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Analyzing energetic performance of a tube-in-tube heat exchanger using five different nanofluids

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Abstract: In this study, we analyze the effect of the use of 5 nanofluids on energy performance of a tube-in-tube heat exchanger. After defining a Performance Energy Criterion (PEC) as the ratio of heat flow rate over pumping power, a theoretical analysis is carried out and it is compared to experimental results with SiO_2/water, ZnO/water, and a commercial nanofluid. The thermophysical properties (thermal conductivity, dynamic viscosity) used in calculations as well as in experiments are measured values. We have found a fair agreement between theoretical and experimental values and we have given clues to choose nanofluids in order to improve the PEC.

Keywords: Nanofluid, SiO_2 nanoparticles, ZnO nanoparticles, heat transfer, pressure drop, performance energy criterion.

Nomenclature

Latin letters

B	constant	
C_k, C_μ	constants in formulae 15 & 16	
C_p	specific heat capacity	J kg^{-1} K^{-1}
D	tube diameter	m
k	thermal conductivity	W m^{-1} K^{-1}
G	mass velocity	kg.s^{-1}.m^{-2}
h	convective heat transfer coefficient	W m^{-2} K^{-1}
k	thermal conductivity	W m^{-1} K^{-1}
L	tube length	m
$\dot{m}$	mass flow rate	kg s^{-1}
Nu	Nusselt number	
p	Pressure	Pa
PEC	Performance Energy Criterion	
P	Power	W
Pr	Prandtl number	
Q_v	volume flow rate	m^3.s^{-1}
Re	Reynolds number	
T	temperature	K, °C
V	mean velocity	m.s^{-1}

Greek symbols

β	shape factor	
δ	ratio of PECs	
δ_H	enhancement factor of heat transfer coefficient	
Δp	pressure drop	Pa

φ	nanoparticle volume fraction	
φ_w	nanoparticle mass fraction	
Λ	Darcy coefficient	
μ	dynamic viscosity	Pa s
ξ	friction factor coefficient	
ρ	density	kg m^{-3}
ψ	sphericity	

Subscripts

exp	experimental
f	base fluid
i	internal
in	inlet
nf	nanofluid
out	outlet
s	nanoparticle
t	thermocouple
p	wall
R	reference

1. Introduction

Improving energy efficiency in reducing energy consumption has become a major challenge in all activity sectors. In this objective, different methods have been proposed to decrease the size of heat exchangers or to increase their transferred thermal power [1-3]. However, when the available space is limited by the process, such as for example, in electronic cooling or automotive applications, it is interesting to use heat exchangers of the same size (or smaller) with better performance. This can be achieved by modifying the heat exchanger geometry and the wall surfaces by means of inserts, corrugations, coating or specific devices. Another way is, together with the preceding methods, to use fluids with enhanced thermal conductivity in order to enhance the heat transfer coefficient when compared with that of classical fluids for the same geometry. Nanofluids are colloidal suspensions of nanoparticles which are engineered to have thermal conductivity higher than that of the base fluid and which can be used in this purpose [4, 5]. However, together with thermal conductivity enhancement, viscosity generally is increased and the gain in transferred heat is paid in terms of pumping power. There is a competition between heat transfer rate and pumper power.

Many research works have been carried out to define criteria of interest to evaluate potential improvement of heat exchangers or to compare different heat exchangers for the same thermal load. In this article, we remind different criteria which have already been used. With the chosen criterion, we calculate the performance of a tube-in-tube heat exchanger with different nanofluids and base fluids (water). Theoretical results are compared with experimental ones.

2. Nanofluid properties

Many experimental and theoretical studies have been carried out to determine the physical properties of nanofluids. The main theoretical results are reminded.

2.1 Density

The density of the nanofluid is evaluated according to the standard formula:

$$\rho = (1-\varphi)\rho_f + \varphi\,\rho_s \qquad (1)$$

where, φ is the volume fraction of the nanofluid.

ρ_f is the density of the base fluid

ρ_s is the density of the nanoparticles.

2.2 Specific heat

The formula of the specific heat for a mixture is given by,

$$C_p = (1-\varphi_w) C_{p_f} + \varphi_w C_{p_s} \tag{2}$$

φ_w is the mass fraction of the nanofluid.

C_{p_f} is the specific heat capacity of the base fluid

C_{p_s} is the specific heat capacity of the nanoparticles

2.3 Thermal conductivity

Concerning thermal conductivity many controversial results have been published. A benchmark study with 34 laboratories has been conducted in the frame of the International Nanofluid Property Benchmark Exercise (INPBE) [6]. The conclusion was that the experimental results could be accurately reproduced by an extended theory of the classic effective medium theory [7]. In the pioneering work of Maxwell [8] the effective thermal conductivity k for a mixture with spherical particles is given by

$$k = k_f \frac{k_s + 2k_f - 2(k_f - k_s)\varphi}{k_s + 2k_f + (k_f - k_s)\varphi} \tag{3}$$

φ is the volume fraction of the nanofluid.

k_f is the thermal conductivity of the base fluid

k_s is the thermal conductivity of the nanoparticles

This model has been extended by Hamilton and Crosser [9] to non-spherical particles by introducing a shape factor β given by β = 3 / ψ, where ψ is the particle sphericity, defined as the ratio of the surface area of a sphere with the same volume as that of the particle and the surface area of the particle. The effective conductivity is expressed as follows:

$$k = k_f \frac{k_s + (\beta-1)k_f - (\beta-1)(k_f - k_s)\varphi}{k_s + (\beta-1)k_f + (k_f - k_s)\varphi} \tag{4}$$

φ is the volume fraction of the nanofluid.

k_f is the thermal conductivity of the base fluid

k_s is the thermal conductivity of the nanoparticles

The Maxwell formula corresponds to sphericity equals one.

This model has been completed by Nan et al. [7] which is valid for ellipsoidal nanoparticles. It takes into account an interfacial resistance between the particle and the surrounding medium. In this work, the thermal conductivity of every studied nanofluid has been measured to avoid any controversy.

2.4 Dynamic viscosity

The addition of solid particles to a liquid can alter its rheological behaviour. The limiting case for dilute suspensions of small, rigid, spherical particles was treated by Einstein (1906) [10] and extended to ellipsoidal particles. The viscosity is given by:

$$\mu_{nf} = \mu_f (1 + B\varphi) \tag{5}$$

where *B* depends on the ratio of the revolution ellipsoid axes and is equal to 2.5 for spherical particles. However, in most published results and, in particular, in a benchmark carried out by the INPBE, it has been found that the dependence of viscosity on particle volume fraction was significantly stronger than predicted for dilute suspensions [11]. Many scattered results having been published, for each nanofluid considered in this work, the dynamic viscosity was measured.

3. Performance criteria

It is difficult to give a general criterion to characterize the performance of a heat exchanger using different fluids. Many types of criteria which take into account both heat transfer enhancement and pumped power have been suggested, generally for evaluating enhancement due to specific surfaces. To characterize the performance of the nanofluid use in a specific device (as for the rating problem of a heat exchanger), there are several ways and some of them are reminded hereafter.

3.1 Heat transfer enhancement

The first one is to only consider the heat transfer performance in comparing heat transfer coefficient h or the Nusselt number Nu with a reference h_R and Nu_R respectively, for the same geometry and experimental conditions. So, we define an enhancement factor δ_H as

$$\delta_H = h/h_R \tag{6}$$

In the case of nanofluids, these reference values generally are those of the base fluid either from experiments or from standard or specific correlations. It is generally found that h is greater than the heat transfer coefficient of the base fluid. However, this method does not take into account the eventual extra energy due to increase of pumping power.

3. 2 Heat transfer compared with pumping power

As heat transfer and pressure drop are the most critical factors, they can be compared through several approaches. Many criteria have been proposed. Some of them are reminded. Studies using nanofluids are also cited.

a - The heat transfer per unit of pressure drop is represented by the ratio of the j - Colburn factor by the f - friction factor (or Darcy coefficient Λ) [12]

$$j = \frac{h}{C_p\, G} Pr^{2/3} \tag{7}$$

G is the mass velocity equal to $\dot{m}/S$, where S is the cross section of the flow duct.
The Darcy coefficient is defined from the pressure drop which can be written for a tube with an internal diameter D:

$$\Delta p = \Lambda \frac{L}{D} \frac{\rho V^2}{2} \tag{8}$$

Pira et al. [13] have compared 4 Performance Energy Criteria. They have shown that, to compare existing heat exchangers with equal duty, the two best criteria were (i) the j/Λ ratio, (ii) the ratio between the transferred heat flow rate and the pumping power. This is the last one we have chosen in this study.

b - Gosselin and Da Silva [14] have defined a heat transfer enhancement parameter Ω as the ratio between the heat transfer rate with the solid–liquid mixture and the heat transfer rate with the liquid phase only, at constant pumping power. By using classical published correlations for thermophysical properties of nanofluids they conclude that the Ω parameter could be maximized for a given volume fraction of nanoparticles.

c - For a given heat flow rate, a high pressure drop is the parameter which penalizes an installation. Corcione et al. [15] have evaluated the effect of the volume fraction on the pumping power requirement in terms of relative friction loss diminution δ_p defined as:

$$\delta_p = 1 - \frac{P_{m,nf}}{P_{m,f}} \tag{9}$$

$P_{m,nf}$ and $P_{m,f}$ are the pumping power required for the nanofluid and for the base fluid respectively, at same heat flow rate.
Pumping powers are given by:

$$P_{m,nf} = Q_{vnf}\ \Delta p_{nf} \tag{10}$$

$$P_{m,f} = Q_{vf}\ \Delta p_f \tag{11}$$

Q_v being the volume flow rate.
Writing the effective thermal conductivities and dynamic viscosities as a function of volume concentration, these authors have calculated the percentage optimal nanoparticle loadings for the three regimes, laminar, transitional and turbulent. Three formulae have been determined, one for each regime, under the form:

$$\varphi_{opt}(\%) = f\left(Re_f, \theta_m, d_p, L, D\right)$$

$$500 \le Re_f \le 5.10^6, 30\,°C \le \theta_m \le 70\,°C, 25\,nm \le d_p \le 100\,nm, 50 \le L/D \le 1000 \quad (12)$$

These authors have concluded in remarking the strong influence of the base fluid. It is probably due to their modelling of thermal conductivity depending on the base fluid Reynolds number.

d – Another criterion used by Prasher et al. [16] was to compare the pressure drop of a nanofluid with that of the base fluid for the same heat transfer coefficients

$$h_{nf} = h_f \quad (13)$$

they have found

$$\frac{\Delta p_{nf}}{\Delta p_f} = \left(\frac{\mu_{nf}}{\mu_f}\right)\left(\frac{k_f}{k_{nf}}\right)^4 \left(\frac{Nu_f}{Nu_{nf}}\right)^4 \quad (14)$$

Expressing the conductivity and the viscosity under the simplified form

$$\frac{k_{nf}}{k_f} = 1 + C_k\,\varphi \quad (15)$$

$$\frac{\mu_{nf}}{\mu_f} = 1 + C_\mu\,\varphi \quad (16)$$

they write

$$\frac{\Delta p_{nf}}{\Delta p_f} = \frac{(1 + C_\mu\,\varphi)}{(1 + C_k\,\varphi)^4}\left(\frac{Nu_f}{Nu_{nf}}\right)^4 \quad (17)$$

They conclude that, if it is desired that Δp_{nf} should not exceed Δp_f then

$$C_\mu \le 4\,C_k \quad (18)$$

e –As already employed in many studies [1-3], we can use the energy PEC (Performance Evaluation Criterion) defined below and based on an energy global approach. It is defined as the ratio of heat flow rate transferred to the required pumping power in the test section:

$$PEC = \frac{\dot{m}.C_p(T_{out} - T_{in})}{Q_v.\Delta p} \quad (19)$$

Where $\dot{m}$ is the mass flow rate (kg/s), T_{in} and T_{out} the heat exchanger inlet and outlet temperatures and Δp the pressure drop (Pa).

This criterion is directly related to gains and losses of energy in an industrial plant.

4. Theoretical expression of the Performance Energy Criterion (PEC)

4.1 General remarks

This criterion allows us to compare the transferred thermal energy P_{th} with the mechanical energy needed for pumping (pumping power) P_{mech}:

$$PEC = \frac{P_{th}}{P_{mech}} \quad (20)$$

It is important to know whether the PEC of a nanofluid is greater than the PEC of the base fluid. In this viewpoint, we write the following ratio:

$$\delta = \frac{PEC_{nf}}{PEC_f} \quad (21)$$

where PEC_{nf} stands for a given nanofluid and PEC_f for its base fluid. Taking into account definitions, this ratio becomes:

$$\delta = \frac{P_{th,nf}}{P_{mech,nf}} \Big/ \frac{P_{th,f}}{P_{mech,f}} = \frac{P_{mech,f}}{P_{mech,nf}} \frac{P_{th,nf}}{P_{th,f}} \tag{22}$$

If $\delta > 1$, use of the nanofluid is energetically favourable.

To compare theoretical and experimental results, we have treated the case of a fluid flowing inside a cylindrical tube with a volume flow rate Q_v . This tube represents the internal tube of a tube-in-tube heat exchanger. The tube length is L and its internal diameter D. The inlet and outlet temperature are respectively T_{in} and T_{out} and the wall temperature is T_p.

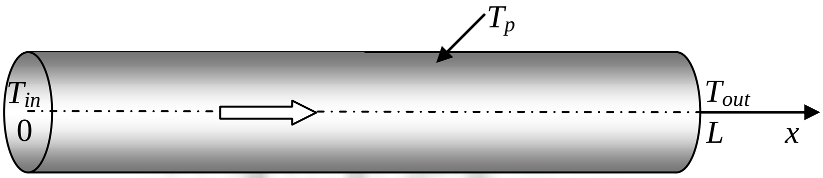

Figure 1. Geometry considered for calculations.

To make the comparison with experimental results easier we have adopted two approaches: (i) to calculate δ as a function of Q_v for the same thermal power transferred by the base fluid and the nanofluid (ii) to calculate δ as a function of the Reynolds number whatever the heat flow rate. We will give the theory in the two cases. The comparison experiment – theory will be carried out as a function of the Reynolds number only first because we don't know the transferred thermal power a priori in our experiments.

4.2 - PEC as a function of the volume flow rate Q_v

For the same thermal energy transferred with the nanofluid as that with the base fluid, we have

$$P_{th,nf} = P_{th,f} \tag{23}$$

and δ can be written

$$\delta = \frac{P_{mech,f}}{P_{mech,nf}}. \tag{24}$$

The pumping power is given by:

$$P_{mech} = Q_v \ \Delta p \tag{25}$$

where Q_v is the volume flow rate and Δp the pressure drop which must be overcome by the pumping system. Then:

$$\delta = \frac{Q_{v,f} \ \Delta p_f}{Q_{v,nf} \ \Delta p_{nf}} \tag{26}$$

Pressure drop

The pressure drop generally is written as

$$\Delta p = \xi \frac{\rho V^2}{2} \tag{27}$$

ξ being a coefficient which depends on Reynolds number *Re* and on geometry. To make the method clearer the ξ coefficient is given for a straight smooth tube of *D* diameter and length *L*:

$$\Delta p = \Lambda \frac{L}{D} \frac{\rho V^2}{2} \tag{28}$$

In this formula Λ is the Darcy friction factor depending on the flow regime.

Laminar regime

$$\Lambda = \frac{64}{Re} \tag{29}$$

knowing that

$$Re = \frac{\rho V D}{\mu} = \frac{\rho Q_v D}{S \mu} \tag{30}$$

where *S* is the cross section of the tube

Then

$$\Delta p = \frac{64}{D^2 \, 2 S} L \mu Q_v \tag{31}$$

or

$$\Delta p = \frac{128}{\pi D^4} L \mu Q_v \tag{32}$$

Then, the mechanical power is

$$P_{mech} = \frac{128}{\pi D^4} L \mu Q_v^2 \tag{33}$$

Turbulent regime

For a smooth tube, the Darcy friction factor follows the Blasius law:

$$\Lambda = \frac{0.316}{Re^{1/4}} \tag{34}$$

and the pressure drop in function of the volume flow rate is

$$\Delta p = 0.241 \frac{L}{D^{19/4}} \rho^{3/4} \mu^{1/4} Q_v^{7/4} \tag{35}$$

Then, in turbulent regime, the mechanical power is

$$P_{mech} = Q_v \Delta p = 0.241 \frac{L}{D^{19/4}} \rho^{3/4} \mu^{1/4} Q_v^{11/4}$$

Nanofluid flow rate

To obtain δ for identical thermal powers we must compare the volume flow rates of the base fluid with the volume flow rate of the nanofluid needed to have the same thermal power. Writing that the heat flow rates are the same for the base fluid and the nanofluid with the same inlet temperature T_{in}:

$$P_{th} = \rho_f \, Q_{v,f} \, C_{p,f} \, (T_{fout} - T_{in}) = \rho_{nf} \, Q_{v,nf} \, C_{p,nf} \, (T_{nfout} - T_{in}) \tag{36}$$

The nanofluid volume flow rate is

$$Q_{v,nf} = Q_{v,f} \, \frac{\rho_f}{\rho_{nf}} \, \frac{C_{p,f}}{C_{p,nf}} \, \frac{T_{f,out} - T_{in}}{T_{nf,out} - T_{in}} \tag{37}$$

At this point, it is necessary to calculate the outlet temperatures. For that we must know how heat is transferred to/from the fluids.
For a fluid flowing inside a tube whose wall temperature T_p is imposed, the energy balance in a length dx gives:

$$-\rho \, Q_v \, C_p \, dT = h \, \pi \, D \, dx \, (T - T_p) \tag{38}$$

Assuming h is constant and integrating this equation between $x = 0$ and $x = L$ gives:

$$T_{out} - T_p = (T_{in} - T_p) \, e^{-\frac{h \, \pi \, D \, L}{\rho \, Q_v \, C_p}} \tag{39}$$

T_{out} is the outlet temperature (for $T = T_{out}$).

This equation can be written as:

$$T_{out} - T_{in} = (T_{in} - T_p) \left[e^{-\frac{h \, \pi \, D \, L}{\rho \, Q_v \, C_p}} - 1 \right] \tag{40}$$

Or

$$T_{out} - T_{in} = (T_{in} - T_p) \left[e^{-NTU} - 1 \right] \tag{41}$$

The ratio $\frac{T_{in} - T_{out}}{T_{in} - T_p}$ is nothing but the effectiveness of the heat exchanger

$$\varepsilon = \frac{T_{in} - T_{out}}{T_{in} - T_p} = 1 - e^{-NTU} \tag{42}$$

The heat transfer coefficient can be defined as:

$$h = \frac{Nu \, k}{D} \tag{43}$$

The Nusselt number also depends on Q_v. Its value is different according the flow regime, laminar or turbulent.

Laminar regime

Firstly, we consider the laminar regime and we have used the Sieder & Tate correlation [17]

$$Nu = 1.86 \left(\mathrm{Re} \, \mathrm{Pr} \, \frac{D}{L} \right)^{1/3} \tag{44}$$

Where Re and Pr are the Reynolds and Prandtl numbers respectively. Expressing these numbers as a function of Q_v, it comes:

$$Nu = 1.86 \left(\frac{4\,\rho\, C_p}{\pi\; L\; k} Q_v \right)^{1/3} \tag{45}$$

And, the exponent of the exponential becomes:

$$\frac{h\,S}{\rho\, Q_v\, C_p} = 1.86\;\; 4^{1/3}\, \pi^{2/3}\, L^{2/3}\, \rho^{-2/3}\, C_p^{-2/3}\, k^{2/3}\, Q_v^{-2/3} \tag{46}$$

This last expression can be written as the product of three terms, a constant term depending on the geometry, a constant term depending on physical properties (therefore on φ for the nanofluid) and a term depending on Q_v.

Then, we write this exponent under the form:

$$\frac{h\,S}{\rho\, Q_v\, C_p} = \;\; B\;\; C\, Q_v^{-2/3} = F\; Q_v^{-2/3} \tag{47}$$

Where

$$B = 1.86\;\; 4^{1/3}\, \pi^{2/3}\, L^{2/3} \tag{48}$$

$$C = \rho^{-2/3}\, C_p^{-2/3}\, k^{2/3} \tag{49}$$

The nanofluid volume flow rate can be written as:

$$Q_{v,nf} = Q_{v,f}\; \frac{\rho_f}{\rho_{nf}}\, \frac{C_{p,f}}{C_{p,nf}}\, \frac{\left(e^{-F_f\, Q_{v,f}^{-2/3}} - 1 \right)}{\left(e^{-F_{nf}\, Q_{v,nf}^{-2/3}} - 1 \right)} \tag{50}$$

This is a transcendental equation which has to be solved numerically.

Turbulent regime

Secondly, we consider the turbulent regime and we choose a Dittus-Boelter or Colburn type correlation [18]:

$$Nu = A'\, \mathrm{Re}^{0.8}\; \mathrm{Pr}^{1/3} \tag{51}$$

Expressing the Nusselt number as a function of Q_v, it comes:

$$Nu = A' \left(\frac{4\,\rho\, Q_v}{\pi\, D\, \mu} \right)^{0.8} \left(\frac{\mu\, C_p}{k} \right)^{1/3} \tag{52}$$

The exponent of the exponential becomes:

$$\frac{h\,S}{\rho\, Q_v\, C_p} = A'\, 3.03\, L\;\; \pi^{0.2}\; D^{-0.8}\; \rho^{-0.2} \mu^{-0.467}\; C_p^{-2/3}\; k^{2/3}\; Q_v^{-0.2} \tag{53}$$

The last expression is again the product of three terms, a constant depending on geometry, a constant depending on physical properties (therefore on φ for the nanofluid) and a term with Q_v.

We write the exponent under the form

$$\frac{h\,S}{\rho\,Q_v\,C_p} = B'\ C'\,Q_v^{-0.2} = F'\,Q_v^{-0.2} \tag{54}$$

where

$$B' = A\,3.03\,L\,\pi^{0.2}\ D^{-0.8} \tag{55}$$

$$C' = \rho^{-0.2}\mu^{-0.467}\,C_p^{-2/3}\,k^{2/3} \tag{56}$$

The nanofluid volume flow rate can be written:

$$Q_{v,nf} = Q_{v,f}\ \frac{\rho_f}{\rho_{nf}}\,\frac{C_{p,f}}{C_{p,nf}}\,\frac{\left(e^{-F_f^{'}\,Q_{v,f}^{-0.2}}-1\right)}{\left(e^{-F_{nf}^{'}\,Q_{v,nf}^{-0.2}}-1\right)} \tag{57}$$

This is again a transcendental equation which has to be solved numerically.

Then,we obtain

Laminar regime:

$$\delta = \frac{P_{mech,f}}{P_{mech,nf}} = \frac{\mu_f\ Q_{v,f}^2}{\mu_{nf}\ Q_{v,nf}^2} \tag{58}$$

$$\delta = \frac{\mu_f}{\mu_{nf}}\left[\frac{\rho_{nf}\ C_{p,nf}}{\rho_f\ C_{p,f}}\ \frac{\left(e^{-F_{nf}\ Q_{v,nf}^{-0.2}}-1\right)}{\left(e^{-F_f\ Q_{v,f}^{-0.2}}-1\right)}\right]^2 \tag{59}$$

Turbulent regime

$$\delta = \frac{P_{mech,f}}{P_{mech,nf}} = \left(\frac{\mu_f}{\mu_{nf}}\right)^{\frac{1}{4}}\left(\frac{\rho_f}{\rho_{nf}}\right)^{\frac{3}{4}}\frac{Q_{v,f}^{11/4}}{Q_{v,nf}^{11/4}} \tag{60}$$

$$\delta = \frac{P_{mech,f}}{P_{mech,nf}} = \left(\frac{\mu_f}{\mu_{nf}}\right)^{\frac{5}{4}}\left(\frac{\rho_{nf}}{\rho_f}\right)^{\frac{14}{4}}\left(\frac{C_{p,nf}\ (e^{-F_{nf}\ Q_{v,nf}^{-0.2}}-1)}{C_{p,f}\ \ (e^{-F_f\ Q_{v,f}^{-0.2}}-1)}\right)^{\frac{11}{4}} \tag{61}$$

In these formulae, the physical properties are function of the particle volume concentration φ

$$\frac{\mu_{nf}}{\mu_f} = f(\varphi) \tag{62}$$

$$\frac{\rho_f}{\rho_{nf}} = \frac{1}{1-\varphi+\dfrac{\rho_f}{\rho_s}\varphi} \tag{63}$$

$$\frac{C_{p,nf}}{C_{p,f}} = (1-\varphi_w) + \varphi_w \frac{C_{ps}}{C_{p,f}} \quad \text{where} \tag{64}$$

$$\varphi_w = \frac{\rho_s\, \varphi}{\rho_s\, \varphi + \rho_f\, (1-\varphi)} \tag{65}$$

4.3 - PEC as a function of Re

In this case, we don't assume that the transferred thermal power is the same for the base fluid as for the nanofluid. The PEC is calculated for the same Reynolds number whatever the transferred power. Recalling the PEC expression:

$$\delta = \frac{P_{th,nf}}{P_{mech,nf}} \Big/ \frac{P_{th,f}}{P_{mech,f}} = \frac{P_{mech,f}}{P_{mech,nf}} \frac{P_{th,nf}}{P_{th,f}} \tag{66}$$

we express this formula as a function of the Reynolds number by writing

$$Re = \frac{4\, \dot{m}}{\pi\, D\, \mu} \tag{67}$$

$$\delta = \frac{\dfrac{\pi\, D\, \mu_f\, Re_f}{4\, \rho_f}}{\dfrac{\pi\, D\, \mu_{nf}\, Re_{nf}}{4\, \rho_{nf}}} \frac{\Delta p_f}{\Delta p_{nf}} \frac{\dfrac{\pi\, D\, \mu_{nf}\, Re_{nf}}{4}}{\dfrac{\pi\, D\, \mu_f\, Re_f}{4}} \frac{C_{p,nf}}{C_{p,f}} \frac{(T_{out,nf} - T_{in})}{(T_{out,f} - T_{in})} \tag{68}$$

$$\delta = \frac{\rho_{nf}}{\rho_f} \frac{\Delta p_f}{\Delta p_{nf}} \frac{C_{p,nf}}{C_{p,f}} \frac{(T_{out,nf} - T_{in})}{(T_{out,f} - T_{in})} \tag{69}$$

knowing that

$$\frac{T_{out} - T_{in}}{T_p - T_{in}} = 1 - e^{-NTU} \tag{42}$$

we obtain as a function of the Reynolds number

$$NTU = 4 \frac{L}{D} \frac{Nu\, k}{Re\, \mu\, C_p} \tag{70}$$

or, knowing that

$$Pr = \frac{\mu\, C_p}{k} \tag{71}$$

$$NTU = 4 \frac{Nu}{\left(Re\, Pr \dfrac{D}{L}\right)} \tag{72}$$

Laminar regime

In this case, with the same correlation as in the previous paragraph, we obtain:

$$\frac{T_{s,nf}-T_e}{T_{s,f}-T_e}=\frac{1-e^{-7.44\left(Pr_{nf}\frac{D}{L}\right)^{-2/3}Re_{nf}^{-2/3}}}{1-e^{-7.44\left(Pr_f\frac{D}{L}\right)^{-2/3}Re_f^{-2/3}}} \tag{73}$$

Concerning the pressure drop, the relations giving the Darcy friction coefficient are the same for the base fluid and the nanofluid:

$$P_{mech}=Q_v\ \Delta p=\frac{\pi}{8}\Lambda\frac{L}{D^2}\frac{\mu^3\ Re^3}{\rho^2} \tag{74}$$

$$\frac{P_{mech,f}}{P_{mech,nf}}=\frac{\dfrac{\mu_f^3\ Re_f^3}{\rho_f^2}}{\dfrac{\mu_{nf}^3\ Re_{nf}^3}{\rho_{nf}^2}}\frac{\Lambda_f}{\Lambda_{nf}} \tag{75}$$

Assuming that the Reynolds numbers of the base fluid and the nanofluid are identical, whatever the transferred heat flow rate, it comes:

$$\frac{P_{mech,f}}{P_{mech,nf}}=\frac{\mu_f^3\rho_{nf}^2}{\mu_{nf}^3\rho_f^2} \tag{76}$$

and finally

$$\delta=\frac{\mu_f^2\rho_{nf}^2}{\mu_{nf}^2\ \rho_f^2}\frac{C_{p,nf}}{C_{p,f}}\frac{(T_{out,nf}-T_{in})}{(T_{out,f}-T_{in})} \tag{77}$$

or

$$\delta=\frac{\mu_f^2\rho_{nf}^2}{\mu_{nf}^2\ \rho_f^2}\frac{C_{p,nf}}{C_{p,f}}\frac{1-e^{-7.44\left(Pr_{nf}\frac{D}{L}\right)^{-2/3}Re_{nf}^{-2/3}}}{1-e^{-7.44\left(Pr_f\frac{D}{L}\right)^{-2/3}Re_f^{-2/3}}} \tag{78}$$

Turbulent regime

With the same correlation as previously, the δ expression is:

$$\delta=\frac{\mu_f^2\rho_{nf}^2}{\mu_{nf}^2\ \rho_f^2}\frac{C_{p,nf}}{C_{p,f}}\frac{1-e^{-4A\frac{L}{D}\left(Pr_{nf}\right)^{-2/3}Re_{nf}^{-0.2}}}{1-e^{-4A\frac{L}{D}\left(Pr_f\right)^{-2/3}Re_f^{-0.2}}} \tag{79}$$

This expression is weakly dependent on Prandtl and Reynolds numbers but strongly dependent on the viscosity and density ratios. In particular, accurate values of viscosities are needed to determine exact values of the PEC ratios.

What it is surprising is that δ depends very little on the thermal conductivity.

5. Comparison with experimental results

Calculations have been compared with experimental results obtained in this work and those of Ferrouillat et al. [19, 20]. Performances of several nanofluids have been studied. Four were formulated in our laboratory and their composition was known: two with SiO_2 nanoparticles, two with ZnO nanoparticles. For each nanoparticle material two types of nanoparticles have been used, spherical or particles with a shape factor (rods or curved cylinders). The fifth is a commercial fluid with ZnO nanoparticles. The known characteristics are summarized in table 1. Values of viscosity and thermal conductivity have been measured in an experimental way. Heat transfer coefficients and Darcy friction factors have been determined for Reynolds numbers varying from 2000 to 15000 for Silica and, between 2000 and 9000 for Zinc Oxyde.

Table I

Base fluid	Nanoparticle	Volume fraction	Sphericity factor	Remarks
Water	SiO_2	1.08	1	
Water	SiO_2	2.28	0.4 - 0.5	
Water	ZnO	0.82	1	
Water	ZnO	0.93	0.5 – 0.6	
Unknown	ZnO	Unknown	Unknown	Commercial product

4.1 PEC as a function of the Reynolds number

4.1.1 SiO_2 nanoparticles in water

To compare our calculations to experimental results we have first considered the effect of the use of SiO_2 nanoparticles manufactured in the same laboratory whose fabrication process is controlled and which has been described in [20]. Two types of nanoparticles have been incorporated in a base fluid (water plus additives), one approximately spherical (sphericity equal to 1), the other with a shape factor (with sphericity of 0.4 – 0.5) which were called "bananas" due to their appearance. The nanofluid has been either cooled or heated and its temperature varies approximately between 20 to 50 °C. Comparison of theoretical δ values with results of Ferrouillat et al. is given on Figure 2. Plain and dotted lines are the calculation results. When the nanofluid is cooled, it is labelled "cooling" in the figure when it is heated it is labelled "heating". It is shown that, firstly, the model can reasonably represent the experiments. Nevertheless, the theoretical results are higher than of the experimental ones. This could be due to the simplicity of the model in which we have assumed constant values of the physical properties. Secondly, in each case (experiment and theory), δ is less than one. This indicates that the energy balance is not favourable.

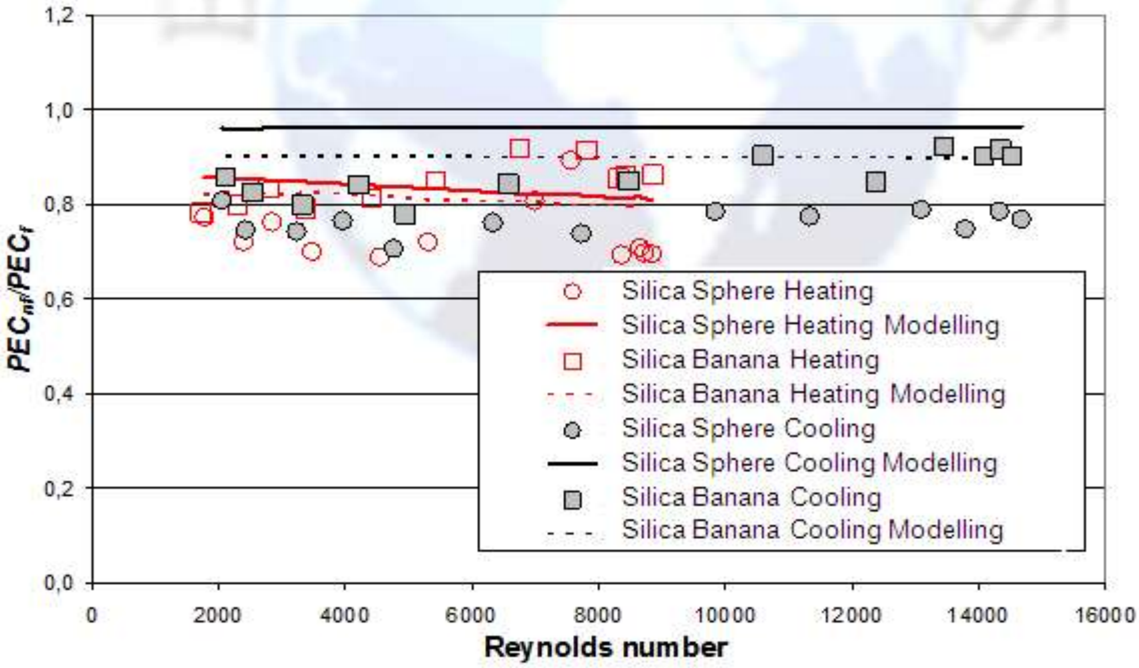

Figure 2: Comparison of the Performance Energy Criterion obtained with a $Si0_2$/water nanofluid with that obtained with the base fluid (water)

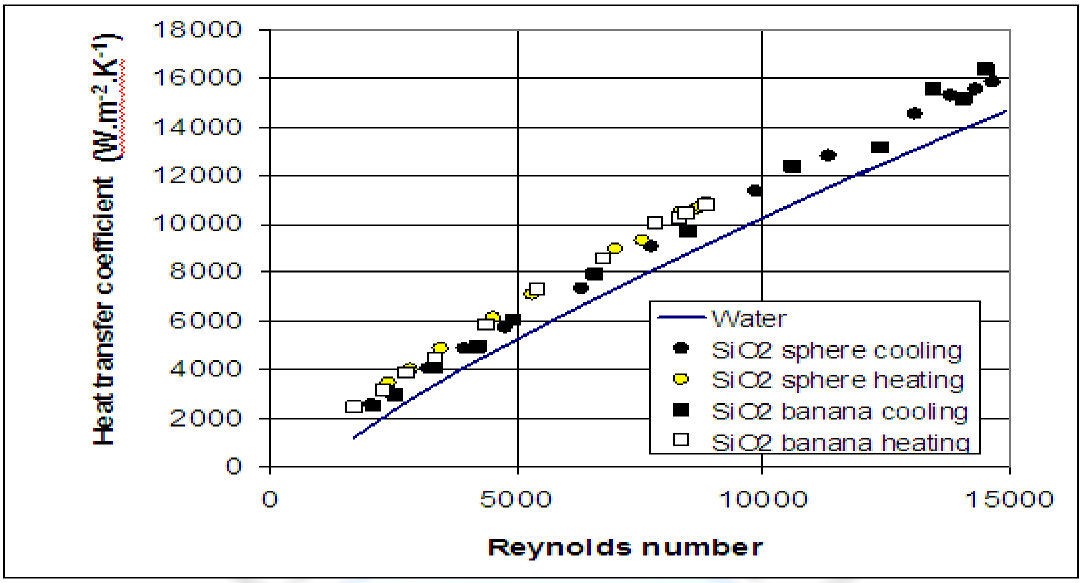

Figure 3: Heat transfer coefficients obtained with colloidal suspensions of SiO_2 nanoparticles compared to those obtained with water.

It must be remarked that if we are interested by the heat transfer only, the heat transfer coefficients of the two nanofluids are higher than that of the base fluid as shown in fig. 3. The enhancement factor δ_H (ratio between the two nanofluid heat transfer coefficients and the heat transfer coefficient of water) is shown figure 4. It varies between 1.5 to 1.2 approximately.

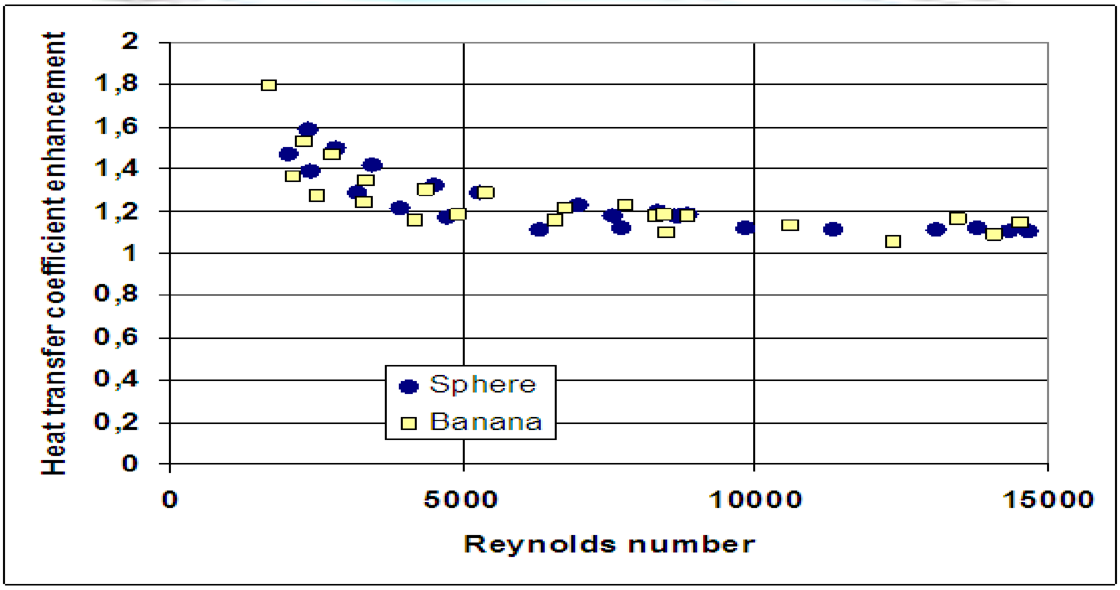

Figure 4: Enhancement factor δ_H of SiO2/water nanofluid.

The experimental results has already been discussed [19, 20], in particular the uncertainty. The accuracy is better in fully turbulent flow (Re > 10000) than in transitional flow (2300 < Re < 10000). Nevertheless, we can observe that in laminar regime two behaviours can be depicted. In laminar regime, when the nanofluid is heated the heat transfer coefficient is slightly higher than for a cooled nanofluid. This twofold behaviour is reduced as the Reynolds number increases in the transitional regime. In turbulent regime a single behaviour is observed. This is probably due to the modification of velocity and temperature profiles in laminar regime. In turbulent regime streamlines are strongly mixed and this behaviour no longer appears.

4.1.2 ZnO nanoparticles in water

Two nanofluids were studied in these experiments. One is formulated with ZnO nanoparticles whose sphericity is about 1which were supplied by Nyacol. The second is formulated with nanoparticles whose sphericity is about 0.5 which were supplied by Evonik. In both cases the base fluid is water. The preparation mode is described in [20]. As for SiO_2 nanofluids, comparison is made between theory and results of Ferrouillat et al. [20]. On figure 5, it is seen that the energetic balance is not favourable. The PEC is slightly better for spherical nanoparticles. It is probably due to the volume concentration which is higher (2.28 % compared to 1.08 %) leading to higher pressure drop.

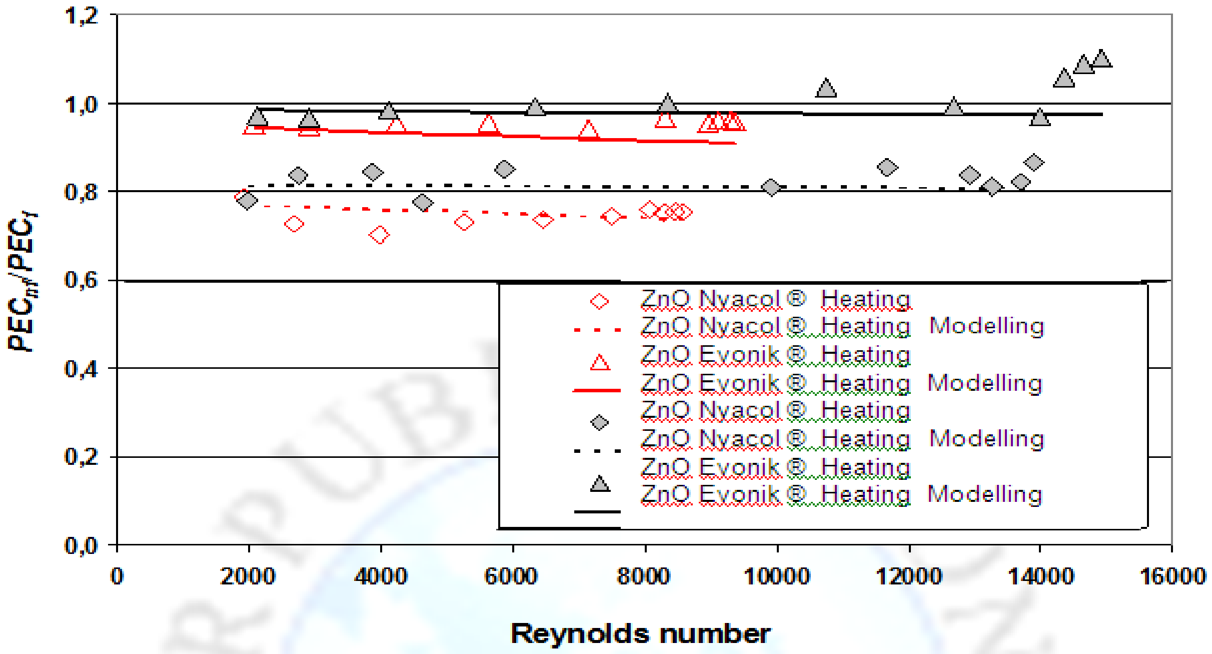

Figure 5: Comparison of the Performance Energy Criterion obtained with ZnO/water nanofluids with that obtained with the base fluid (water)

If we are interested by the heat transfer only, the heat transfer coefficients of the two nanofluids are higher than that of the base fluid as shown in fig. 6. The heat transfer coefficient enhancement is nearly constant and equal to about 1.4 (Figure 7). As observed with SiO_2 nanofluids, accuracy is better for fully turbulent flows.

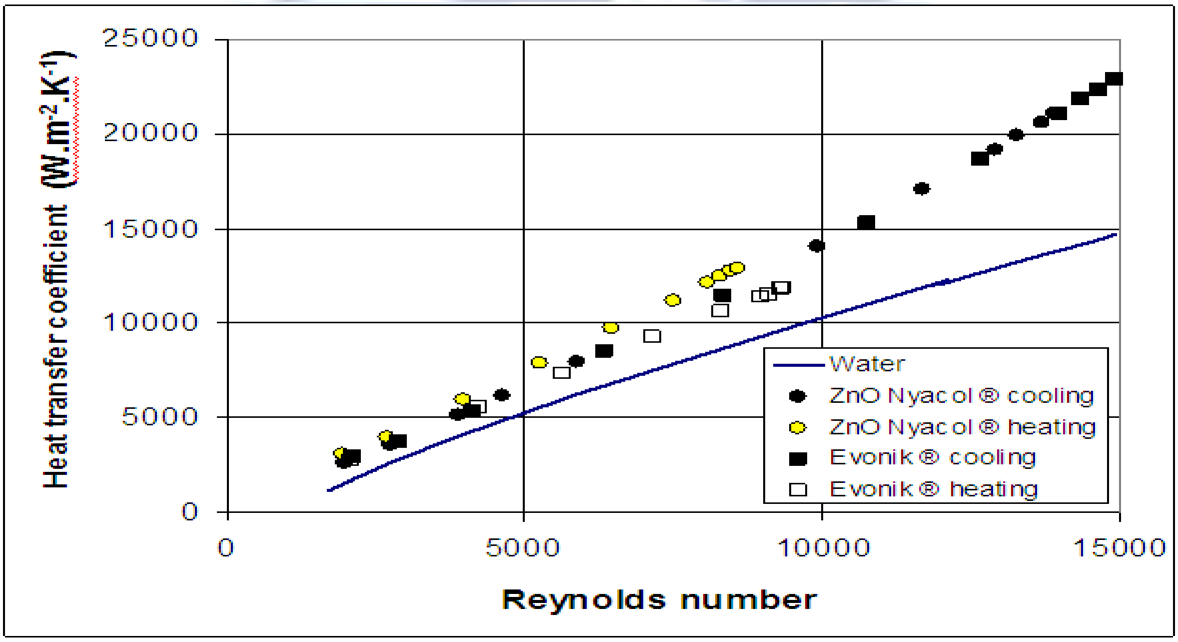

Figure 6: Heat transfer coefficients obtained with colloidal suspensions of ZnO nanoparticles compared to those obtained with water.

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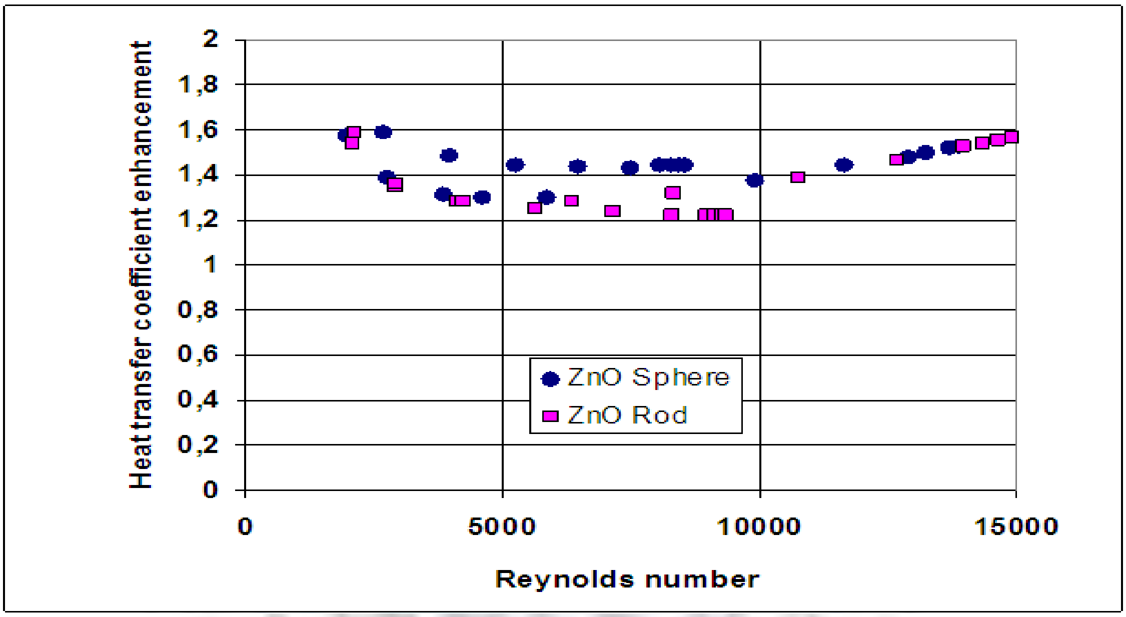

Figure 7: Enhancement factor of ZnO/water nanofluids.

`4.1.3 Commercial nanofluid (ZnO nanoparticles)

A commercial nanofluid with ZnO nanoparticles has been tested. The base fluid is unknown.
The physical properties have been measured to allow us to carry out both the reduction of experimental data and the theoretical calculations of the PEC. It must be noticed that, in the absence of the base fluid knowledge we have made the comparison with water.

We have first reported the heat transfer coefficient obtained with this fluid (figure 8). It is observed a strong enhancement of the heat transfer coefficient compared with that of water (about 1.5 – 1.65, Figure 9). The general trend for Reynolds numbers between 2000 and 6000 is similar to that observed with the home made nanofluid. However obtained PEC is very low due to the strong pressure drop of this fluid (Figure 10).

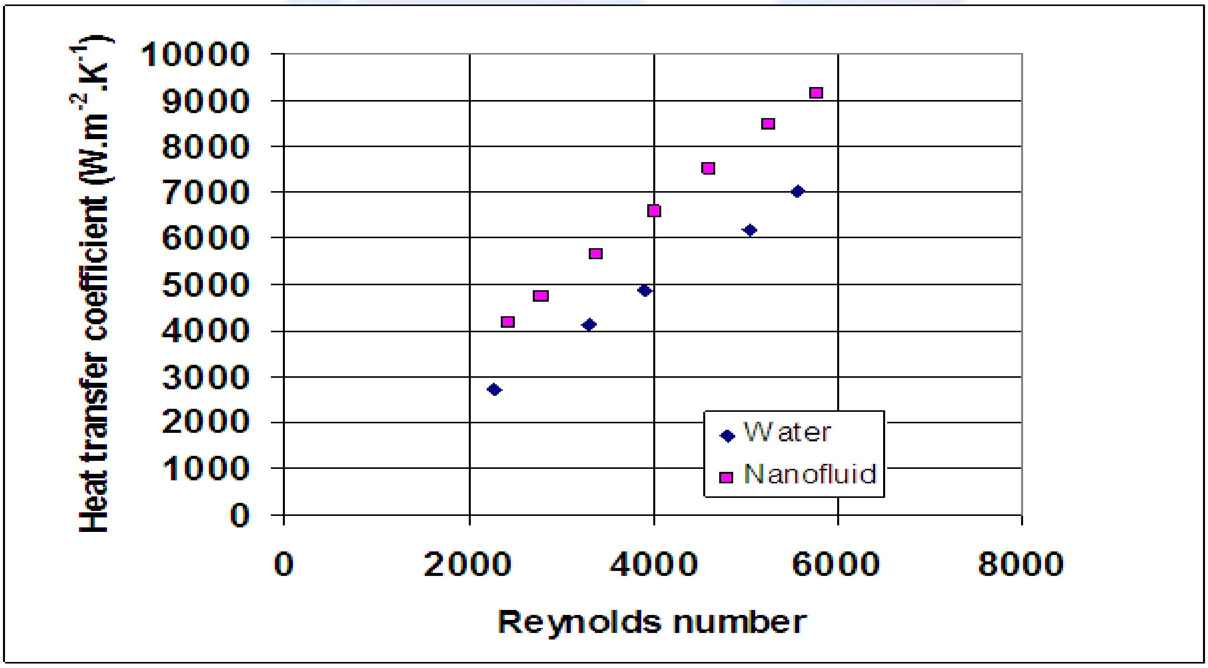

Figure 8: Heat transfer coefficient of the commercial fluid with ZnO nanoparticles as a function of Reynolds number.

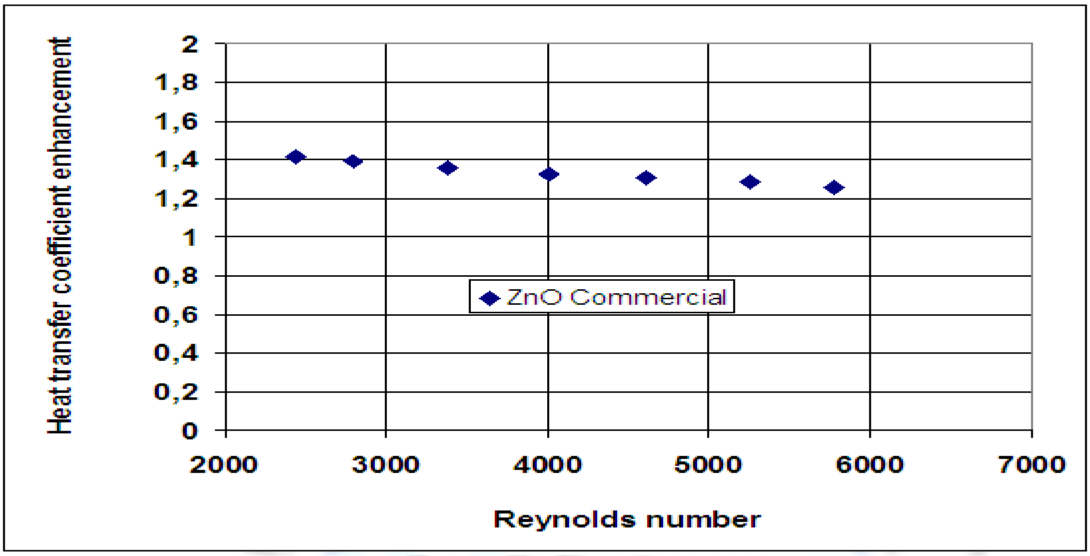

Figure 9: Enhancement factor of the heat transfer coefficient of a nanofluid with ZnO nanoparticles. The reference fluid is water.

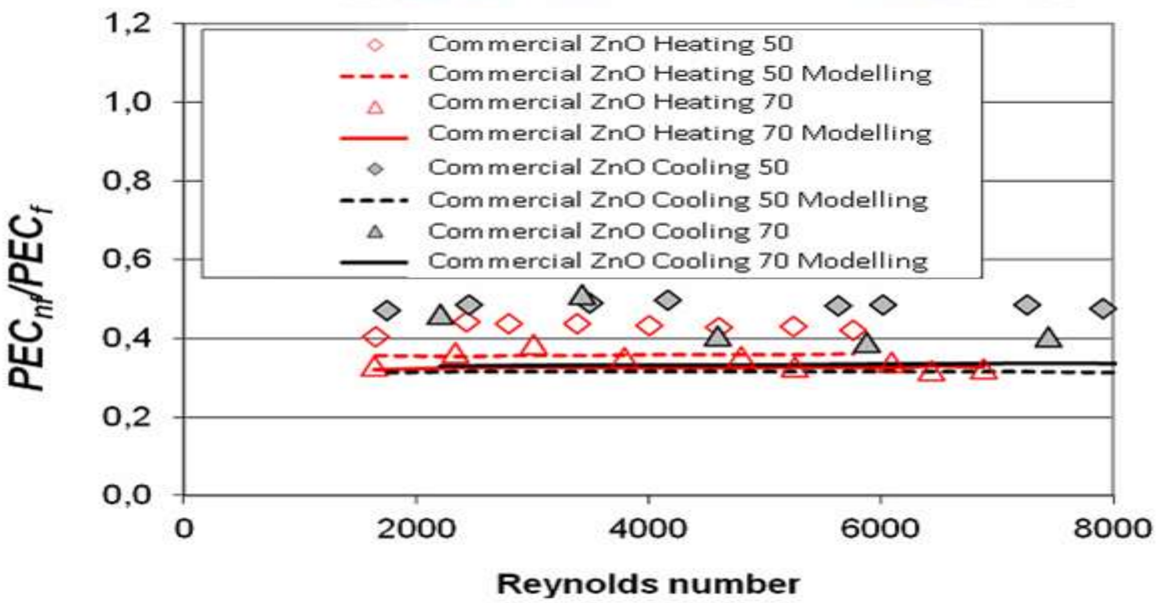

Figure 10: Comparison of the Performance Energy Criterion obtained with a commercial fluid with ZnO nanoparticles with that obtained with water.

4.1.4 Summary

Our results are summarized in table II.

Table II. Summary of results

Base fluid	Nanoparticle	Volume fraction	δ	δ_H	Remarks
Water	SiO_2	1.08	0.7 – 0.8	1.05-1.59	range
Water	SiO_2	2.28	0.8 - 0.9	1.09 – 1.8	range
Water	ZnO	0.82	0.75 – 0.8	1.44	average
Water	ZnO	0.93	0.95	1.36	average
Unknown	ZnO	Unknown	0.4	1.25 – 1.41	range

In all the studied cases, an enhancement of the heat transfer coefficient is observed. This enhancement is generally higher for nanofluids with ZnO nanoparticles whose thermal conductivity is higher. However, the enhancement of the heat transfer coefficient is paid in terms of PEC, i-e in terms of pressure drop.

Conclusion

In this work we have defined a Performance Evaluation Criterion (PEC) whose analytical expression has been determined in the case of a nanofluid flowing inside a cylindrical tube. This chosen PEC is the most general but necessitates the knowledge of transferred heat flow rate and pumping power. We have compared the criterion obtained for 5 nanofluids (PEC_{nf}) with that obtained with pure water in the same experimental conditions (PEC_f). So, we have defined a coefficient δ which is the ratio PEC_{nf} / PEC_f. If this ratio is more than one the energy balance is favourable. These ratios have been calculated for flows with the same Reynolds number. Calculations have been compared with experimental results obtained with four nanofluids formulated in our laboratory and one commercial nanofluid. One can see that the experimental results are reasonably represented. For all studied nanofluids the δ coefficient is less than one and in all studied cases, water remains the most energetic favourable fluid. However, if we only consider the heat transfer gain it is observed that all studied nanofluids allow the heat transfer coefficient to be enhanced. The choice of a thermal fluid (if possible) must be guided by the targeted objective. If we are concerned by a global energy gain, the obtained results show that further work must be undertaken to find a favourable fluid. If we look for heat transfer enhancement only, many nanofluids can fulfil such a requirement.

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