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Performance of Aluminium Silicon Carbide (AlSiC) Metal Matrix Composite as a Fin Material

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Abstract— The paper investigates the performance of Aluminium Silicon Carbide (AlSiC) as a fin material against other traditional fin materials namely –aluminium and copper. Its performance is also compared to steel which is rarely but sometimes used as fin material. Theoretical approaches using the effectiveness number of transfer units ε-NTU is used to obtain the rate of heat transferred. The results of the analysis and study shows that fin efficiency and heat transferred by AlSiC compares favourably with that of traditional fin materials while offering lesser weight and higher strength than any of these materials.

Index Terms— AlSiC, Composite, Extended surfaces, Heat exchanger, Fin, Heat transfer, Materials.

1 Introduction

TRADITIONAL extended surface materials are made up of highly conductive metals notably copper and aluminium [1]. Extended surfaces (fins) are used to enhance the heat transfer from a given surface [2] and they (extended surfaces) can increase the rate of heat transferred several fold [1]. The rate of heat transferred is obtained from Newton's law of cooling given as [3],

$$q = h * A_t * \Delta T \quad (1)$$

To increase heat transferred from a surface therefore, the surface area needs to be increased. Attaching an extended surface increases the surface area A_t of heat transferred without necessarily increasing the surface area of the body itself.

Extended surfaces are widely used in radiators of automobiles and industrial generators; for cooling electrical appliances such as motors and transformers; as heat sinks in electronics such as motherboards; in air-cooling of internal combustion engines such as the motor cycle IC engine; and application in various heat exchangers.

The choice of one traditional fin material over the other for use in a heat exchanger is often a trade-off between desirable qualities or material properties. Copper for instance, has the highest rate of heat rejection per unit volume than any other material due to its high thermal conductivity. Aluminium on the other hand has a cost and weight advantage, it is also very abundant making it the second most important metal on earth after steel [4] and as Niem [5] put it, "at one third the cost and density, the choice is clear". However, the choice is not that clear if strength is a major design consideration as aluminium is temperature challenged. At an elevated temperature of say 250°C, the strength of copper is over six times that of aluminium. The following figures show the densities and strength of fin materials considered in this report.

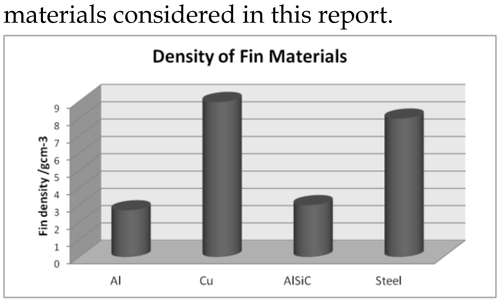

Fig. 1: Density of fin materials

The large weight of copper is its big disadvantage in designs where weight is a major design criterion.

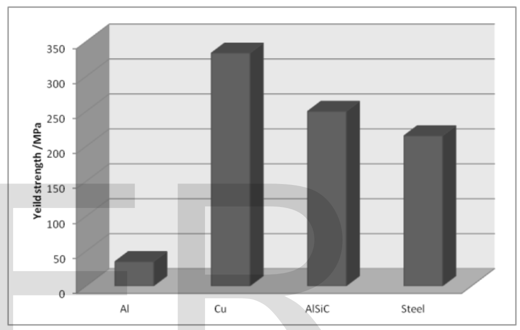

Fig. 2: Strength of fin materials

Aluminium is clearly not an ideal material where strength is an essential design criterion.

2 Mathematical Model

A mathematical model was developed using the *ε-NTU* method to determine the heat transferred from hot coolant of a radiator whose tube surface area of heat transferred is increased by triangular plate fins as illustrated in fig. 3 below.

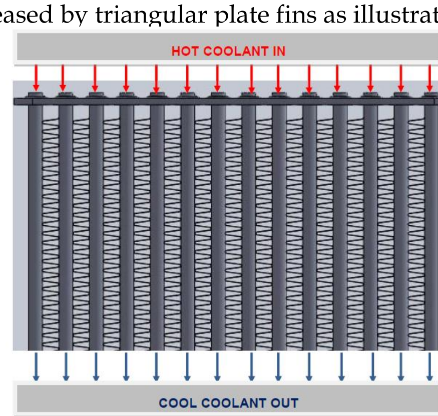

Fig. 3: Section of radiator showing tubes and triangular fins

2.1 Heat Transfer Coefficient from Nusselt Number

A cross section of the radiator unit is shown in fig. 3 with rectangular tubes containing coolant and fins as shown. The parameters are as follows:

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Tube inner diameter or coolant thickness (d_i) = 0.9mm;
Length of tube (L) = 509mm; Length of fin (l_f) = 5.72mm;
Tube or fin width (w) = 35mm;
Distance of fins apart (f_d) = 1.96mm;
Perpendicular distance of fin (f_h) = 5.64mm;
Thickness of fin (t) = 0.07mm;
Number of tubes in design (n_t) = 57;
Number of fins column (n_f) = 58
Total number of air passages (n_a) = $n_f * L/f_d$ (2)
Total coolant surface area
$A_c = (2d_i.L + 2w.L) * n_t$ (3)
Total air surface area
$A_a = \left(2l_f.w + f_d.f_h\right) * n_a$ (4)

2.1.1 Coolant Side Heat Transfer

The coolant used for the model is ethylene-glycol (50:50) at inlet temperature of 120^{0}C;
Thermal conductivity $K_c = 0.4151W/(m.K)$;
Specific heat $C_{p,c} = 3681.92J/(kg.K)$;
Density $\rho_c = 1015.57kg/m^3$;
Dynamic viscosity $\mu_c = 74.4082 * 10^{-5}\ Pa.s$;
and Volume flow rate $\dot{Q}_c = 0.0018927m^3/s$
Wetted perimeter of tube $P = 2(d_i + w)$ (5)
Pipe cross-sectional area $A = d_i.w$ (6)
Hydraulic diameter $D_H = 4A/P$ (7)
Velocity of flow can be obtained from the coolant volumetric flow rate $\dot{Q}_c$ and the pipe cross sectional area for all tubes as
$v_c = \dot{Q}_c/(n_t.A)$ (8)
Reynolds number for the flow is obtained by substituting values from (7) and (8) into (9)
$Re = (\ \rho.v.D_H/\mu)_{coolant}$ (9)

The Nusselt number is important in evaluating heat transfer based on the fluid properties. In term of enthalpy, Nusselt number is given as $Nu = hD/k$ (10)
It is can also be given as the function of Reynolds number *Re*, Prandtl number *Pr* and a constant *K* which is based on the heat exchanger geometry. Generally, the relation is of the form
$Nu \propto K(Re)^m(\mathrm{Pr})^n$
where K is a generic scalar variable.
For laminar flow across flat plates, Thirumaleshwar [2] gives Nusselt number as
$Nu = 0.664 * \sqrt{Re} * \sqrt[3]{Pr}$ (11)

If a flow is in a transitional region however as it is in this case (coolant side), both its laminar and turbulent layers are to be considered when obtaining Nusselt number. The Nusselt number for transitional flow is obtained using Gnielsinki correlation for the turbulent flow component (Nu_{turb}) and aspect ratio concept for fully developed laminar flow component ($Nu_{laminar}$). These are then combined as suggested by Taborek *et al* [6] as cited by Shah & Sekulic [7] to obtain the Nusselt number for the flow as follows.
$Nu_{turb} = [(f/2)(Re - 1000) * \mathrm{Pr}\,)/(1 + 12.7 * \sqrt{(f/2)}(\sqrt[3]{\mathrm{Pr}^2} - 1)]$ (12)
Where f is the correction factor given as
$f = A + B * Re^{-1/m}$ (for 2,100 ≤ Re ≤ 4,100) (13)
The equation below combines both flow components
$Nu = \emptyset Nu_{laminar} + (1 - \emptyset)Nu_{turb}$ (14)
Where $\emptyset = 1.33 - (Re/6000)$ (15)
To obtain the Prandtl number,
$Pr = C_p\mu/k$ (16)

2.1.2 Air Side Heat Transfer

From air property table, the following air-side parameters were obtained corresponding to air temperature at 45^{0}C and atmospheric pressure:
Air temperature $T_{air} = 45^0C$;
Density $\rho_{air} = 1.1373kg/m^3$;
Dynamic viscosity $\mu_{air} = 1.9120 * 10^{-5}\ Pa.s$
Specific heat $C_{p,air} = 1004.16J/(kg.K)$;
Thermal conductivity $K_{air} = 0.0266W/(m.K)$;
Volume flow rate $\dot{Q}_{air} = 1.1086m^3/s$
To calculate the Reynolds number of the air at fin Re, Hydraulic diameter D_H of the fin is required
Wetted perimeter of tube $P = 2f_l$ (17)
Area of triangular fin section $A = (f_d/2).f_h$ (18)
The air side Reynolds number is found to be laminar therefore (11) is used to obtain Nu.

2.2 Fin Efficiency and Overall Surface Efficiency

Overall surface efficiency η_o characterises the performance of a single fin. It is given by Incropera and Dewitt [8] as the ratio of total heat transfer from the fin and its base area to the ideal heat transfer if the fin were at base temperature. This should not be confused with the effectiveness ε which is the ratio of actual heat transfer rate to the maximum possible heat transfer rate.
$\eta_o = 1 - (nA_f/A)(1 - \eta_f)$ (19)
Where η_f is the efficiency of a single fin which is dependent on geometry of the heat exchanger. η_f for this model i.e straight fins of triangular profile (fig. 4) is obtained from [8] and [7] as
$\eta_f = (tanhml)/(ml)$ (20)
Where $m = \sqrt{2h/(k_f\delta)}$ (21)

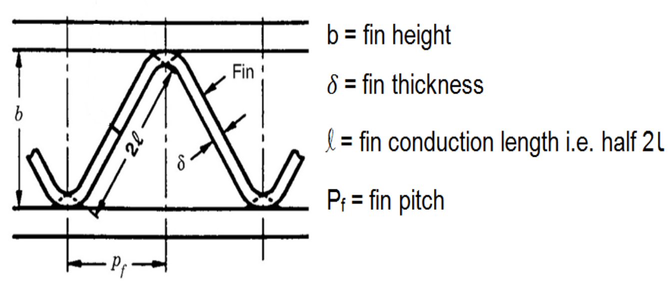

Fig. 4: Triangular plate fin of uniform thickness

2.3 Thermal Resistance & Overall Heat Transfer Coefficient

For the analytical model earlier developed, the thermal resistance across the model is shown in fig. 5 below.

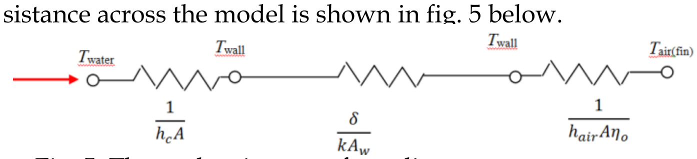

Fig. 5: Thermal resistance of a radiator

Total thermal resistance of the thermal circuit is the sum of resistance on each node
$R = R_{water} + R_{wall} + R_{air(fin)}$ (22)
Overall heat transfer coefficient mathematical relation [8], [1]
$UA = 1/\Sigma R\ = \left((1/h_cA_c) + (1/n\eta_oh_fA_f) + \delta/(kA_w)\right)^{-1}$ (23)

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2.4 Heat Transfer Rate

The rate of heat transferred using the *ε-NTU* method is obtained as follows.
From energy balance on the hot and cold fluid, the heat transfer rate of a heat exchanger [1], [8] is

$$\dot{q} = \dot{m}c_{p,c}\left(T_{c,e} - T_{c,i}\right) = \dot{m}c_{p,h}\left(T_{h,i} - T_{h,e}\right) \quad (24)$$

The exit temperatures for the cold and hot side $T_{c,e}$ and $T_{h,e}$ are unknown therefore the rate of heat transferred cannot be obtained directly. Using the effectiveness NTU method, the maximum heat transfer rate possible q_{max} is first obtained and this compared to actual heat transfer $\dot{q}$ by a ratio ε, heat transfer effectiveness. Heat transfer effectiveness is a dimensionless quantity given as $\varepsilon = \dot{q}/q_{max}$
Therefore the actual heat transfer

$$\dot{q} = \varepsilon . q_{max} \quad (25)$$

Where q_{max} is the maximum possible heat transfer rate between the extreme temperatures. The maximum heat is transferred when hot fluid is cooled to the inlet temperature of the cold fluid or when a cold fluid is heated to the inlet temperature of a hot fluid. Therefore, maximum change in temperature would be the difference in temperature between the hot fluid and cold fluids at inlet i.e

$$\Delta T_{max} = T_{h,i} - T_{c,i}$$

Also, it can be noticed that for the above described scenario to take place, the specific heat capacities must be the same otherwise maximum heat is transferred at the fluid with lowest specific heat capacity indicated as $c_{p,min}$. Maximum heat transfer rate is therefore given as

$$\dot{q}_{max} = \dot{m}c_{p,min}(T_{h,i} - T_{c,i}) \quad (26)$$

Combining (25) and (26),

$$\dot{q} = \varepsilon . \dot{m}c_{p,min}(T_{h,i} - T_{c,i}) \quad (27)$$

The heat transfer effectiveness ε depends on the geometry and flow type of a heat exchanger. For a cross flow (single phase) heat exchanger having both fluids unmixed, which is the case considered in this design, the effectiveness is given by Kays and London as cited by Cengel [1],

$$\varepsilon = 1 - \exp\left((NTU^{0.22}/c)(exp(-c * NTU^{0.78}) - 1)\right) \quad (28)$$

Where thermal ratio $= (\dot{m}c_{p,min})/(\dot{m}c_{p,max})$ (29)

And $\dot{m} = \dot{Q} * \rho$ (30)

The number of heat transfer unit (NTU) is a dimensionless parameter used extensively in heat transfer analysis and is given by [8] and [1] as

$$NTU = UA_t/\dot{m}c_{p,min} \quad (31)$$

3 Results and Discussions

Results obtained for the heat transferred, surface efficiency and fin efficiency are presented.

3.1 Heat Transferred

Fig. 6 shows the heat transfer comparative data over a range of air velocities. Aluminium, copper and AlSiC are seen here to offer reasonably close heat transfer rate.

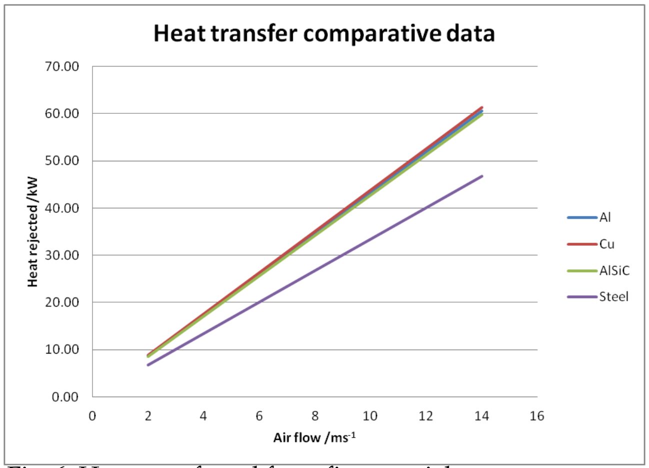

Fig. 6: Heat transferred from fin materials

3.2 Surface Efficiencies

Similarly, fig. 7 also indicates that the surface efficiency of AlSiC compares favourably with that of traditional fin materials.

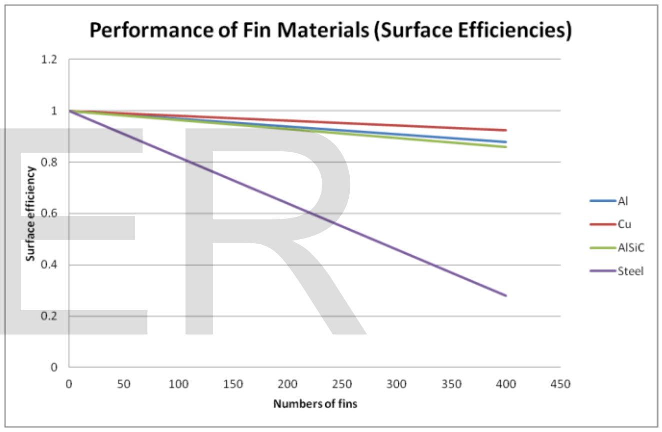

Fig. 7: Comparing surface efficiencies

3.3 Fin Efficiencies

Graph of comparative data of fin efficiencies for various fin materials considered is presented in fig. 8 below. Again, the result shows AlSiC, though having lower fin efficiency, still compares favourably with traditional fin materials.

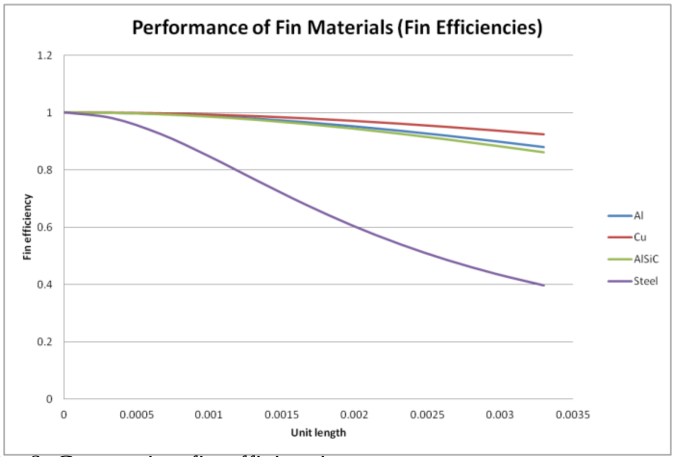

Fig. 8: Comparing fin efficiencies

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4 Conclusion

Aluminium silicon carbide (AlSiC) is a metal matrix composite made up of aluminium (matrix) and SiC fibre has been shown in this report to compare favourably with traditional fin materials.

Due to its low weight and high strength, AlSiC composite is suitable for low weight applications where aluminium has been the favourite; and high strength applications where copper has been used as choice material. The significance of this is that strength can no longer be compromised for low weight and vice versa as AlSiC offers both though at a slightly lower performance. Another major advantage of AlSiC over other materials compared in this report is its low coefficient of expansitivity which makes it a choice material for applications sensitive to geometrical changes.

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NOMENCLATURE

Symbols	Description	Units
A	Surface area	m^2
C	Thermal ratio (C_{min}/C_{max})	
C_p	Specific heat	$J/(kg.K)$
D_H	Hydraulic diameter	m
E	Young modulus	Pa
f	Friction factor	
f_d	Distance of fins apart	m
f_h	Perpendicular distance of fin	m
h	Convection heat transfer coefficient	$W/(m^2K)$
L	Length of tube	m
m	standard extended surface parameter	m^{-1}
$\dot{m}$	Mass flow rate	kgs^{-1}
n	Number of fins	
n_a	Numbers of air passages	
n_f	Numbers of fins column	
n_t	Numbers of tubes in design	
Nu	Nusselt number	
P	Perimeter	m
Pr	Prandtl number	
$\dot{Q}$	Volume flow rate	m^3/s
R	Thermal resistance	K/W
Re	Reynold number	
t	Thickness of fin	m
T	Temperature	K
U	Overall heat transfer	$W/(m^2K)$
v	Poisson ratio, volume flow rate, velocity	$-, m^3s^{-1}, ms^{-1}$
w	Tube or fin width	m
ε	Heat exchanger effectiveness	
μ	Dynamic viscosity	$Pa.s$
ρ	Density	kg/m^3
Subscripts		
a	Air	
c	Coolant	
C	Cold	
f	Fin property	
H	Hot	
o	Overall surface	
max	Maximum	
p	Prime	
t	Total	

References

[1] Y. C¸engel, *Heat and mass transfer*, 3rd ed. New Delhi: Tata McGraw-Hill, 2007, pp. 159, 609-646.
[2] M. Thirumaleshwar, *Fundamentals of heat and mass transfer*, 1st ed. New Delhi: Dorling Kindersley, 2009, pp. 221, 469.
[3] J. Holman and S. Bhattacharyya, *Heat transfer in SI units*, 9th ed. New Dehli: Tata McGraw-Hill, 2008, pp. 10, 538.
[4] K. Budinski and M. Budinski, *Engineering materials - properties and selection*, 8th ed. New Delhi: Prentice-Hall India, 2007, pp. 653.
[5] A. Niem, 'At one-third the cost—and density—the choice is clear | AMM | 21/08/2008', *Amm.com*, 2008. [Online]. Available: http://www.amm.com/Article/2531053/At-one-third-the-costand-densitythe-choice-is-clear.html. [Accessed: 12-Jun- 2014].
[6] J. Taborek, G. Hewitt and N. Afgan, *Heat exchangers*, 1st ed. Washington: Hemisphere Pub. Corp., 1983.
[7] R. Shah and D. Sekulic´, *Fundamentals of heat exchanger design*, 1st ed. Hoboken, NJ: John Wiley & Sons, 2003, pp. 283, 475, 482, 483.
[8] F. Incropera and D. DeWitt, *Fundamentals of heat transfer*, 5th ed. New Delhi: John Wiley & Sons (Asia) Pte, 2006, pp. 136-144, 645-647.

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