PERFORMANCE STUDY ON AUTOMOTIVE RADIATOR BY THE MODIFICATION OF FIN GEOMETRY AND COOLANT

BADISETTI SAIKUMAR¹, N. HARI BABU², M. V. RAMANA³ & B. V. V. PRASADA RAO⁴

¹Research Scholar, Department of Thermal Engineering, Aditya Institute of Technology and Management,
Tekkali, Andhra Pradesh, India

²Professor, Department Mechanical Engineering, Aditya Institute of Technology and Management,
Tekkali, Andhra Pradesh, India

³Associate Professor, Department Mechanical Engineering, Aditya Institute of Technology and Management,
Tekkali, Andhra Pradesh, India

⁴Assistant Professor, Department Mechanical Engineering, Aditya Institute of Technology and Management,
Tekkali, Andhra Pradesh, India

ABSTRACT

The present work is the study of a conventional radiator for the improvement of thermal performance through modification of fin design. In the automotive industry all the radiators are designed by conventional Fins because of their ease of manufacturing and brazing to the tubes. Heat dissipation in such Fins takes place by convection between the ambient air and fins, but the conduction is neglected between tube panel and fins. In this study, it introduces conduction heat transfer by the addition of the conduction fin element to the panel. The additional fin element is in direct contact with the panel, thereby enhancing conduction heat transfer. Further, the heat dissipation rate improved by change of coolant using Water (35%) -Ethylene Glycol (65%) mixture as a liquid coolant in the radiator. In this paper ϵ -NTU method is described to do heat transfer calculations. Modelling and simulation of the conventional as well as trapezoidal fins are performed by CATIA and CFD Tools. The outlet temperature of the liquid coolant is evaluated and compared for both conventional and the modified design. The modified fin design improves thermal performance by increase in heat dissipation rate.

KEYWORDS: Radiator, Fin, ϵ -NTU Method, Ethylene Glycol & Thermal Performance

Received: Jul 03, 2018; **Accepted:** Jul 24, 2018; **Published:** Aug 03, 2018; **Paper Id.:** IJMPERDAUG201899

1. INTRODUCTION

Automotive radiator is a key component of the engine cooling system. To cool down the engine, a coolant passes through the engine block, where it absorbs heat from the engine. The hot coolant is then fed into the inlet tank of the radiator (located either on the top of the radiator, or along one side), from which it is distributed across the radiator core through tubes to another tank on the opposite end of the radiator. As the coolant passes through the radiator tubes on its way to the opposite tank, it transfers much of its heat to the tubes which, in turn, transfer the heat to the fins that are lodged between each row of tubes. The fins, then release the heat to the ambient air. Fins are used to greatly increase the contact surface of the tubes to the air, thus increasing the exchange efficiency. The standard radiator tube is a thin walled tube that stretches the length of the core. When the radiator is in use, coolant flows from tank to tank via the tubes in the core allowing it to dissipate heat through the tube walls and the

fins. There is one difference between aluminium tubes and copper-brass tubes. Due to the weakness of copper-brass, the standard tubes are only 9/16 inch, commonly referred to as 1/2-inch tube. There can be other tube widths in copper-brass, but 1/2 inch is the most common. If copper-brass tubes wider, then tubes should be thicker to withstand the additional pressure of the coolant passing through. The result would be an extremely heavy and thick radiator that would be difficult to use. By comparison, standard aluminium tubes are one inch wide, but can also be built $1\frac{1}{4}$ inch or $1\frac{1}{2}$ inch-wide without adding much weight. A standard aluminium radiator is made with only two rows of tubes, but will have more surface contact with the fins than a four-row copper-brass radiator using 1/2-inch tubes. The additional surface area gives the aluminium radiator core a boost in its ability to cool. In addition to this, aluminium tubes can be made through an extrusion process to create a heavy-duty tube that is able to take significantly more pressure and heat.

Generally, LMTD or ϵ -NTU method is used to do heat transfer calculations of the radiator. Both methods have its own advantages and preferred according to data availability. When the radiator inlet and outlet temperature are known LMTD gives faster solution. When any of the temperature is unknown LMTD method undergoes iterations to find a solution. In this case ϵ -NTU is better. In this paper ϵ -NTU method is described to do heat transfer calculations.

Many researchers had devoted efforts to fin design optimization by varying the fin geometry for the past decade. Raja Suryan G T et al.[1] studied the heat dissipation rates and outlet temperature by the modification fin design. Kraus et al. [2] had addressed important design guidelines for cooling electric devices. Chen and Wang [3] had presented a study related to a novel step or trapezoid surface design applicable to air-cooled heat sink under cross flow condition, the resultant thermal resistance of the proposed trapezoid design reveals a 10% lower thermal resistance than that of the plate fin surface at a specified pumping power. A. R. Khot et al. [4] performed various methods for radiator performance evaluation and testing of the radiator are considered because all internal combustion engines produce heat as a by product of combustion and friction. Studies of extended surfaces of longitudinal fins of rectangular, triangular, trapezoidal, and concave & convex parabolic profiles, and the like were well documented in the monograph by Kraus et al. [5]. Mokheimer [6] studied the effect of temperature dependent heat transfer coefficient h on the fin efficiency of annular (radial) fins having rectangular, triangular, concave parabolic and convex parabolic profiles. Dul'kin and Garas'ko [7] developed closed-form inverse solutions (fin parameter as a function of fin temperature) applicable for a straight fin of constant cross-section area subject to a power law type temperature dependent h . A generalized analytical solution that took into account the effect of variable heat transfer coefficient, variable cross-sectional area, and curvature of the lateral surface of an extended surface was presented by Agwu Nnanna et al. [8].

Aziz and Beers-Green [9] provided information about the temperature distribution, heat transfer rate, fin efficiency, and optimum dimensions of a rectangular fin subject to examine various effects such as a convective fin base, wall conduction resistance, and contact resistance between the wall and the fin base. They concluded that contact resistance is essential in designing heat sink. The earlier work of Aziz and Enamul-Huq [10] used a regular perturbation method to develop an analytical solution for a straight fin with a temperature dependent thermal conductivity; Kim and Moon [11] showed that a stepped circular pin array of optimum dimensions may be superior in thermal performance of the uniform diameter pin fin array. Yazıcıoğlu [12] performed an experimental study on steady state natural convection heat transfer from vertical rectangular fins & found that the rate of heat transfer from fin arrays depends on the geometric parameters and the base-to-ambient temperature differences. Güvenç [13] investigated natural convection heat transfer from vertically oriented rectangular fin arrays experimentally & found that fin spacing is the most important variable for

maximum natural heat transfer rate. Starner and McManus [14] studied the natural convection heat transfer from four different fin array configurations with three base types and heat transfer coefficients were calculated & concluded that fin height, fin spacing and base orientation has a significant effect on the rate of heat transfer from fin arrays

2. PRESENT WORK

Figure 2.1 (a): Drawing of V-Fin

Figure 2.1 (b): Drawing of Trapezoidal-Fin

Figure 2.1 (c): Mesh Modal : (V- Fins)
Structured Mesh with Tetra Elements

Figure 2.1 (d): Mesh Modal: (Trapezoidal- Fins)
Structured Mesh with Tetra Elements

2.1. Theoretical Calculations

Core dimensions: 365*340*24 mm

Tubes =34/row, Number of Fins per inches – 10 (For conventional Fin)

Number of Fins per inches – 7(For Trapezoidal Fin)

2.1.1. Total Heat transfer Area of the Radiator with Conventional fin Design (V-Fin):

Total tubes area of Heat Transfer = 1.20m²

Surface area of Fins=2.02 m²

Total Heat Transfer Area= 2.02+1.20=3.2 m²

Known data: $T_{h,1}=87^{\circ}\text{C}=360\text{K}$, $T_{c,1}=27^{\circ}\text{C}=300\text{K}$, $U_m=250\text{ W/m}^2\text{K}$,

$$C_{p,c}=1.005 \text{ kJ/kg K}, \dot{m}_h=0.225 \text{ kg/sec}, \dot{m}_c=1.25 \text{ kg/sec}$$

2.1.1(a). Calculation of Outlet Temperature and Heat Dissipation Rate when Water as Coolant

Specific heat of water at a temperature of 360K is, $C_{p,h}=4.18 \text{ kJ/kg K}$

$$C = C_{\min}/C_{\max} = \dot{m}_h c_h / \dot{m}_c c_c = (0.225 \times 4180) / (1.25 \times 1005) = 942.17/1256.25 = 0.74$$

$$NTU = AUm/C_{\min} = 3.2 \times 250/942.17 = 0.84$$

From the figure: at $NTU=0.84$ & $C=0.74$, the Effectiveness (ϵ) = 0.48

$$\text{Effectiveness-NTU method, } Q = \epsilon C_{\min} (T_{h,i} - T_{c,i}) = 0.48 \times 942.17 \times (360 - 300) = 27.134 \text{ kW}$$

$$T_{h,o} = T_{h,i} - Q/C_h = 360 - (27134/942.17) = 331.2\text{K (or)} 58^\circ\text{C}$$

$$T_{c,o} = T_{c,i} + Q/C_c = 300 + (27134/1256.25) = 321.599\text{K (or)} 48.44^\circ\text{C}$$

2.1.1(b). Calculation of Outlet Temperature and Heat Dissipation Rate when Ethylene Glycol as Coolant

Specific heat of ethylene glycol at a temperature of 360K is, $C_{p,h}=3.8 \text{ kJ/kg K}$

$$C = C_{\min}/C_{\max} = \dot{m}_h c_h / \dot{m}_c c_c = (0.225 \times 3800) / (1.25 \times 1005) = 856.52/1256.25 = 0.68$$

$$NTU = AUm/C_{\min} = 3.2 \times 250/856.52 = 0.93$$

From the figure at $NTU=0.93$ & $C=0.68$, the Effectiveness (ϵ) = 0.55

$$\text{Effectiveness-NTU method } Q = \epsilon C_{\min} (T_{h,i} - T_{c,i}) = 0.55 \times (856.52) \times (360 - 300) = 28.265 \text{ kW}$$

$$T_{h,o} = T_{h,i} - Q/C_h = 360 - 28265/856.52 = 327\text{K (or)} 54^\circ\text{C}$$

$$T_{c,o} = T_{c,i} + Q/C_c = 300 + 28265/1256.25 = 322.49\text{K (or)} 49.34^\circ\text{C}$$

Figure 2.2: Effectiveness for Cross Flow Exchanger with Fluids Unmixed

2.1.2. Calculations of the Total Surface Area of the Radiator with Trapezoidal Fin

$$\text{Total tubes area of Heat Transfer} = 1.20\text{m}^2$$

$$\text{Total surface area of Fins} = 1.8\text{m}^2$$

Additional fin element Dimensions: 24 mm*2.5 mm

$$\text{Total area of the additional fin element} = 0.5\text{m}^2$$

$$\text{Total Heat transfer area of the radiator with Trapezoidal fin} = 1.20 + 1.8 + 0.5 = 3.5\text{m}^2$$

2.1.2(a). Calculation of Outlet Temperature and Heat Dissipation Rate when Water as Coolant

$$C = C_{\min}/C_{\max} = \dot{m}_h c_h / \dot{m}_c c_c = (0.225 \times 4180) / (1.25 \times 1005) = 942.17 / 1256.25 = 0.74$$

$$NTU = AU_m / C_{\min} = 3.5 \times 250 / 942.17 = 0.92$$

From the figure at $NTU=0.92$ & $C=0.74$ the Effectiveness (ϵ) = 0.54

$$\text{Effectiveness-NTU method } Q = \epsilon C_{\min} (T_{h,i} - T_{c,i}) = 0.54 \times 942.17 \times (360 - 300) = 30.526 \text{ kW}$$

$$T_{h,o} = T_{h,i} - Q/C_h = 360 - 30526 / 942.17 = 327.6K \text{ (or) } 54.45^\circ C$$

$$T_{c,o} = T_{c,i} + Q/C_c = 300 + 30526 / 1256.25 = 324.29K \text{ (or) } 51.14^\circ C$$

2.1.2(b). Calculation of Outlet Temperature and Heat Dissipation Rate when Ethylene Glycol as Coolant

$$C = C_{\min}/C_{\max} = \dot{m}_h c_h / \dot{m}_c c_c = (0.225 \times 3800) / (1.25 \times 1005) = 856.52 / 1256.25 = 0.68$$

$$NTU = AU_m / C_{\min} = 3.5 \times 250 / 856.52 = 1.02$$

From the figure at $NTU=1.02$ & $C=0.68$ the Effectiveness (ϵ) = 0.62

$$\text{Effectiveness-NTU method } Q = \epsilon C_{\min} (T_{h,i} - T_{c,i}) = 0.62 \times (856.52) \times (360 - 300) = 31.862 \text{ kW}$$

$$NTU = AU_m / C_{\min} = 3.5 \times 250 / 856.52 = 1.02$$

From the figure at $NTU=1.02$ & $C=0.68$ the Effectiveness (ϵ) = 0.62

$$\text{Effectiveness-NTU method } Q = \epsilon C_{\min} (T_{h,i} - T_{c,i}) = 0.62 \times (856.52) \times (360 - 300) = 31.862 \text{ kW}$$

$$T_{h,o} = T_{h,i} - Q/C_h = 360 - 31862 / 856.52 = 322.79K \text{ (or) } 49.64^\circ C$$

$$T_{c,o} = T_{c,i} + Q/C_c = 300 + 31862 / 1256.25 = 325.36K \text{ (or) } 52.36^\circ C$$

3. SIMULATION: (CATIA MODELS)

Tube Dimensions: Width – 22 mm, Height – 2 mm, Thickness – 0.26 mm

Fin Dimensions : Width – 24 mm, Height – 8.17 mm, Thickness – 0.07 mm

Number of Fins per inches – 10 (For conventional Fin)

Number of Fins per inches – 7 (For Trapezoidal Fin)

Figure 3.1: Conventional Design V-Arrangement of Fins

Figure 3.2: Modified Design Trapezoidal -Arrangement of Fins

3.1. CFD Boundary Conditions

Table 3.1: CFD Boundary Conditions

Air Inlet: Velocity(m/s)	Air Outlet: Static Pressure (Pa)	Coolant Inlet: Velocity(m/s)	Coolant Outlet: Static Pressure (Pa)	Air Inlet Temperature(K)	Coolant Inlet Temperature(K)
16.666	0	0.001	0	300	360

3.1.1. Properties Table

Table 3.2: Properties Table

Properties	Air	Ethylene Glycol	Aluminium
Density(kg/m ³)	1.225	1073	2719
Specific heat(j/kg-K)	1005	3800	871
Thermal conductivity(W/m-K)	0.0242	0.318	202.4

4. RESULTS

4.1. Theoretical Results Table

Table 4.1: Theoretical Results Table

Type of Fin Design	Coolant (Water/Ethylene Glycol)	Inlet Temperature(K)	Outlet Temperature(K)	Heat Dissipation (kJ)
Conventional (A=3.2m ²)	Water	360	331.2	27.134
	Ethylene Glycol	360	327	28.265
Trapezoidal (A=3.5m ²)	Water	360	327.6	30.526
	Ethylene Glycol	360	322.79	31.862

4.2. Conventional Design Results

Figure 4.1: Temperature Distribution Contour

Figure 4.2: Stream Line

Figure 4.3: Residual Plot-Conventional Design

4.3. Trapezoidal Design Results

Figure 4.4: Temperature Distribution Contour

Figure 4.5: Stream Line

Figure 4.6: Residual Plot-Trapezoidal Design

5. CONCLUSIONS

Numerical simulation has been carried out for conventional design and Trapezoidal design for heat dissipation rate and outlet temperature, where trapezoidal design gives better results.

The modified fin design with lesser fin density lowers the outlet coolant temperature, thereby it improves thermal performance of the radiator.

The modified coolant used Ethylene Glycol also gives lower outlet temperature for both conventional design and modified design.

Hence the present study with modified fin design and coolant improves the performance of the radiator there by it improves the engine efficiency.

REFERENCES

1. Raja Suryan G T, Amarkarthik A, *Improving the Thermal Performance of Automobile Radiators - The Role of Internal Radiator Fins*. *International Journal For Trends In Engineering & Technology* Volume 14 Issue 1 – June 2016 - Issn: 2349 – 9303
2. A. Kraus, A. Bar-Cohen, A. A. Wative, *Cooling electronic equipment*, in: *Mechanical Engineers' Handbook: Energy and Power*, vol. 4, third ed., 2006.
3. C. H. Chen, C. C. Wang, *A novel trapezoid fin pattern applicable for air-cooled heat sink*, *Heat Mass Transf.* 51 (2015) 1631–1637.
4. Prof. A. R. Khot, Prof. D. G. Thombare, “An Overview of Radiator Performance Evaluation and Testing”, *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)* ISSN: 2278-1684, PP: 07-14
5. A. D. Kraus, A. Aziz, J. R. Welty, *Extended Surface Heat Transfer*, John Wiley, NY, 2001.
6. E. M. A. Mokheimer, *Performance of annular fins with different profiles subject to variable heat transfer coefficient*, *Int. J. Heat Mass Transf.* 45 (2002) 3631–3642.
7. I. N. Dul'kin, G. I. Garas'ko, *Analytical solutions of 1-D heat conduction problem for a single fin with temperature dependent heat transfer coefficient – I. Closed-form inverse solution*, *Int. J. Heat Mass Transf.* 45 (2002) 1895–1903.
8. A. G. Agwu Nnanna, A. Haji-Sheikh, D. Agonafer, *Effect of variable heat transfer coefficient, fin geometry, and curvature on the thermal performance of extended surfaces*, *ASME J. Electr. Pack.* 125 (2003) 456–460.

9. A. Aziz, A. Beers-Green, *Performance and optimum design of convective radiative rectangular fin with convective base heating, wall conduction resistance, and contact resistance between the wall and the fin base*, *Energy Convers. Manag.* 50 (2009) 2622–2631.
10. KM, S., GL, S., BK, A. K., & Anand, A. (2016). *Foam Finned Down Flow Type Automotive Radiator Rating and Sizing*.
11. A. Aziz, S. M. Enamul-Huq, *Perturbation solution for a convecting fin with variable thermal conductivity*, *ASME J. Heat Transf.* 97 (1975) 300–302.
12. K. Y. Kim, M. A. Moon, *Optimization of a stepped circular pin-fin array to enhance heat transfer performance*, *Heat Mass Transf.* 46 (2009) 63–74.
13. Yazicioğlu B., “*Performance of Rectangular Fins on a Vertical Base in Free Convection Heat Transfer*”, M. S. Thesis in Mechanical Engineering, Middle East Technical University, Ankara (2005).
14. Güvenc A., “*An Experimental Investigation on Performance of Rectangular Fins on a Vertical Base in Free Convection Heat Transfer*”, M. S. Thesis in Mechanical Engineering, Middle East Technical University, Ankara (1999).
15. Starner K. E. and McManus H. N. “*An Experimental Investigation of Free Convection Heat Transfer from Rectangular Fin Arrays*”, *Journal of Heat Transfer*, 273-278, (1963).
16. A. Rajabi, *Homotopy perturbation method for fin efficiency of convective straight fin with temperature dependent thermal conductivity*, *Phys. Lett. A* 364 (2007)33–37.
17. C. Arslanturk, *A decomposition method for fin efficiency of convective straight fins with temperature dependent thermal conductivity*, *Int. Commun. Heat MassTransf.* 32 (2005) 831–841.