EXPERIMENTS ON LAMINAR FORCED CONVECTION OF NANOFLUIDS AND MODELING WITH HEAT LOSSES AND TEMPERATURE DEPENDENT THERMOPYSICAL PROPERTIES

Ivana G. Cerqueira, igabi@ufrj.br
Thiago C. de Carvalho, thiagomec@poli.ufrj.br
Laboratory of Transmission and Technology of Heat – LTTC, POLI & COPPE/UFRJ, Rio de Janeiro, RJ
Jeziel S. Nunes, jeziel@inpi.gov.br
INPI, Rio de Janeiro, RJ
Renato M. Cotta, cotta@mecanica.coppe.ufrj.br
Laboratory of Transmission and Technology of Heat - LTTC,
Mechanical Eng. Dept. – POLI & COPPE/UFRJ, Rio de Janeiro, RJ, Brazil

Abstract. *The objective of the present work is to theoretically and experimentally analyze laminar forced convection of nanofluids for flows in straight circular tubes, involving the determination of local and average heat transfer coefficients, as a function of the Reynolds number, for a given concentration of nanoparticles in acquired commercial nanofluids (water-aluminium oxide). The theoretical model for the thermal entrance involves temperature dependent thermophysical properties and heat losses to the external environment, and the applied methodology for the solution of the modeled problem consists of the Generalized Integral Transform Technique (GITT), employing a dedicated routine prepared in the Mathematica 7.0 platform. An available thermohydraulic circuit was modified and tested with a test section made of a copper tube with external diameter of* $9.525x10^{-3}$ *m, with electrical resistance strip heating, seeking an uniform prescribed heat flux condition. Experimental runs for various mass flow rates, both for deionized water and the water-alumina nanofluid at 20% nominal mass concentration, allowed for obtaining a representative set of experimental results offering comparisons and validation of the proposed model, and illustrating the heat transfer enhancement effect promoted by the nanofluid.*

Keywords*: Nanofluids, Heat transfer enhancement, Integral Transforms, Forced convection, Hybrid methods*

1. INTRODUCTION

Due to the industry urge in enhancing the thermal performance of their most common working fluids such as water, ethylene glycol, and heat exchanger oil, several studies were developed during the last decades to improve the performance of the so-called thermal fluids. Recent research in nanotechnology has enabled the development of a new class of thermal fluids in the form of liquid suspensions containing metal or metal oxide particles, mostly in sizes smaller than 50 nanometers, which have higher thermal conductivities than those of traditional heat transfer fluids, and are known as nanofluids. The name "nanofluid" was first used by a group of the Argonne National Laboratory in the U.S. (Choi, 1998; Choi and Eastman, 2001).

Since then, several experiments with nanofluids have advertised a significant increase in thermal conductivity compared to the regular fluid without suspended particles, and with a marked temperature dependence of the thermophysical properties (Eastman et al. 2004; Keblinski et al., 2005, Das et al., 2006). Correlations for thermal conductivity with explicit temperature dependence have been recently proposed for water based nanofluids (Mintsa et al., 2009). Results in general confirm the increase in the effective thermal conductivity with an increase in particle volume fraction and with a decrease in particle size. Also, the relative increase in thermal conductivity was found to be more important at higher temperatures. Nguyen et al. (2008) also studied the effect of temperature and volume concentration of nanoparticles in the dynamic viscosity of the nanofluid water-Al_2O_3. They concluded that the viscosity of the nanofluid depends on temperature and concentration, while the particle size has an effect only at high fractions of nanoparticles, and established a critical temperature that is dependent on the concentration and size of nanoparticles, in which the properties of nanoparticles in suspension are changed and induced into a hysteresis effect. Several correlations were also proposed in the literature for the viscosity of nanofluids. The Einstein formula and the expressions so derived, originated from the classical theory of linear fluids, are useful but limited to small volumetric concentrations of nanoparticles.

With respect to convective heat transfer in nanofluids, a thorough review of the available work is provided in (Kakaç and Pramuanjaroenkij, 2009), related to the heat transfer enhancement of internal forced convection with the dispersion of metallic and metal oxide nanoparticles in common liquids. Xuan and Li (2003) measured the convective heat transfer coefficient of Cu-water nanofluid, and obtained a significant increase with respect to the base fluid. For a given Reynolds number, the nanofluid heat transfer coefficient containing 2% volumetric concentration of Cu nanoparticles proved to be 60% higher than that of pure water. A few other studies, such as in Heris et al. (2006), Nguyen et al. (2007), Rea et al. (2009), also confirm the better thermal performance presented by various nanofluids in forced convection, in different configurations of internal flow.

The Laboratory of Heat Transmission and Technology, LTTC, Mechanical Engineering, POLI & COPPE / UFRJ, started the research on nanofluids around 2004 with support from CENPES / Petrobras and, in collaboration with the Materials Division of INMETRO, nanofluids were synthesized and characterized (Cotta et al., 2007b; Fonseca et al., 2009). This research also included the first efforts on numerically and experimentally analyzing the thermal behavior of the available alumina-water nanofluids in laminar forced convection within circular tubes (Cotta et al., 2009; Cerqueira et al., 2010). The nanofluid then analyzed was made by dispersing different concentrations of aluminum oxide (Al_2O_3) in water, with an appropriate dispersant, while the nanoparticles were purchased from Nanostructured and Amorphous Materials, USA. The simulations were performed using a hybrid numerical-analytical technique known as the Generalized Integral Transform - GITT (Cotta, 1993; Cotta, 1998; Cotta and Mikhailov, 2006), implemented in the symbolic computation platform *Mathematica* 7.0 (Wolfram, 2008), for solving the nonlinear partial differential equations that govern the laminar forced convection of nanofluids within circular tubes (Cotta et al., 2007a). A thermohydraulic circuit was built and tested to obtain experimental results in forced convection that were analyzed and discussed in (Cerqueira et al., 2010), as well as verified against the proposed model with temperature dependent thermophysical properties (Cotta et al., 2007a).

The present work is an extension of the research presented in (Cerqueira et al., 2010), including modifications in both the experimental setup and in the proposed theoretical model. In the theoretical model, emphasis has been placed in taking simultaneously into account the heat losses along the test section and the temperature variation of all the thermophysical properties. Critical comparisons with well established correlations (Churchill and Ozoe, 1973; Shah, 1975) for regular fluids are also undertaken. In addition, in order to evaluate the performance of commercial nanofluids, which recently have been made available for purchase, a water-alumina (alpha) nanofluid was acquired from Nanostructured and Amorphous Materials, USA, with a 20%wt concentration, and employed throughout the experiments.

2. THEORETICAL MODEL AND CORRELATIONS

The proposed mathematical formulation of the problem has the objective of modeling a forced convection experiment in a circular tube to evaluate the heat transfer enhancement due to the use of a nanofluid in laminar flow. It is then considered forced convection heat transfer inside a circular tube of a Newtonian fluid with temperature dependent thermophysical properties, in light of the marked variations with temperature that have been encountered in the literature. The wall of the tube is heated by an electrical resistance along its length with uniform heat flux, while the heat losses to the external environment are accounted for via an effective heat transfer coefficient that includes conduction across the insulation and the natural convection losses at the external surface. The flow is assumed to be fully developed at the test section entrance, after flowing isothermally through a sufficiently long duct of the same diameter, and the effects of viscous dissipation and axial heat diffusion are neglected. The corresponding energy balance equation is written as:

$$\rho(T)c_p(T)\left[u(r,z,T)\frac{\partial T(r,z)}{\partial z}+v(r,z,T)\frac{\partial T(r,z)}{\partial r}\right]=\frac{1}{r}\frac{\partial}{\partial r}\left[rk(T)\frac{\partial T(r,z)}{\partial r}\right],\ 0<r<r_w,\ 0<z<L \tag{1.a}$$

with the following boundary conditions

$$\frac{\partial T(r,z)}{\partial r}=0,\ \ r=0 \tag{1.b}$$

$$-k(T)\frac{\partial T(r,z)}{\partial r}=-q_w+h_{ef}\left(T(r,z)-T_\infty\right),\ \ r=r_w,\ z>0 \tag{1.c}$$

and the fluid inlet condition

$$T(r,0)=T_0,\quad 0\le r\le r_w \tag{1.d}$$

The longitudinal velocity component is obtained from the simplified formulation for fully developed flow and determined by direct integration of the z-momentum equation with variable viscosity (Yang, 1962; Oliveira Filho et al., 2001):

$$\frac{1}{r}\frac{\partial}{\partial r}\left[r\mu(T)\frac{\partial u(r,z)}{\partial r}\right]=\frac{dp(z)}{dz}, 0<r<r_w, z>0 \tag{2.a}$$

$$\frac{\partial u(r,z)}{\partial r}=0, r=0\,; \quad u(r,z)=0, r=r_w, z>0 \tag{2.b, c}$$

The solution methodology adopted in solving the temperature problem, Eqs.(1), is based on the Generalized Integral Transform Technique (GITT) and borrowed from the approach proposed in (Cotta et al., 2007a), here including heat losses to the external environment. Due to space limitations, this approach is not detailed here, but can be found in (Cerqueira et al., 2010).

The literature provides a few different correlations, either theoretical or empirical, to predict the Nusselt number for laminar flow as a function of dimensionless axial distance, in the dimensionless form of Graetz number (Gz) (Kakaç, 1987). However, the above reviewed work on forced convection with nanofluids reported heat transfer rates augmentation that do not seem to be explainable only by changing the effective thermophysical properties in comparison with the base fluid, and thus might not be predictable by the classical correlations for heat transfer coefficients in laminar flow. Therefore, we have recalled two correlations for laminar forced convection that are well accepted in the literature, the first one a theoretical correlation based on the analytical solution of the so called Graetz problem for prescribed heat flux (Shah, 1975), and the correlation of experimental data performed in (Churchill and Ozoe, 1973), which also provides a correction on the heat transfer coefficient for the variation of viscosity with temperature. The two expressions, respectively for local and average Nusselt numbers, respectively, are given by:

– Churchill & Ozoe (1973)

$$\frac{Nu_l+1.7}{5.357[1+(Gz/97)^{8/9}]^{3/8}}=\left[1+\left(\frac{Gz/71}{[1+(\mathrm{Pr}/0.0468)^{2/3}]^{1/2}[1+(Gz/97)^{8/9}]^{3/4}}\right)^{4/3}\right]^{3/8} \tag{3.a}$$

– Shah (1975)

$$Nu_m=\begin{cases}\dfrac{1.953}{(\chi)^{1/3}}, & \chi\le 0.03\\[2mm] 4.354+\dfrac{0.0722}{\chi}, & \chi>0.03\end{cases} \tag{3.b}$$

where,

$$\chi=\frac{1}{Gz}=\frac{x/D_h}{\mathrm{Re}\,\mathrm{Pr}} \tag{3.c}$$

Eq.(3.a) may also be numerically integrated to provide the average Nusselt number as a function of the dimensionless axial position (inverse of the Graetz number).

3. EXPERIMENTAL SETUP AND PROCEDURE

Results were obtained for a nanofluid made of aluminum oxide nanoparticles (Al_2O_3-α, APS 30±10 nm, purity 99.9%) dispersed in pure water at 20%wt and pH of 6.3, purchased from Nanostructured & Amorphous Materials, Inc, USA (US$87. per kg).

The thermohydraulic circuit is divided into five parts: Heater system, Test section, Hydraulic circuit, Heat rejection system and Data acquisition. The heater system consists of an electrical resistance in the form of a metallic tape that was installed over the surface of an electrically insulated copper tube treated with electrostatic painting, to allow for an uniform heat flux along the pipe wall. The pipe is thermally insulated over its length after being covered with Kapton tape, as illustrated in Figure 1 below.

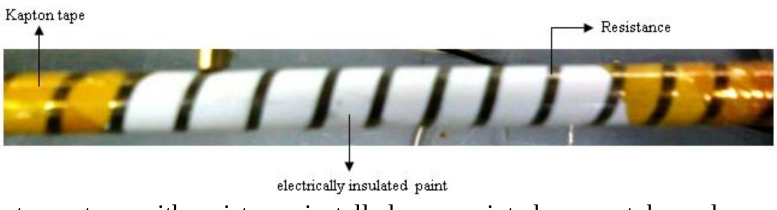


Figure 1. Detail of heater system, with resistance installed over painted copper tube and covered with Kapton tape.

The heater system also comprises a Variac which gives a variable output AC voltage from 0 to 300VAC. Upon leaving the Variac, the alternating current is rectified, filtered and applied to the resistance tape wound over the copper tube. The DC supply has eliminated small electromagnetic induction noise caused in the reading of the thermocouples.

This setup allows applying to the tube a heating power adjustable from 0 to 1500W. The circuit has a 20 A circuit breaker to protect it and also serves as a key drive. The reading of electrical parameters is performed by a digital ampmeter (A) which measures the current within the heater, and a digital voltmeter (V) which measures the supplied voltage.

The test section consists of a copper tube with outer diameter of 3/8 "and wall thickness of 1/16." The copper tube is connected by C-PVC unions to provide flexibility to the test section, allowing for quick exchange if necessary. The wall temperature measurements along the tube are provided by type K (Chromel-Alumel) thermocouples placed on the external copper wall. To measure the temperature within the fluid, type E (chromel-constantan) thermocouples were placed inside mixing connections at the inlet and outlet of the test section. For the thermal insulation of the test section we have chosen an anti-fire foam made of expanded low density polyethylene, in a double layer.

The hydraulic circuit is composed of the fluid storage tank, the pump, hoses, valves for flow control, the return pipe and a solenoid valve for mass flow rate measurements. The peristaltic pump is made by PROVITEC model AWG 5000-A. This pump type prevents the contamination of the fluid, employing a non-toxic hose and avoiding contact with the mechanical seals and other pump parts. For mass flow rate measurements we have used a precision digital scale for low flow rates, where readings of mass versus time are automatically acquired through the RS232 output of the precision scale. For higher mass flow rates a turbine type flow sensor is installed at the exit of the test section.

The heat rejection system consists of a shell and coil heat exchanger, made of about twenty coiled sections of 18 cm in diameter of a 3/8" copper tube, subjected to a shell side water stream at room temperature. The acquisition system was based on the data logger manufactured by Agilent, model 34070A. Illustrative photos of the thermohydraulic circuit are shown in Figures 2, while the experimental apparatus is schematically described in Figure 3.

(a) (b)

Figures 2. a)Overview of the thermo-hydraulic circuit for experiments on forced convection of nanofluids (LTTC, COPPE / UFRJ); b) Detailed view of the flow control, pump and mass flow rate acquisition.

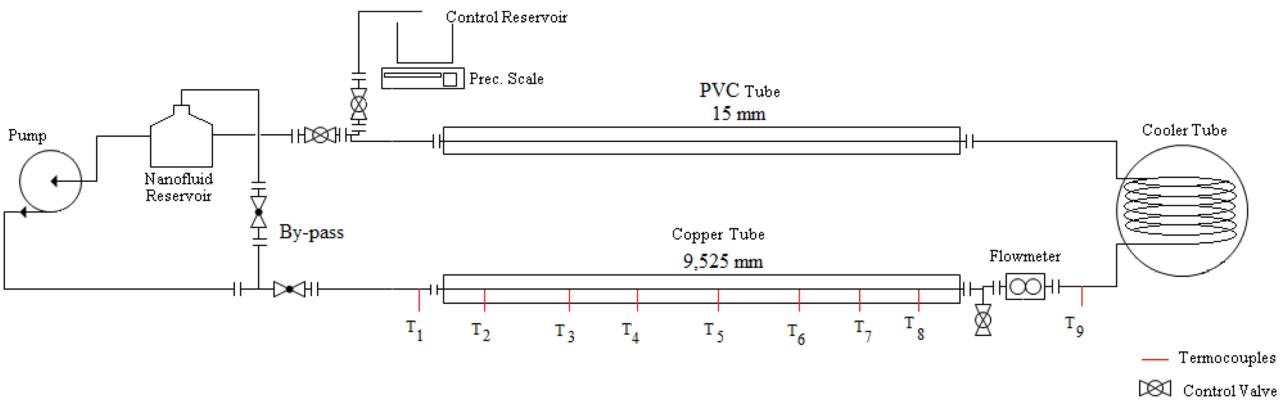


Figure 3. Schematic representation of the experimental apparatus for analysis of forced convection of nanofluids.

The implemented experimental procedure allowed for steady measurements repeatability and reducing errors with respect to the previous setup. In synthesis, we first check the fluid level in the reservoir to power on the pump, adjusting its rotation. The data acquisition system is connected and the control valve for the heat exchanger cooling water is adjusted. Only then the heater system is turned on. When the system achieves its steady state, we power on and tare the precision scale, measure the voltage and current in the resistance, and measure the mass flow rate. At the end of each test the system is cooled back again, after the shutdown of the heating power. In the case of exchange of fluid, we use the purge valve located before the entry to the coil and shell heat exchanger, the lowest point of the thermal circuit.

In the results here reported we have utilized 3 liters of the acquired alumina-water nanofluid in the circuit, with a nominal volumetric concentration of 5.03% of aluminum oxide nanoparticles. In the technical report of its manufacturer there is no indication of the dispersant used, time and dispersion method or even date of manufacturing. Unfortunately, due to the high concentration of nanoparticles and certainly the not fully adequate dispersion method or dispersant employed, a significant sedimentation of nanoparticles in the thermohydraulic circuit could be observed after each test batch was concluded, which can be illustrated in detail in Figure 4 below, after we have installed a transparent tubing to allow for the flow visualization. This behavior has alerted us to measure concentrations after each set of runs, which would then be more representative of the actual concentrations during the tests. In fact, the measured values of 2,33% and 3,35% in volume concentration are more representative of the actual flowing fluid conditions in each batch of experimental runs.

Figure 4. Sedimentation of alumina nanoparticles along the acrylic tube placed after the test section for flow visualization.

To calculate the effective properties of the nanofluid we have used the following expressions for the density and specific heat as obtained from the conventional mixtures theory:

$$\rho_{nf} = (1-\phi)\rho_{fb} + \phi\rho_p \tag{4.a}$$

$$c_{p,nf} = [(1-\phi)\rho_{fb}c_{p,fb} + \phi\rho_p c_{p,p}]/\rho_{nf} \tag{4.b}$$

For the thermal conductivity, we have adopted the Hamilton and Crosser (1962) correlation, with the empirical form factor defined as n=3 for nanoparticles with spherical shape, since it has provided the best agreement with experimental results obtained in our previous work, using both the Hot wire and Flash techniques (Fonseca et al., 2009):

$$k_{nf} = k_{fb}(k_p + (n-1)k_{fb} - (n-1)\varphi(k_{fb} - k_p)/(k_p + (n-1)k_{fb} - \varphi(k_{fb} - k_p)) \tag{4.c}$$

As for the viscosity, we have used the correlation in Nguyen et al. (2007), which is also in good agreement with experimental results previously obtained (Cotta et al., 2007):

$$\mu_{nf} = \mu_{fb}(123\varphi^2 + 7.3\varphi + 1) \tag{4.d}$$

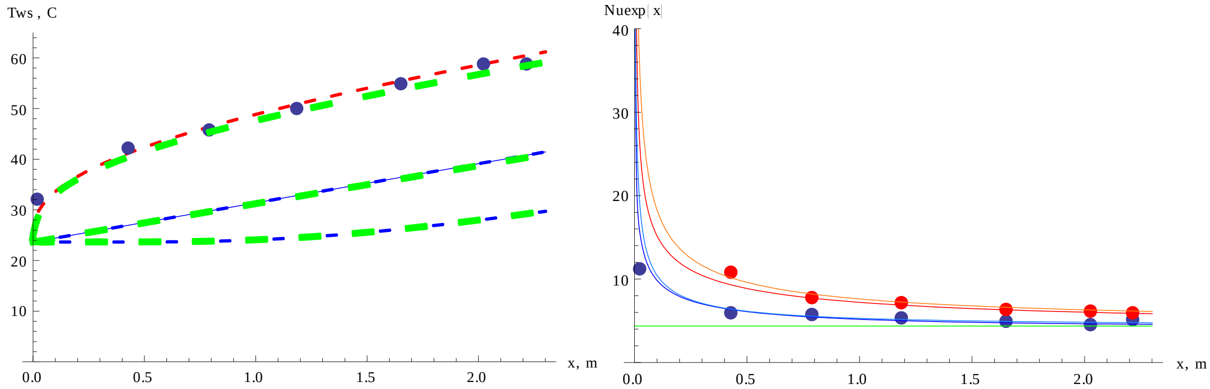
The temperature dependence of the thermophysical properties was considered to closely follow that of the base fluid (pure water) in the range of temperatures achieved in these experiments.

4. RESULTS & DISCUSSION

Before undertaking the nanofluid forced convection experiments, several runs with pure water were performed to verify and validate the experimental setup and procedure in laminar flow. Figures 5.a, b illustrate this validation effort for a Reynolds number equal to 1531. In Fig.5.a the temperatures measured along the tube wall (blue dots) are compared with the solution of the linear model (dashed red line for the wall temperature and blue lines for the bulk and centerline temperatures) and the nonlinear model with temperature dependent thermophysical properties (dashed green lines for wall, bulk and centerline temperatures). In Fig.5.b, the local (blue dots) and average (red dots) Nusselt numbers determined from the measured temperatures are compared with theoretical predictions for the local Nusselt (blue line) and average Nusselt (red line) numbers. The dots in each case are the experimental results and show good agreement with theoretical predictions, with a slightly better adherence of the nonlinear model results.

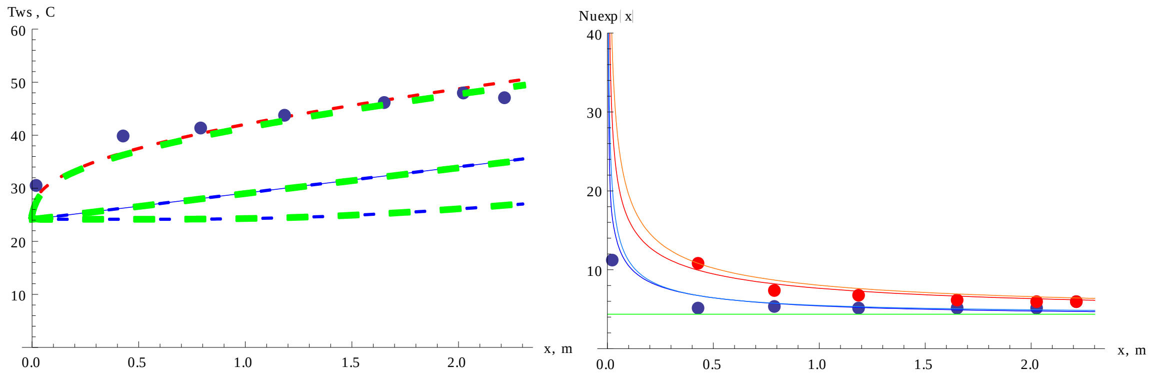
As previously discussed, due to the high concentration of nanoparticles, there was a significant sedimentation of nanoparticles in the thermohydraulic circuit along the experimental runs with the nanofluid, which were organized in two independent sets of measurements. Figures 6 and 7 illustrate the comparison between the theoretical and experimental results for each sequence of runs using the nanofluid, respectively, with the volumetric concentrations of

3.35% at Re=1518 and 2.33% at Re=1891. The temperatures measured on the tube wall by the thermocouples were directly compared with the GITT solution of the linear model (dashed red line) and of the nonlinear model (dashed green line). In addition, the Nusselt numbers were calculated from the measured temperatures and compared with their theoretical predictions for the local (blue line) and average values (red line), again for both models. The dots in each case represent the experimental results, which have good agreement with theoretical predictions in the range of Reynolds numbers examined.



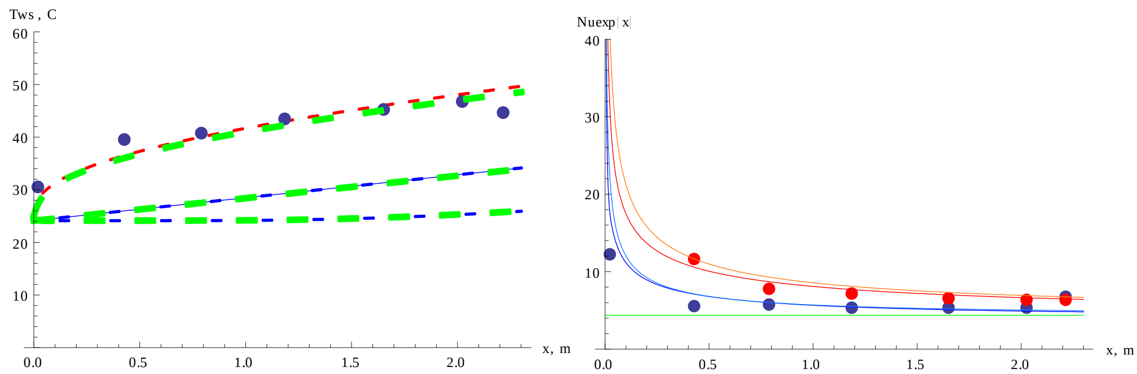
(a) (b)

Figures 5.a,b. Experimental and theoretical results (linear and nonlinear models) for (a) wall, bulk and centerline temperatures and (b) local and average Nusselt numbers, for pure water in laminar flow with Re = 1531.



(a) (b)

Figures 6.a,b. Experimental and theoretical results (linear and nonlinear models) for (a) wall, bulk and centerline temperatures and (b) local and average Nusselt numbers, for the nanofluid with 3.35% volumetric concentration in laminar flow with Re = 1518.



(a) (b)

Figures 7.a,b. Experimental and theoretical results (linear and nonlinear models) for (a) wall, bulk and centerline temperatures and (b) local and average Nusselt numbers, for the nanofluid with 2.33% volumetric concentration in laminar flow with Re = 1891.

Tables 1 and 2 show a direct comparison between the experimental results for the average heat transfer coefficient of pure water against the alumina nanofluid, respectively with 3.35% and 2.33% volume concentrations. The achieved heat transfer enhancement is within 3 to 11% with respect to the base fluid case, which approximately falls within the expected behavior due to the thermal conductivity augmentation in these situations. It should be noted that the results for the nanofluid with lower concentration are apparently demonstrating a higher heat transfer enhancement, but in fact, due to the difficulties in achieving exactly the same values of Reynolds number in each experimental run, the values are slightly more different in this case (Re=1891 for the nanofluid and Re=1857 for pure water) than in Table 1. It should be mentioned that the propagation of uncertainties in the experimental determination of the local Nusselt numbers have resulted in an uncertainty of up to 7.5% in such cases. Also, the tabulated results below are within the same order of enhancement, but slightly worse, than those achieved in our previous research dealing with the water-alumina nanofluid manufactured at INMETRO (Cotta et al., 2007b, Cerqueira et al., 2010).

Table 1. Comparison of the experimental results for the average heat transfer coefficient of pure water and of the nanofluid with 3.35% volumetric concentration.

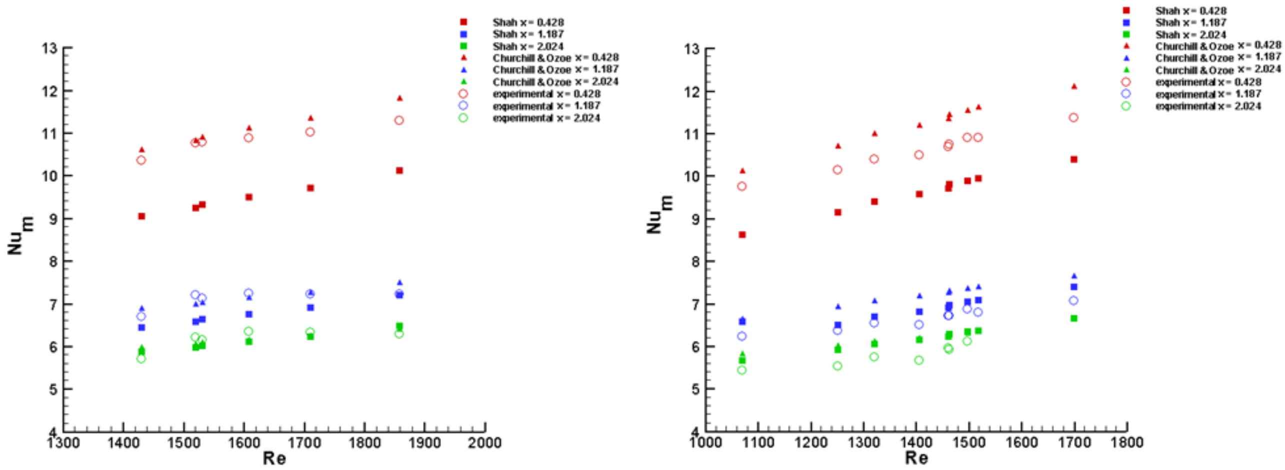
z (m)	h_m experimental (W/m²°C)		Deviation (%)
	Nanofluid, Re = 1518	Water, Re = 1519	
0.428	1152.2	1037.6	11.0
0.787	782.2	757.1	3.3
1.187	722.3	699.6	3.2
1.651	658.8	637.2	3.4
2.024	640.4	607.8	5.4

Table 2. Comparison of the experimental results for the average heat transfer coefficient of pure water and of the nanofluid with 2.33% volumetric concentration.

z (m)	h_m experimental (W/m²°C)		Deviation (%)
	Nanofluid. Re = 1891	Water. Re = 1857	
0.428	1194.4	1085.7	10.0
0.787	810.1	757.4	7.0
1.187	744.8	700.4	6.3
1.651	675.3	634.5	6.4
2.024	657.5	613.0	7.3

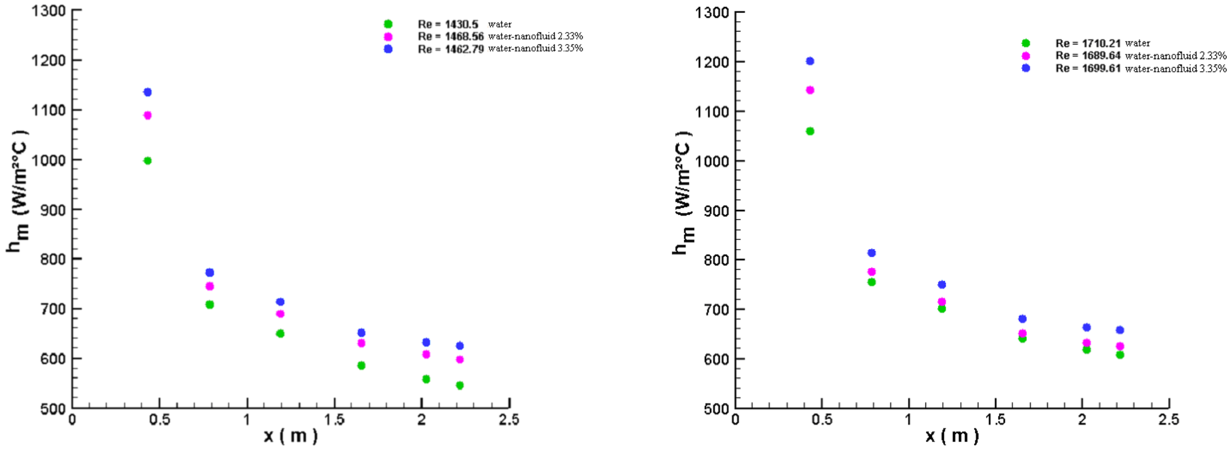
Figures 8 a,b show a comparison between the experimental results for the average Nusselt number of pure water and for the water-alumina nanofluid with 3.35% volumetric concentration. respectively. against the predictions of the two adopted correlations. for various values of the Reynolds number and at three different positions along the tube (x = 0.428; 1.187 and 2.024 m). Clearly. the experimental results for pure water are in excellent agreement with the Churchill & Ozoe correlation throughout the ranges of axial positions and Reynolds numbers considered. while a more significant deviation from the Shah correlation is observed at the position closer to the channel inlet. The nanofluid experimental results are also reasonably predicted by the correlations. especially for regions not so close to the channel inlet, where the agreement with the Churchill & Ozoe correlation is still better than with respect to the theoretical correlation by Shah.

Figures 9 a,b provide a direct graphical comparison of the average heat transfer coefficient along the channel for the three fluids, pure water, nanofluid with 3.35% and 2.33% concentrations, at the closest available values of Reynolds number from each set of experimental runs, a)Re=1430, 1469 and 1463 and b)Re=1710, 1690, and 1700, respectively. The overall trend confirms the general observation found in the literature, that the increase on particles concentration leads to a heat transfer enhancement effect, at least in the range here considered. The results for the lower Reynolds number seem to present a more significant heat transfer enhancement with respect to the other case, but it should be noted that the values of Re for the two nanofluids are again slightly more apart from the value for water in this case. Nevertheless, we have not been able to observe any sort of anomalous behavior on the heat transfer coefficients that could not be predicted by the effective thermophysical properties and classical theoretical models and correlations.



(a) pure water (b) nanofluid (3.35% volumetric concentration)

Figures 8.a.b. Experimental results of average Nusselt numbers for laminar flow of pure water and nanofluid compared with the Churchill & Ozoe and Shah correlations at three positions in the test section (x = 0.428; 1.187 and 2.024 m).



(a) (b)

Figures 9. Comparison of the average heat transfer coefficient along the channel for the three fluids, pure water, nanofluid with 3.35% and 2.33% concentrations, at the closest available values of Reynolds number from each set of experimental runs, a)Re=1430, 1469 and 1463; b)Re=1710, 1690, and 1700.

5. CONCLUSION

Heat transfer enhancement in laminar forced convection of water-alumina nanofluids within circular tubes subjected to a prescribed wall heat flux has been investigated, both experimentally and theoretically. The experimental runs have span a range of Reynolds number from approximately 1000 to 1900. The wall channel was heated by an electrical resistance that allowed for wall temperature variations from 22 ºC up to around 60ºC at the exit of the channel. A commercial nanofluid has been acquired, with a nominal volumetric concentration of nanoparticles around 5%, but due to an inadequate dispersion method and/or dispersant employed, significant sedimentation of the nanoparticles could be observed after shutdown of the circuit. Therefore, losses of particles in the filling, flow measurement and regular operation of the circuit have led to sensible reductions of the nanoparticles concentration during each batch of experimental runs. For this reason, the concentrations were experimentally determined at the completion of each set of runs, yielding the values of 3.35% and 2.33% for each one. It has been concluded that a heat transfer enhancement effect is observable, in the range of 3% to 11%, depending on the concentration, axial position and Reynolds number. In general, the enhancement is more noticeable for the larger concentration, larger values of Re and at regions closer to the inlet. The enhancement was found to be less pronounced than with a previous alumina-water nanofluid previously tested, manufactured at INMETRO with a lower concentration and with the use of an appropriate dispersant, when deposits of nanoparticles in the circuit were much less pronounced. Therefore, the stability of the nanofluid is a major aspect in the successful use of the nanoparticles as a heat transfer enhancement agent.

It has also been observed that a classical correlation of experimental results in laminar forced convection (Churchill & Ozoe, 1973) can fairly well predict the behavior of the tested nanofluids, once effective thermophysical properties are employed according to available expressions in the literature. Therefore, at least in this range of concentrations and Reynolds numbers, we have found it unnecessary to develop specific correlations for the water-alumina nanofluid that employ the concentration as an adjustment factor. In addition, both the linear and nonlinear models of laminar forced convection with prescribed wall heat flux and heat losses were able to reliably reproduce the wall temperature distributions and Nusselt numbers experimentally obtained. At least in the present range of wall and fluid temperatures, the variation of thermophysical properties with temperature provides only a slight deviation in the theoretical results for the heat transfer coefficients, though yielding a better agreement with the experimental results. Future work should be directed to exploring higher Reynolds numbers (turbulent flow) and higher temperature differences (pressurized systems), in order to clarify in more definitive terms the achievable heat transfer enhancement with this class of nanofluids.

6. ACKNOWLEDGEMENTS

The authors would like to acknowledge the financial support provided by CNPq and CENPES/Petrobras, both in Brazil.

7. REFERENCES

Cerqueira. I. G.. Mota. C. A. A.. Nunes. J. S.. Cotta. R. M.. 2010. “Experimental-Theoretical Analysis of Laminar Internal Forced Convection With Nanofluids”. Brazil. 13rd Brazilian Congress of Thermal Sciences and Engineering - ENCIT 2010.

Choi. S. U. S.. 1998. “Nanofluid technology: Current Status and Future Research”. Korea-U.S. Technical Conference on Strategic Technologies. Vienna. VA. USA.

Choi. S. U. S. and Eastman. J.A.. 2001. U.S. Patent No. 6.221.275 B1.” Enhanced Heat Transfer Using Nanofluids”. Apr. 24. 2001.

Churchill. S.W.. Ozoe. H.. 1973. “Correlations for Laminar Forced Convection in Flow over an Isothermal Flat Plate and in Developing and Fully Developed Flow in an Isothermal Tube”. ASME J. Heat Transfer**.** Vol. 95 pp.416-23.

Cotta. R.M.. 1993.” Integral Transforms in Computational Heat and Fluid Flow”. CRC Press. USA.

Cotta. R.M.. 1998.” The Integral Transform Method in Thermal and Fluids Sciences and Engineering”. Begell House. New York

Cotta. R.M.. Mikhailov. M.D.. 2006.” Hybrid Methods and Symbolic Computations”. in: W.J. Minkowycz. E.M. Sparrow. and J.Y. Murthy (Eds.). Handbook of Numerical Heat Transfer. second ed.. Wiley. NY. pp. 493-522.

Cotta. R. M.. Santos. C.A.C.. and Kakaç. S.. 2007a. "Unified Hybrid Theoretical Analysis of Nonlinear Convective Heat Transfer”. Proceedings of IMECE2007. ASME. Paper no. IMECE2007-41412. Nov.11-15.

Cotta. R.M.. Orlande. H.R.B.. Couto. P.. Balbo. A.. Mota. C.A.A.. Massard. H.. Naveira. C.P.. Pereira. A.G.F.. and de Sousa G.P.. 2007b.” Nanofluids: Fabrication. Caracterization. Thermophysical Properties. and Forced Convection”. Final Project Report. COPPETEC no. PEM7393. under contract for Petrobras Research Center. Rio de Janeiro. Brasil. July 2007.

Cotta. R.M.. C.A.A. Mota. C.P. Naveira-Cotta. J.S. Nunes. M.R. Avelino. F.V. Castellões. and J.N.N. Quaresma. 2009. “Heat Transfer Enhancement in Laminar Forced Convection: Nanofluids. Microchannels. Structured Surfaces”. Invited Keynote Lecture. ICCHMT International Symposium on Convective Heat and Mass Transfer in Sustainable Energy. CONV-09. Yasmine Hammamet. Tunisia. April.

Das. S. K.. Choi. S. U. S.. Patel. H.. 2006. “Heat Transfer in Nanofluids – A Review. Heat Transfer Engineering”. Vol. 27. No. 10. pp. 3-19.

Eastman. J. A.. Phillpot. S. R.. Choi. S. U. S. and Keblinski. P.. 2004.” Thermal Transport in Nanofluids”. Annu. Rev. Mater. Res.. Vol. 34. pp. 219-246.

Fonseca. H.M.. Orlande. H.R.B.. Tavman. I.. and Cotta. R.M.. 2009. “Measurements of Nanofluids Physical Properties. High Temperatures-High Pressures. Int. J. of Thermophysical Properties Research”. V.38. no.2. pp.187-197.

Heris. S.Z.. Esfahany. M.N.. & Etemad. G.. 2006. “Investigation of CuO/Water Nanofluid Laminar Convective Heat Transfer through a Circular Tube”. J. Enhanced Heat Transfer. Vol.13. no.4. pp.279-289.

Kakac. S.. 1987. “The Effect of Temperature-dependent Fluid Properties on Convective Heat Transfer”. in: Handbook of Single-Phase Convective Heat Transfer**.** S. Kakac¸. R.K. Shah. W. Aung (Eds.). Wiley. New York. Chapter 18.

Kakaç. S. and Pramuanjaroenkij. A.. 2009. “Review of convective heat transfer enhancement with nanofluids”. Int. Journal of Heat and Mass Transfer. v.52. pp. 3187–3196.

Keblinski. P.. Eastman. J. A.. Cahill. D. G.. 2005. ”Nanofluids for Thermal Transport”. Materials Today. June. pp. 36-44.

Mintsa. H.A.. Roy. G.. Nguyen. C.G.. Doucet. D.. 2009. "New Temperature Dependent Thermal Conductivity Data for Water-Based Nanofluids". Int. J. Thermal Sciences. Vol.48. pp.363-371.

Nguyen C.T.. Desgranges F.. Galanis N.. Roy. Maré T.. Boucher S.. Palm. S.J.. (2004). “Heat Transfer Enhancement in a Radial Flow Cooling System Using Nanofluids”. In: Proceedings of the CHT-04 ICHMT. International Symposium on Advances Computational Heat Transfer. Paper Nº CHT-04-121.18p.

Nguyen. C.T.. Roy. G.. Gauthier. C.. Galanis. N.. 2007. "Heat Transfer Enhancement Using Al2O3–Water Nanofluid for an Electronic Liquid Cooling System". Applied Thermal Engineering. Vol. 27. pp. 1501–1506.

Oliveira Filho. F.S.. Guedes. R.O.C.. Scofano Neto. F.. 2001.” Temperature-Dependent Viscosity Flow”. In: Convective Heat Transfer in Ducts: The Integral Transform Approach. C.A.C. Santos. J.N.N. Quaresma. and J.A. Lima (Eds.). Editora E-Papers. Rio de Janeiro. Brasil.

Rea. U.. McKrell. T.. Hu. L.W.. and Buongiorno. J.. 2009. “ Laminar Convective Heat Transfer and Viscous Pressure Loss of Alumina-Water and Zirconia-Water Nanofluids”. Int. J. Heat & Mass Transfer Vol. 52. Issues 7-8. Pages 2042-2048.

Shah. R. K.. 1975. “Thermal Entry Solutions for the Circular Tube and Parallel Plates”. Proc. of the 3rd National Heat and Mass Transfer Conf.. Vol.1. Indian Institute of Technology. paper no. HMT-11-75. Bombay. India.

Wolfram. S.. 2008. “The Mathematica Book”. version 7.0. Cambridge-Wolfram Media.

Xuan. Y.. Li. Q.. 2003. "Investigation on Convective Heat Transfer and Flow Features of Nanofluids". Journal of Heat Transfer. Vol. 125. pp. 151–155.

Yang. K. T.. 1962. “Laminar Forced Convection of Liquids in Tubes with Variable Viscosity”. ASME J. Heat Transfer. Vol. 84. pp.353-362.

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.