PERFORMANCE COMPARISON OF VARIOUS COOLANTS FOR LOUVERED FIN TUBE AUTOMOTIVE RADIATOR

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Abstract

In the present study, screening of various coolants (water, ethylene glycol, propylene glycol, brines, nanofluid and sugarcane juice) for louvered fin automotive radiator has been done based on different energetic and exergetic performance parameters. Effects on radiator size, weight and cost as well as engine efficiency and fuel consumption are discussed as well. Results show that the sugarcane juice seems to be slightly better in terms of both heat transfer and pumping power than water and nanofluid, whereas significantly better than EG and PG. For same heat transfer capacity, the pumping power requirement is minimum and vice-versa with sugarcane juice, followed by nanofluid, water, EG and PG. Study on brines shows an opportunity to use water+25%PG based nanofluids for improvement of performance as well as operating range. Replacement of water or brines by using sugarcane juice and water or water+25%PG based nanofluids will reduce the radiator size, weight and pumping power, which may lead to increase in compactness and overall engine efficiency or reduction in radiator cost and engine fuel consumption. In overall, both sugarcane juice and nanofluid seem to be potential substitutes of water. However, both have some challenges such as long term stabiliy for practical use.

Keywords: louvered fin tube radiator, sugarcane juice, nanofluid, performance, size, cost.

1. Introduction

Due to the increasing power requirement and the limited available space in the vehicles, it is extremely difficult to increase the size of the heat exchangers (HEXs) placed in the front of the vehicles. The overall aim of this study is to increase the performance of the automotive radiator. There are few methods which can applicable for that- (i) by using various plate fin heat transfer surfaces such as wavy fins, louvered fins, etc., (ii) by using high thermal conductive materials in the formation core of automotive radiator such as aluminum, copper, graphite etc., (iii) by using coolant having low freezing point, high boiling point and high heat transfer coefficient, (iv) by changing the position of the heat exchanger. With respect to fin, the multi-louvered fin has the highest heat transfer enhancement relative to pressure drop in comparison with most other fin types [1]. An important aspect of louvered fin performance is the degree to which the flow follows the louver. Flat tubes are more popular for

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automotive applications due to their lower profile drag compared with round tubes. Louvered aluminum fins and flat tubes are widely used in automotive radiator [2-3].

This study is focused on the searching of alternative heat transfer fluids for overall performance improvement. To enhance the cooling rate, increasing the surface area by addition of fins is the earliest approach but this approach of increasing heat transfer already reached to their limit. Water and water mixed with anti-freezing agents such as ethylene glycol (EG) and propylene glycol (PG) are the traditional coolants for automotive radiator. Recently nanofluid have been proposed as coolant for automotive radiator [4-5]. However, operation and long term stability are major challenges for nanofluid [6]. Hence, the searching of alternative fluid is not ending. In this respect, sugar cane juice, which has very similar freezing and boiling points with water (Table 1), may be an alternative.

In the present study, the energetic and exergetic performance analyses of louvered fin and flat tube automotive radiator using various coolants (water, ethylene glycol, propylene glycol, sugar-cane juice and alumina-water nanofluid) has been done. Effect of temperature on various coolant properties is also discussed. Effects of various operating parameters on the heat transfer rate, effectiveness, pumping power, performance index and second law efficiency are discussed. Radiator performance by using water based EG and PG brines are studied as well.

Table 1: Freezing and boiling temperatures of various fluids

Fluids	**Freezing point**	**Boiling point**
Water	0 ^{o}C	100 ^{o}C
Ethylene glycol	−59 ^{o}C	187.4 ^{o}C
Propylene glycol	−12.9 ^{o}C	197.3 ^{o}C
Sugarcane Juice	−12 ^{o}C	107^{o}C

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

Description	Air side	Coolant side
Core Width, W_c	382mm	
Core height, H_c	491mm	
Core depth, F_d	44mm	
Fin metal thickness	0.8mm	
Hydraulic diameter	1.008mm	3.378mm
Tube thickness		0.32mm
Total heat transfer area/total volume	926 m^2/m^3	175 m^2/m^3
Louvered fin parameters	s1 = s2 = 4.1, L_a = 25°, L_p = 0.9, L_h = 1, F_p=2.6, T_p=10, T_w=2.5, L_l=6.8 (all in mm)	

2. Theoretical modeling and Simulation

Louvered fin-tube radiator considered here is cross flow type and the core portion consists of vertical flat coolant tubes and multi-louvered fins, and its dimension as shown in Table 2 is taken from [7]. The formulation with various coolants is based on energy and exergy balance including heat transfer and fluid flow effects. The following assumptions have been made for the modeling:

1) Steady flow for both air and coolant.
2) All the heat rejected from coolant absorbed by air flow through radiator.

3) Properties have been taken based on mean fluid temperature.

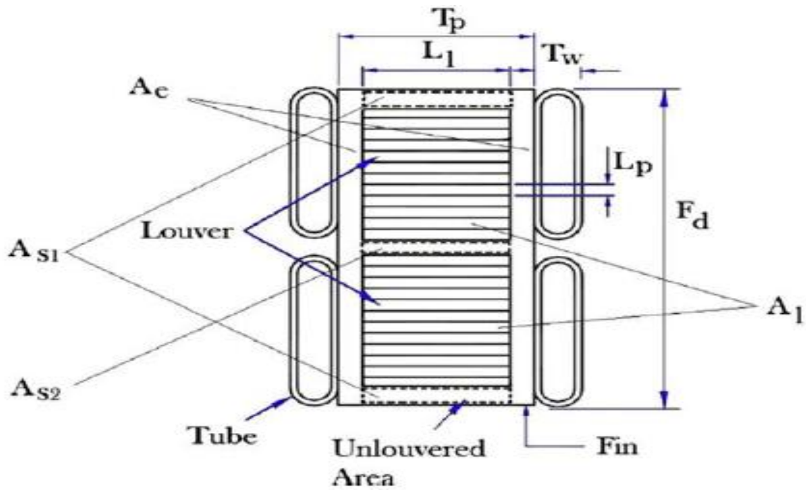

Figure1: Geometric construction details of louvered fin

For air-side heat transfer coefficient calculation, different zones have been considered as shown in Figure 1 and individual heat transfer coefficient of each zone has been calculated and combined them. Hence, Air side heat conductance is given by,

$$\eta_o h_a A_a = \eta_{f,l} h_l A_l + \eta_{f,s1} h_{s1} A_{s1} + \eta_{f,s2} h_{s2} A_{s2} + h_e A_e \quad (1)$$

Where, zonal heat transfer coefficients are given by,

$$h_l = 0.664 k_a \rho_a u_l \mathrm{Re}_l^{-0.5} \mathrm{Pr}_a^{0.33} / \mu_a \quad (2)$$

$$h_{s1} = 0.664 k_a \rho_a u_c \mathrm{Re}_{s1}^{-0.5} \mathrm{Pr}_a^{0.33} / \mu_a \quad (3)$$

$$h_{s2} = 0.664 k_a \rho_a u_c \mathrm{Re}_{s2}^{-0.5} \mathrm{Pr}_a^{0.33} / \mu_a \quad (4)$$

$$\frac{h_e D_{h_e}}{k_a} = 7.541(1 - 2.61A_r + 4.97A_r^2 - 5.119A_r^3 + 2.702A_r^4 - 0.548A_r^5) \quad (5)$$

and fin efficiencies and other details are given in [7]. Now, air-side heat capacity rate is given by:

$$C_a = \rho_a u_a H_c W_c c_{p,a} \quad (6)$$

Coolant-side heat transfer coefficient can be expressed as:

$$h_f = \frac{Nu_f k_f}{D_{h,f}} \quad (7)$$

Where, Nu for water, EG, PG and sugarcane juice are given by,

$$Nu_f = \frac{(f_f/2) Re_f Pr_f}{1.07 + 12.7\sqrt{f_f/2}(Pr_f^{(2/3)} - 1)} \quad (8)$$

Whereas, Nusselt number for nanofluid is expressed as [8]

$$Nu_{nf} = 0.0222\left(\mathrm{Re}_{nf}^{0.8} - 60\right)\mathrm{Pr}_{nf}^{0.4}\left(1 + 0.32178\phi^{0:64788}\right) \quad (9)$$

Where, Reynolds number has been calculated using hydraulic diameter.

The effective density and the effective specific heat of the nanofluid have been evaluated by,

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p \quad (10)$$

$$\left(\rho c_p\right)_{nf} = (1-\phi)\left(\rho c_p\right)_{bf} + \phi\left(\rho c_p\right)_p \tag{11}$$

Viscosity of nanofluid is given by [9],

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

The effective thermal conductivity of the nanofluid has been evaluated using following equation [10],

$$k_{nf} = \frac{k_p + 2k_{bf} + 2(k_p - k_{bf})(1+\beta)^3\phi}{k_p + 2k_{bf} - (k_p - k_{bf})(1+\beta)^3\phi} k_{bf} \tag{13}$$

Now, the coolant-side heat capacity rate is given by,

$$C_f = \rho_f V_f c_{p,f} \tag{14}$$

Coolant pressure drop is given by:

$$\Delta p_f = \left[G^2{}_f \times f_f \times H_c\right] \Big/ \left[2 \times \rho_f \times \left(D_{h,f} / 4\right)\right] \tag{15}$$

Where, the friction factor has been calculated from suitable correlations [8,11].
Now, overall heat transfer co-efficient is given by

$$\frac{1}{UA} = \frac{1}{\eta_o h_a A_a} + \frac{1}{h_f A_f} \tag{16}$$

Effectiveness for cross-flow unmixed fluid is given by [11],

$$\varepsilon = 1 - \exp\left[\frac{NTU^{0.22}}{C^*} \exp\left(-C^* NTU^{0.78} - 1\right)\right] \tag{17}$$

Total heat transfer rate is given by,

$$Q = \varepsilon C_{\min}\left(T_{f,in} - T_{a,in}\right) \tag{18}$$

Coolant pumping power is given by,

$$P = V_f \Delta p_f \tag{19}$$

Now, the performance index can be defined by,

$$PI = Q/P \tag{20}$$

The exergy loss by the coolant is given by [4],

$$\Delta Ex_f = Q - T_0\left[\dot{m} c_p \ln\left(T_{in}/T_{out}\right) - \dot{m}\,\Delta p/(\rho T)\right]_f \tag{21}$$

Whereas, the exergy gain rate by air is calculated by [4],

$$\Delta Ex_a = Q - T_0\left[\dot{m} c_p \ln\left(T_{out}/T_{in}\right) + \dot{m} R \ln\left(p_{in}/p_{out}\right)\right]_a \tag{22}$$

Now, irreversibility and second law efficiency are given by, respectively,

$$I = \Delta Ex_f - \Delta Ex_a \tag{23}$$

$$\eta_{II} = \Delta Ex_a \big/ \Delta Ex_f \tag{24}$$

For implementing the analysis, an EES code was written for the studied louvered fin radiator. Thermo-physical and transport properties of both air and coolants have been calculated based on mean temperature. As the exit temperatures are output parameters of simulation, iteration has been done to use of mean temperature based properties. In-build subroutines have been used for the temperature dependent properties of water (for nanofluid also) and air. Web site based data set has been used for the temperature dependent properties of EG and PG. Temperature dependent properties of sugarcane juice have been taken from research work by [12-13]. Properties of alumina nanoparticles have been taken from [4]. Particle volume fraction in alumina-water nanofluid has been taken as 1.5%.

The numerical code has been verified with experimental data [7]. A comparison is shown in Figure 2 for variations of heat transfer rate and water inlet temperature with inlet air velocity for same geometry and operating conditions ($T_{in,a} = 20.3^oC$, $T_{in,f} = 80^oC$ and $V_f = 120\,Lit/\min$). Similar trend has been observed and showed maximum 3% and 2% deviations between the predicted and experimental data for heat transfer rate and water outlet temperature, respectively.

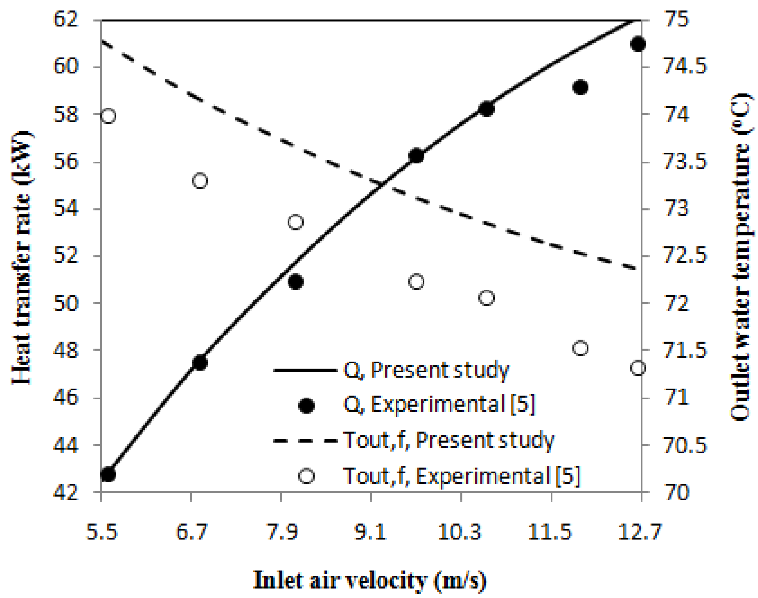

Figure 2: Validation of simulation code with experimental data [7] for water

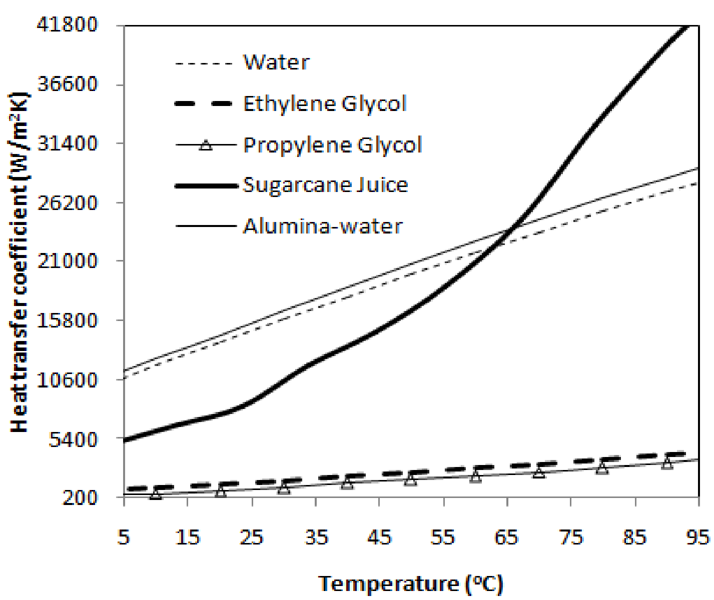

Figure 3: Variation of heat transfer coefficient with temperature

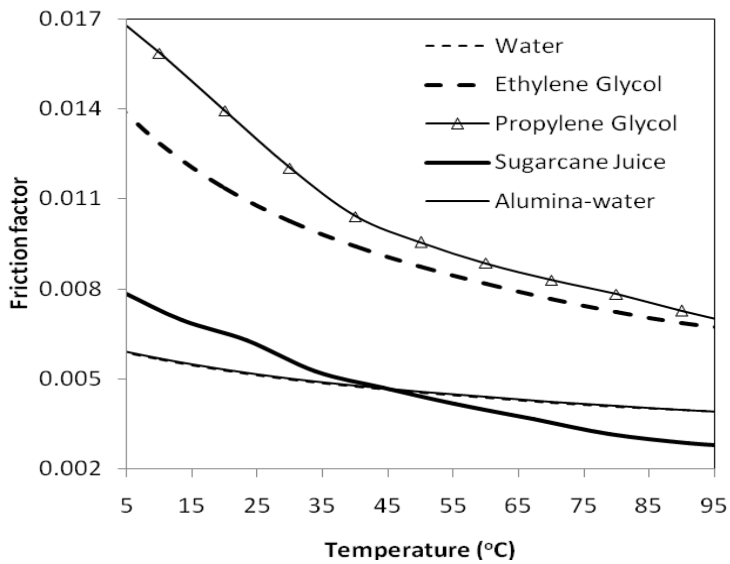

Figure 4: Variation friction factor with temperature

3. Results and Discussions

Variations of heat transfer coefficient and friction factor with temperature are shown in Figures 3 and 4, respectively, for mass velocity of 5000 kg/m^2s and hydraulic diameter of 0.01m. As shown, the heat transfer coefficient of sugarcane juice highly increases as the temperature increases compared to others and this is only due to fast decrease of viscosity. Hence, alumina-water and water are having higher heat transfer coefficient at lower temperature, whereas sugarcane is having higher heat transfer coefficient at higher temperature (approximately above 60^{o}C). On the other hand, friction factor of Propylene Glycol is higher as compared to other coolants. Although, sugarcane juice is having higher friction factor at lower temperature but at higher temperature it decreases highly as compared to water. Hence, the sugarcane juice is better than other fluids in terms of both heat transfer and pressure drop at higher temperature. In general, the automotive radiator is operated at coolant mean temperature of above 60^{o}C and hence it is expected to get better performance with sugarcane juice. This interesting fact has motivated the present simulation study using sugarcane juice as automotive engine coolant. However, boiling of water in sugarcane juice may start at 91-95^{o}C leading to change of characteristics [14] and hence it is safe to use sugarcane juice as coolant for sensible heating upto about 91^{o}C. For the simulation, coolant inlet temperature, air inlet temperature and air frontal velocity have been taken as 90^{o}C, 30^{o}C and 10m/s, respectively. Aluminum alloy of thermal conductivity 177W/mK has been taken as fin materials.

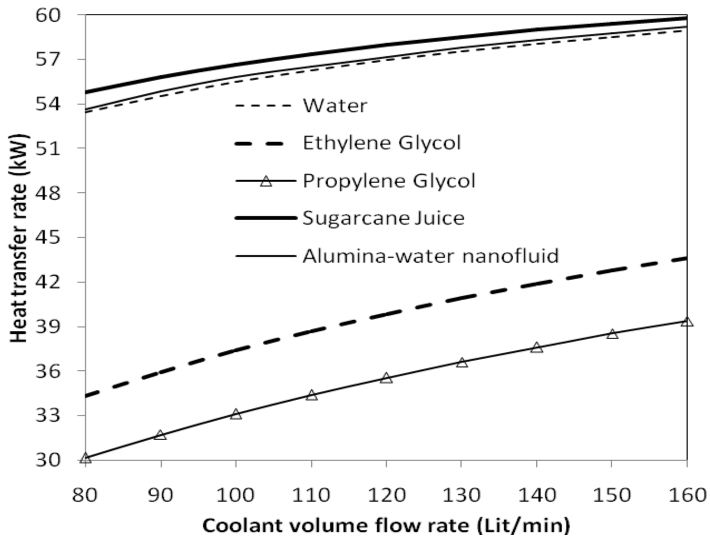

Figure 5: Variation of heat transfer rate with coolant volume flow rate

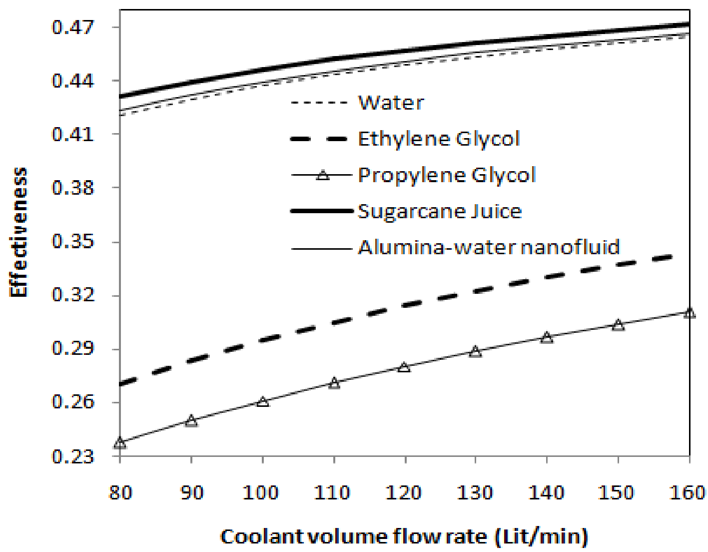

Figure 6: Variation of heat exchanger effectiveness with coolant flow rate

Variations of the heat transfer rate, effectiveness, pumping power, performance index and second law efficiency with various coolant volume flow rate are shown in Figures 5-9. It has been observed that heat transfer rate, effectiveness and pumping power goes on increasing with coolant flow rate due to dual effects of heat transfer coefficient and heat capacity increments and sugarcane juice yields slightly better heat transfer rate and effectiveness than water and nanofluid, whereas significantly better than EG and PG mainly due to lower dynamic viscosity. On the other hand, due to same reason, pumping power of sugarcane juice is slightly lower than water and nanofluid, whereas significantly lower than EG and PG. As a result, sugarcane juice yields slightly better performance index and second law efficiency than water and nanofluid, whereas significantly better than EG and PG. However, performance index highly decreases (as the effect of flow rate on pumping power is more predominant than that on heat transfer rate), whereas second law efficiency increases with increase in coolant volume flow rate for all studied coolants. Performance characteristic of various coolant is illustrated in Figure 10. For same heat transfer capacity, the pumping power requirement is minimum with sugarcane juice, followed by nanofluid, water, EG and PG. Similarly, for same pump power supply, heat transfer rate is maximum with sugar cane juice, followed by nanofluid, water, EG and PG.

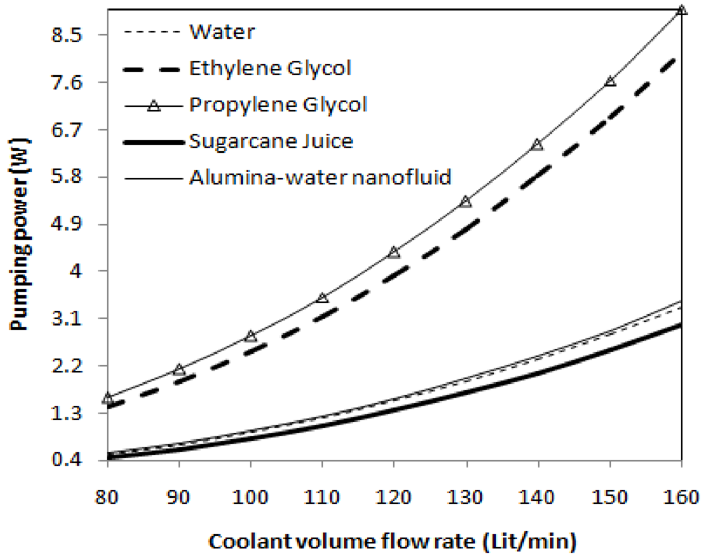

Figure 7: Variation of pumping power with coolant flow rate

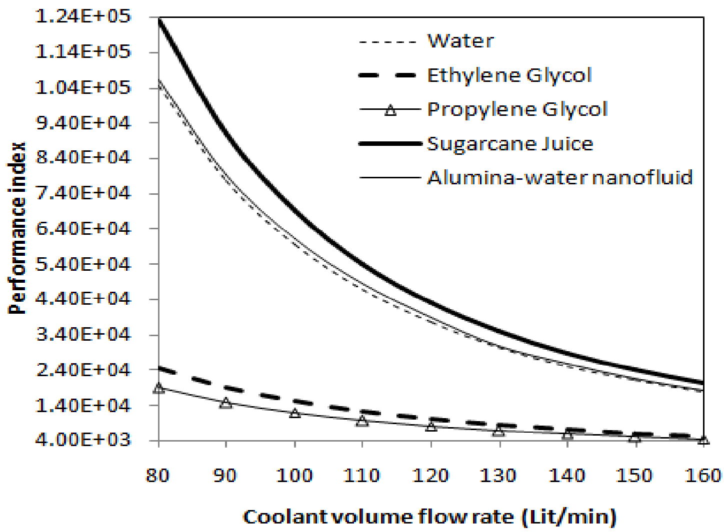

Figure 8: Variation of performance index with coolant flow rate

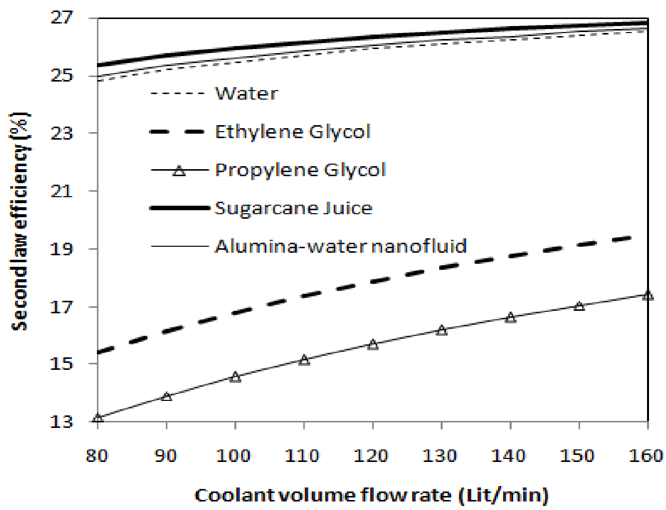

Figure 9: Variation of second law efficiency with coolant flow rate

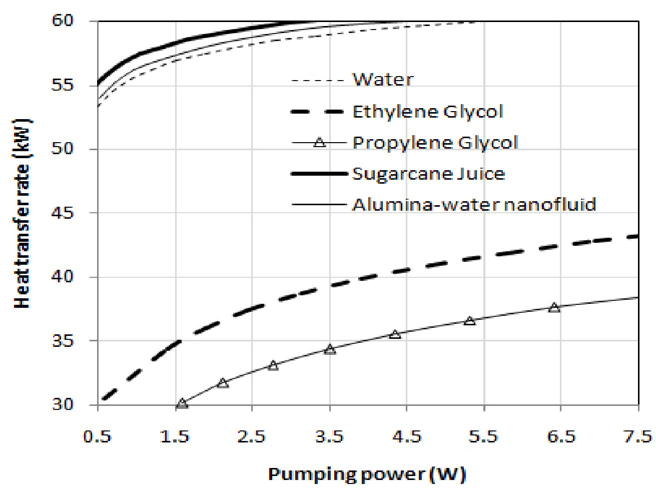

Figure10: Performance graph (heat transfer rate with pumping power)

Variations of heat transfer rate, effectiveness, pumping power and performance index with EG or PG mass faction in water based brines are shown in Figures 11 and 12. With increase in mass fraction, the heat transfer rate and effectiveness gradually decrease for EG, whereas, for PG, it seems to be decreases initially, then increases and again decreases by yielding some maximum values corresponding to optimum mass fraction of about 25%. This abnormal behavior of perfomance with PG mass fraction may be due to the typical change of dynamic viscosity. It may be noted that the performance values are similar to the that with pure water. Due to same reason (viscosity change behavior), pumping power yields minimum value at about 25% whereas as increases continuously for EG. As a results, performance index decreases monotonically for EG with mass fraction whereas decreases, increases and again decreases with PG mass fraction yielding maximum value at about 25% mass fraction.

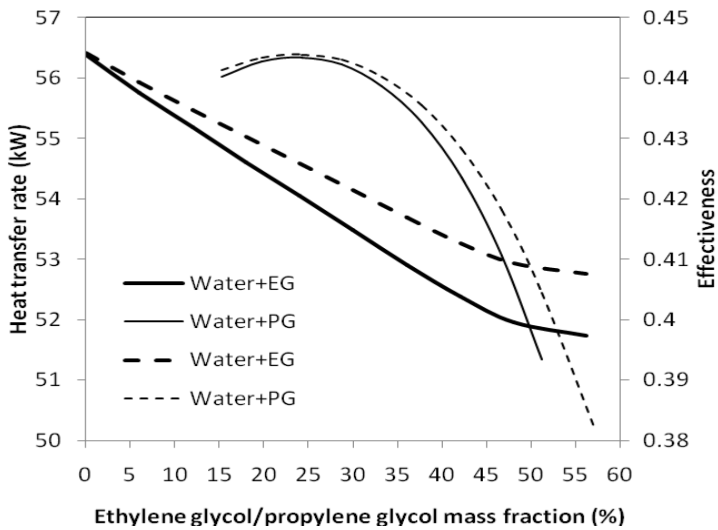

Figure 11: Variation of heat transfer rate and effectiveness for brines

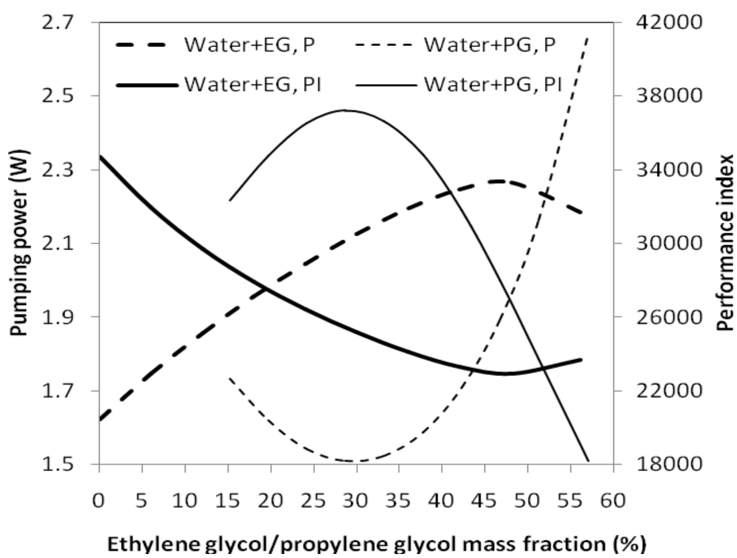

Figure 12: Variation of pumping power and performance index for brines

Table 3: Performance comparison of various heat transfer fluids (coolants)

Parameters	Water	EG	PG	Nanofluid (ϕ=1.5%)	Water + 25% EG	Water + 25% PG	Sugarcane Juice
Heat transfer rate (kW)	56.96	39.85	35.56	57.58	54.54	56.92	58.01
Effectiveness (%)	44.88	31.40	28.02	45.26	42.97	44.85	45.70
Pumping power (W)	1.526	3.921	4.361	1.572	1.961	1.449	1.350
Performance index	37334	10164	8155	39573	27812	39276	42963
Second law efficiency (%)	25.93	17.88	15.71	26.07	24.77	25.89	26.33

Comparision of various fluids are summerized in Table 3 coolant volume flow rate of 120lpm and air frontal velocity of 10m/s. As shown, sugarcane juice yields maximum performance followed by alumina-water nanofluid, water+25%PG and water. Recent many studies showed 5-10% radiator performaance improvement using nanofluid. Hence, it seems to be similar radiator performance by using nanofluid and sugarcane juice. Furthermore, there is an another opportunity to use water+25%PG based nanofluids for performance improvement of radiator. However, both sugarcane juice and nanofluids have some challenges such as long term stabiliy to use in radiator.

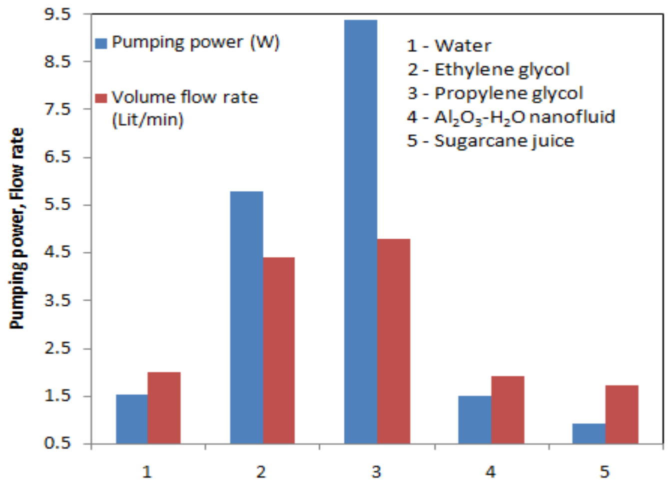

Figure 13: Comparison for same heat transfer rate and radiator size

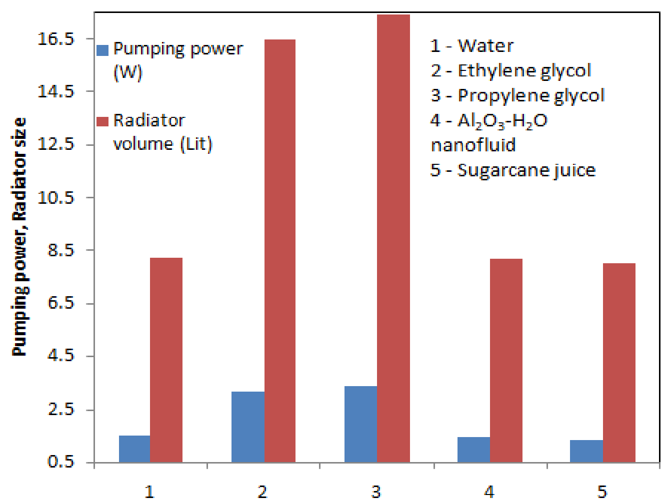

Figure 14: Comparison for same heat transfer rate and mass flow rate

Reduction in size and weight of the radiators are among the achievements of this type of researches. In addition to reducing the production cost, better designation of cars are possible when the radiator becomes smaller in size. On the other hand, better cooling has positive effects on fuel consumption and the amount of fuel consumption decreases. Compared to water, the coolant flow rate and pumping power reduce by 13% and 41% respectively, by using sugar cane juice, whereas, only 5% both by using alumina nanofluid for same cooling capacity and radiator size (Figure 13). Reductions of coolant flow rate and pumping power lead to decrease of coolant cost and increase of overall engine efficiency or decrease of fuel consumption, respectively. On the other hand, for same cooling capacity and mass flow rate, the radiator size and pumping power reduce by 2.5% and 13.5%, respectively, by using sugar cane juice, whereas, about 2% both by using alumina nanofluid compared to water (Figure 14). Reduction of radiator size may lead to compactness as well as decrease of radiator weight and cost. As discussed earlier, values of above effects may change by using other nanofluids. However, in overall, both sugarcane juice and nanofluids are potential substitutes of water.

4. Conclusions

The energetic as exergetic performance analyses of louvered fin and flat tube automotive radiator have been done using various coolants (water, EG, PG, water-EG brine, water-PG brine, sugar-cane juice and alumina-water nanofluid). Based on the results and discussion, the following conclusions can be made:

- Sugarcane juice yields better heat transfer and pressure drop characteristics at higher temperature (approximately above 60^{o}C).
- Heat transfer rate, effectiveness, pumping power and exergetic efficiency go on increasing whereas performance index goes on decreasing with coolant flow rate.
- Sugarcane juice is slightly better in terms of both heat transfer pumping power than water and nanofluid, whereas significantly better than EG and PG.
- For same heat transfer capacity, the pumping power requirement is minimum and vice-versa with sugarcane juice, followed by nanofluid, water, EG and PG.
- For brines, performance index decreases monotonically with EG mass fraction whereas decreases, increases and again decreases with PG mass fraction yielding maximum value at about 25% mass fraction.
- There is an opportunity to use water+25%PG based nanofluids for improvement of performance as well as operating temperature range of radiator.
- Compared to water, the coolant flow rate and pumping power reduce by 13% and 41% respectively, by using sugar cane juice, whereas, only 5% both by using alumina nanofluid for same cooling capacity and radiator size.
- For same cooling capacity and mass flow rate, the radiator size and pumping power reduce by 2.5% and 13.5%, respectively, by using sugar cane juice, whereas, about 2% both by using alumina nanofluid compared to water.
- Use of sugarcane juice, water or water+25%PG based nanofluids may lead to reduction in radiator size, weight and cost, and engine fuel consumption.

Nomenclature

A	heat transfer area [m^2]
C	heat capacity rate, [WK^{-1}]
c_p	specific heat, [Jkg^{-1}K^{-1}]
C*	heat capacity ratio
D_h	hydraulic diameter [m]
f	friction factor
F_l	fin length [m]
F_{th}	fin thickness [m]
F_h	fin height [mm]
F_p	fin pitch [mm]
G	mass velocity [kgm^{-2}s^{-1}]
h	heat transfer coefficient [Wm^{-2}K^{-1}]
I	irreversibility [W]
k	thermal conductivity [Wm^{-1}K^{-1}]
L_a	louver angle [degree]
L_d	fin length [mm]
L_h	louver height [mm]
L_p	louver pitch [mm]
$\dot{m}$	mass flow rate [kgs^{-1}]
NTU	number of heat transfer units
Nu	Nusselt number
P	pumping power
Pr	Prandtl number
Q	heat transfer rate [W]
Re	Reynolds number
T	temperature [K]
T_0	dead state temperature [K]
u	fluid velocity [ms^{-1}]
U	overall heat transfer coefficient [Wm^{-2}K^{-1}]
V	volume flow rate [lpm]

Δp	pressure drop [Pa]	***Subscripts***	
ΔEx	exergy gain or loss rate [W]	a	air
η_f	fin efficiency	bf	base fluid
η_o	total heat transfer surface effectiveness	c	core
η_{II}	second law efficiency	f	fin, fluid (coolant)
μ	fluid viscosity [Nsm^{-2}]	in	inlet
φ	nanoparticle volume fraction	nf	nanofluid
ρ	fluid density [kgm^{-3}]	ext	exit
ε	heat exchange effectiveness	np	nanoparticle

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