IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 17, Issue 4 Ser. III (Jul. – Aug. 2020), PP 24-40
www.iosrjournals.org

Determination of the best configuration of a given shell and tube heat exchanger for water cooling using nanofluid (CuO) and concepts of efficiency, effectiveness, and irreversibility

Élcio Nogueira
Department of Mechanical Engineering and Energy - DME, State University of Rio de Janeiro - UERJ, Brazil

Abstract:*The purpose is to determine the best configuration option for a shell and tube heat exchanger for water cooling, with negligible irreversibility associated with pressure drop in the flow. Two different situations are analyzed: hot water flows in the tube and the cold fluid in the shell, and vice versa, with the cold nanofluid (CuO) flowing in the counter current. The water enters at a temperature of 90 ºC and the nanofluid enters at 27 ºC. The volumetric fractions of copper oxide (CuO) are considered as a parameter, using the concepts of efficiency, effectiveness, and irreversibility that provide a precise measure of the thermal performance of a given heat exchanger.*
Key Word:*Shell and tube heat exchanger; Copper oxide nanofluid; Ethylene Glycol; Second law of thermodynamics.*

Date of Submission: 11-08-2020 Date of Acceptance: 27-08-2020

I. Introduction

The purpose is to determine the best configuration option for a shell and tube heat exchanger for water cooling, with negligible irreversibility associated with pressure drop in the flow. Two different situations are analyzed: hot water flows in the tube and the cold fluid in the shell, and vice versa, with the cold nanofluid (CuO) flowing in the counter current. The water enters at a temperature of 90 ºC and the nanofluid enters at 27 ºC. The volumetric fractions of copper oxide (CuO) are considered as a parameter, using the concepts of efficiency, effectiveness, and irreversibility that provide a precise measure of the thermal performance of a given heat exchanger.

Adrian Bejan [1] published several articles during the 1980s of the 19th century on the generation of entropy and application of the second law of thermodynamics in the heat and mass flows. In his work, he selected some applications in the areas of heat exchangers, thermal energy storage, and mass exchanger design. The studies aimed to analyze the balance between irreversibilities associated with heat transfer and flow and to evaluate the generation of entropy at the level of the system components. As a hypothesis, he presented an analysis of heat transfer in systems with negligible irreversibility associated with pressure drop in the flow.

Ahmad Fakheri [2] defined the concept of efficiency for heat exchangers based on the second law of thermodynamics. Efficiency is defined as the ratio of the actual heat transfer rate to the optimal heat transfer rate. It is commonly determined by the general expression $\eta=\tanh(Fa)$, where Fa is the dimensionless group that characterizes the thermal performance of different heat exchangers. According to the author, this way of defining efficiency provides a convenient way for and analysis of heat exchangers.

The use of nanofluids in shell and tube, double tube and spiral tube heat exchangers was presented by Maysam Molana [3], to demonstrate the advantages of this type of application. The study included experimental results and confirmed the increase in the number of Nusselt and the required pumping power.The article presents a review of the literature on the use of nanofluids in tubular heat exchangers with nanofluids. It concludes that the nanoparticles of alumina, copper oxide and carbon nanotubes are the most used in the applications.

The injection of air bubbles into shell and tube heat exchangers was implemented by Gaurav Thakur and Gurpreet Singh [4]. Air bubbles increase the turbulence in the fluid and, consequently, the heat exchange. The study results revealed that the heat transfer rate increases by about 25-40%.

In work developed by Pankaj K Pandey et al. [5], nanofluid was applied in a shell and tube heat exchanger with aluminum oxide and silicon carbide nanoparticles with water. It has been experimentally proven that Al2O3 acts as a better nanoparticle compared to SiC and that there is a significant increase in the heat transfer coefficient when the distribution of the nanoparticles is homogeneous and stabilized. Its effectiveness usually measures the thermal performance of a heat exchanger. Still, the effectiveness does not provide any information about the efficiency and irreversibility, which represents the entropy generation of the system.

In the article presented by Roopesh Tiwari and Govind Maheshwari [6], the second law of thermodynamics was utilized into the shell and tube heat exchanger design. The objective was to determine maximum effectiveness with an efficiency based on the analogy of the constant fin area with the insulated tip.

Ali Heydari et al. [7] present study in a three-dimensional CFD modeling simulation to investigate the effect of nanofluids with vary volume fractions for application in baffled shell and tube heat exchanger. The results indicate, considering a constant heat transfer rate, using some nanofluids will increase the outlet temperature of the shell, that is, the outlet temperature is increased by enhancing volume fraction for nanoparticles with higher conductivity.

Yue Sun et al. [8] developed a numerical simulation of a shell and tube heat exchanger with tilted baffles to investigate the flow and heat transfer characteristics. Shell and tube heat exchanger with segmental baffles is used for comparison. The standard shell and tube heat exchanger with segmental bases (STHX-SG) has many advantages. However, there are disadvantages such as high-pressure drop, dead flow zones, fouling, and vibration. The comparisons show that the inclined deflectors can generate oblique flow and, as a result, flow patterns and temperature distributions uniform are observed through the numerical results. A better uniform distribution in the fields of temperature and flow leads to higher overall performance for the heat exchanger, characterized by the relationship between the rate of heat transfer and pressure drop.

K.Y. Leong et al. [9] concluded that the arrangement of baffles influences the performance of the shell and tube heat exchangers. They developed a study that focuses on the heat transfer and entropy analysis in the segmental helical baffles. This study, realized between three different heat exchangers, indicates that 50° helical baffles exhibit the lowest entropy generation.

Omid Mahian et al. [10] present reviews of works on the generation of entropy in nanofluids and discuss the effects of the thermal flow of nanofluids on the entropy generation rate for different applications. Water was used as the base fluid in most analyzed works, followed by ethylene glycol. Aluminum oxide ($Al2O3$) and copper oxide (Cu) are the most used nanoparticles. In a global approach, the generation of total entropy is obtained for the entire system, and the Nusselt number and the friction factor are calculated based on the formulations developed, mainly by Bejan. They concluded, mostly, that depending on the volume fraction of the nanoparticles, geometric arrangement, and flow regime, the use of nanofluids can be beneficial in reducing irreversibility. However, minimizing entropy generation can lead to inefficient processes that are not suitable for specific heat exchanger devices.

M. Karuppasamy et al. [11] presents a numerical study on a double tube heat exchanger with a twisted plate insert to maximize the heat transfer and minimize the pumping power. They suggest the use of $SiO2$-based nanofluid with twisted ribbon insertion to achieve maximum amplification in heat exchange since the thermal performance factor is 2.4 times higher in laminar flow.

Jaafar Albadr et al. [12] present an experimental study in a horizontal shell and tube heat exchanger in counterflow under turbulent flow conditions of a nanofluid consisting of water and different values of flowing $Al2O3$ (0.3–2)%. It was demonstrated that a particle volume concentration of 2% the use of $Al2O3$/water nanofluid gives significantly higher thermal performance and that the overall heat transfer coefficient of the nanofluid is 57% greater than that of distilled water.

Ahmed A. H. Mostafa et al. [13] deal with the optimization of the deflector angle of a particular shell and tube heat exchanger using the fluent CFD software package. They conclude that the heat exchanger with 30º deflectors offers the highest conversion efficiency.

Salah Almurtaji et al. [14] provides a systematic review of the knowledge towards using nanofluids for enhancing thermal-hydraulic performance, especially for plate and plate-fin heat exchangers. They present a general operational theory of heat exchangers and emphasize the importance of replacing conventional working fluids with nanofluids as a means of significantly increasing thermal efficiency. They state that the available literature makes it possible to conclude that carbon nanotubes (CNT) are highly effective for thermo-hydraulic performance since they cause higher heat transfer and less pressure drop.

An analytical solution is presented for shell and tube heat exchangers. The objective is in the heat exchange between fluids consisting of hot water in the tube and cold nanofluid in countercurrent flow in the shell. Ethylene glycol-water (EG50%) is the base fluid, and the analysis focuses mainly on the outlet temperature of the water to be cooled. The study shows that the most favorable result for the hot fluid outlet temperature occurs when the efficiency is high Élcio Nogueira [15].

II. Methodology

The Shell and Tube heat exchanger [Figure 1] considered in this analysis has the following physical characteristics: LS = 0.762 m; DS = 0.508 m; DT = 0.0127 m; NT = 32; one pass in the tube, one pass in the shell and 3 baffles.

LS is the length of the heat exchanger, DS is the diameter of the shell, DT is the diameter of the tube and NT is the number of tubes.

Inlet temperatures for hot and cold fluids are, respectively, Thi = 90 ºC and TCi = 27 ºC.

Table 1 – Properties of the fluids and nanoparticles [15]

	Cold Fluid	**Hot Fluid**	**CuO**	**EG50%**
k W/(m ºC)	**0.60**	**0.67**	**400**	**1830.4223**
Cp J/(Kg ºC)	**4180**	**4216**	**8933**	**3878.9317**
μKg/(m s)	**$0.758\ 10^{-3}$**	**$0.3031\ 10^{-3}$**	-	**$5.277983\ 10^{-4}$**
ρKg/m^3	**997**	**970**	**385**	**1058.33**
νm^2/s	**$0.8\ 10^{-6}$**	**$0.334\ 10^{-6}$**	-	**$4.987\ 10^{-7}$**
αm^2/s	**$1.430\ 10^{-7}$**	**$1.680\ 10^{-7}$**	**$1.163\ 10^{-4}$**	**$4.865\ 10^{-4}$**
Pr	**5.68**	**1.98**	-	**975.49**

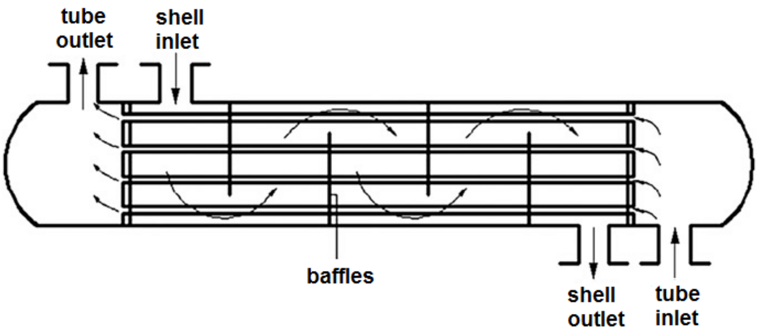

Figure 1– The basic configuration of Shell and Tube [15]

The properties of the base fluid can be obtained by the equations below and using the values in Table 1:

$$\mu_{solution} = \mu_{EG} V_{EG} + (1 - V_{EG})\mu_{wc} \qquad 01$$

$$\rho_{solution} = \rho_{EG} V_{EG} + (1 - V_{EG})\rho_{wc} \qquad 02$$

$$k_{solution} = k_{EG} V_{EG} + (1 - V_{EG})k_{wc} \qquad 03$$

$$Cp_{solution} = Cp_{EG} V_{EG} + (1 - V_{EG})Cp_{wc} \qquad 04$$

at where:

$\mu_{solution}$ is the absolute viscosity of the base fluid consisting of Ethylene glycol (50%) and water;

$\rho_{solution}$ is the specific mass of the base fluid consisting of Ethylene glycol (50%) and water;

$k_{solution}$ is the thermal conductivity of the base fluid consisting of Ethylene glycol (50%) and water;

$Cp_{solution}$ is the specific heat of the base fluid consisting of Ethylene glycol (50%) and water;

V_{EG} is the properties of the ethylene glycol (50%).

The properties of the nanofluid is given by:

$$\rho c = \emptyset \rho particle + (1 - \emptyset)\rho solution \qquad 05$$

$$\mu c = \mu solution(1 - 0.19\emptyset + 306\emptyset 2) \qquad 06$$

$$Cpc = (\emptyset \rho particle Cpparticle + (1 - \emptyset)\rho solution Cpsolution)/\rho c \qquad 07$$

$$kc = [(kparticle + 2ksolution + 2(kparticle - ksolution)(1 - 0.1)3\emptyset)/(kparticle + 2ksolution(kparticle - ksolution)(1 + 0.1)2\emptyset)]\, ksolution \qquad 08$$

at where:

μ_c is the absolute viscosity of cold fluid;

ρc is the specific mass of the cold fluid;

$k c$ is the thermal conductivity of the cold fluid;

$Cp c$ is the specific heat of the cold fluid;

$\emptyset$ is the volume fraction of the nanoparticles.

The heat exchange area is obtained by the following equation:

$$A_T = \pi D_T L_S N_T \qquad 09$$

The Reynolds number in the tube is given by:

$$\mathrm{Re}_T = \frac{4\left(\dot{m}_T/N_T\right)}{\pi D_T \mu_T} \qquad 10$$

The Nusselt number in the tube is given by:

$$Nu_S = 4.364 + 0.0722 Re_S Pr_S \frac{d_e}{L_S} \quad for\ Re_S < 2100 \qquad 11$$

or

$$\mathrm{Nu}_S = 0.023 \mathrm{Re}_S{}^{0,8} \mathrm{Pr}_S{}^{n} \quad for\ Re_T \geq 2100 \qquad 12$$

d_e is the shell equivalent diameter:

$$d_e = \frac{4x(free - flow\ area)}{wetted\ perimeter} \qquad 13$$

Then,

$$d_e = \frac{4\left(P_T{}^2 - \frac{\pi D_T{}^2}{4}\right)}{\pi D_T} \quad for\ square\ pitch \qquad 14$$

at where, the tube pitch for square pitch is given by:

$$\mathrm{P}_T = 1.25 D_T \qquad 15$$

Then, the convection heat transfer coefficient in the shell is a function of Nu_S:

$$h_S = \frac{Nu_S k_S}{d_e} \qquad 16$$

D_B and B_S, the bundle diameter and baffles space, are given respectively by:

$$D_B = D_s\left(\frac{N_T}{0.125}\right)^{1/2.207} \qquad 17$$

$$\mathrm{B}_S = 0.4 \mathrm{D}_B \qquad 18$$

Then, the number of baffles is given by:

$$N_B = \frac{L_S}{B_S} \qquad 19$$

The overall convection heat transfer coefficient is given by:

$$U_O = \frac{1}{\frac{1}{h_T} + \frac{1}{h_S}} \qquad 20$$

The thermal capacities of the cold and hot fluids is given by:

$$\mathrm{C}_c = \dot{\mathrm{m}}_c \mathrm{Cp}_c \text{ and } \mathrm{C}_h = \dot{\mathrm{m}}_h \mathrm{Cp}_h \qquad 21$$

Then, we have the Number of Thermal Units (NTU):

$$NTU = \frac{A_T U_O}{C_{min}} \qquad 22$$

at where, C_{min} is the lowest value among thermal capacities.

The effectiveness (ε) of the shell and tube heat exchanger is given by:

$$\varepsilon = \frac{1}{\frac{1}{\eta NTU} + \frac{(1+C^*)}{2}} \quad 23$$

The efficiency (η) of the heat exchanger is:

$$\eta = \frac{Tanh(Fa)}{Fa} \quad 24$$

at where,

$$Fa = \frac{NTU(1 - C^*)}{2} \quad 25$$

and

$$C^* = \frac{C_{max}}{C_{min}} \quad 26$$

The heat transfer rate is given by:

$$Q_{Actual} = \frac{(Th_i - Tc_i)C_{min}}{\frac{1}{\eta NTU} + \frac{(1+C^*)}{2}} \quad 27$$

The maximum heat transfer rate is:

$$Q_{max} = C_{min}\,(Th_i - Tc_i) \quad 28$$

The outlet temperatures is given by:

$$Th_O = Th_i - \frac{Q_{Actual}}{\dot{m}_h Cp_h} \quad 29$$

$$Tc_O = Tc_i + \frac{Q_{Actual}}{\dot{m}_c Cp_c} \quad 30$$

The thermal irreversibility is given by:

$$\eta_T = \frac{C_h}{C_{min}} \ln\left(\frac{Th_O}{Th_i}\right) + \frac{C_c}{C_{min}} \ln\left(\frac{Tc_O}{Tc_i}\right) \quad 31$$

III. Result and discussion

III.1 - Cold nanofluid (CuO) with Ethylene Glycol water-based fluid (50%) in the shell and hot water in the tube

Figure 2 shows the Reynolds number as a function of the mass flow rate of the nanofluid in the shell, with fractions of nanoparticles as a parameter. It is possible to observe the flow laminarization process since the Reynolds number decreases when the fraction volume, for the same mass flow of the nanofluid, increase. This flow-related aspect has significant effects on the heat exchange process and the overall thermal performance of the heat exchanger.

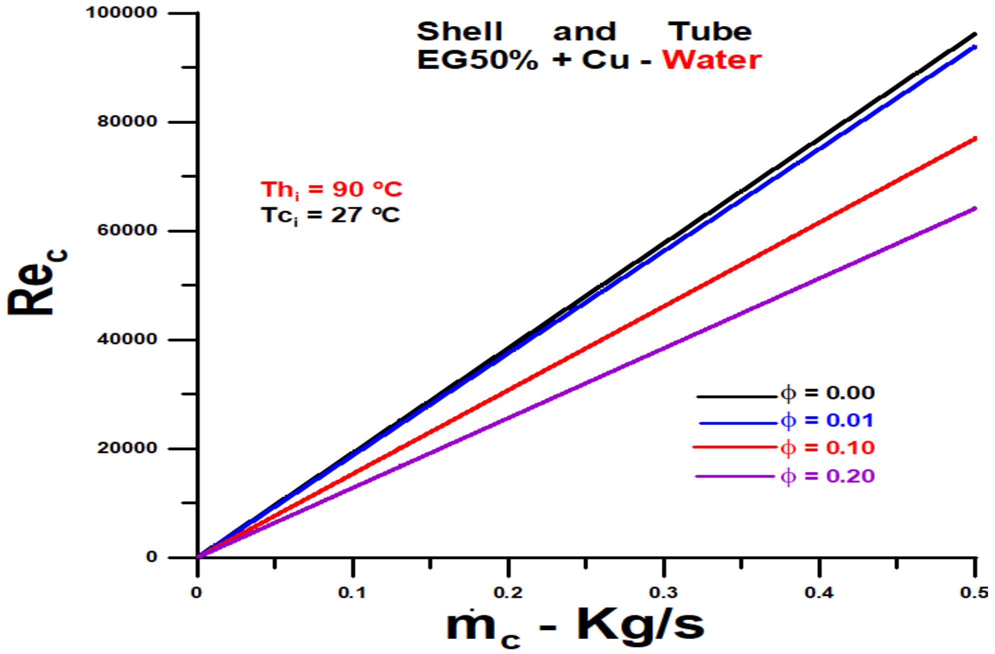

Figure 2– Reynolds number versus mass flow rate in the shell

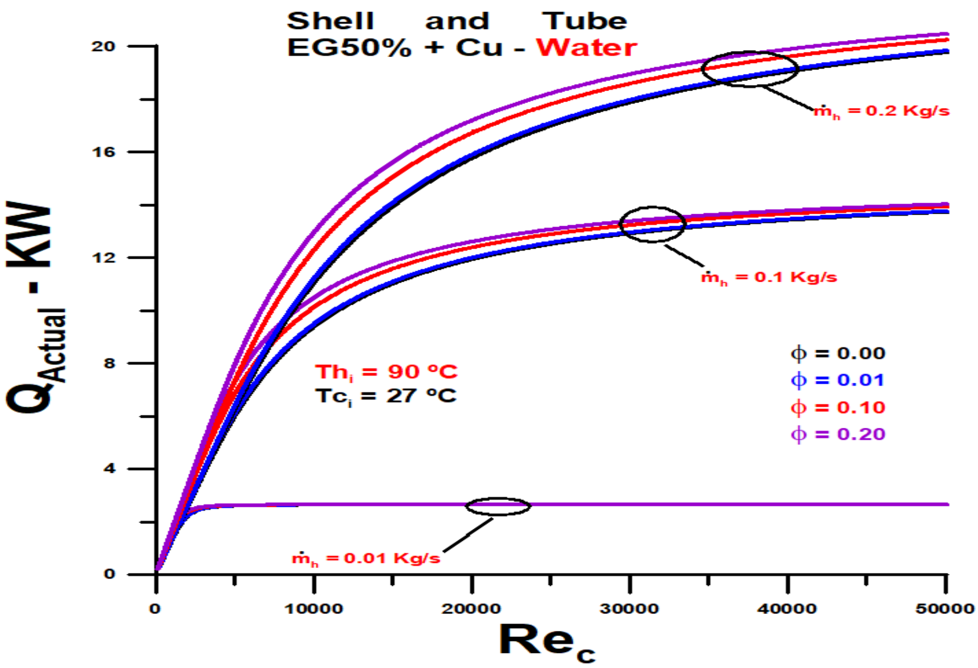

Figure 3 – Heat transfer rate versus Reynolds number in the shell

Figure 3 shows results for the heat transfer rate as a function of the Reynolds number associated with the nanofluid in the shell, with a volume fraction of the nanoparticles as a parameter. It can be seen that the volume fraction of the nanoparticles has no significant effect on the variation of the heat transfer rate to the mass flow of the hot fluid in the tube. However, considering the higher flow of water in the tube, equal to 0.2 Kg/s, it can be seen that the heat transfer rate is increasing to the increase in the fraction of the nanofluid.

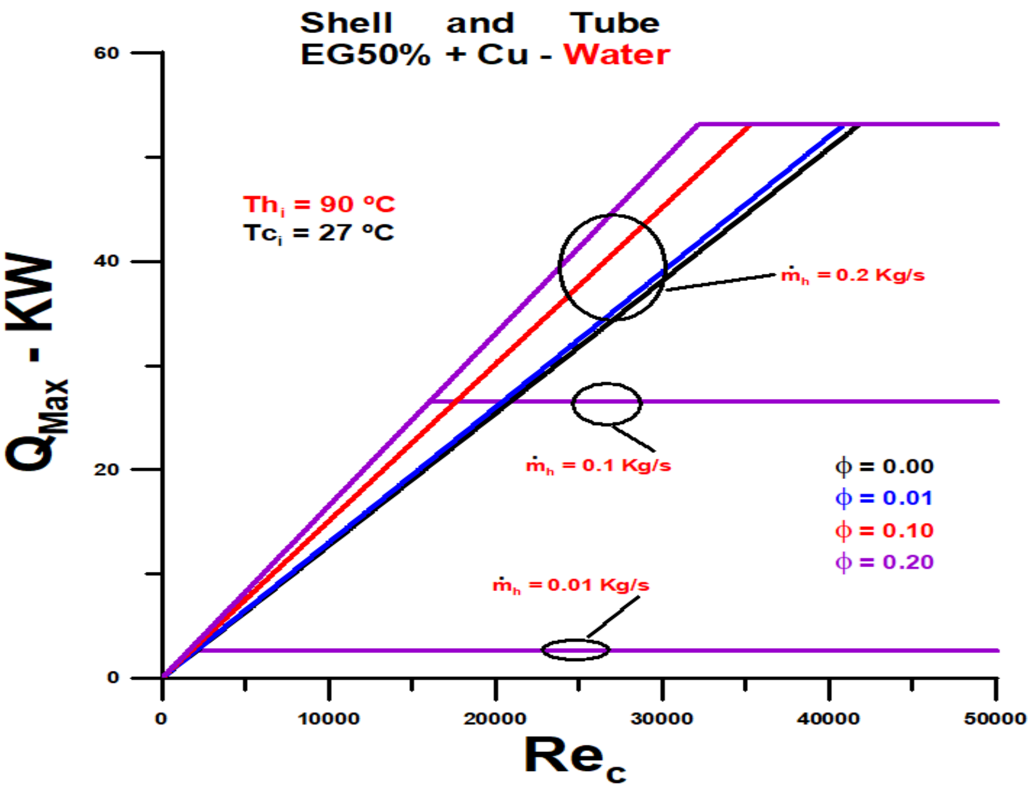

Figure 4– Maximo heat transfer rate versus Reynolds number in the shell

Figures 3 and 4 show that the heat transfer rate for mass flow in the pipe equal to 0.01 Kg/s is remarkably close to the maximum. This result has a significant influence on the outlet temperature of the hot fluid, as will be discussed below. For water flow in the tube equal to 0.2 Kg / s, the influence of the nanofluid is clear, with the maximum heat transfer rate being achieved for smaller Reynolds numbers with the addition of the volume fraction, that is, with the laminarization process of the flow already mentioned.

Efficiency is shown in Figure 5. The lowest efficiency is observed for mass flow in the tube equal to 0.01 Kg / s, for Reynolds number in the hull above 10000. However, there is a local maximum around 2000 for the Reynolds number. For higher water flow rates in the pipe, the efficiency increases progressively. It reaches a maximum flow rate equal to 0.1 Kg / s around 10000 for the Reynolds number and volume fraction equal to 0.2. When the mass flow in the tube is higher, that is, equal to 0.2 Kg / s, the maximum is obtained for Reynolds number close to 25000. Once again, it is evident that the results related to thermal performance are strongly influenced by the flow of water in the tube, supported by the influence of the volume fraction.

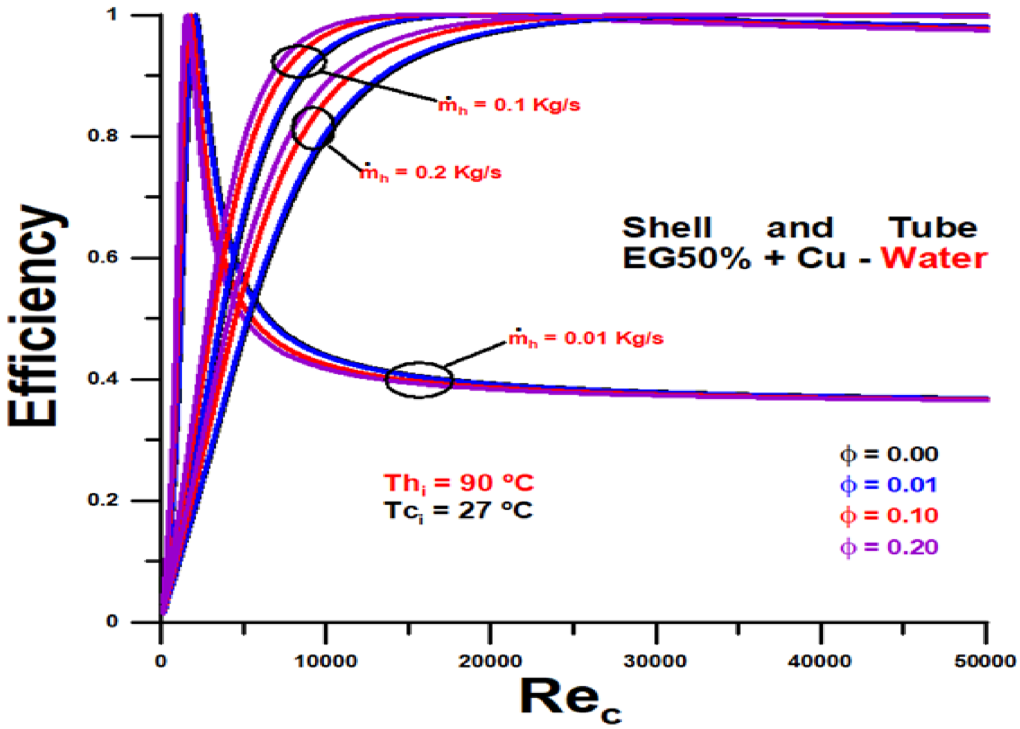

Figure 5 – Thermal efficiency versus Reynolds number in the shell

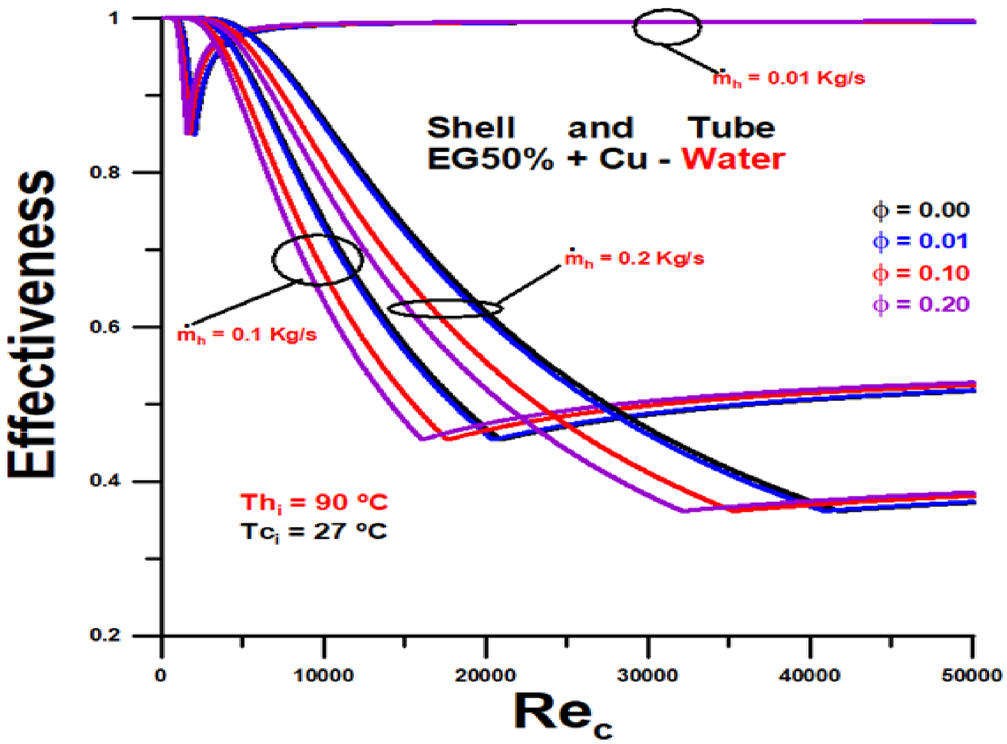

Figure 6– Effectiveness versus Reynolds number in the shell

The effectiveness, Figure 6, is maximum for flow in the pipe equal to 0.01 Kg / s, and, for this flow, there is no significant difference for different fractions in volume in the hull. The effectiveness decreases with the increase in mass flow in the tube, with evident influence for different fractions in volume in the shell. When the thermal capacity, Cmin, changes from one fluid to another, there is a precise inflection in the results. The inflection point is dependent on the mass flow rates and the volume fraction of the nanoparticles. For higher volume fractions, lower Reynolds number values in the hull. It is important to note that the effectiveness is high in all situations analyzed when the number of Reynolds in the shell is in the laminar flow range.

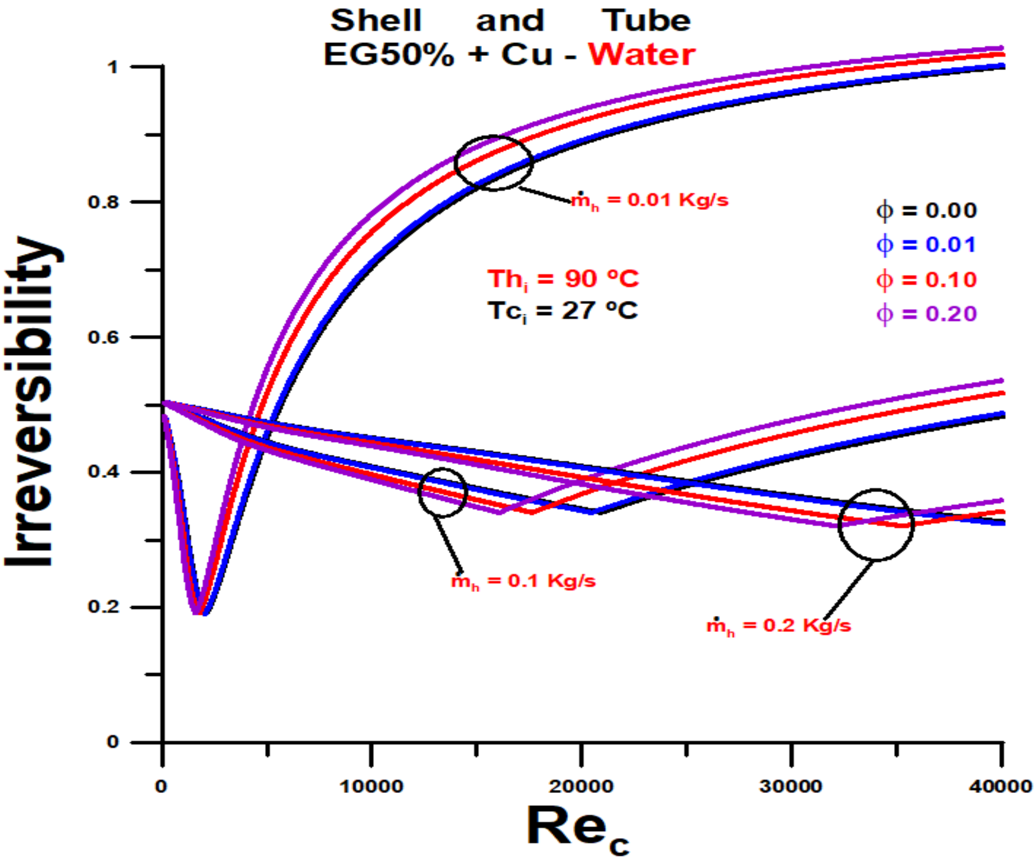

Figure 7– Irreversibility versus Reynolds number in the shell

For mass flow equal to 0.01 Kg / s in the shell, the irreversibility, Figure 7, decreases in the laminar flow region and increases in the turbulent flow region, reaching a maximum close to 40,000 for the Reynolds number. For higher flow rates in the tube, the irreversibility is relatively low, presenting an inflection point when the thermal capacity changes, Cmin, from one fluid to another. The influence of the volume fraction of the nanoparticles is noticeable in all situations under analysis.

The outlet temperature in the shell is increased in a laminar regime for all situations under analysis, Figure 8, and decreases in the turbulent regime. For lower flow in the tube, equal to 0.01 Kg / s, and high flow in shell, with Reynolds number above 20000, the outlet temperature is close to the inlet temperature of the nanofluid. The temperature difference increases with increasing water flow in the pipe. The influence of the volume fraction of the nanoparticles is noticeable for higher mass flow rates in the tube.

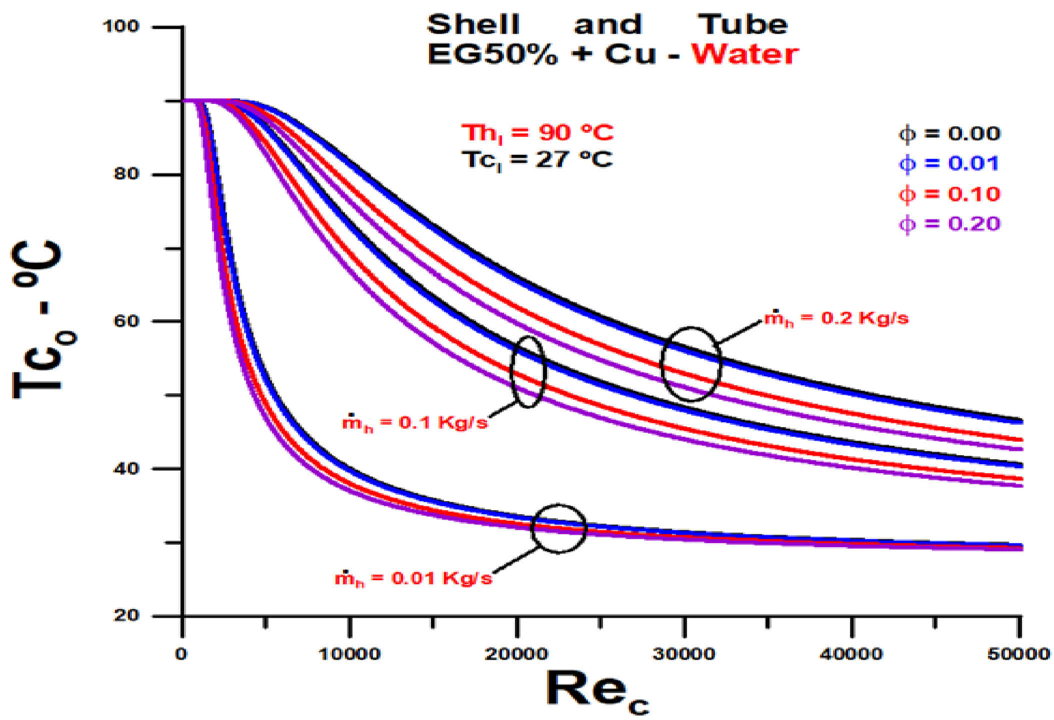

Figure 8 – Cold output temperature versus Reynolds number in the shell

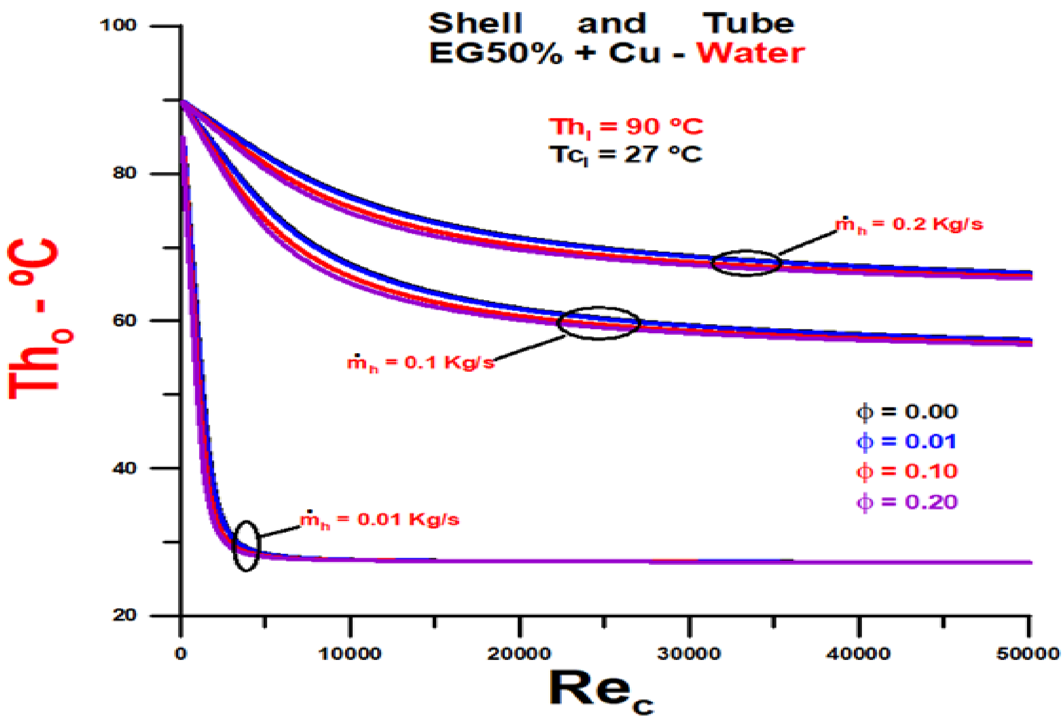

Figure 9 –Hot output temperature versus Reynolds number in the shell

Figure 9 shows the results for the leaving water temperature. The hot fluid outlet temperature is remarkably close to the cold fluid inlet temperature for mass flow in the tube equal to 0.01 Kg / s. For higher mass flow rates in the tube, the temperature difference, to the cold fluid inlet temperature, is increasing, without significant influence, in all cases, with the volume fraction of the nanoparticles.

In summary:

a - When it comes to cooling the water in the tube, using nanofluid in the hull, the best result is obtained when the flow in the tube is relatively low, and the flow is turbulent in the shell. The influence of the volume fraction of the nanoparticles is not significant. It is important to note that the effectiveness and irreversibility are high for lower water flows in the pipe, as well as the heat transfer rate is remarkably close to the maximum. The efficiency, however, is lower for lower water flows in the tube and the turbulent regime in the hull. Evidently, for lower outlet temperatures in the pipe, the most relevant quantities are effectiveness and irreversibility, when the heat transfer rate approaches the maximum;

b - When the flow of hot water to be cooled is significantly low, it is evident that an excellent option is to use the hot fluid in the pipe. Also, the use of nanoparticles does not change the outlet temperature values substantially, and the ideal Reynolds number is approximately equal to 10,000, that is, a relatively low mass flow in the hull, between 0.05 kg / s and 0.1 kg / s (Figure 2).

III.2 - Cold nanofluid (CuO) with Ethylene Glycol water-based fluid (50%) in the tube and hot water in the shell

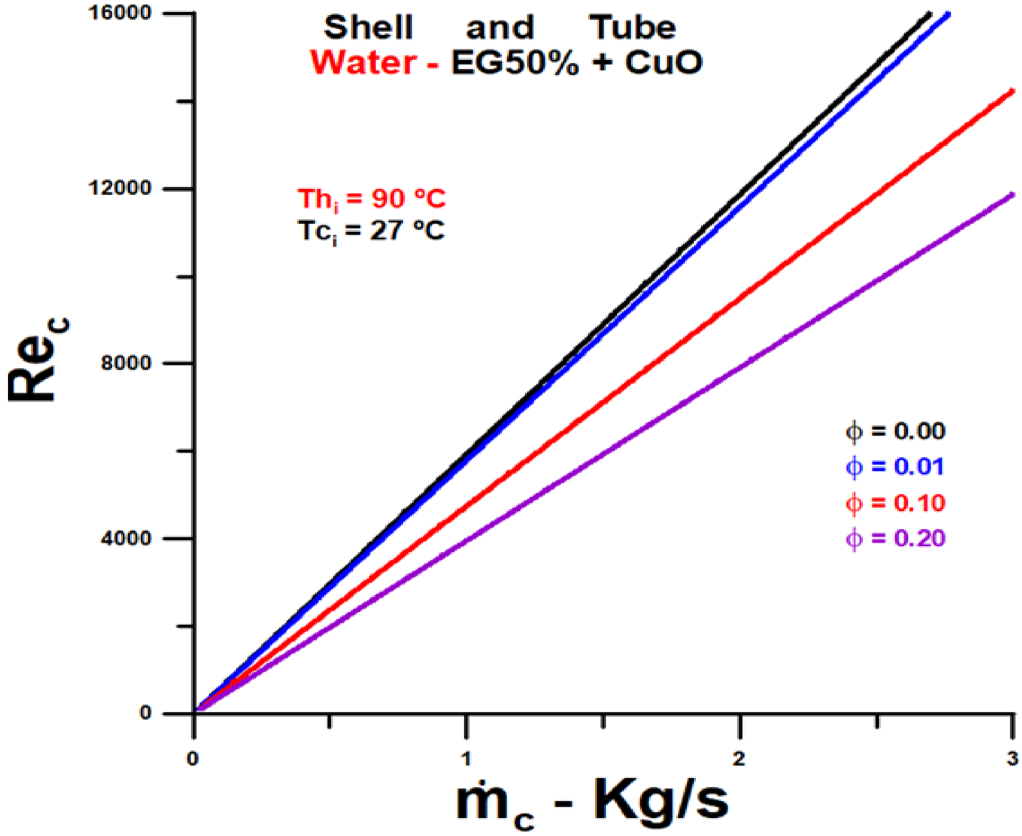

Figure 10 – Reynolds number versus mass flow rate in the tube

Figure 10 shows the number of Reynolds as a function of the mass flow of the nanofluid in the tube. Volume fractions of nanoparticles are introduced as a parameter. The flow laminarization process is noticeable, since the Reynolds number decreases when the volume fraction is increased, for the same mass flow of the nanofluid. This flow-related aspect has significant effects on the heat exchange process and the overall thermal performance of the heat exchanger.

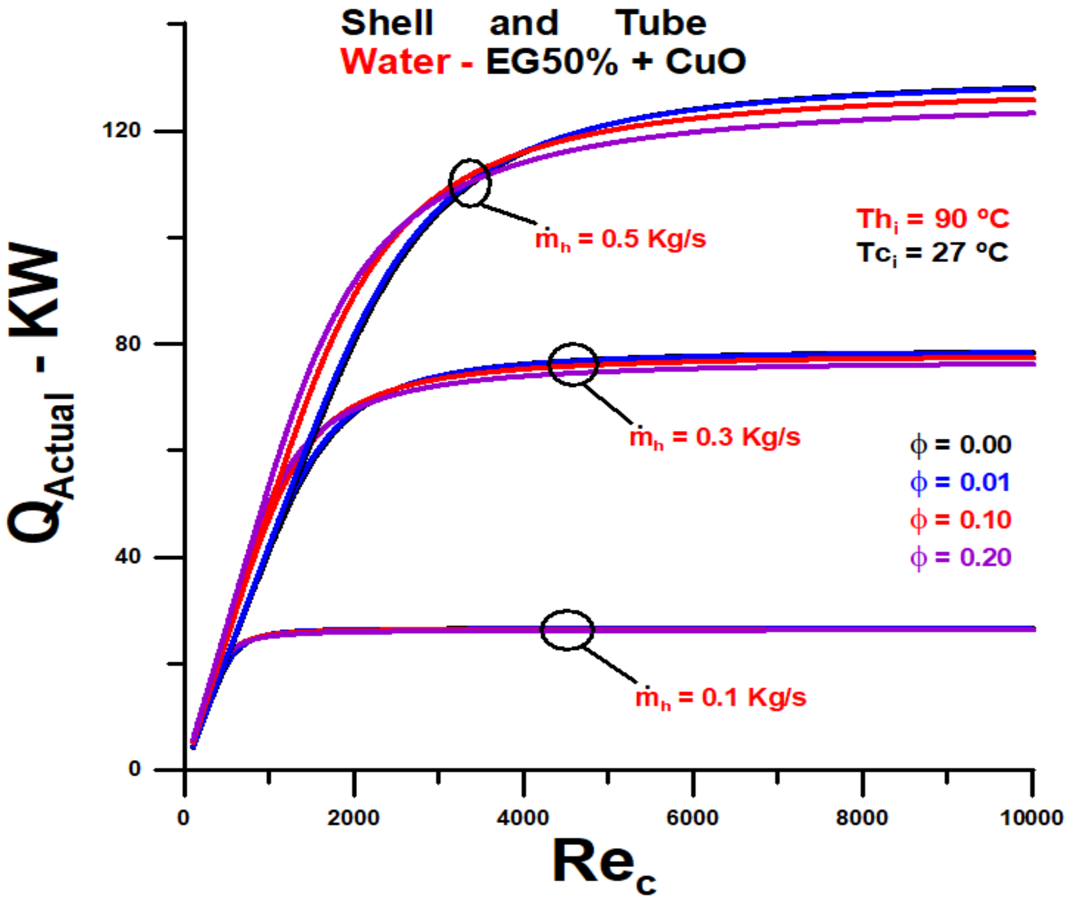

Figure 11 – Heat transfer rate versus Reynolds number in the tube

Figure 11 shows results for the heat transfer rate as a function of the Reynolds number associated with the nanofluid in the tube, with the volume fraction of the nanoparticles as a parameter. It can be seen that the volume fraction of the nanoparticles has no significant effect on the variation of the heat transfer rate to the change in the mass flow of the hot fluid in the hull. However, considering the higher flow of water in the shell, equal to 0.5 Kg / s, it can be seen that the heat transfer rate is higher for higher volume fraction when the flow is laminar, and the reverse occurs when the flow is in turbulent regime. This result reflects the fact that the conductive effect is more significant in the laminar regime, and the convective impact is in the turbulent regime.

Figures 11 and 12 show that the heat transfer rate for mass flow in the shell equal to 0.1 Kg/s is remarkably close to the maximum. This result has a significant influence on the outlet temperature of the hot fluid, as will be discussed below. For water flow in the hull equal to 0.5 Kg / s, the influence of the nanofluid is clear, with the maximum heat transfer rate being achieved for smaller Reynolds numbers with the addition of the volume fraction, that is, with the laminarization process of the flow already mentioned.

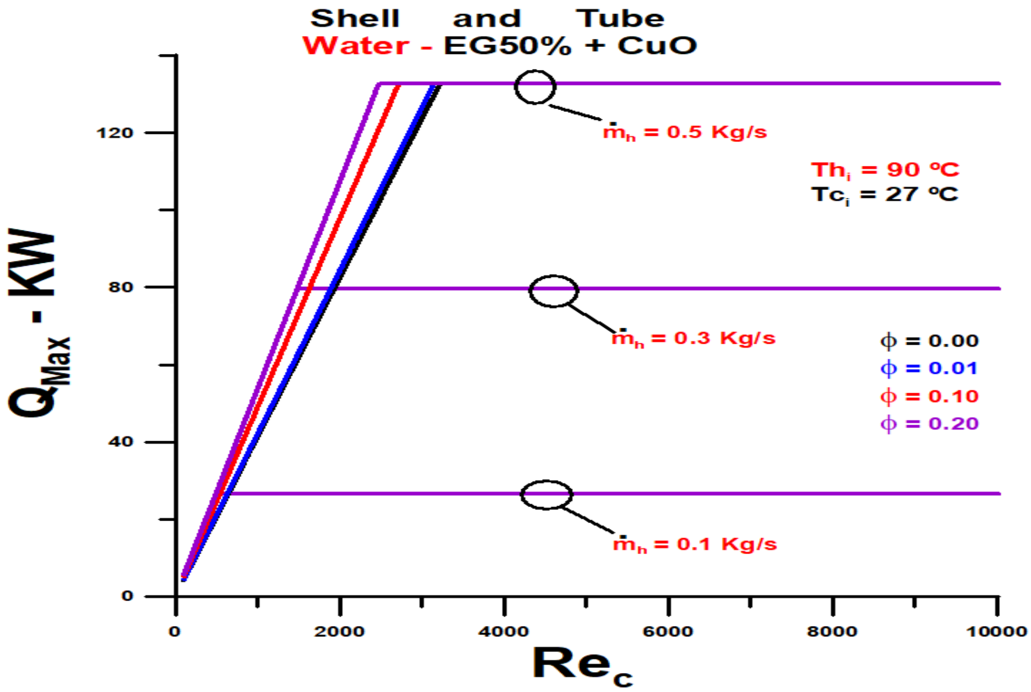

Figure 12 – Maximo heat transfer rate versus Reynolds number in the tube

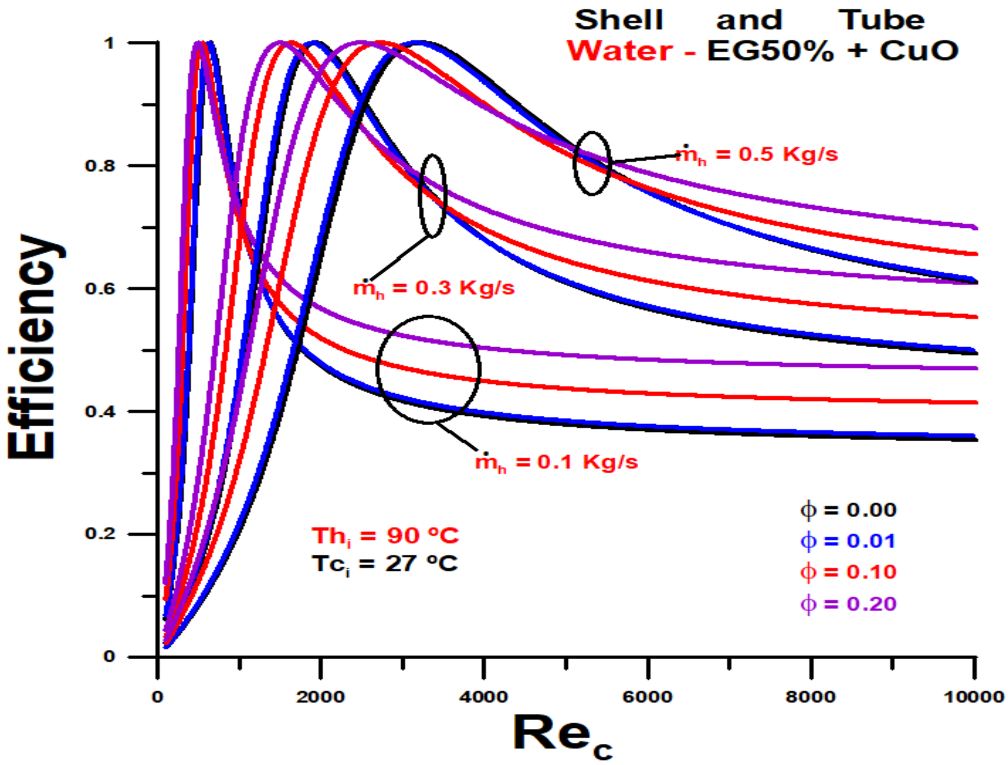

Figure 13 – Thermal efficiency versus Reynolds number in the tube

Efficiency is shown in Figure 13. The lowest efficiency is observed for mass flow in the tube equal to 0.1 Kg / s, for Reynolds number in the tube above 3000. However, there is a local maximum of around 1000 for the Reynolds number. For higher water flows in the hull, efficiency increases progressively. It reaches a maximum flow rate equal to 0.2 Kg / s between 1000 and 1500 for the Reynolds number and volume fraction equal to 0.2. When the mass flow in the tube is higher, that is, equal to 0.5 Kg / s, the maximum is obtained for Reynolds number between 3000 and 4000. The results related to the thermal performance are strongly influenced by the water flow in the hull, seconded by the influence of the volume fraction. However, for the situation under analysis, that is, the flow of hot water in the shell, the impact of nanoparticles is more accentuated, with a higher fraction in volume translating into greater thermal efficiency for the turbulent regime.

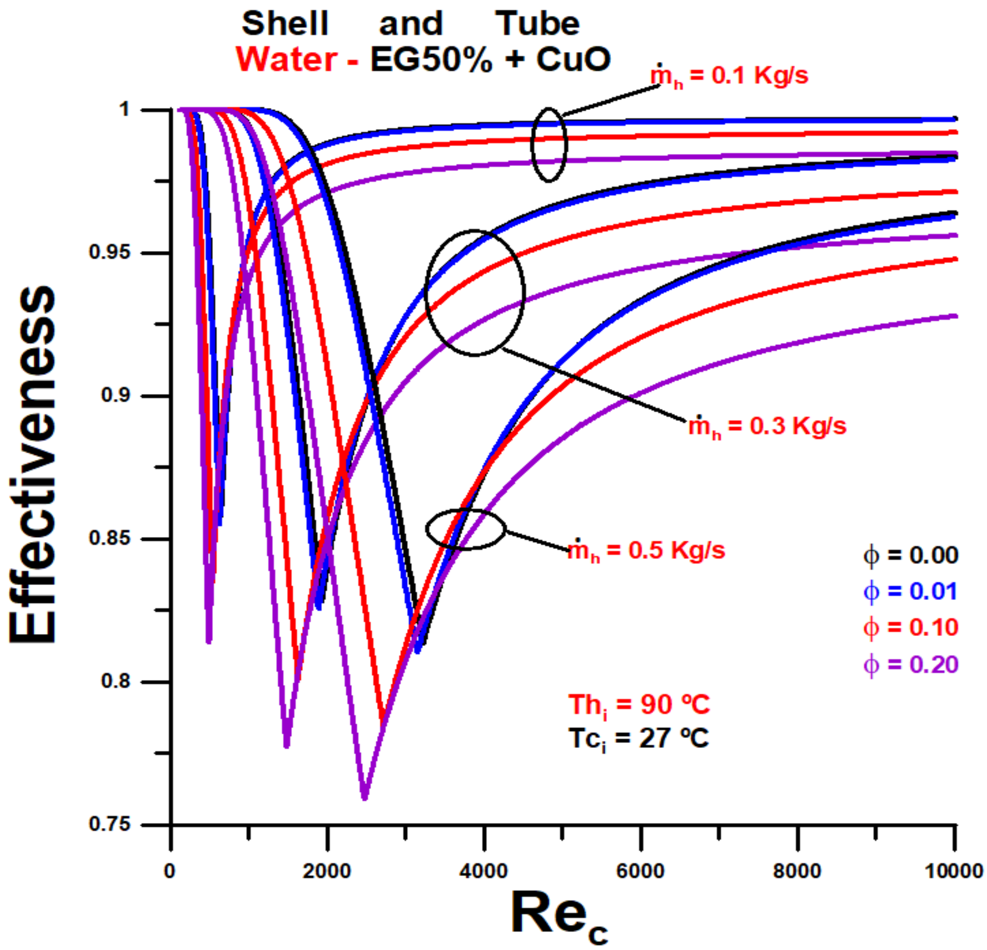

Figure 14 – Effectiveness versus Reynolds number in the tube

The effectiveness, Figure 14, is maximum for flow in the hull equal to 0.1 Kg / s. The effectiveness decreases with the increase in mass flow in the shell, with significant influence for different fractions in volume in the tube. When the thermal capacity, Cmin, changes from one fluid to another, there is a precise inflection in the results. The inflection point is dependent on the mass flow rates and the volume fraction of the nanoparticles. For higher volume fractions, lower values for the Reynolds number in the tube. The effectiveness is high in all situations analyzed when the number of Reynolds in the shell is in the laminar flow range.

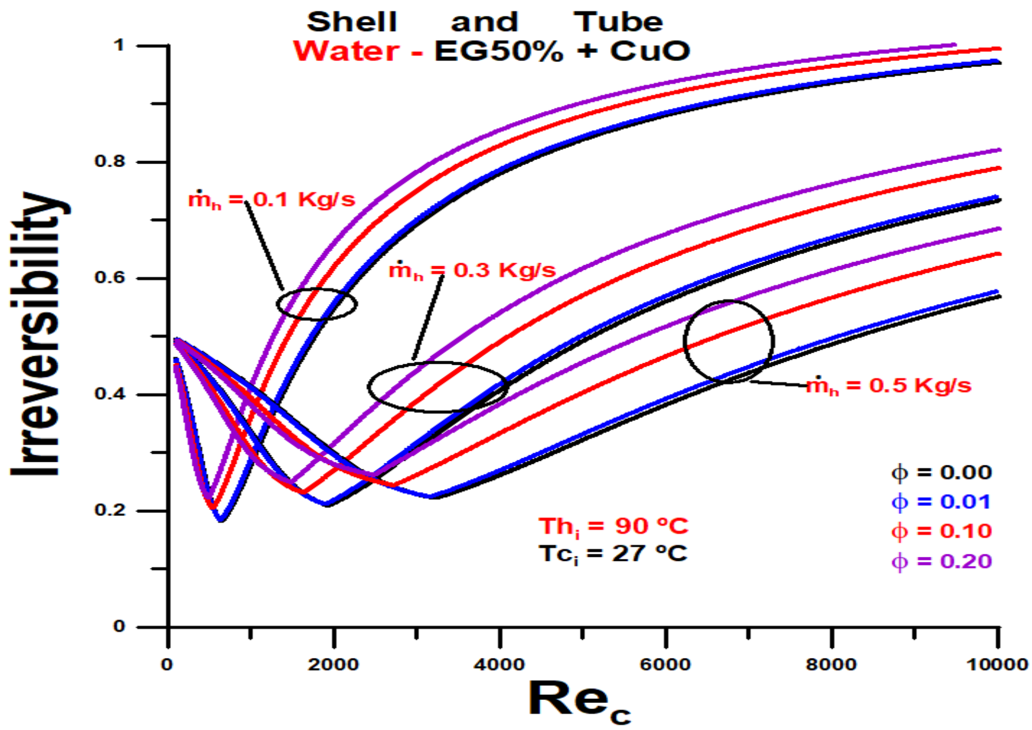

Figure 15 – Irreversibility versus Reynolds number in the tube

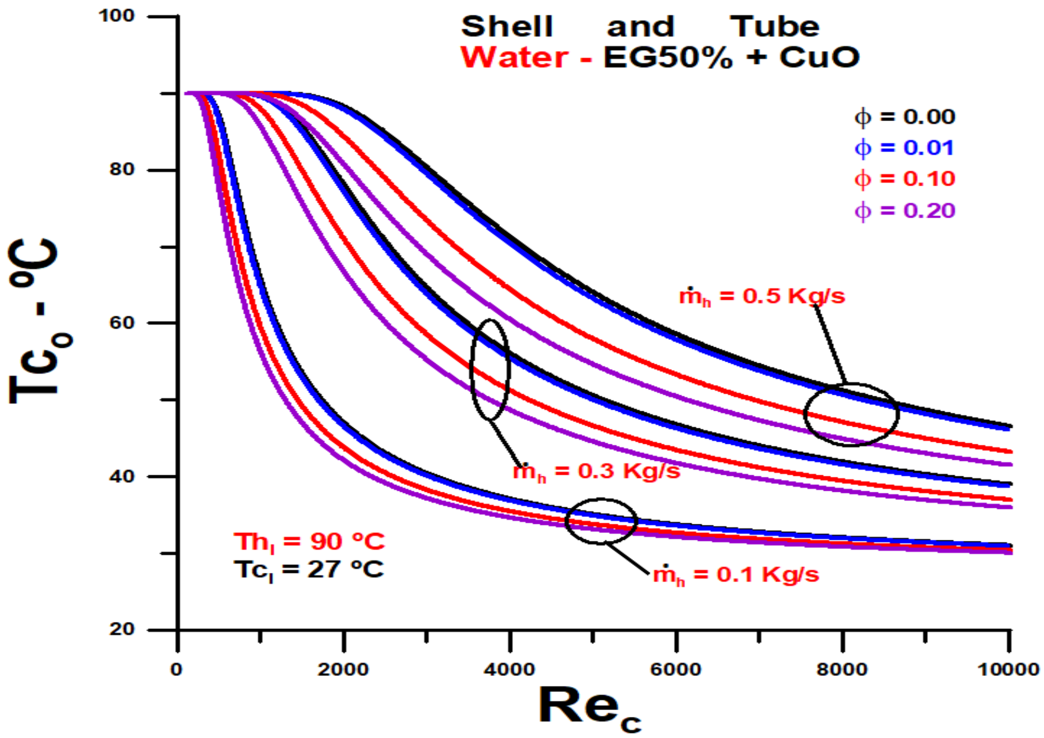

Figure 16 – Cold output temperature versus Reynolds number in the tube

For all mass flow rates in the tube, the irreversibility, Figure 15, is decreasing in the region of laminar flow. In the turbulent flow region, irreversibility increases, reaching a maximum close to 10,000 for the Reynolds number when the mass flow is equal to 0.1 Kg / s in the shell. For higher flows, the irreversibility is significantly lower. The influence of nanoparticles is noticeable in all situations analyzed, with irreversibility presenting an inflection point when there is an exchange of thermal capacity, Cmin, from one fluid to another.

The outlet temperature in the tube, cold fluid, is increased in a laminar regime for all situations under analysis, Figure 16, and decreases with the increase in the number of Reynolds. For lower flow in the shell, equal to 0.1 Kg / s, and high flow in the tube, with Reynolds number above 5000, the outlet temperature is close to the nanofluid inlet temperature. The temperature difference increases with increasing water flow in the shell. The influence of the volume fraction of the nanoparticles is relevant for higher mass flow rates in the tube.

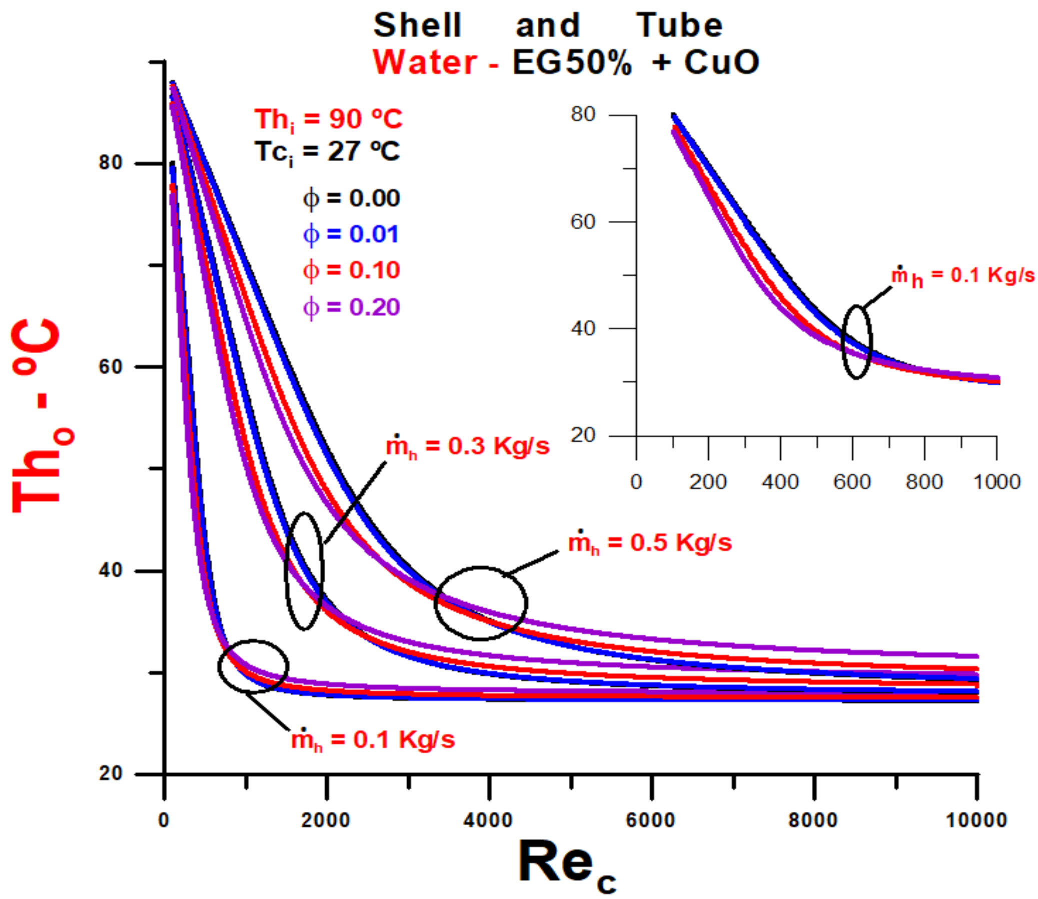

Figure 17 – Hot output temperature versus Reynolds number in the tube

Figure 17 shows the results for the hot water outlet temperature in the shell. The outlet temperature of the hot fluid is close to the inlet temperature of the cold fluid for mass flow in the tube equal to 0.1 Kg / s in the turbulent regime in the tube. For higher mass flow rates in the hull, the temperature difference, to the cold fluid inlet temperature, is increasing with a significant influence of the volume fraction of the nanoparticles. In the highlight, in the laminar regime and less hot water flow in the shell, it can be observed that the outlet temperature is lower for more significant fractions in the volume of the nanoparticles. In the latter case, the conductive effect is predominant in the heat exchange, due to the higher diffusivity of the nanoparticles. In a turbulent regime, where the convective influence is dominant, the lowest hot water outlet temperature is obtained for smaller volume fractions of the nanoparticles.

In summary:

a - when it comes to cooling the water in the shell, an excellent result is obtained when the flow is turbulent in the tube. The influence of the fraction by volume of the nanoparticles is significant, with emphasis on the laminar regime. The effectiveness and irreversibility are high for lower water flow in the hull, as well as the heat transfer rate is remarkably close to the maximum. Efficiency, however, is lower for lower water flows in the shell and turbulent flow in the pipe. Evidently, for lower outlet temperatures in the shell, the most relevant quantities are effectiveness and irreversibility, when the heat transfer rate approaches the maximum;

b - When the mass flow of hot water to be cooled is significantly high, it is evident that the best option is to use the hot fluid in the shell and, preferably, a turbulent regime in the tube.

IV. Conclusion

When it comes to obtaining lower outlet temperatures for the hot fluid, when using the type of hull and tube exchanger under analysis, one should consider the situation in which the heat transfer rate approaches the maximum. In this situation, effectiveness and irreversibility are high, compared to thermal efficiency.

For the situations under analysis, the heat transfer rate approaches the maximum for lower flows of the hot fluid in a turbulent regime for the cold fluid.

If the flow rate of the hot fluid is below 0.1 Kg / s and close to 0.01 Kg / s, the best option is that the hot fluid is in the tube, with Reynolds number in the hull between 5000 and 10000.

For hot fluid flow rates equal to or above 0.1 Kg / s, the best option is for the hot fluid to be in the hull, with Reynolds number in the tube between 2000 and 6000.

The influence of nanoparticles is greater when the flow of the hot fluid occurs in the shell, with higher temperatures of the hot fluid leaving for more significant fractions in the volume of the nanofluid in the turbulent regime in the tube.

The turbulent flow of nanoparticles in the tube does not translate into an advantage when what you want to achieve is the lowest possible temperature for the water.

References

[1]. A. Bejan (1987). "The thermodynamic design of heat and mass transfer processes and devices." Heat and Fluid Flow, Vol. 8, No. 4, pp. 258-276.

[2]. Ahmad Fakheri (2007). "Heat Exchanger Efficiency." Transactions of the ASME, Vol. 129, pp. 1268- 1276.

[3]. Maysam Molana (2016). "A Comprehensive Review on the Nanofluids Application in the Tubular Heat Exchangers." American Journal of Heat and Mass Transfer, Vol. 3 No. 5 pp. 352-381.

[4]. Gaurav Thakur and Gurpreet Singh (2017). "An experimental investigation of heat transfer characteristics of water-based Al2O3 nanofluid operated shell and tube heat exchanger with air bubble injection technique." International Journal of Engineering & Technology, 6 (4), pp. 83-90.

[5]. Pankaj K Pandey, Palak K Lakhani, Kapil Kumar, Parth Bohra, and Rounak Mishra (2017). "Heat Transfer Analysis of Shell and Tube Heat Exchanger using Al2O3/SiC Nanofluid." European Journal of Advances in Engineering and Technology, 2017, 4(8): 608-616.

[6]. Roopesh Tiwari and Govind Maheshwari (2017). "Effectiveness and Efficiency Analysis of Parallel Flow and Counter Flow Heat Exchangers." IJAIEM, Volume 6, Issue 6, pp. 314-319.

[7]. Ali Heydari, Mostafa Shateri, and Sina Sanjari (2015). "Performance investigation of baffled shell and tube heat exchanger using different nanofluids." 16th Conference on Fluid Dynamics, fd2015, Nov. 17-19, Razi University, Kermanshah, Iran.

[8]. Yue Sun, Xinting Wang, Rui Long, Fang Yuan, and Kun Yang (2019). "Numerical Investigation and Optimization on Shell Side Performance of a Shell and Tube Heat Exchanger with Inclined Trefoil-Hole Baffles." Energies, 12, 4138.

[9]. K.Y. Leong, R. Saidur, M. Khairulmaini, Z. Michael, A. Kamyar (2012). "Heat transfer and entropy analysis of three different types of heat exchangers operated with nanofluids." International Communications in Heat and Mass Transfer, 39, pp. 838–843.

[10]. Omid Mahian, Ali Kianifar, Clement Kleinstreuer, Moh'd A. Al-Nimr, Ioan Pop, Ahmet Z. Sahin, Somchai Wongwises (2013). "A review of entropy generation in nanofluid flow." International Journal of Heat and Mass Transfer 65 (2013) 514–532.

[11]. M. Karuppasamy, R. Saravanan, M. Chandrasekaran, V. Muthuraman (2019). 'Heat transfer magnification in a double tube heat exchanger with nanofluid and tube insert." International Journal of Mechanical Engineering and Technology (IJMET), Volume 10, Issue 09, September 2019, pp. 120-127.

[12]. Jaafar Albadr, Satinder Tayal, and Mushtaq Alasadi (2013). "Heat transfer through heat exchanger using Al2O3 nanofluid at different concentrations." Case Studies in Thermal Engineering, 1, pp. 38–44.

[13]. Ahmed A. H. Mostafa, Essam E. Khalil, Gamal El-Hariry Waleed A. Abdelmaksoud, and Emad M. Saad (2017). "Shell and Tube Heat Exchanger Performance." AIAA Propulsion and Energy Forum, 10-12 July, Atlanta, GA, 15th International Energy Conversion Engineering Conference, pp. 1-13.

[14]. Salah Almurtaji, Naser Ali, Joao A. Teixeira, and Abdulmajid Addali (2020). "On the Role of Nanofluids in Thermal-Hydraulic Performance of Heat Exchangers—A Review." Nanomaterials, 10, 734; DOI:10.3390/nano10040734.

[15]. Élcio Nogueira (2020). "Efficiency and Effectiveness Concepts Applied in Shell and Tube Heat Exchanger Using Ethylene Glycol-Water Based Fluid in the Shell with Nanoparticles of Copper Oxide (CuO)." Journal of Materials Science and Chemical Engineering, 2020, 8, 1-12. DOI: 10.4236/msce.2020.88001

Élcio Nogueira , et. al. "Determination of the best configuration of a given shell and tube heat exchanger for water cooling using nanofluid (CuO) and concepts of efficiency, effectiveness, and irreversibility." *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)*, 17(4), 2020, pp. 24-40.