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Coolants Performance in Louvered Fin Tube Automotive Radiator

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Abstract—*In the present study, screening of various coolants (water, ethylene glycol, propylene glycol, brines, nanofluids and sugarcane juice) for louvered fin automotive radiator have been done based on different energetic and exergetic performance parameters. 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.. Replacement of water or brines by using sugarcane juice, water 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 nanofluids 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, nanofluids, performance.*

Nomenclature

A	heat transfer area, m^2
C	heat capacity rate, W/K
c_p	specific heat, J/kg K
C*	heat capacity ratio
D_h	hydraulic diameter, m
EG	ethylene glycol
f	fanning 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, kg/m^2s
h	heat transfer coefficient, W/m^2 K
I	irreversibility, W
k	thermal conductivity, W/m K
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, kg/s
NTU	number of heat transfer units
Nu	Nusselt number
P	pumping power
PG	propylene glycol
Pr	Prandtl number
Q	heat transfer rate, W
Re	Reynolds number
T	temperature, K
T_0	dead state temperature, K
u	fluid velocity, m/s
U	overall heat transfer coefficient, W/m^2 K
V	volume flow rate, lpm
Δp	pressure drop, Pa
ΔEx	exergy gain or loss, W
η_f	fin efficiency
η_o	total surface temperature effectiveness
η_{II}	second law efficiency
μ	dynamic viscosity, Ns/m^2
φ	volume fraction of particles
ρ	density, kg/m^3
ε	heat exchanger effectiveness
Subscripts	
a	air
bf	base fluid
c	core
f	fin, fluid (coolant)
in	inlet
l,s1,s2,e	zones (Fig.1)
nf	nanofluids
out	outlet
p	nanoparticles

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. By using coolant having low freezing point, high boiling point and high heat transfer coefficient with louvered fin the highest heat transfer enhancement relative to pressure drop occur 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 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].

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. The conventional fluids, such as water, ethylene glycol (EG) and propylene glycol (PG) have been proven to have poor convective heat transfer performance due to comparatively lower thermal conductivity, In order to achieve higher rate of cooling capacity with the use of these conventional fluid, larger size heat exchange unit has been required which enhances pumping power requirement.

In order to reduce pumping power requirement and to achieve higher cooling capacity, higher compactness and effectiveness of heat transfer systems are necessary. Within last two decades, extensive researches have proven that nanofluids (a suspension of nanometre-sized metallic particles in a base fluid) are superior as a heat transfer agent over conventional fluids [4]. However, operation and long term stability are major challenges for nanofluids. 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.

Table 1: Freezing and boiling temperatures of various fluids

Fluids	Freezing point	Boiling point
Water	0 oC	100 oC
Ethylene glycol	−59 oC	187.4 oC
Propylene glycol	−12.9 oC	197.3 oC
Sugarcane Juice	−12 oC	107oC

In the present study, the energetic as well as 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) have 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.

2. THEORETICAL MODELING AND SIMULATION

The automotive radiator consists of coolant inlet tank, outlet tank, pressure cap and core. The major components of the core are coolant tubes and fins. Louvered fin radiator consider in this study is cross flow type and consists of vertical flat coolant tubes and multi-louvered fins, and its dimension as shown in Table 2 is taken from[5]. The mathematical model has been developed based on first law and second law of thermodynamics including heat transfer and fluid flow effects. The following assumptions have been made for analysis:

1) Steady state process.
2) All the heat rejected from nanofluid absorbed by air flow through radiator.
3) Properties have been taken based on mean fluid temperature.

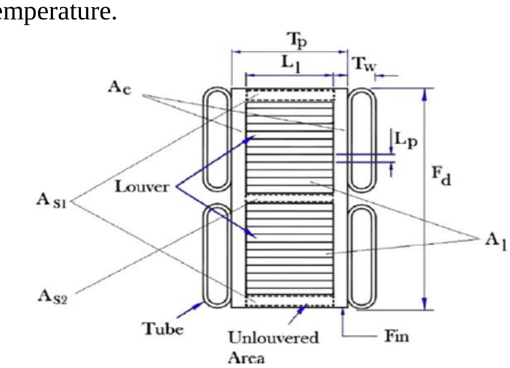

Fig. 1: Geometric construction details of louvered fin

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

Description	Air side	Coolant side
Core Width, Wc	382mm	
Core height, Hc	491mm	
Core depth, Fd	44mm	
Fin metal thickness	0.8mm	
Hydraulic diameter	1.008mm	3.378mm
Tube thickness		0.32mm
Total heat transfer area/total volume	926 m2/m3	175 m2/m3
Louvered fin parameters	s1 = s2 = 4.1, La = 25°, Lp = 0.9, Lh = 1, Fp=2.6, Tp=10, Tw=2.5, Ll=6.8 (all in mm)	

2.1 Mathematical modeling

For air-side heat transfer coefficient calculation, different zones have been considered as shown in Fig. 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 are given as follows:

$$\eta_{f,l} = k_f F_{th} \frac{\left(\frac{1}{\frac{\tanh(m_e a)}{m_e}} - \frac{1}{\frac{m_l \sinh^2(m_e a)\tanh\left(m_l\left(\frac{F_l}{2}-a\right)\right)}{m_e^2} + \frac{\cosh(m_e a)\sinh(m_e a)}{m_e}}\right)}{2(a h_e + (\frac{F_l}{2} - a) h_l)} \quad (6)$$

$$\eta_{f,s1} = k_f F_{th} \frac{\left(\frac{1}{\frac{\tanh(m_e a)}{m_e}} - \frac{1}{\frac{m_{s1} \sinh^2(m_e a)\tanh\left(m_{s1}\left(\frac{F_l}{2}-a\right)\right)}{m_e^2} + \frac{\cosh(m_e a)\sinh(m_e a)}{m_e}}\right)}{2(a h_e + (\frac{F_l}{2} - a) h_{s1})} \quad (7)$$

$$\eta_{f,s2} = k_f F_{th} \frac{\left(\frac{1}{\frac{\tanh(m_e a)}{m_e}} - \frac{1}{\frac{m_{s2} \sinh^2(m_e a)\tanh\left(m_l\left(\frac{F_l}{2}-a\right)\right)}{m_e^2} + \frac{\cosh(m_e a)\sinh(m_e a)}{m_e}}\right)}{2(a h_e + (\frac{F_l}{2} - a) h_{s2})} \quad (8)$$

where, other details are given in [5].

Now, air-side heat capacity rate can be expressed as:

$$C_a = \rho_a u_a H_c W_c c_{p,a} \quad (9)$$

Coolant-side heat transfer coefficient can be expressed as:

$$\boldsymbol{h_f = \frac{Nu_f k_f}{D_{h,f}}} \quad (10)$$

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

$$Nu_f = \frac{(f_f/2) Re_f Pr_f}{1.07 + 12.7\sqrt{f_f/2}(Pr_f^{(2/3)} - 1)} \quad (11)$$

Nusselt number for nanofluid is expressed as [6]

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

Where, Reynolds number has been calculated using hydraulic diameter.

The effective density and the effective specific heat of the nanofluid have been evaluated using the following relations:

$$\left(\rho c_p\right)_{nf} = (1-\phi)\left(\rho c_p\right)_{bf} + \phi\left(\rho c_p\right)_p \quad (12)$$

Viscosity of nanofluid is calculated based on correlation from[7]

$$\mu_{nf} = \mu_{bf}\left(1 - 0.19\phi + 306\phi^2\right) \quad (13)$$

The effective thermal conductivity of the nanofluid has been evaluated using following equation [8] given by ($\phi \le 5\%$),

$$k_{nf} = \frac{k_p + 2k_{bf} + 2(k_p - k_{bf})(1+\beta)^3\phi}{k_p + 2k_{bf} - (k_p - k_{bf})(1+\beta)^3\phi} k_{bf} \quad (14)$$

Now, overall heat transfer co-efficient is given by

$$\frac{1}{UA} = \frac{1}{\eta_o h_a A_a} + \frac{1}{h_f A_f} + R_w \quad (15)$$

Here, R_w is tube wall resistance and fouling factors are neglected.

Effectiveness for cross-flow unmixed fluid can be expressed as[9].

$$\varepsilon = 1 - \exp\left[\frac{NTU^{0.22}}{C^*}\exp\left(-C^* NTU^{0.78} - 1\right)\right] \quad (16)$$

Total heat transfer rate can be expressed as:

$$Q = \varepsilon C_{\min}\left(T_{f,in} - T_{a,in}\right) \quad (17)$$

The exergy gain by cold fluid (air) is given by,

$$\Delta Ex_a = Q - T_0\left[\dot{m}c_p \ln\left(\frac{T_{out}}{T_{in}}\right) + \dot{m}R\ln\left(\frac{p_{in}}{p_{out}}\right)\right]_a \quad (18)$$

At the outlet of the heat exchanger, pressure can be considered to be atmospheric. The second law efficiency are given by,

$$\eta_{II} = \Delta Ex_a / \Delta Ex_f \quad (19)$$

For implementing the analysis, a EES code is developed for the compact heat exchanger. For given inlet conditions and surface core geometry of cross flow heat exchanger, this program is useful in estimating the heat transfer coefficients, pressure drops, heat transfer rate and exergetic performances. In-build subroutines have been used for the temperature dependent properties of water (for nanofluid also) and air. Web site based data base 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 [10-11]. Properties of alumina nanoparticles have been taken from [4]. Nanoparticle volume fraction has been taken as 1.5%.

The simulation code has been validated with the result of [5]. The variation of heat transfer rate with water volume flow rate from the analysis have been compared with the result shown by [5] for same geometry and operating conditions. Similar trend has been observed and about ±7% deviation has been occurred during

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3. RESULTS AND DISCUSSIONS

3.1 Properties comparison of various coolants

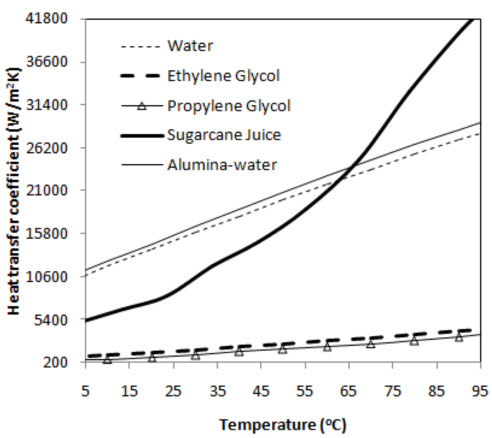

Fig. 2: Variation of heat transfer coefficient with temperature

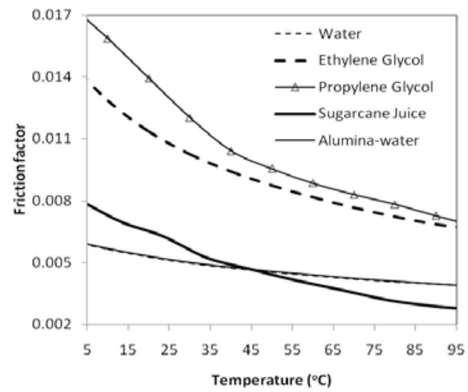

Fig. 3 Variation friction factor with temperature

Variations of heat transfer coefficient and friction factor with temperature are shown in Figs. 2 and 3, respectively, for mass velocity of 5000 J/m^{2}s 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. 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. Fin is assumed to be made of Aluminum alloy whose thermal conductivity is taken as 177W/m K.

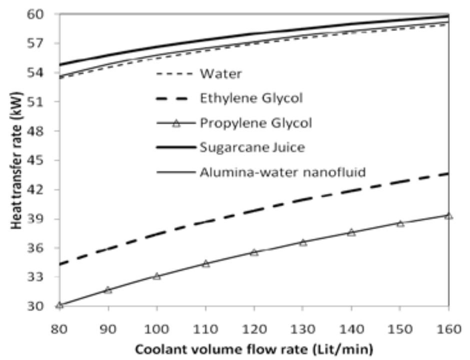

Fig. 4 Variation of heat transfer rate with coolant volume flow rate

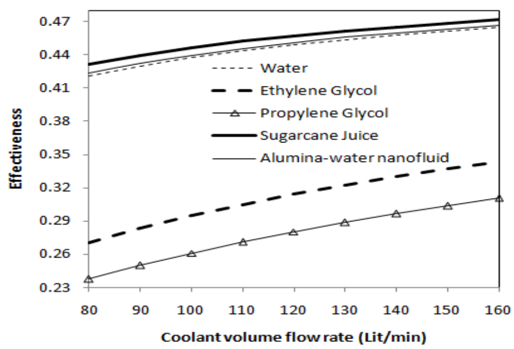

Fig. 5 Variation of heat exchanger effectiveness with coolant flow rate

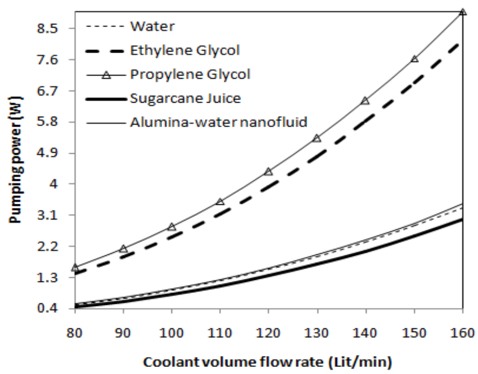

Fig. 6. Variation of pumping power with coolant flow rate

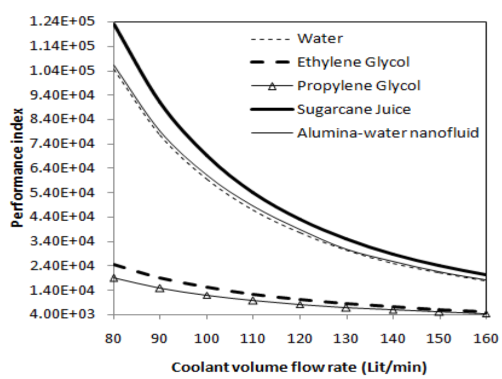

Fig. 7 Variation of performance index with coolant flow rate

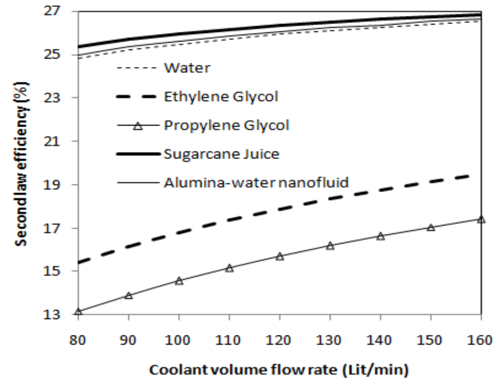

Fig. 8. Variation of second law efficiency with coolant flow rate

3.2 Performance comparison of various coolants

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 4-8. It has been observed that heat transfer rate, effectiveness and pumping power go on increasing with coolant flow rate and sugarcane juice yields slightly better heat transfer rate and effectiveness than water and nanofluid, whereas significantly better than EG and PG. On the other hand, 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.

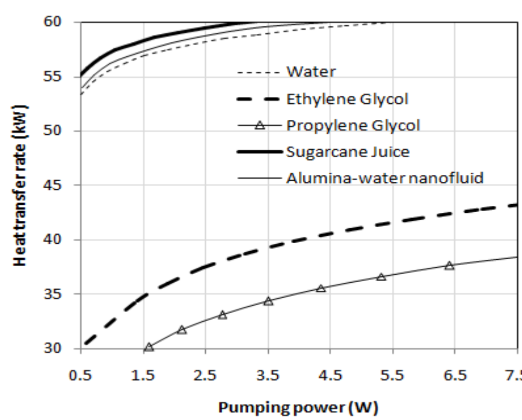

Fig. 9: Heat transfer rate with pumping power for various coolants

Performance characteristic of various coolant is illustrated in Fig. 9. 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.

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

Parameters	Water	EG	PG	Nanofluid	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 120 lpm and air frontal velocity of 10m/s. As shown, sugarcane juice yields maximum performance followed by alumina-water nanofluid. Recent many studies showed 5-10% radiator performaance improvement using nanofluids. Hence, it seems to be similar radiator performance by using nanofluids and sugarcane juice. However, both sugarcane juice and nanofluids have some challenges such as long term stabiliy to use in radiator

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.
- 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.

REFERENCES

[1] Yan WM and Sheen PJ. Heat transfer and friction characteristics of fin and tube heat exchangers. International Journal of Heat and Mass Transfer 2000,43,1651-1659.

[2] Torregrosa AJ, Broatch A, Olmeda P and Romero C. Assessment of the influence of different cooling system configurations on engine warm-up, emissions and fuel consumption. International Journal of Automotive Technology 2008,**9**(4), 447-458.

[3] Oliet C, Oliva A , Castro J and Pe´rez-Segarra CD. Parametric studies on automotive radiators, Applied Thermal Engineering .2007,27, 2033–2043.

[4] Sarkar J, Tarodiya R. Performance analysis of louvered fin tube automotive radiator using nanofluids as coolants. International Journal of Nanomanufacturing, 2013, 9(1),51-65.

[5] Vaisi A, Esmaeilpour M and Taheria H. Experimental investigation of geometry effects on the performance of a compact louvered heat exchanger. Applied Thermal Engineering2011, 31,3337-3346.

[6] Vajjha RS, Das DK and Kulkarni DP. Development of New Correlations for Convective Heat Transfer and Friction Factor in Turbulent Regime for Nanofluids. Int. J. Heat Mass Transfer2010,53(21–22), 4607–4618.

[7] Maiga SEB, Nguyen CT, Galanis N and Roy G. Heat transfer behaviours of nanofluids in a uniformly heated tube. Superlattices Microstructures2004,Vol. 35, 543-557.

[8] Yu W and Choi SUS. The role of interfacial layers in the enhanced thermal conductivity of nanofluids: a renovated Maxwell model. Journal of Nanoparticle Research2003, 5, 167-171.

[9] Leong KY, Saidur R, Kazi SN and Mamun AH. Performance investigation of an automotive car radiator operated with nanofluid-based coolants (nanofluid as a coolant in a radiator).Applied Thermal Engineering 2010, Vol. 30, 2685–2692.

[10] Filho ZA, Minim LA, Romero JT, MinimVPR, Telis VRN. Thermophysical Properties of Industrial Sugar Cane Juices for the Production of Bioethanol. J. Chem. Eng. Data 2010, 55, 1200–1203.

[11] Filho ZA, Telis VRN, Oliveira EB, Coimbra JSR and Romero JT. Rheology and fluid dynamics properties of sugarcane juice. Biochemical Engineering Journal 2011, 53, 260–265.