transfer. The most suitable fin spacing placed on the plate was determined. It has been observed that the narrow or very wide fin spacing affects the rate of heat transfer to the environment. Looking at Fig. 1, 100 fins on the plate transfer less heat than 66 fins with the same dimensions and the same heat convection values. The reason for this is that the fin spacing is narrowed due to the high number of fins. There is less flow and heat convection from the narrower fin spacing. Therefore, less heat load is thrown into the environment. On the other hand, the low number of fins on the rectangular finned cooling plate causes the fin spacing to be too wide. A very wide range reduces the heat load to the environment. In this design, the maximum heat transfer is realized at the fin spacing $D = 2.5$ mm, the wing wall thickness $t = 0.5$ mm, and the number of fins 66. In Fig. 3, as the fin spacing decreases, the number of fins increases as the surface dimensions are kept constant, and it has been measured that the total heat load from the fins increases. Another factor is the flow velocity (U), shown in Fig. 4. As the airflow velocity increases, the total heat load emitted from the fins also increases.

REFERENCES

1. Q. LUO, P. LI, L. CAI, X. CHEN, H. YAN, H. ZHU, P. ZHAI, P. LI, Q. ZHANG, *Experimental investigation on the heat dissipation performance of fl ared- fin heat sinks for concentration photovoltaic modules*, Appl. Therm. Eng., **157**, art. 113666, 2019.
2. J.C. WANG, M.S. LİAO, Y.C. LEE, C.Y. LİU, K.C. KUO, C.Y. CHOU, C.K. HUANG, J.A. JİANG, *On enhancing energy harvesting performance of the photovoltaic modules using an automatic cooling system and assessing its economic benefits of mitigating greenhouse effects on the environment*, J. Power Sources, **376**, *1*, pp. 55–65, 2018.
3. E.Z. AHMAD, A. FAZLİZAN, H. JARİMİ, K. SOPİAN, A. IBRAHİM *Enhanced heat dissipation of truncated multi-level fin heat sink (MLFHS) in case of natural convection for photovoltaic cooling*, Case Studies in Thermal Engineering, **28,** art. 101578, 2021.
4. J.G. HERNANDEZ-PEREZ, J.G. CARRİLLO, A. BASSAM, M. FLOTA-BANUELOS, L.D. PATİNO-LOPEZ, *A new passive PV heatsink design to reduce efficiency losses: a computational and experimental evaluation*, Renew. Energy, **147**, pp. 1209–1220, 2020.
5. L. IDOKO, O. ANAYA-LARA, A. MCDONALD, *Enhancing PV modules efficiency and power output using multi-concept cooling technique*, Energy Rep. **4**, pp. 357–369, 2018.
6. A.A. RAHEİM AMR, A.A.M. HASSAN, M. ABDEL-SALAM, A.H.M. EL-SAYED, *Enhancement of photovoltaic system performance via passive cooling: theory versus experiment*, Renew. Energy, **140**, pp. 88–103, 2019.
7. A. IBRAHİM, S. MAT, A. FUDHOLİ, A.F. ABDULLAH, K. SOPİAN, *Outdoor performance evaluation of building integrated photovoltaic thermal (BIVPT) solar collector with spiral flow absorber configurations*, Int. J. Power Electron. Drive Syst., **9**, pp. 1918–1925, 2018.
8. H. MOUSAVİ, A.A,R. DARZİ, M. FARHADİ, M. OMİDİ *A novel heat sink design with interrupted, staggered and capped fins*, International Journal of Thermal Sciences, **127**, pp. 312–320, 2018, https://doi.org/10.1016/j.ijthermalsci.2018.02.003.
9. Q. SHEN, D. SUN, Y. XU, T. JIN, X. ZHAO, *Orientation effects on natural convection heat dissipation of rectangular fin heat sinks mounted on LEDs*, International Journal of Heat and Mass Transfer, **75**, pp. 462–469, 2014, https://doi.org/10.1016/j.ijheatmasstransfer.2014.03.085.
10. R. RAY, A. MOHANTY, P. PATRO, K.C. TRIPATHY, *Performance enhancement of heat sink with branched and interrupted fins*, International Communications in Heat and Mass Transfer, **133**, art. 105945, 2022, https://doi.org/10.1016/j.icheatmasstransfer.2022.105945.
11. S. JAWAIRIA, J. RAZA, *Optimization of heat transfer rate in a moving porous fin under radiation and natural convection by response surface methodology: Sensitivity analysis*, Chemical Engineering Journal Advances, **11**, art. 100304, 2022, https://doi.org/10.1016/j.ceja.2022.100304.
12. P. RAZELOS, X. KAKATSIOS, *Optimum dimensions of convecting-radiating fins: Part I – longitudinal fins*, Appl. Therm. Eng., **20**, *13*, pp. 1161-1192, 2000, https://doi.org/10.1016/S1359-4311(99)00089-7.
13. S. KİWAN, *Effect of radiative losses on the heat transfer from porous fins***,** Int. J. Therm. Sci., **46**, *10*, pp. 1046–1055, 2007, https://doi.org/10.1016/j.ijthermalsci.2006.11.013.
14. V.D. RAO, S.V. NAIDU, B.G. RAO, K.V. SHARMA, *Heat transfer from a horizontal fin array by natural convection and radiation-A conjugate analysis*, Int. J .Heat Mass Transf., **49**, *19–20*, pp. 3379–3391, 2006, https://doi.org/10.1016/j.ijheatmasstransfer.2006.03.010.
15. V. CHOUDHARY, M. KUMAR, A.K. PATİL, *Experimental investigation of enhanced performance of pin fin heat sink with wings*, Applied Thermal Engineering, **155**, pp. 546–562, 2019, https://doi.org/10.1016/j.applthermaleng.2019.03.139.
16. B. KUNDU, *Performance and optimum design analysis of longitudinal and pin fins with simultaneous heat and mass transfer: Unified and comparative investigations*, Applied Thermal Engineering, **27**, *5–6*, pp. 976–987, 2007, https://doi.org/10.1016/j.applthermaleng.2006.08.003.

17. D.Q. KERN, A.D. KRAUS, *Extended surface heat transfer*, McGraw-Hill, New York. 1972.
18. A. ZHUKAUSKAS, *High-performance single-phase heat exchangers*, Hemisphere Publishing, New York, 1989, Chapter 14.
19. R.K. SHAH, A.D. KRAUS, D. METZGER, *Compact heat exchangers: A festschrift for A.L. London*, Hemisphere Publishing, New York. 1990.
20. A.E. BERGLES, *Electronic and microelectronic equipment*, Hemisphere Publishing, New York, 1990.
21. R.M. MANGLIK, A.D. KRAUS, *Process, enhanced, and multiphase heat transfer*, Begell House, New York, 1996.
22. S. KAKAC, A.E. BERGLES, F. MAYİNGER, H. YUNCU, *Heat transfer enhancement of heat exchangers*, Kluwer Academic, Dordrecht, The Netherlands, 1999.
23. A.D. KRAUS, A. AZİZ, J. WELTY, *Extended surface heat transfer*, Wiley, New York, 2001.
24. B. YAZICIOĞLU, H. YÜNCÜ, *Optimum fin spacing of rectangular fins on a vertical base in free convection heat transfer*, Heat Mass Transfer, **44**, pp. 11–21, 2007, https://doi.org/10.1007/s00231-006-0207-6.
25. J. TU, W.W. YUEN, Y. GONG, *An assessment of direct chip cooling enhancement using pin fins*, Heat Transfer Engineering, **33**, *10*, pp. 845–852, 2012, https://doi.org/10.1080/01457632.2012.654445.
26. A.S. LAVINE, T.L. BERGMAN, F.P. INCROPERA, D.P. DEWITT, *Isı ve Kütle Geçişinin Temelleri*, Literatür Yayıncılık, Basım, 2015.
27. A. BEJAN, G. TSATSARONIS, M.J. MORAN, *Thermal design and optimization*, John Wiley & Sons, 1996, p. 250.
28. A. BEJAN, E. SCIUBBA, *The optimal spacing of parallel plates cooled by forced convection*, International Journal of Heat and Mass Transfer, **35**, *12*, 1992, pp. 3259–3264. https://doi.org/10.1016/0017-9310(92)90213-C.
29. A. BEJAN, A.D. KRAUS, *Heat transfer handbook*, John Wiley & Sons, Inc., 2003.

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