IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 16, Issue 5 Ser. IV (Sep. - Oct. 2019), PP 01-12
www.iosrjournals.org

Thermohydraulic Performance in the Flow of Copper Oxide (CuO) or Aluminum Oxide (Al_2O_3) Water-Borne Nanofluids in a Finned Flat Tube Heat Exchanger (Automotive Radiator)

Élcio Nogueira[1]
[1]*Department of Mechanical and Energy – DME, FAT/UERJ – Brazil*

***Abstract**: Theoretical thermohydraulic performance analysis of a compact heat exchanger, type finned flat tube, used in automotive radiators is performed in based-water nanofluids. The theory of effectiveness (ε-NTU), and experimental data for water flow, are used for comparison. The oxides used in the research are CuO (copper oxide) and Al_2O_3 (aluminum oxide). Results were obtained for air and water outlet temperatures, heat transfer rate, the pressure drop in finned channels and tubes, as a function of the volume fraction of the oxides. It has been shown that there is an effective improvement in heat exchange and little influence on total pressure drop when using oxides in volume fractions ranging from 0.01 to 0.05, and when Newtonian fluid flow is allowed in theoretical analysis. It can be observed from the data obtained that the performance of copper oxide is significantly superior to aluminum oxide. The laminarization effect of the flow was observed in the analysis, and it is more significant with increasing the volume fraction of oxide.*

Keywords: *Nanofluids; Copper Oxide; Aluminum Oxide; Compact heat exchanger; Automotive radiator; ε-NUT theory.*

Date of Submission: 09-10-2019 Date of acceptance: 25-10-2019

I. Introduction

The text by Kays and London (1984) provides an excellent introduction for the analysis of compact flat plate type heat exchangers and contains heat transfer coefficient and friction factor data for various geometries. Saidi, M.L. et. al (2006) present an experimental study to determine the performance of radiators used in passenger vehicles. The effectiveness method (ε-NTU) was used to determine the Nusselt number, heat transfer coefficient and global coefficient of heat exchange. The trial error method was used to determine the quantities of interest, from the available experimental data. Thus, the Colburn factor and the coefficient of friction were estimated as results of the procedure. The utility of the presented method is justified because it provides empirical data that can be used in projects of compact heat exchangers.

Research involving compact heat exchangers of all types, mainly car radiators, has been developed over the years and automotive companies invest high resources in all sorts of techniques that can optimize energy performance. As a result, new experimental apparatuses are built with the aim of improving automotive radiators through the incorporation of new technologies such as heat pipes and thermosyphons Pabón, N. Y. L. (2014).

Recently, Aroucha, A. L. C. and Pereira, F. L. (2019) presented a final monograph of the undergraduate course in Production Engineering with an emphasis in Mechanics on compact heat exchangers of the types used in the automotive and aeronautical industries. The compact heat exchanger, type finned flat tube, was analyzed in greater depth since it is widely used in automotive radiators.

Datil, V. R. et. al (2017) state that more powerful and smaller vehicles create problems in the dissipation of the heat exchanged in automotive radiators. They discuss the various different procedures recently used to optimize the thermal performance of ever-smaller radiators.

Bharathi Mahanti, V. D. M., Ravi Kumar. (2017) present an experimental study using nanofluids (CuO-Water) as a refrigerant for an automotive radiator. They conclude, mainly, that nanofluids increase the thermal performance of an automotive radiator.

Nagar, U. T., and Trivedi, B. M. (2017) state that the main aspect for the cooling of air and oil in an automotive vehicle is in the design of the radiator. They claim that nanofluids are being used for effective cooling.

Parashurama, M.S., Dhananjaya D.A., Naveena Kumar R.R (2015) present a study aimed at investigating the heat transfer characteristics of a nanofluid automobile radiator compared to a radiator using conventional refrigerant. They conclude that the heat transfer coefficient of nanofluids is higher than pure water and therefore the total radiator area can be reduced.

Rahul A. Bhogare, B. S. Kothawale (2014) performed experiments to analyze the effects of nanoparticle volume fraction on heat exchange. Results show that Nusselt number, total heat transfer, effectiveness, and total heat transfer coefficient increase with increasing nanoparticles, Reynolds number associated with air and mass flow of refrigerant circulating through the radiator.

V. L. Bhimani, P. P. Rathod, A. S. Sorathiya (2013) presents work where a forced convective heat transfer in water-based nanofluids has experimentally been compared to that of pure water in an automobile radiator. Results demonstrate that the application of nanofluid with low concentration can have enhanced heat transfer efficiency up to 45% in comparison with pure water.

II. Objectives

To analyze the effect of the volumetric fraction associated with copper and aluminum oxides on the performance of a compact finned flat tube heat exchanger (automotive radiator). To perform comparisons between heat transfer rate and pressure drop when using nanofluids compared to water flow. To observe the effect of nanoparticles on the fluid flow when they are used in low volume fraction and Newtonian fluid can be admitted.

III. Methodology

We propose an extension, as a first approximation, regarding the Colburn factor and coefficient of friction in Élcio Nogueira, André L. C. Aroucha e Fernando Lamin Pereira (2019):

$$\begin{aligned} \mathrm{J} = {} & 0.02619862019 - 3.626028274^{-5}\mathrm{Re_a} + 3.16047951^{-8}\mathrm{Re_a}^2 - 1.568380435^{-11}\mathrm{Re_a}^3 \\ & + 4.633774581^{-15}\mathrm{Re_a}^4 - 8.056435353^{-19}\mathrm{Re_a}^5 + 7.225312269^{-23}\mathrm{Re_a}^6 \\ & - 8.564813179^{-28}\mathrm{Re_a}^7 - 4.452384011^{-31}\mathrm{Re_a}^8 + 3.807970933^{-35}\mathrm{Re_a}^9 \\ & - 1.015364192\mathrm{d}^{-39}\mathrm{Re_a}^{10} \qquad (01) \end{aligned}$$

and

$$\mathrm{f} = 1.199866203\mathrm{Re_a}^{(-0.4747697361)} \qquad (02)$$

The model developed, through the procedures specified, was used the experimental data of the dissertation on Radiator Automotive of Ribeiro, L.N. (2007). In order to compute the data available, interpolations were performed within the available Reynolds number range and equations were used by Élcio Nogueira, André Aroucha e Fernando Lamim (2019). Equivalent to that done for the dimensionless coefficients J and f, we performed interpolations of the above data and the approximated equations obtained are given by the equations:

$$\begin{aligned} \mathrm{QK_{exp}} = {} & 74.96904025 + 53.18082817\mathrm{m_a} - 3.148106367\mathrm{m_a}^2 + 0.131605291\mathrm{m_a}^3 \\ & - 0.002308469675\mathrm{m_a}^4 \ (03) \end{aligned}$$

and

$$\begin{aligned} \Delta\mathrm{P_{Exp}} = {} & 5.833333333 - 2.91045066\mathrm{dm_a} + 2.769230769\mathrm{m_a}^2 - 0.2812742813\mathrm{m_a}^3 \\ & + 0.008741258741\mathrm{m_a}^4 \qquad (04) \end{aligned}$$

$\mathrm{QK_{exp}}$ is the ratio between the experimental heat transfer rate and the Mean Logarithmic Temperature Difference – MLTD.

$\Delta\mathrm{P_{Exp}}$ is the experimental loss of charge and ma being the air mass flow rate.

The experimental data relevant to the comparisons, heat transfer rate and pressure drop, were obtained by Ribeiro, L.N. (2007) in the wind tunnel of Behr Brasil Ltda, the constructor of automotive radiators, are represented by Tables 1 and 2. Thermophysical properties of base fluid and nanoparticles are presented in Table 3.

Table 1 Dimensions of finned flat tubes heat exchanger

Features	*Nomenclature*	*Value*
Heat Exchanger Width	B	396 mm
Heat Exchanger Height	H	436 mm
Heat Exchanger Thickness	L	55 mm
Dimensions of tubes	a x b x c	13,3 x 2,6 x 448 mm
Wet perimeter of each tube	Pc	31,8 mm
Hydraulic diameter	D_H	4,36 mm
Number of pipe rows	Nf	3
Tubes per row	Tf	43
Total number of tubes	Qt	129
Cross-tube spacing	ST	8,8 mm
Distance between tubes	SL	18,3 mm
Thickness of fins	e	0,05 mm
Spacing between fins	E	2,8 mm
Number of fins	Qa	155

Dissertation of Ribeiro, L.N. (2007; Behr Brasil Ltda)

Table 2 Thermo physical properties of air

Property	*Value*
cp	1,008 kJ/(kg.K)
k	28,816 W/(m.K)
v	19,31 x 10^{-6} m^2/s
ρ	1,048 kg/m^3
Pr	0,702

Dissertation of Ribeiro, L.N. (2007; Behr Brasil Ltda)

Table 3 Thermophysical properties of base fluid and nanoparticles

SI	Property	Copper Oxide – CuO	Aluminum – Al_2O_3	Water – H_2O
2	ρ Kg/m3	8933	3950	1000
3	Cp [J/ (Kg . K)	385	873.34	4184
4	μ [Kg/m3]	-	-	0.478 10^{-3}
5	ϑ [m2/s]	-	-	10^{-6}

Theoretical analysis

Determination of heat transfer rate

The theoretical determination of the heat transfer rate depends on the overall heat transfer coefficient, which in turn depends on the heat transfer coefficients, ha and hw, on the airside and the waterside, respectively. To begin the calculations, it becomes necessary to determine the physical properties in the function of the average temperatures of the fluids. However, the exit temperatures, in theory, are unknown a priori, and the average temperatures should be initially estimated.

With the initially stipulated output temperatures, defined physical properties, and the geometric quantities of the exchanger supplied, we have,

For the air:

$$G_a = \frac{m_a}{A_{mín}} = \frac{m_a}{\sigma_a A_{fr}} \qquad (05)$$

$$Re_a = \frac{G_a D_{ha}}{\mu_a} \qquad (06)$$

$$J = \frac{h_a}{G_a c_{pa}} {Pr_a}^{2/3} \qquad (07)$$

The Prandtl number for air, Pra, is obtained by interpolating the data, valid for air as the ideal gas, published by Çengel and Boles (2013, page 934):

$$Pr_a = 1.005351636d0 + 0.01292094145 Tmed_a + 2.524174317^{-5} {Tmed_a}^2 - 5.074647769^{-8} {Tmed_a}^3 + 1.564763295^{-8} {Tmed_a}^4 \qquad (08)$$

then,

$$h_a = J \frac{G_a c_{pa}}{{Pr_a}^{2/3}} \qquad (09)$$

For water-based nanofluid properties we have:

$$\rho_{nano} = \emptyset \rho_{particle} + (1-\emptyset)\rho_w \qquad (10)$$

$$\mu_{nano} = \mu_w (1 + 2.5\emptyset) \qquad (11)$$

$$Cp_{nano} = (\emptyset \rho_{particle}\, Cp_{particle} + (1-\emptyset)\rho_w Cp_w)/\rho_{nano} \qquad (12)$$

$$k_{nano} = [(k_{particle} + 2k_w + 2(k_{particle} - k_w)(1-0.1)^3\emptyset)/(k_{particle} + 2k_w(k_{particle} - k_w)(1+0.1)^2\emptyset)]\, k_w \ (13)$$

$$\alpha_{nano} = \frac{k_{nano}}{\rho_{nano}\, Cp_{nano}} \qquad (14)$$

Where, Equation 11 is the Einstein equation, used for very dilute suspension, and ∅ is the volume fraction of nanoparticles.

Other quantities associated with the flow are obtained by:

$$Re_{nano} = \left[4\left(\frac{m_{nano}}{N_{tubes}}\right)\right] / (\pi D_{hnano}\, \mu_{nano}) \qquad (15)$$

$$f_{nano} = \frac{64}{Re_{nano}} for Re_{nano} \leq 2100 \quad (16)$$

or

$$f_{nano} = (0.79 Ln(Re_{nano}) - 1.69)^{-2} for Re_{nano} > 2100 \quad (17)$$

Considering the flow regime of the Newtonian flow of water-base nanofluid in the tube as completely developed, we have, for turbulent flow, approximately:

$$\mathrm{N}u_{nano} = 0,023\, Re_{nano}{}^{0,8} Pr_{nano}{}^{0,4} \quad (18)$$

If the flow regime in the water-base nanofluid is laminar, it is used to interpolate the data of the Master Thesis of Nogueira, E. (1988, page 130), for the thermal input region under development:

$$Nu_{nano} = 1.409019812d0 Z_{nano}{}^{(-0.3511653489)} \quad para\ 10^{-5} \leq Z_{nano} < 10^{-3} \quad (19.1)$$

$$Nu_{nano} = 1.519296981d0 Z_{nano}{}^{(-0.3395483303\, d0)} \quad para\ 10^{-3} \leq Z_{nano} < 10^{-2} \quad (19.2)$$

$$\begin{aligned} Nu_{nano} = {} & 10.8655 - 570.4671787 Z_{nano} + 28981.67578 Z_{nano}{}^{2} - 950933.9838 Z_{nano}{}^{3} \\ & + 20237498.47 Z_{nano}^{4} - 276705269.6 Z_{nano}{}^{5} + 2340349265 Z_{nano}{}^{6} \\ & - 1.112482493^{10} Z_{nano}{}^{7} + 2.269345238^{10} Z_{nano}{}^{8} para 10^{-2} \leq Z_{nano}\ 10^{-1} \quad (19.3) \end{aligned}$$

$$\begin{aligned} Nu_{nano} = {} & 5.261d0 - 19.93019048 nano + 139.4921627 Z_{nano}{}^{2} - 605.9954034 Z_{nano}{}^{3} \\ & + 1716.100694 Z_{nano}^{4} - 3217.96875 Z_{nano}{}^{5} + 3954.86111 Z_{nano}{}^{6} - 3056.051587 Z_{nano}{}^{7} \\ & + 1344.246031 nano^{8} - 256.2830687 Z_{nano}{}^{9}\ para\ 10^{-1} \leq Z_{w} = 10^{0} \quad (19.4) \end{aligned}$$

Then, we have:

$$h_{nano} = Nu_{nano} \frac{k_{nano}}{D_{hnano}} \quad (20)$$

The overall heat transfer coefficient is obtained in relation to the air exchange area and, in order to perform the calculations, it is necessary to determine the efficiency of the fin since there is variation of temperature between the entrance of the plate of the exchanger (base of the fin) and its outlet:

$$\eta = \frac{tgh(mL)}{mL} \quad (21)$$

where

$$mL = \sqrt{2h_a/k_a t} \quad (22)$$

The efficiency of the fin, weighted by area, is determined by:

$$\eta' = \beta\eta + 1 - \beta \quad (23)$$

where

$$\beta = \frac{Finarea}{totalarea} \quad (24)$$

Then, we have:

$$\frac{1}{U_a} = \frac{1}{\eta' h_a} + \frac{1.0}{A_{med} K_{aleta}} + \frac{1}{(A_w/A_a) h_{nano}} \quad (25)$$

where

$$A_{med} = \frac{A_a + A_{nano}}{2.0} \quad (26)$$

and

$$\frac{A_w}{A_a} = \frac{water - based\ nanofluid\ sideheattransferarea}{airsideheattransferarea} \quad (27)$$

By the theory of effectiveness (ε-NUT) we have:

$$N = NTU = \frac{A_a U_a}{C_{mín}} \quad (28)$$

The thermal capacities of air and nanoparticles water-based are calculated by:

$$Ca = m_a * Cp_a \quad (29)$$

and

$$Cnano = m_{nano} * Cp_{nano} \quad (30)$$

Cmin is the lowest value between the thermal capacities of water and air. Finally,

$$Q = \varepsilon C_{mín} (T_{h,af} - T_{c,af}) \quad (31)$$

$$\Delta T_{Ln} = \frac{Q}{U_a A_{total}} \quad (32)$$

and

$$QK_{teo} = \frac{Q}{\Delta T_{Ln}} \quad (33)$$

where

$$\varepsilon = 1 - exp\left[\left(\frac{c_{min}}{c_{max}}\right)^{-1} (NTU)^{0,22} \left\{exp\left[-\frac{c_{min}}{c_{max}} (NTU)^{0,78}\right] - 1\right\}\right] \quad (34)$$

according to Kakaç, S. (1991, page 35).

QK_{teo} is the theoretical value for the ratio between the heat transfer rate in the air and the mean logarithmic temperature difference – MLTD.

With the heat transfer rate determined, as the first approximation, one can calculate the air and water exit temperatures, through the energy balance equations:

$$Q = \varepsilon C_{min}\left(T_{h,af} - T_{c,af}\right) \quad (35)$$

and

$$Q = m_a c_{pa}\left(T_{a,af} - T_{a,ef}\right) \quad (36)$$

The average outlet, air and water temperatures can then be determined and compared to the initially set temperatures:

The mean air and water temperatures can then be determined and compared to the initially defined temperatures:

$$T_{m,a} = \frac{T_{a,af} + T_{a,ef}}{2} \quad (37)$$

and

$$T_{m,nano} = \frac{T_{nano,af} + T_{nano,ef}}{2} \quad (38)$$

With the average temperatures finally calculated, the values obtained for the heat transfer rate were compared and, if they are outside of an admissible value, when compared with experimental values or empirical expressions, the calculations for thermophysical properties can be re-started, until a satisfactory convergence is obtained for the problem.

Determination of dissipated power in the air

In the calculation of pressure drop in a finned heat exchanger the main losses are related to the friction factor (f_a), and the air-side pressure drop can be determined by [Kakaç, S. (2002)]:

$$\Delta P_a = \left[\frac{\left(G_a^{\,2}\right)}{2\rho_{ai}}\right]\left[\left(1.0 + \sigma_a^{\,2}\right)\left(\frac{\rho_{ai}}{\rho_{ao}} - 1.0\right) + \frac{4.0 f_a L_{aleta}\,\rho_{ai}}{Dh_a\,\rho_{med}}\right] \quad (39)$$

where

$$\frac{1.0}{\rho_{med}} = \left(\frac{1.0}{\rho_{ai}} + \frac{1.0}{\rho_{ao}}\right) \quad (40)$$

The friction factor, f_a, is determined by equation 02, obtained through the experimental values of Kays and London (1984), and the specific mass of air at the outlet of the heat exchanger, ρao, can be estimated initially as a function of outlet temperature of the air, through the equation:

$$\rho_{ao} = 1.28123142 - 0.004142716793xT_{oa} + 1.921703199xTo_a^{\,2} - 1.340288713xT_{oa}^{\,3} + 3.583356643xT_{oa}^{\,4} \quad (41)$$

The pressure drop determination procedure is also iterative and as complex as that defined for determining the heat transfer rate, since a very refined approximation to the value of the specific mass of the air at the heat exchanger outlet is required.

The value obtained for the specific mass of the air outlet, through the polynomial interpolation, equation 41, allows the determination of an approximate value for the pressure drop, but does not initially represent the correct value accepted as solution. The stopping criterion, also here, is defined by comparing the result obtained for the experimental pressure drop, within an acceptable error.

For the airflow, we have:

$$Pot_a = \frac{m_a \Delta P_a}{\rho_a} \quad (42)$$

Determination of dissipated power in nanofluid flow

$$Pot_{nano} = \frac{m_{nano}\,\Delta P_{nano}}{N_{tubes}\,\rho_{nano}} \quad (43)$$

$$\Delta P_{nano} = \frac{8 f_{nano}\,L_{tube}\,Q_{nano}^{\;2}}{Dh_{nano}^{\;5}\pi^2 g} \quad (44)$$

$$Pot_{nano} = Q_{nano}\,\Delta P_{nano} \quad (45)$$

Determination of dissipated total power

$$Pot_{total} = Pot_a + Pot_{nano}\,N_{tubes} \quad (46)$$

IV. Results and Discussions

In Figure 1 we have graphical results for the Colburn factor and coefficient of friction. The results presented are obtained by extending the data from Kays and London (1984), according to the procedure recommended by Élcio Nogueira, André Aroucha and Fernando Lamin Pereira (2019) and Equations 01 and 02.

In Figures 2 and 3 we present results obtained for the properties of nanofluid, as a function of nanoparticle volume fraction. It can be observed that there is a significant difference in specific mass and specific heat between oxides. The differences observed for thermal conductivity and dynamic viscosity are not significant. Regarding thermal diffusivity, Figure 3, there is a slight difference for oxides.

In Figure 4 we have results for friction factor in tube and air, as a function of the mass flow of nanofluid. It can be observed that the friction factor, in this case, is much higher than the friction factor for water flow and that there are also significant differences for the oxides considered in this analysis when the volume fraction varies.

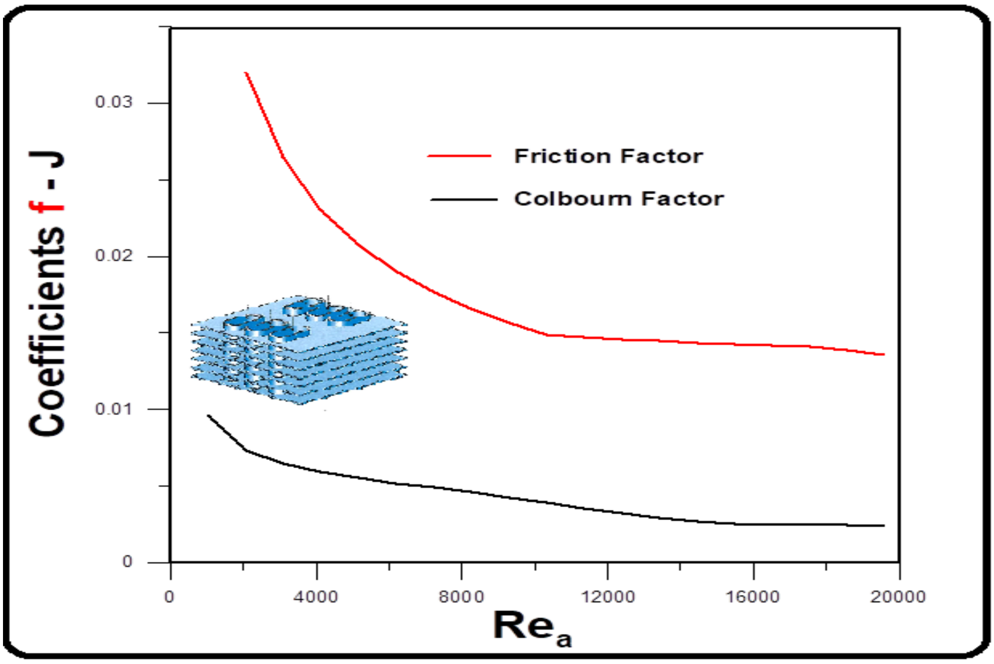

Figure 1 Friction factor and Colburn factor for finned flat-tube compact heat exchanger

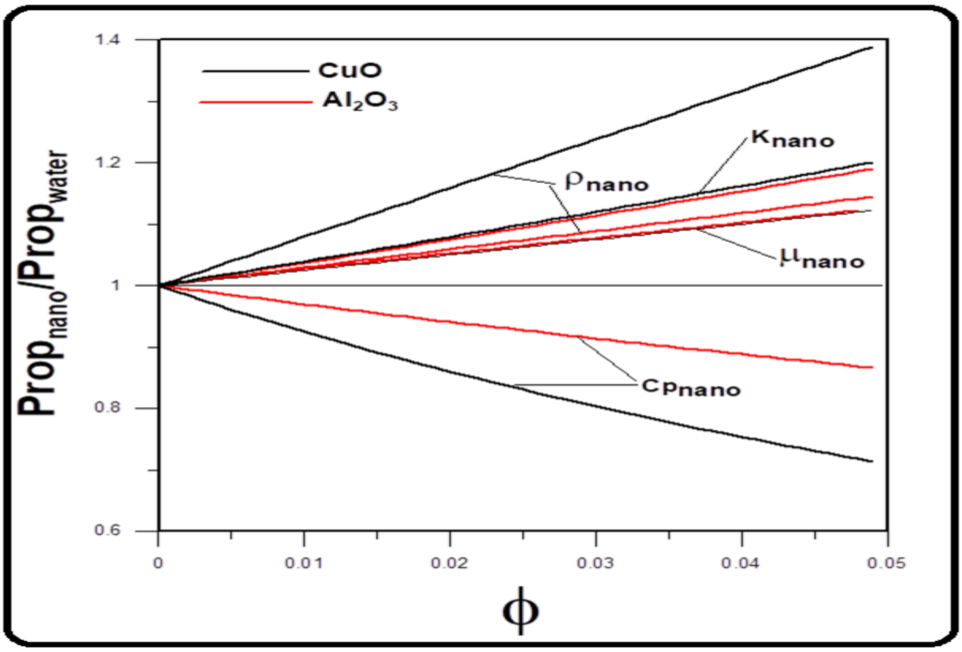

Figure 2 Properties of water-based nanofluids in relation of water

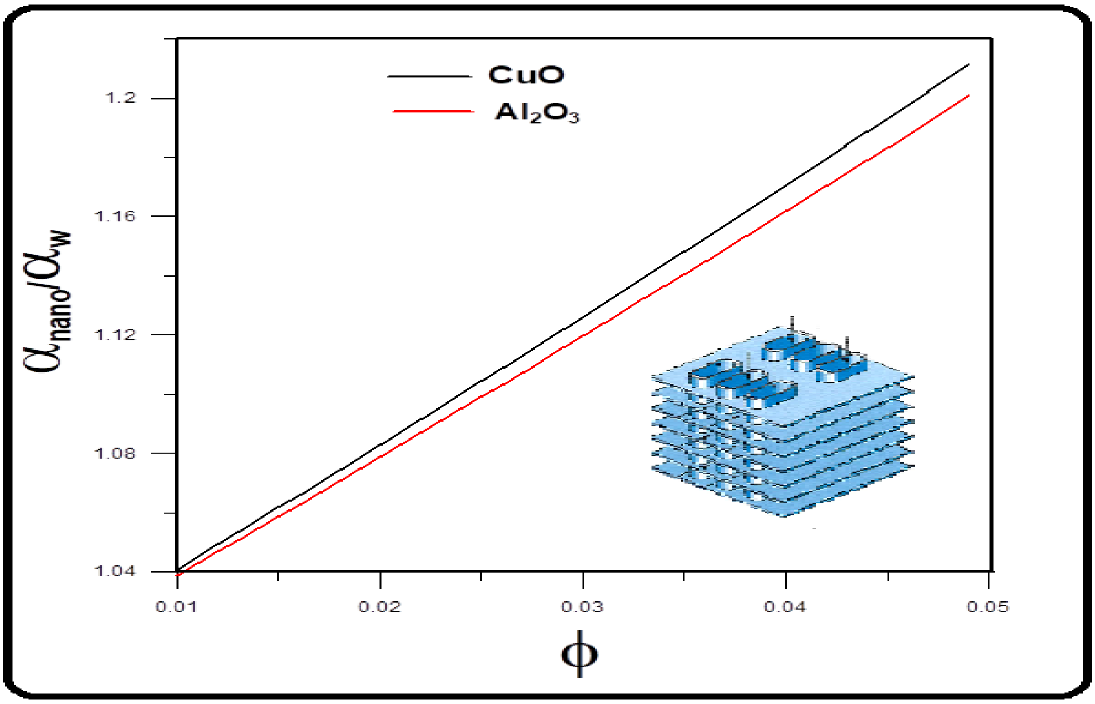

Figure 3 Thermal diffusivity of water-based nanofluids in the relation of water

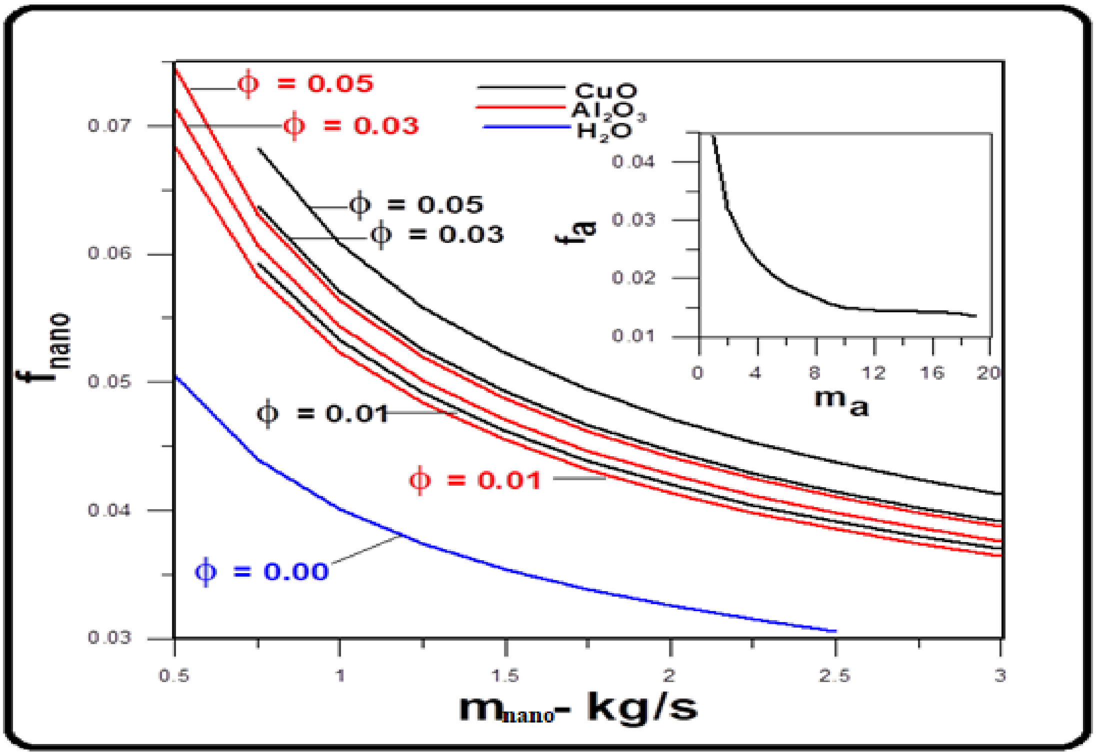

Figure 4 Friction factor for water-based nanofluids and volume fraction as a parameter

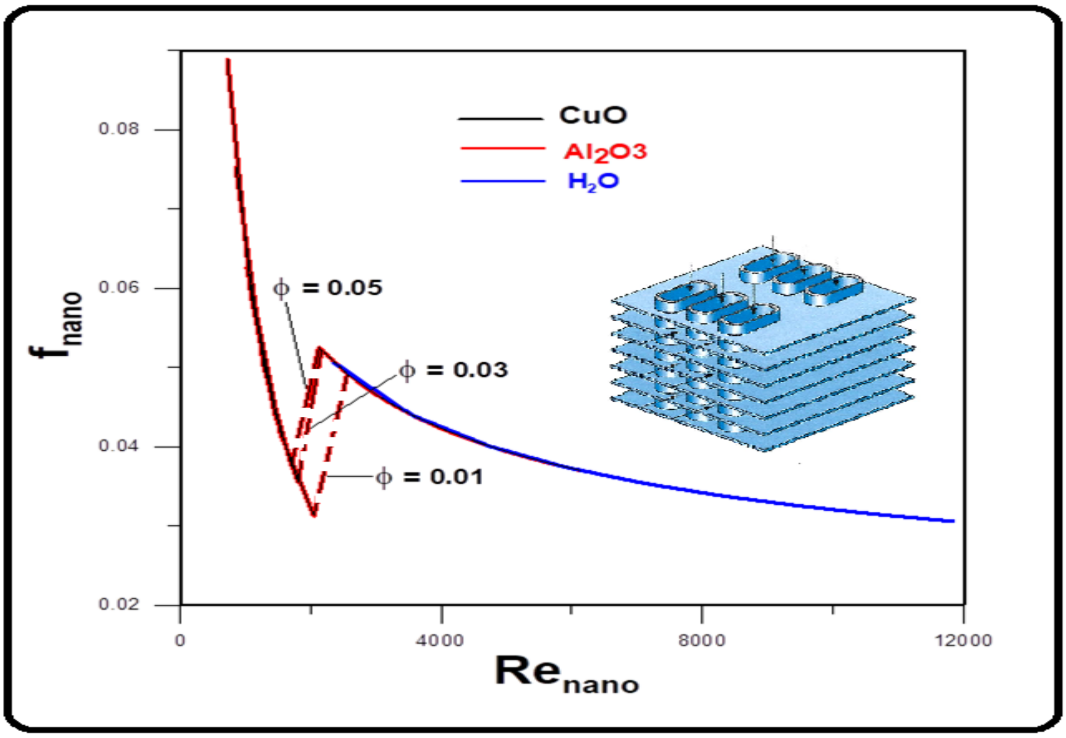

Figure 5 Friction factor and the laminarization of the water-based nanofluids flow

When the presented data are arranged in the dimensionless form, Figure 5, we realize that there is no significant difference for the friction factor regarding the oxide volume ratio. However, there is a very interesting aspect to consider when analyzing the results in relation to the Reynolds number for the flow: there is a progressive laminarization of the flow when the volume fraction is changed in relation to the water flow. While the water flow is turbulent for all flows considered in the analysis, it becomes progressively laminar for larger fractions in oxide volume.

Through Figures 6 and 7 we analyze the air temperature variation at the radiator outlet. It can be observed that there is a clear temperature variation for oxides in relation to water. The higher the volume fraction of the oxide, the higher the air outlet temperature. As might be expected, the lowest the nanofluid flow rate, the higher the air outlet temperature. Copper oxide has a significantly higher effect than aluminum oxide with respect to the increase in air outlet temperature in the radiator.

Through Figure 8 we analyze the temperature variation at the radiator tube outlet, for a given mass flow rate of nanofluid. The higher the volume fraction of the oxide, the lower the outlet temperature. Similar to that observed for Figures 6 and 7, copper oxide has better performance compared to aluminum oxide.

The heat transfer rate on the radiator is shown in Figure 9 for a given nanofluid flow. The best performance of oxides in relation to water flow is evident. To corroborate what has been observed previously, copper oxide outperforms aluminum oxide.

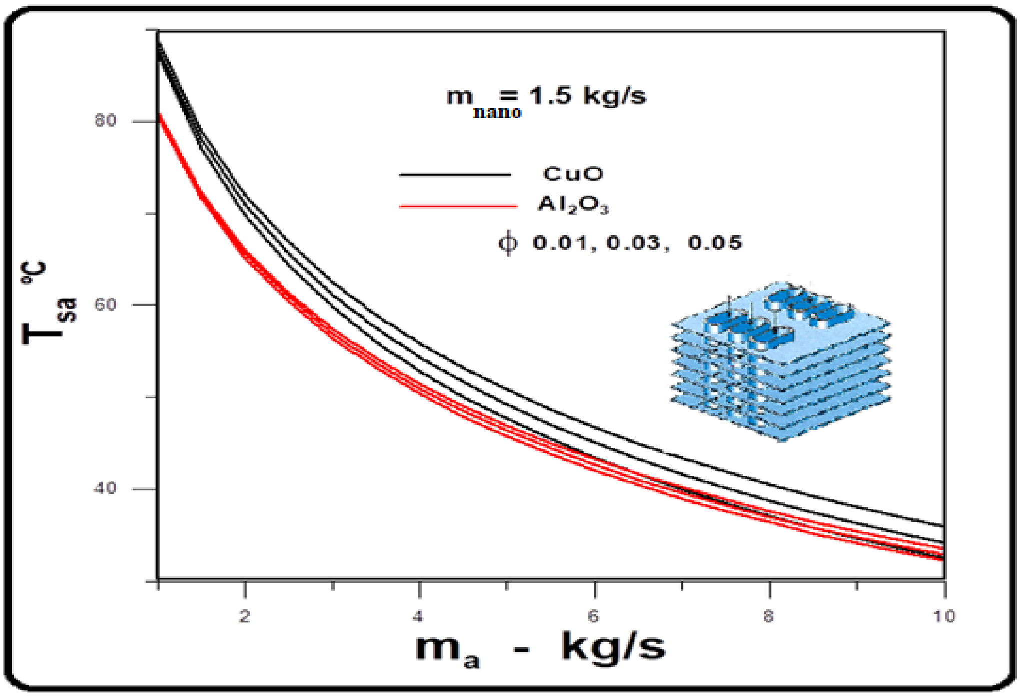

Figure 6 Outlet air temperature with volume fraction of nanoparticles as a parameter

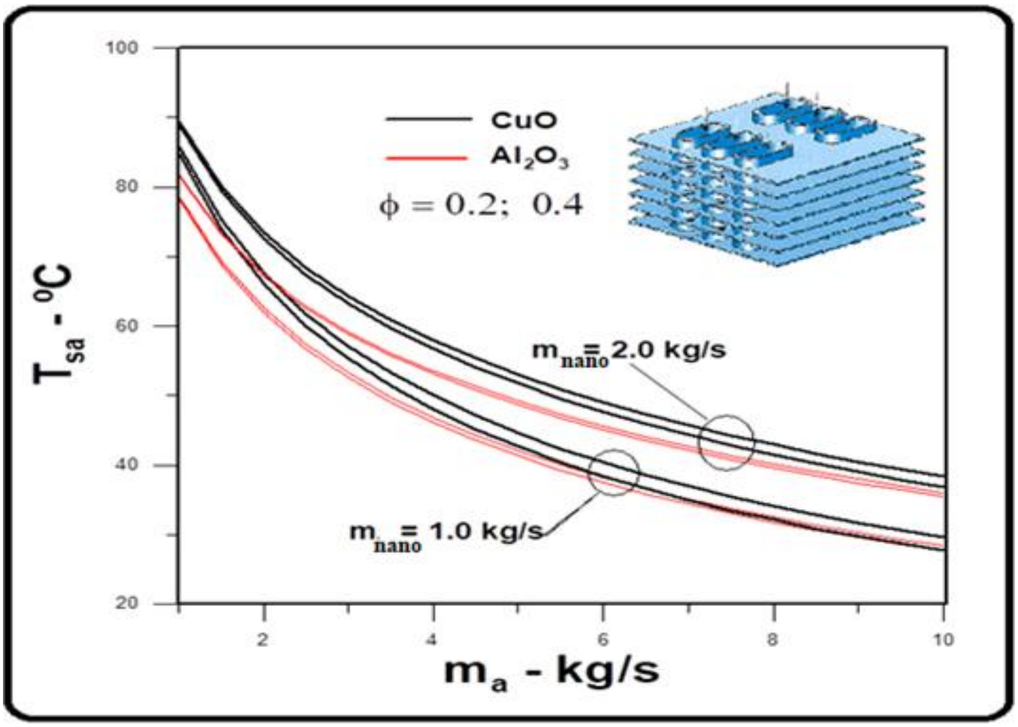

Figure 7 Outlet air temperature with volume fraction of nanoparticles as a parameter

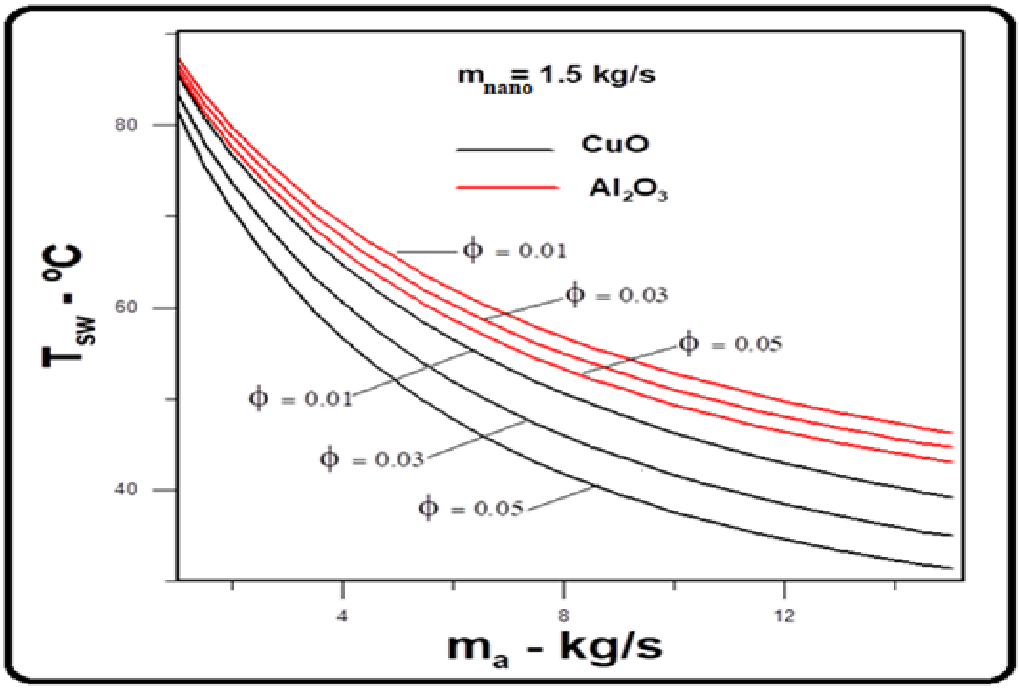

Figure 8 Nanofluid output temperature with volume fraction as a parameter

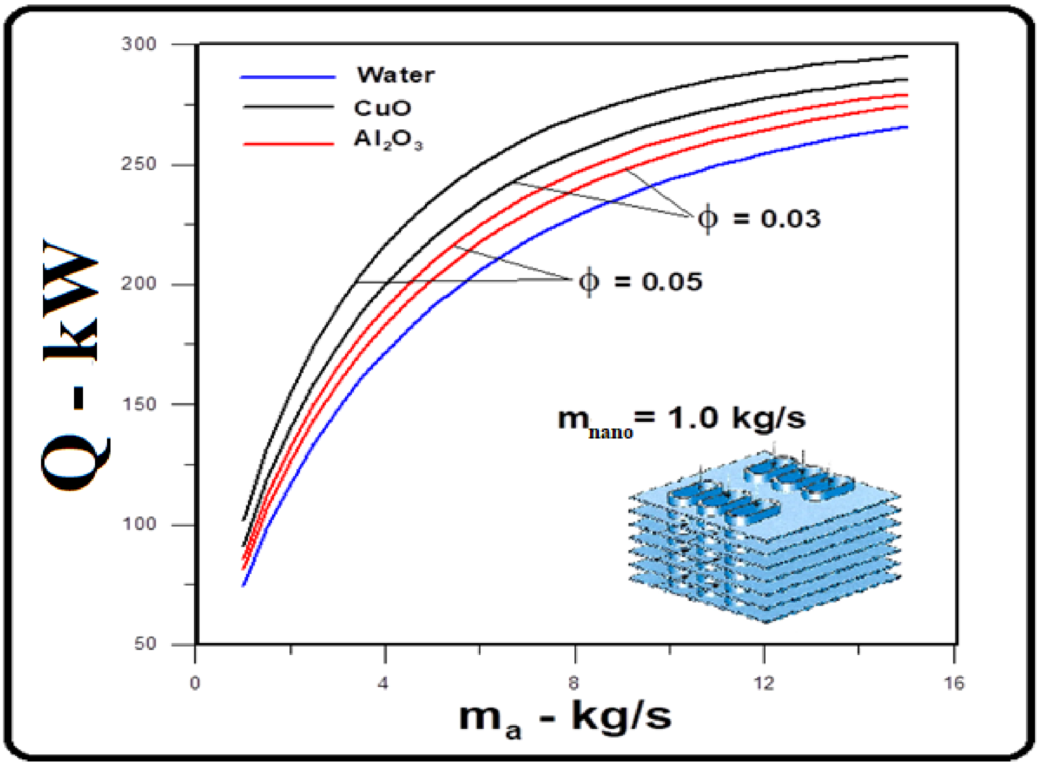

Figure 9 Heat transfer rate with oxide volume fraction as a parameter

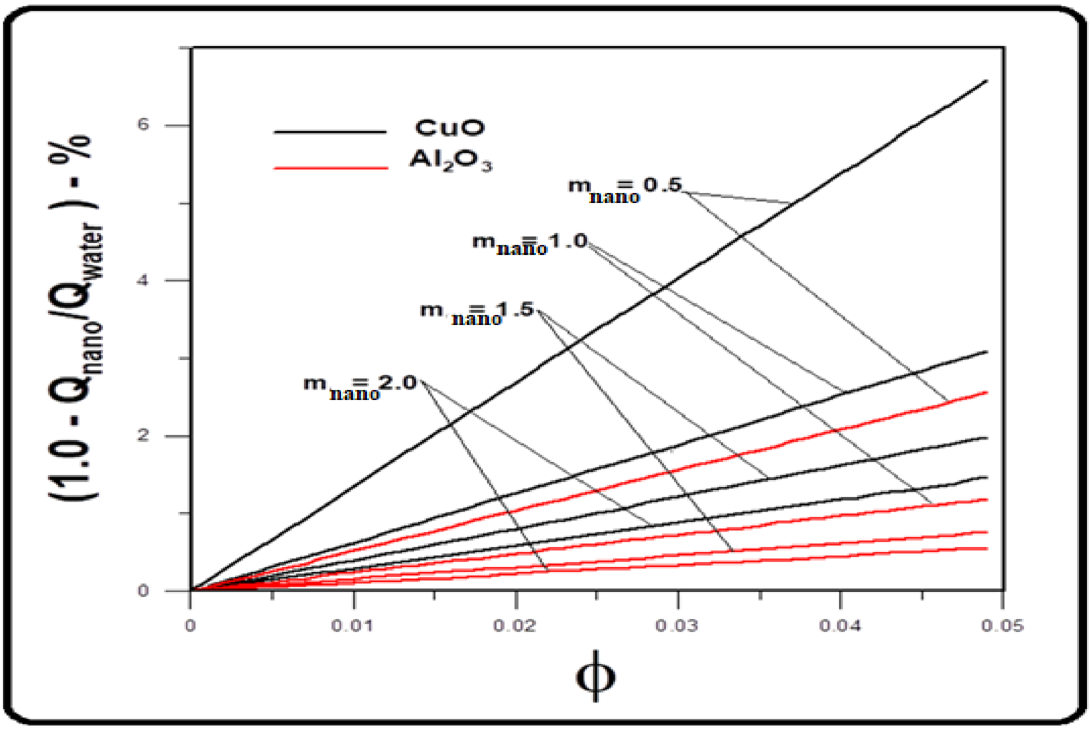

Figure 10 Effectiveness of nanoparticle addition

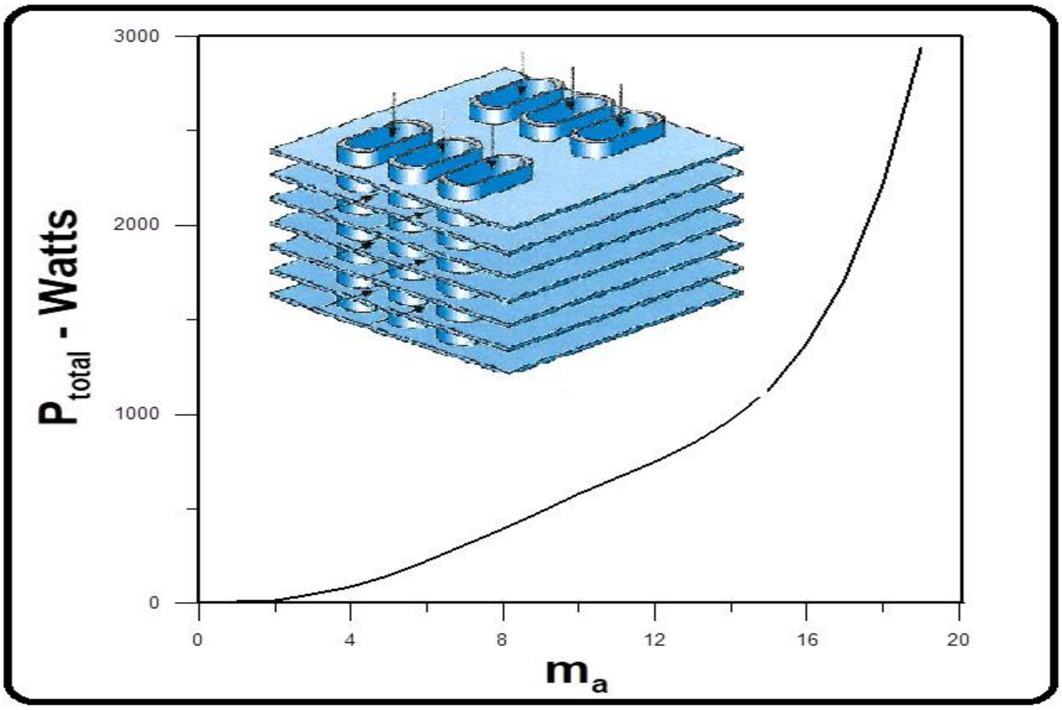

Figure 11 Total dissipated power

Figure 10 demonstrates the effectiveness of oxide performance in heat transfer rate relative to volume fraction. It can be observed that the performance is higher for smaller nanofluid flow rates, for the same volume fraction. In fact, the higher the nanofluid flow rate, the lower the water performance. This must be associated with the fact that the lower the flow in the tubes, the longer the heat absorption by the nanoparticles.

Through Figure 11 we present the total power dissipation by the flow. It is important to note that the power dissipated by the flow in the pipes is very small in relation to the dissipation in the finned channels.

V. Conclusions

The results show that the addition of nanoparticles in automotive radiators presents significant thermal performance and does not significantly influence energy dissipation due to viscous effects, for low oxide volume fraction values and when the flow in the tubes is considered Newtonian.
Addition of Copper Oxide (CuO) nanoparticles presents better thermal performance compared to the addition of Aluminum Oxide (Al2O3).
The effectiveness of adding nanoparticles is higher for lower nanofluid flow rates, for the same volume fraction of oxide.

There is a significant laminarization of the flow when nanoparticles are added to the flow.
Dimensional analysis can help conclude whether simply adding nanoparticles might be sufficient to invest in reducing the automotive radiator once their effective use was proved.

References

[1]. AROUCHA, A.L.C. AND PEREIRA, F.L. "Study and theoretical and experimental analysis of a compact heat exchanger type finned tube (Automotive Radiator)". Graduation Final Project by the Course of Production Engineering with emphasis in Mechanics, Faculty of Technology of the State University of Rio de Janeiro - FAT / UERJ. Advisor: Prof. Dr. Élcio Nogueira, Resende, Brazil, 2019.
[2]. ÉLCIO NOGUEIRA, ANDRÉ LUÍS DO CARMO AROUCHA, FERNANDO LAMIM PEREIRA, "Compact Heat Exchanger in Automotive Radiators: Theoretical Versus Experimental Analysis of Coefficient of Friction and Colburn Factor". International Journal for Research in Applied Science & Engineering Technology (IJRASET) Volume 7 Issue II, 2019,
[3]. ÇENGEL, Y. A., BOLES, M. A. "Thermodynamics". AMGH Editora Ltda, Porto Alegre, RS, 2013.
[4]. KAKAÇ, S. "Boilers, Evaporators, and Condensers". John Wiley & Sons, INC. New York, 1991.
[5]. KAYS, W. M., LONDON, A. L. "Compact Heat Exchangers". McGraw-Hill, New York, 1984.
[6]. NOGUEIRA, E. "Laminar Flow and Heat Transfer in Immiscible Fluids without Stratification". Thesis presented to the Postgraduate Division of the Technological Institute of Aeronautics as part of the requirements to obtain the title of Master of Mechanical Engineering, in the Aeronautics, Propulsion and Energy area of the Aeronautical and Mechanical Engineering Course. Advisor: Prof. Dr. Renato Machado Cotta, 1988.
[7]. ÖZISIK, M. N. "Heat Transfer - A Basic Text". Editora Guanabara Koogan S.A., Rio de Janeiro, 1990.
[8]. PARASHURAMA M. S., DHANANJAYA D. A., NAVEENA KUMAR R.R. "Experimental Study of Heat Transferi na a Radiator using Nanofluid". IJERDR, Volume 3, Issue 2, 2015.
[9]. PABÓN, N. Y. L. "Design and Manufacture of a Wind Tunnel and Thermal Characterization of an Automotive Radiator". Dissertation submitted to the Graduate Program in Mechanical Engineering of the Federal University of Santa Catarina to obtain the Master Degree in Mechanical Engineering. Advisor: Marcia B. H. Mantelli. Florianópolis, 2014.
[10]. RAHUL A. BHOGARE, B.S. KOTAWALE. "Performance investigation of Automobile Radiator operated with Al2O3 based nanofluid". Journal of Mechanical and Civil Engineering. Volume 11, Issue 3, Ver. V (May – Jun, 2014), pp. 23-30.
[11]. RIBEIRO, L. N. "Optimization of Flat Fin Compact Heat Exchangers through Limit Layer Analysis". Dissertation defended by the Department of Mechanical Engineering of Taubaté, as part of the requirements to obtain the Master's Degree by the Graduate Program in Mechanical Engineering. Advisor: Dr. Sebastião Cardoso, 2007.
[12]. Saidi, M.L. et. Al. "Experimental Study of Thermal Performance and Pressure Drop in Compact Heat Exchanger Installed in Automotive". ASME Internal Combustion Engine Division, Spring Technical Conference May 8-10, 2006, Aachen, Germany, 2006.
[13]. SHAHAD FALIH HASSAN AND THAHSEEN A. AL-HATTAB. "Enhancement of Thermal Performance of Nanofluids in Heat Pipe". International Journal of Advances in Engineering & Technology, Dec. 2016.
[14]. V. L. BHIMANI, Dr. P. P. RATHOD. "Experimental Study of Heat Transfer Enhancement Using Water Based Nanofluids as a New Coolant for Car Radiators". International Journal of Emerging Technology and Advanced Engineering". Volume 3, Issue 6, June 2013.

Élcio Nogueira" Thermohydraulic Performance in the Flow of Copper Oxide (CuO) or Aluminum Oxide (Al2O3) Water-Borne Nanofluids in a Finned Flat Tube Heat Exchanger (Automotive Radiator)" IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) , vol. 16, no. 5, 2019, pp. 01-12