

Analytical Performances Evaluation of Compact Heat Exchanger Using Nano-fluids



Engineering

KEYWORDS : Nanofluid, compact heat exchanger, louvered fin geometry, Performance.

Dr. Suwarna Torgal

Department of Mechanical Engineering, Institute of Engineering & Technology, Devi Ahilya University, Indore, MP-452017, India

Ashish Mishra

Department of Mechanical Engineering, Institute of Engineering & Technology, Devi Ahilya University, Indore, MP-452017, India

ABSTRACT

Researches in heat transfer have been carried out over the previous several decades, leading to the development of the currently used heat transfer enhancement techniques. Water and ethylene glycol as conventional coolants have been widely used in a compact heat exchanger for many years. These heat transfer fluids offer low thermal conductivity. With the advancement of nanotechnology, the new generation of heat transfer fluids called, "nanofluids" have been developed and researchers found that these fluids offer higher thermal conductivity compared to that of conventional coolants. This study is focused on the application of mixture of water (80%) and ethylene glycol (20%) based nanofluids (Cu, SiC, Al₂O₃, and TiO₂) are presented in this article. Use of nanofluid as coolant in compact heat exchanger improves the effectiveness, cooling capacity with the reduction in pumping power. Effects of various operating parameters such as air inlet temperature, coolant inlet temperature, air and coolant mass flow rates, effectiveness, pumping power and exergy analysis are studied as well. SiC-80% H₂O-20% EG (base fluid) yields best performance in compact heat exchanger having louvered fin geometry followed by Al₂O₃-base fluid, TiO₂-base fluid and Cu-base fluid. The maximum cooling improvement for SiC is 18.36%, whereas that for Al₂O₃ is 17.39%, for TiO₂ is 17.05% and for Cu is 13.41% as coolants and enhancement in second law efficiency for SiC is highest (21.89%) followed by Al₂O₃ (21.53%), TiO₂ (21.3%), Cu (18.97%) nanofluids compare to base fluid as a coolant alone. Present study reveals that the nanofluids may effectively use as coolant in compact heat exchanger to improve the performance.

NOMENCLATURE

A total heat transfer area ($A_f + A_s$), m²
 A_f fin surface area, m²
 A_l louver surface area, m²
 A_t external tube surface area, m²
 D_h hydraulic diameter of fin array, mm
 D_m major diameter, mm
 F_p fin pitch, mm
 F_d fin depth, mm
 F_l fin length, mm
 H_o heat transfer coefficient, W/m² k
 j Colburn factor, dimensionless
 k thermal conductivity, W/m K
 L fin length for heat conduction from primary to Midpoint between plates for symmetric heating, mm
 L_h louver height, mm
 L_l louver length, mm
 l_p louver pitch, mm
 m ($\sqrt{2h_o k_f \delta_f}$), m⁻¹
 C heat capacity rate, W/K
 C_p specific heat J/kg K
 C^* C_p , min/ C_p , max
 DP pressure drop, Pa
 EG ethylene glycol
 Ex exergy (W)
 f fanning friction factor, dimensionless
 G mass velocity, kg/m²s
 H total coolant flow length, m
 NTU number of heat transfer units
 Nu Nusselt number
 P pumping power
 Pr Prandtl number
 Re Reynolds number
 t fin thickness, m
 T temperature, K
 U overall heat transfer coefficient W/m² K
 W mass flow rate
 S_1 non-louvered inlet and exit fin regions, mm
 S_2 re-direction length, mm
 σ minimum free flow area/frontal area
 η_f fin efficiency
 η_o total surface temperature effectiveness
 μ Dynamic viscosity, Ns/m²
 v volumetric flow rate, m³/s
 ρ density, kg/m³
 ϕ volume fraction of particles

ϵ heat exchanger effectiveness
 α total one side of heat transfer area/total volume

Subscripts

fr frontal area
 a air
 nf nanofluids
 f fluid (base fluid)
 p particles
 in inlet

1. Introduction

To enhance the cooling rate by addition of fins is the earliest approach but this approach of increasing cooling rate already reached to their limit. It has been proven that conventional fluids, such as water and EG have poor convective heat transfer performance and therefore high compactness and effectiveness of heat transfer systems are necessary to achieve the required heat transfer [1]. In recent years, extensive research has proven that nanofluids (a suspension of nanometer-sized metallic particles in a base fluid) are superior as a heat transfer agent over conventional fluids [2]. Nanofluids have the potential to reduce such thermal resistances, and the industrial groups that would benefit from such improved heat transfer fluids are quite varied. They include transportation, electronics, medical, food, defence, nuclear, space, biomedical and manufacturing of many types [3]. Recently there has been considerable research highlighting superior heat transfer performances of nanofluids in automotive radiator. Leong *et al.* [2] reported that about 3.8% of heat transfer enhancement and almost 18.7% reduction of air frontal area is achieved by adding 2% copper nanoparticles at Reynolds number of 6000 and 5000 for air and coolant respectively. The performance of finned tube heating units with nanofluids has been compared mathematically with a conventional heat transfer fluid which comprised of 60% EG and 40% water by Strandberg and Das [4]. Their model predicted an 11.6% increase in finned tube heating output under certain conditions with the 4% Al₂O₃/60% EG nanofluid and an 8.7% increase with the 4% CuO/60% EG nanofluid compared to heating output with the base fluid. A three-dimensional laminar flow and heat transfer with two different nanofluids, Al₂O₃ and CuO, in an ethylene glycol and water mixture circulating through the flat tubes of an automobile radiator and evaluate their superiority over the base fluid studied by Das *et al.*[5]. Their analyses showed substantial increase in the average heat transfer coefficient with concentration. The simulation of cooling effects of water, TiO₂

nanofluid, Al₂O₃ nanofluid and CuO nanofluid have done by Peng *et al.* [6]. They showed that compared to water by using TiO₂, Al₂O₃ and CuO nanofluid, the average surface heat transfer coefficient is increased by 10.82%, 8.43% and 11.24%, and correspondingly the pump power is increased only by 1.06%, 1.30% and 1.98%, respectively. Peyghambarzadeh *et al.* [1] experimentally performed the forced convection heat transfer in a car radiator and they reported that by the addition of only 1 vol. % of Al₂O₃ nanoparticle into the water or EG, an increase in Nusselt number of about 40% in comparison with the pure water and pure EG. However, the analytical analysis of louver finned tube radiator using nanofluid is scarce.

In the present work, nanofluids have been taken as a coolant for compact heat exchanger such as automotive radiators, and numerically analyzed the radiator performance by considering louver fin geometry. Also the performance of radiator has been compared by considering different nanoparticle (Cu, SiC, Al₂O₃, TiO₂) in a base fluid like 80% water-20% Ethylene glycol. Effects of various operating parameters such as air inlet temperature, coolant inlet temperature, air and coolant mass flow rates on radiator heat transfer, effectiveness, pumping power and irreversibility are studied as well.

II. Theoretical modeling and Simulation:

The automotive radiator which is of compact heat exchanger type is made of four major components, coolant inlet tank, outlet tank, pressure cap and core. Coolant tanks are positioned either on top and bottom of the core. The major sub components of the core are coolant tubes and fins. Flat tubes are more popular for automotive applications due to their lower profile drag compared with round tubes. Louver fin radiator consider in this study is illustrated in Fig.1-1c, whereas its dimension is shown in Table.1.

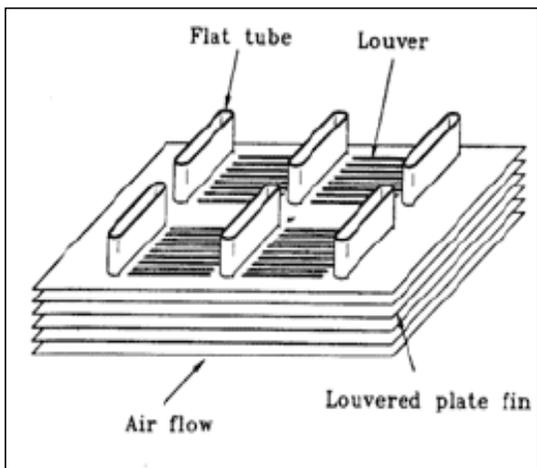


Figure 1a Plate and Tube Louver fin Geometry

The mathematical model has been developed based on first law of thermodynamics including heat transfer and fluid flow effects. Following assumptions have been made for analysis:

- 1) Properties of nanofluid as well as air assumed to be constant.
- 2) Steady state process.
- 3) All the heat rejected from nanofluid absorbed by air flow inside radiator.

Initially, air side calculations were performed to determine air heat capacity, air heat transfer coefficient, fin efficiency and total surface temperature effectiveness. These data were needed to calculate heat exchanger effectiveness: NTU number and overall heat transfer coefficient for the nanofluids' side calculation. The mathematical formulations are shown below.

Figure 1b Definition of geometrical parameters for a multi-louvered fin heat exchanger.

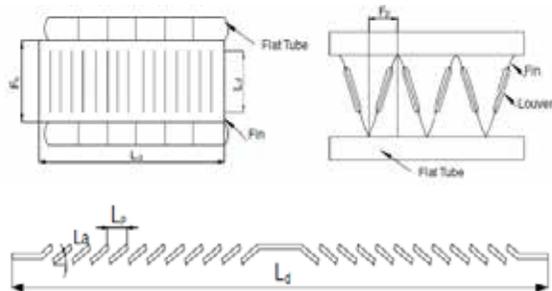


Figure 1c Cross section of multi-louvered fin geometry.

1 Air side calculation

(a) Air heat capacity rate, Ca can be expressed as:

$$C_a = W_a c_{p,a} \tag{1}$$

Core mass velocity of air is expressed as

$$G_a = \frac{W_a}{A_f \sigma_a} \tag{2}$$

(b) Heat transfer coefficient, ha can be expressed as

$$h_a = \frac{j_a G_a c_{p,a}}{Pr_a^{2/3}} \tag{3}$$

Correlation for the Colburn j factor [18]

$$j_a = 0.26712 Re_a^{-0.1944} \times \left(\frac{La}{90}\right)^{0.257} \times \left(\frac{F_p}{L_p}\right)^{-0.5177} \times \left(\frac{F_h}{L_p}\right)^{-1.9045} \times \left(\frac{L_h}{L_p}\right)^{1.7159} \times \left(\frac{L_d}{L_p}\right)^{-0.2147} \times \left(\frac{t}{L_p}\right)^{-0.05} \tag{4}$$

Reynolds number expression for louver fin is [18]

$$Re_a = \frac{\rho_a u_a L_p}{\mu a} \tag{5}$$

(c) fin efficiency, ηf can be expressed as

$$\eta_f = \frac{\tanh ml}{ml} \tag{6}$$

Where,

$$m = \sqrt{\frac{2h_a}{kt}} \tag{7}$$

(d) Total surface temperature effectiveness, can be expressed as

$$\eta_o = 1 - \frac{A_f}{A} (1 - \eta_f) \tag{8}$$

2 Nanofluid side Calculation

The parameters needed for nanofluid side calculation are nanofluid heat transfer coefficient, nanofluid heat capacity rate. Heat transfer coefficient can be expressed as:

$$h_{nf} = \frac{Nu_{nf} k_{nf}}{D_{h,nf}} \tag{9}$$

Where, knf is calculated Koo & Sleinstruer relation [7]

Nusselt number for nanofluid is expressed as [8]

$$Nunf = 0.021(Re_{nf})^{0.8}(Pr_{nf})^{0.5} \tag{10}$$

Reynolds number expression for nanofluid is

$$Re_{nf} = \frac{G_{nf} D_{h,nf}}{\mu_{nf}} \tag{11}$$

Viscosity of nanofluid is calculated based on correlation from Maiga [9]

$$\mu_{nf} = \mu_f (1 - 0.19\phi + 306\phi^2) \tag{12}$$

C_{p,nf} and ρ_{nf} were calculated based on correlations obtained from Sarkar [3].

$$c_{p,nf} = \frac{(1-\phi)\rho_f c_{p,f} + \phi\rho_p c_{p,p}}{\rho_{nf}} \tag{13}$$

$$\rho_{nf} = (1-\phi)\rho_f + \phi\rho_p \tag{14}$$

Core mass velocity of coolant is expressed as

$$G_{nf} = \frac{W_{nf}}{A_{fr}\sigma_{nf}} \tag{15}$$

Prandtl number expression for nanofluid is

$$Pr_{nf} = \frac{\mu_{nf}c_{p,nf}}{k_{nf}} \tag{16}$$

Heat capacity rate, C_{nf} can be expressed as:

$$C_{nf} = W_{nf}c_{p,nf} \tag{17}$$

Pressure drop can be expressed as:

$$DP_{nf} = \frac{G_{nf}^2 \times f_{nf} \times H}{2 \times \rho_{nf} \times \left(\frac{D_{h,nf}}{4}\right)} \tag{18}$$

Where, the friction factor correlation of nanofluid [3], is given by

$$f_{nf} = 0.3164 \times (Re_{nf})^{-0.25} \left(\frac{\rho_{nf}}{\rho_f}\right)^{0.797} \left(\frac{\mu_{nf}}{\mu_f}\right)^{0.108} \tag{19}$$

3 Performance calculations:

Overall heat transfer coefficient, based on air side can be expressed as below [8], where wall resistance and fouling factors are neglected:

$$\frac{1}{U_a} = \frac{1}{\eta_a h_a} + \frac{1}{\left(\frac{\alpha_{nf}}{\alpha_a}\right) h_{nf}} \tag{20}$$

Number of heat transfer unit is expressed as

$$NTU = \frac{U_a A_{fr,a}}{C_a} \tag{21}$$

(a) Heat exchanger effectiveness for cross-flow unmixed fluid can be expressed as:

$$\epsilon = 1 - \exp\left[\frac{NTU^{0.22}}{C^*} \exp(-C^* NTU^{0.78} - 1)\right] \tag{22}$$

Where, $C^* = \frac{C_{min}}{C_{max}}$ (23)

(b) Pumping power can be expressed as:

$$P = \dot{V}_{nf} \times DP_{nf} \tag{24}$$

Where,

$$\dot{V}_{nf} = \frac{W_{nf}}{\rho_{nf}} \tag{25}$$

(c) Total heat transfer rate can be expressed as:

$$Q = \epsilon C_{min} (T_{nf,in} - T_{a,in}) \tag{26}$$

4 Exergy analyses

The exergy fed to a heat exchanger is destroyed due to two main reasons: (i) Lack of thermal equilibrium arising out of finite temperature difference in and outside the apparatus and (ii) Dissipative effect of fluid friction (dissipative forces arising on account of fluid friction also contribute significantly to irreversibility, in the form of pressure drop).

The Guoy–Stodola theorem provides the basis for calculation of irreversibility in heat exchangers, which is the quantitative measure of the exergy loss in the process and is related to entropy generation as

$$I = T_0 S_{gen} \tag{27}$$

The above-mentioned contributions of exergy loss can be quantified using second law of thermodynamics.

tified using second law of thermodynamics.

Assuming the working fluid follows ideal gas relation, the exergy loss by the hot fluid (nanofluid) is given by,

$$\Delta Ex_{nf} = Q - T_0 \left[\dot{m} C_p \ln\left(\frac{T_{out}}{T_{in}}\right) - \dot{m} R \ln\left(\frac{P_{out}}{P_{in}}\right) \right]_{nf} \tag{28}$$

Similarly, the exergy gain by cold fluid (air) is given by,

$$\Delta Ex_a = Q - T_0 \left[\dot{m} C_p \ln\left(\frac{T_{out}}{T_{in}}\right) + \dot{m} R \ln\left(\frac{P_{in}}{P_{out}}\right) \right]_a \tag{29}$$

At the outlet of the heat exchanger, pressure can be considered to be atmospheric.

The Second law efficiency is the ratio of the minimum exergy which must be consumed to do a task divided by the actual amount amount of exergy consumed in performing the task, is given by,

$$\eta_{II} = \frac{\Delta Ex_a}{\Delta Ex_{nf}} = 1 - \frac{I}{\Delta Ex_{nf}} \tag{30}$$

5 Simulation procedures

For implementing the analysis, a computer program in C++ is developed for the compact heat exchanger. This program is useful in estimating the fluid properties at operating temperatures, surface core geometry of cross flow heat exchanger, heat transfer coefficients, pressure drops, overall heat transfer coefficients and heat transfer rate.

III VALIDATION AND INPUT PARAMETERS:

The simulation code has been validated from the result shown by Vasu [10]. About 5% error has been occurred during comparison of result for louver fin heat exchanger.

Description	Air	Coolant
Fluid mass rate	10-20kg/s	4-6kg/s
Fluid inlet temperature	10-50°C	80-100°C
Core Width	0.6m	
Core height	0.5m	
Core depth	0.4m	

Table 1: Fluid parameters and Normal Operating conditions

Description	Air side	Coolant side
Fin pitch	4.46fin/cm	
Fin metal thickness	0.01cm	
Hydraulic diameter D _h	0.351cm	0.373cm
Min free flow area/frontal area σ	0.780	0.129
Total heat transfer area/total volume α	886 m ² /m ³	138 m ² /m ³
Fin area/Total area β	0.845	

Table 2a: Surface core geometry of flat tubes, continuous fins

Variant	Sample Source	Core Type	Louver Pitch, mm	Louver Length, mm	Louver Angle(deg)
1	C&W(1)	C	1.318	12.44	28

Table 2b: Surface core geometry of the louver fin heat exchanger Continued

Fin pitch, mm	Tube depth,mm	Fin depth,mm	Fin length,mm	Fin thickness,mm
1.8	22	22	16	0.16

Continued

Tube pitch ,mm	Rows of tubes	D _n ,mm
21	1	3.069

Sample source	Fin Material	Louver No	S ₁ ,mm	S ₁ number	S ₂ ,mm
C&W	Al	12	1.815	2	2.55

Table 2c: Geometric Dimension of the louver fin

1	L _a	28°
2	F _n	2 mm
3	L _n	1.2 mm
4	F _h	8 mm
5	L _h	6.5 mm
6	L _{ti}	36.6 mm

Table 2d: Specification of multi-louvered fin parameters [19]

The compact heat exchanger such as radiator which is considered here is mounted on the present turbo-charged diesel engine of type TBD 232V-12 is cross flow compact exchanger with unmixed fluids in (Fig. 1). Radiator consists of 644 tubes make of brass and 346 continuous fins made of Aluminum alloy whose thermal conductivity is 177W/m K [14]. The common geometrical factors and operating conditions are described in the Tables 1 and 2-2d. Properties of base fluid and air are given in Table 3. Properties of nanoparticles are given in Table 4.

Thermal physical properties	Base fluid	Air
Density(kg/m ³)	1008	1.1614
Specific heat (J/kg K)	4020	1007
Viscosity(N-s/m ²)	0.0019	0.00001846
Conductivity(W/m K)	0.58	

Table 3: Thermal physical properties of Base fluid (80%water-20%ethylene glycol) and Air

Thermal physical properties	Copper (Cu)	Silicon Carbide (SiC)	Alumina (Al ₂ O ₃)	Titanium dioxide (TiO ₂)
Density(kg/m ³)	8933	3160	3970	4157
Specific heat (J/kg K)	385	1340	773	692
Conductivity (W/m K)	401	350	40	8.4

Table 4: Thermal physical properties of nanoparticles [11]

IV RESULT AND DISCUSSION

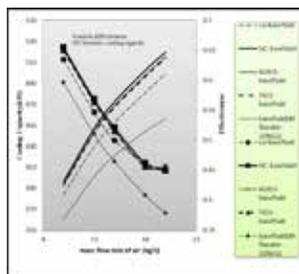


Figure2 Effect of mass flow rate of air on cooling capacity

and effectiveness

a) Influence of varying inlet air mass flow rate:

The variation of cooling capacity and effectiveness with air mass flow rate from 10 to 20 kg/s is shown in Fig.2 keeping constant average values for other input data(mc=5kg/s, Tai=300C, Tnfi=900C). It has been found that with increase in mass flow rate of air, cooling capacity goes on increasing because of increasing heat transfer coefficient and the effectiveness for cooling is goes on decreasing. Also, cooling capacity and effectiveness of nanofluids having base fluid of 80%water-20%EG is much higher as compared to 80%water-20%EG mixture only. On comparing cooling capacity and effectiveness using different nanofluids, it has been observed that nanofluid based on SiC, Al2O3, TiO2 exhibit almost same behavior and they have higher cooling capacity and effectiveness as compared to Cu based nanofluid.

b) Influence of varying coolant mass flow rate:

For the variation of coolant mass flow rate Fig. 3 keeping constant average values for other input data (ma=15kg/s, Tai=300C, Tnfi=900C). It has been observed that both cooling capacity and effectiveness goes on increasing and on comparing cooling capacity and effectiveness of nanofluid using base fluid 80%water-20%EG is much higher when only 80%water-20%EG mixture is used as a coolant.

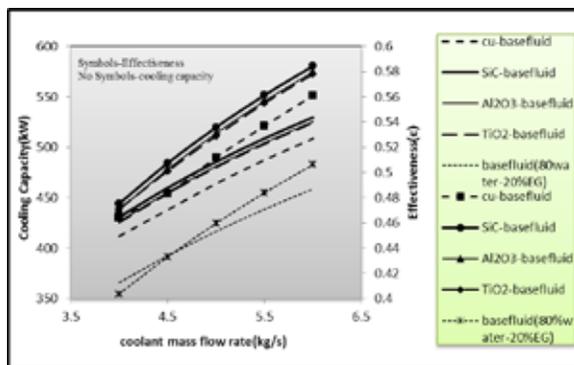


Figure 3 Effect of mass flow rate of coolant on cooling capacity and effectiveness

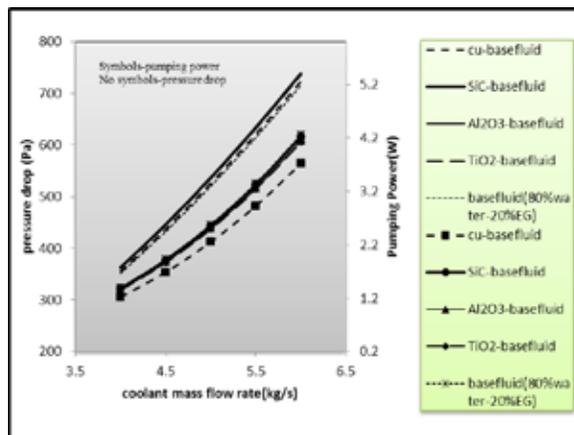


Figure 4 Effect of mass flow rate of coolant on Second law efficiency and pumping power

Result (Fig. 4) also shows that with the variation of coolant mass flow rate, Second law efficiency increases and the pumping power requirement is also increase, but for the Comparison of nano-fluid with base fluid it has been found that requirement of pumping power is reduced for the use of nano-fluid. And also on comparing requirement of pumping power for different nano-fluids Cu based Nano-fluid requires less pumping power as compared with SiC, Al2O3,TiO2 based Nano-fluids.

c) Influence of varying inlet air temperature:

The variation of cooling capacity and effectiveness with air inlet temperature is shown in Fig. 5 for ($m_c=5\text{kg/s}$, $m_a=15\text{kg/s}$, $T_{in}=900\text{C}$). As expected the heat transfer rate clearly decreases with air inlet temperature rise, as the cooling temperature difference is being reduced. It is interesting to point out that Nano-fluid using 80%water-20%EG has higher cooling capacity than that of 80%water-20%EG as coolant.

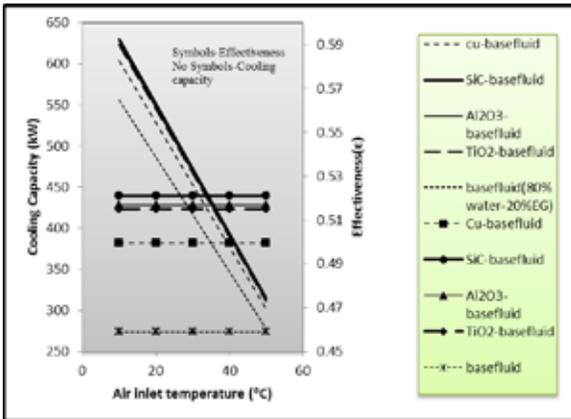


Figure 5 Effect of air inlet temperature on cooling capacity and effectiveness

d) Influence of varying coolant inlet temperature:

The variation of cooling capacity and effectiveness with coolant inlet temperature is shown in Fig.6 for ($m_c=5\text{kg/s}$, $m_a=15\text{kg/s}$, $T_{ai}=300\text{C}$). As expected the heat transfer rate increases with coolant inlet temperature rise due to increment in the cooling temperature difference. Also, for this variation study shows there is very little increment in the effectiveness.

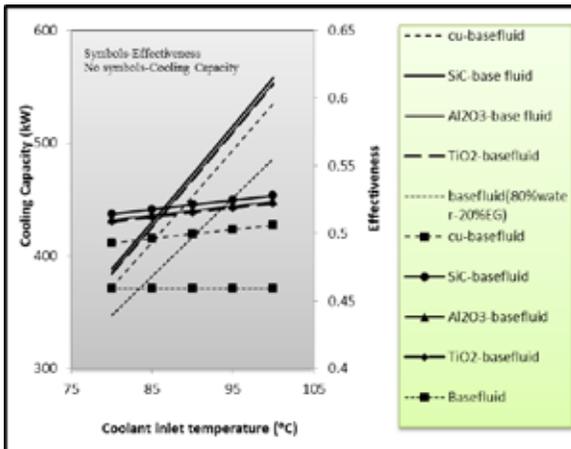


Figure 6 Effect of coolant inlet temperature on cooling capacity and effectiveness

e) Influence of varying volume fraction of Nano-particle:

This study observed that (Fig. 7) for the variation of volume

fraction of nano-particle cooling capacity and effectiveness is increases up to 1% volume fraction and beyond that decrement in the effectiveness and cooling capacity was observed. Also with the increase in volume fraction second law efficiency increases, where pumping power is decreased up to 2% volume fraction and beyond that increment in pumping power was observed(Fig.8). On comparing Cu based nanofluid requires less amount of pumping power.

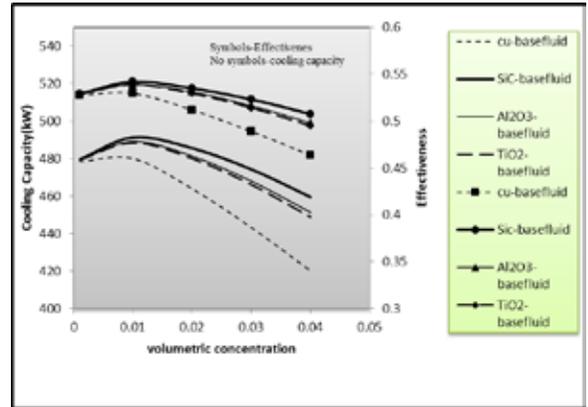


Figure 7 Effect of volume fraction of nanoparticle on cooling capacity and effectiveness

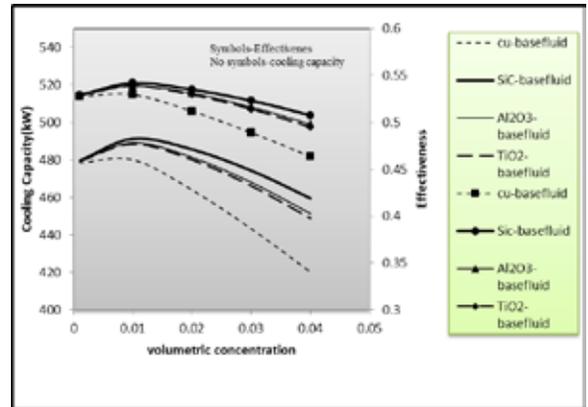


Figure 8 Effect of volume fraction of nanoparticle on pressure drop and pumping power.

V CONCLUSIONS:

A detailed parametric studies on automotive radiator has been done by using ϵ -NTU method using four nanofluids (Cu, SiC, Al2O3, TiO2) in a base fluid 80%water-20%EG as a coolant for louvered fin geometry. Based on results and discussion, following conclusion can be made:

- Cooling capacity and effectiveness increase with increase in mass flow rate of air and coolant.
- Cooling capacity of radiator using nanofluid is greater than radiator using base fluid alone.
- Requirement of pumping power reduce with the use of nanofluid in radiator.
- Pumping power requirement decrease with the increase in volume fraction of nanoparticle up to 2%.
- Second law efficiency increases with the increase in coolant mass flow rate.

REFERENCE

- [1] S. M. Peyghambarzadeh, S. H. Hashemabadi, S. M. Hoseini, M. Seifi Jamnani, Experimental study of heat transfer enhancement using water/ethylene glycol based nanofluids as a new coolant for car radiators, International communication of Heat and Mass transfer, Article in press (2011). | [2] K. Y. Leong, R. Saidur, S. N. Kazi, A. H. Mamun, Performance investigation of an automotive car radiator operated with nanofluid-based coolants (nanofluid as a coolant in a radiator), Applied Thermal Engineering 30 (2010) 2685–2692. | [3] YU-JUEI CHANG, CHI-CHUAN WANG, A generalized heat transfer | correlation for louver fin geometry, Int. J. Heat Mass Transfer. Vol. 40, | No. 3, pp. 53–544, 1997. | [4] R. Strandberg, D.K. Das, Finned tube performance evaluation with nanofluids and conventional heat transfer fluids, International Journal of Thermal Sciences 49 (2010) 580–588. | [5] R. S. Vajjha, D.K. Das, P.K. Namburu, Numerical study of fluid dynamic and heat transfer performance of Al₂O₃ and CuO nanofluids in the flat tubes of a radiator, International Journal of Heat and Fluid Flow 31 (2010) 613–621. | [6] W. G. Peng, Y. C. Liu, Y. W. Hu, Y. R. He, Simulation of heat exchange enhancement using in engine cooling system, Harbin institute of technology 43(1) (2011) 109–113. | [7] J. Koo, C. Kleinstreuer, A new thermal conductivity model for nanofluids. Journal of Nanoparticle Research 6 (2004) 577–588. | [8] B.C. Pak, Y.I. Cho, Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles, Experimental Heat Transfer 11(2) (1998) 151–170. | [9] S.E.B. Maiga, C. T. Nguyen, N. Galanis, G. Roy, Heat transfer behaviours of nanofluids in a uniformly heated tube. Superlattices and Microstructures 35 (2004) 543–557. | [10] V. Vasu, K. Rama Krishna, A.C.S. Kumar, Application of nanofluids in thermal design of compact heat exchanger, International Journal of Nanotechnology and Applications, 2(1) (2008) 75–87. | [11] S. K. Saripella, W. Yu, J. L. Routbort, D. M. France, Rizwan-uddin, Effects of Nanofluid Coolant in a Class 8 Truck Engine, SAE technical paper 2007-01-2141. | [12] E. Ollivier, J. Bellettre, M. Tazerout, G. C. Roy, Detection of knock occurrence in a gas SI engine from a heat transfer analysis. Energy Convers Manage 47(7–8) (2006) 879–93. | [13] W. M. Kays, A.L. London, Compact Heat Exchanger, third ed. McGraw-Hill, Inc., United States, 1984. | [14] W. Yu, D. M. France, S. U. S. Choi, J. L. Routbort, Review and Assessment of Nanofluid Technology for Transportation and Other Applications (No. ANL/ ESD/07-9). Energy System Division, Argonne National Laboratory, Argonne, 2007. | [15] K. V. Wong and O. D. Leon, A Review on Applications of Nanofluids: Current and Future, Advances in mechanical engineering 10.1155/2010/519659 | [16] A. Gupta and S. K. Das, Second law analysis of crossflow heat exchanger in the presence of axial dispersion in one fluid, Energy 32 (2007) 664–672. | [17] Y.J. Chang, C.C. Wang, Air side performance of brazed aluminium heat exchangers, Enhanced Heat Transfer 3, 15–28 (1996). | [18] J. Dong, J. Chen, Z. Chen, W. Zhang, Y. Zhou, Heat transfer and pressure drop correlations for the multi-louvered fin compact heat exchangers, Energy Conversion and Management 48 (2007) 1506–1515. |