IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 18, Issue 1 Ser. II (Jan. – Feb. 2021), PP 56-67
www.iosrjournals.org

Experimental versus Theoretical Analysis of a Compact Flat Finned Tube Heat Exchanger using Thermal Effectiveness.

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Abstract: *The present work uses water as a heat exchanger fluid in a compact finned tube heat exchanger (automotive radiator) as a preliminary experimental procedure to achieve the use of nanoparticles to increase the heat exchanger's thermal efficiency and effectiveness under analysis. The objectives are to carry out measurements of thermal quantities related to water in laminar regime in the tubes; to compare experimental results with theoretical results obtained through the theory of effectiveness (ε - NUT). The air is induced to pass through the free section of the radiator through a wind tunnel. The physical quantity measurement system is coupled with a data acquisition system. The comparison of theoretical and experimental results demonstrates the consistency of the recommended experimental procedure.*

Key Word*: Compact heat exchanger; Automotive radiator; Effectiveness theory (ε – NUT); Flat finned tube heat exchanger.*

Date of Submission: 14-01-2021 Date of Acceptance: 29-01-2021

I. Introduction

The laboratory of Thermal and Hydraulic Machines of the Department of Mechanical and Energy of the University of the State of Rio de Janeiro, located at the Faculty of Technology, unit of Resende, endeavors to implant an experimental system of thermal and hydraulic measures in a compact heat exchanger (automotive radiator). The project's final objective is to use nanoparticles to experimentally analyze the thermal efficiency and effectiveness of the mentioned heat exchanger, with the fluid flowing in a laminar regime in the tubes. We use water as a heat exchanger fluid in the tubes as a preliminary experimental procedure to reach the current work's final objective. The effectiveness theory (ε - NUT) is used to compare the experimental results, demonstrating the recommended experimental procedure's consistency.

An automotive radiator plays an essential role in transferring heat from engine parts to the environment. It is categorized as a compact crossflow heat exchanger designed to transfer heat from the hot coolant to the ambient air. Heat transfer occurs from the refrigerant to the tubes and from the tubes to the air through the fins. The overall heat transfer coefficient is a function of the convective transfer coefficient of air and the refrigerant used. When the engine of an automotive vehicle runs at high speeds, the heat generated increases dramatically, and the cooling process must be efficient and effective. In this case, the radiator must be appropriately sized, and one of the relevant aspects is the refrigerant to be used and the flow regime in the tubes [1].

Elcio Nogueira [2] presents a theoretical method based on effectiveness (ε-NTU) to analyze finned automotive radiators' thermal performance. The refrigerant fluid consists of a suspension of silver nanoparticles in ethylene glycol. The theoretical procedure showed promising results. The study's main conclusion was the demonstration that it is possible to save on using the compact heat exchanger, reducing costs and storage space for the refrigerant.

C. Selvam et al. [3] report an increase in the overall heat transfer coefficient in an automobile radiator using graphene nanoparticles in a mixture of water and ethylene glycol. The results were obtained through experiments on an automobile radiator for a wide range of nanofluid mass flow rates. He concludes that the nanofluid's convective heat transfer coefficient leads to the radiator's better thermal performance. The increase in pressure drop is significantly influenced by nanofluid's mass flow rate based on graphene nanoplates.

Amit S. Gulhaneágua and S.P. Chincholkar [4] report that water and ethylene glycol were widely used as conventional refrigerants in automobile radiators. However, these refrigerants have low thermal conductivity. They present an experimental study focused on applying water-based nanofluid (Al2O3) in lower concentrations in a car radiator. The radiator used 36 vertical tubes with cylindrical cross-section, in crossflow with the air, at a constant speed. In the research carried out, the two-step method is used to prepare the nanofluid, and nanoparticles were dispersed in demineralized water in different volume concentrations, without any dispersant or stabilizer. The results obtained show that the coefficient of heat transfer by convection with nanofluid is higher than that of the base fluid. The experimental study is valid using a mathematical model, which has a confidence level of 96.74%.

D. Ravi and B. K. C. Ganesh [5] present an experimental study on an automotive radiator's performance using water and copper oxide nanofluid. Comparison of radiator performance is performed between pure water and nanofluids.

Madhusree Kole, T.K. Dey [6] Madhusree Kole, T.K. Dey [6] presents experimental results for viscosity in a dispersion of alumina nanoparticles (<50 nm) in commercial vehicle refrigerant using oleic acid. The nanofluid prepared with the surfactant showed stability for more than 80 days. The results demonstrate that the refrigerant is transformed into a non-Newtonian fluid with the addition of alumina nanoparticles. However, they argue that the nanofluid's viscosity can be predicted based on a theoretical model that considers the effect of the nanoparticles' Brownian motion.

Maysam Molana (7) argues that nanofluids in automotive refrigeration systems can achieve high efficiency in automotive radiators. Conclude that nanofluids' use improves thermal efficiency but, on the other hand, reduces the pumping power.

Luiz Carlos Cordeiro Junior and Élcio Nogueira [8] present a theoretical analysis on the influence of the refrigerant flow regime containing nanoparticles in an automotive radiator. It is shown that the heat transfer rate increases by adding silver nanoparticles in a solution of ethylene glycol and water. The study justifies using silver nanoparticles and ethylene glycol as a refrigerant in the radiator of motor vehicles, considering a significant increase in thermal performance.

II. Objectives

To present an experimental procedure, developed to measure thermal quantities related to a compact flat finned tube heat exchanger (Automotive radiator), with a laminar water regime in the tubes. Compare experimental results with theoretical results obtained through the theory of effectiveness (ε - NUT).

III. Methodology

The experimental setup consists of a compact flat finned tube heat exchanger, with essential geometric specifications represented by Table 1 and Figure 1 below. The air is induced to pass through the radiator's free section through a wind tunnel, with flow variation obtained through a frequency inverter coupled to the fan's electric motor. The water flow is obtained through a Venturi meter attached to the radiator inlet pipe. The Reynolds number associated with the mass flow rate of water is in the laminar regime range. The air and water flow rates and some quantities related to the experiment are specified in Table 2. A general scheme of the experimental apparatus is represented in Figure 4. The measurement system of physical quantities is coupled to an acquisition system of data (Figures 2 and 4).

Table 1 – Dimensions of the compact heat exchanger - Automotive radiator

SI. No	Parameter	Value
1	Lenght of the radiator	0,560 m
2	Height of the radiator	0,378 m
3	Width of the radiator	0,025 m
4	Number of tubes	59
5	The hydraulic diameter of the Tube - Dhw	0,00286 m
6	Flow passage hydraulic diameter - Dha	0.00351 m
7	The thickness of the fin	0.00005 m
8	The thickness of the tube	0.000102 m
9	Total air heat transfer area	3.98 m^2
10	Fin heat transfer area	3.16 m^2
11	Fin area/Total area	0.795
12	The hydraulic diameter of the wind tunnel	0.15 m

Table 2 – Variation in some experimental parameters

Parameter	Variation
Mass flow rate of the water	6.8 10-4; 1.4 10-3; 2.56 10-3; 3.49 10-3 – kg/s
Mass flow rate of the air	0.20; 0.24; 0.27; 0.31 – kg/s
The bulk temperature of the wall	(48.12 – 39.71) - ºC
The bulk temperature of the air	(41.66 – 36.53) - ºC
Average air velocity in the wind tunnel	9.81; 11.48; 13.17; 14.85 – m/s
Average ambient air velocity	0.87; 1.02; 1.17; 1.32 – m/s

III.1 - Characterization of the experimental setup

Figure 1- Compact flat finned tube heat exchanger (Automotive radiator)

Figure 2 - Overview of the experimental apparatus

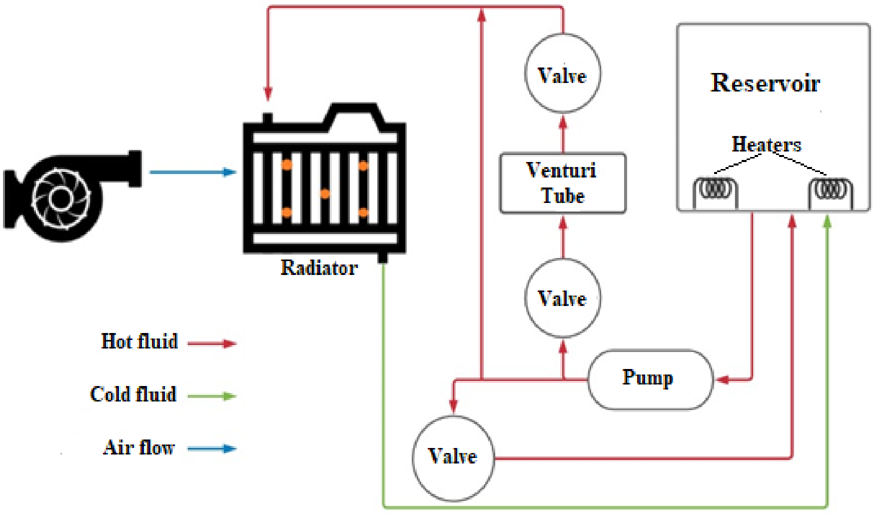

Figure 3 - Physical layout of the experimental setup

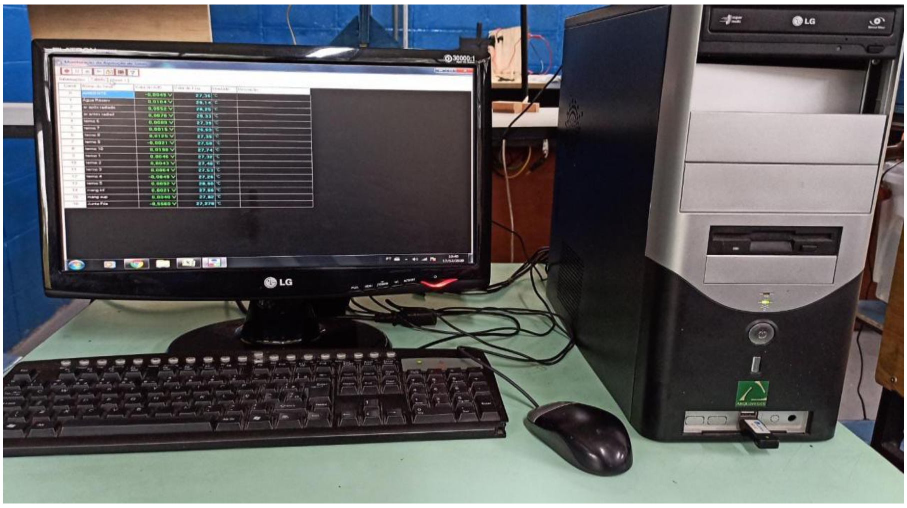

Figure 4 - Data acquisition system (LYNX system)

III.2 – Experimental procedure

The heat transfer rate is given by:

$$\dot{q} = \dot{m}_W Cp_W (T_{wi} - T_{wo}) \quad \mathbf{01}$$

at where T_{wo} and T_{wi} are the output and input water temperature, respectively.

$$\dot{q}_{max} = C_{min}(T_{wi} - T_{ai}) \quad \mathbf{02}$$

at where $\dot{q}_{max}$is the maximum heat transfer rate and $\mathbf{C_{min}}$is theminimum heat capacity between C_w and C_a.

The convective heat transfer of water is given by:

$$h_{Expw} = \frac{\dot{q}}{nA_{sw}(T_{bw} - T_{bs})} \quad \mathbf{03}$$

$$T_{bw} = \frac{T_{wi} + T_{wo}}{2} \quad \mathbf{04}$$

$$T_{bs} = \frac{T_{si} + T_{so}}{2} \quad \mathbf{05}$$

at whereT_{bw}, $T_{bs}$are the bulk temperatures of the water and wall, respectively; T_{so} and T_{si} are the output and input temperature of the surface radiator, respectively; A_{sw} is the surface area and n is the number of tubes.

The convective heat transfer of the airside is given by:

$$h_{Expa} = \frac{\dot{q}}{A_{sa}(T_{bs} - T_{ba})} \quad \mathbf{06}$$

at where A_{sa} and T_{ba} are the internal air side area of the radiator and bulk temperature of the air, respectively.

$$\mathbf{T_{ba}} = \frac{T_{ai} + T_{ao}}{2} \quad \mathbf{07}$$

Nusselt numbers of water and air are determined by:

$$Nu_{Expw} = \frac{h_{Expw} D_{hw}}{k_w} \quad \mathbf{08}$$

$$Nu_{Expa} = \frac{h_{Expa} D_{ha}}{k_a} \quad \mathbf{09}$$

at where D_{hw}, D_{ha}, k_w, k_a are hydraulic diameters and heat conductivity of water and air, respectively.

Finally, the experimental effectiveness is given by:

$$\varepsilon_{Exp} = \frac{\dot{q}}{\dot{q}_{max}} \quad \mathbf{10}$$

III.3 - Theoretical analysis

The physical properties for air and water used in implementing the theoretical model are shown in Table 3 below.

Table 3 - Physical properties of air and water

Properties		Air	Water
Pr	---	0,707	4,34
Cp	J/(kg°C)	1005,7	4178
k	W/(m°C)	0,0265	0,628
ρ	kg/m³	1,177	994,59
μ	kg/(m.s)	0,0000198	0,00065444

For the laminar flow regime in the water, for the thermal input region under development, the theoretical Nusselt number is given by Élcio Nogueira [10]:

$$Nu_w = 1.409019812d0 Z_w^{(-0.3511653489)} \;; \quad 10^{-5} \le Z_w < 10^{-3} \quad \mathbf{11}$$

$$Nu_w = 1.519296981d0 Z_w^{(-0.3395483303d0)} \;; \; 10^{-3} \le Z_w < 10^{-2} \quad \mathbf{12}$$

$$Nu_w = 10.8655 - 570.4671787 Z_w + 28981.67578 Z_w^2 - 950933.9838 Z_w^3 + 20237498.47 Z_w^4 - 276705269.6 Z_w^5 + 2340349265 Z_w^6 - 1.112482493^{10} Z_w^7 + Z_w^8; \; 10^{-2} \le Z_w \le 10^{-1} \quad \mathbf{13}$$

$$Nu_w = 5.261d0 - 19.93019048 Z_w + 139.4921627 Z_w^2 - 605.9954034 Z_w^3 + 1716.100694 Z_w^4 - 3217.96875 Z_w^5 + 3954.86111 Z_w^6 - 3056.051587 Z_w^7 + 1344.246031 Z_w^8 - 256.2830687 Z_w^9; 10^{-1} \le Z_w \le 10^0 \quad \mathbf{14}$$

at where $Z_w = \frac{\alpha_w z}{\bar{u}_w D_{hw}^2}$; α_w is the thermal diffusivity of water; z is the axial coordinate; $\bar{u}_w$ is average velocity.

Then,

$$\mathrm{h_w} = \frac{Nu_w k_w}{D_{hw}} \quad \mathbf{15}$$

The convective heat transfer coefficient of the air, ha, is obtained by:

$$h_a = J \frac{G_a c_{pa}}{Pr_a^{2/3}} \quad \mathbf{16}$$

$$G_a = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma_a A_{fr}} \quad \mathbf{17}$$

$$Re_a = \frac{G_a D_{ha}}{\mu_a} \quad \mathbf{18}$$

and, approximately

$$\mathrm{J} = 0.0942 Re_a^{-0.35925} \;; 400 \le Re_a \le 3000 \quad \mathbf{19}$$

J is the Colburn factor, obtained by Kays and London [9; Figure 10.5].

The fin efficiency is obtained by:

$$\eta = \frac{tg\mathrm{h}(mL)}{mL} \quad \mathbf{20}$$

at were,

$$mL = \sqrt{2h_a/k_a t} \quad \mathbf{21}$$

The efficiency weighted by the area is given by:

$$\eta' = \beta\eta + 1 - \beta \qquad (22)$$

at where:

$$\beta = \frac{Finarea}{totalarea} \qquad (23)$$

The global convective heat transfer is given by:

$$\frac{1}{A_a U_a} = \frac{1}{\eta' h_a} + \frac{1.0}{(A_{med}/A_a)K_{al}} + \frac{1}{(A_w/A_a)h_w} \qquad (24)$$

at where:

$$A_{med} = \frac{A_a + A_w}{2.0} \qquad (25)$$

and

$$\frac{A_w}{A_a} = \frac{water\ side\ heat\ transfer\ area}{air\ side\ heat\ transfer\ area} \qquad (26)$$

The number of thermal units, NTU, is obtained by

$$NTU = \frac{A_a U_a}{C_{min}} \qquad (27)$$

Finally, we have the theoretical effectiveness:

$$\varepsilon_{Teo} = 1 - exp\left[\begin{array}{c}\left(\frac{c_{min}}{c_{max}}\right)^{-1}(NTU)^{0,22} \\ \left\{exp\left[-\frac{c_{min}}{c_{max}}(NTU)^{0,78}\right] - 1\right\}\end{array}\right] \qquad (28)$$

IV. Results and discussion

Figure 5 presents the data's interpolation in Figure 10.95 of the text by W. M. Kays and A. L. London [9], for a Reynolds number range between 400 to 3000. The Colburn factor is essential in determining the coefficient of heat transfer by convection in the air, according to Equation 16. The data provided in the reference cited are those that best match the physical characteristics of the radiator used in the experiment.

The theoretical coefficient of heat transfer by convection in the air is represented by Figure 6. The values presented are average values, considering the Reynolds number range of the water in the tubes.

The average Nusselt number, associated with the heat transfer coefficient by convection in the air, is represented by Figure 7.

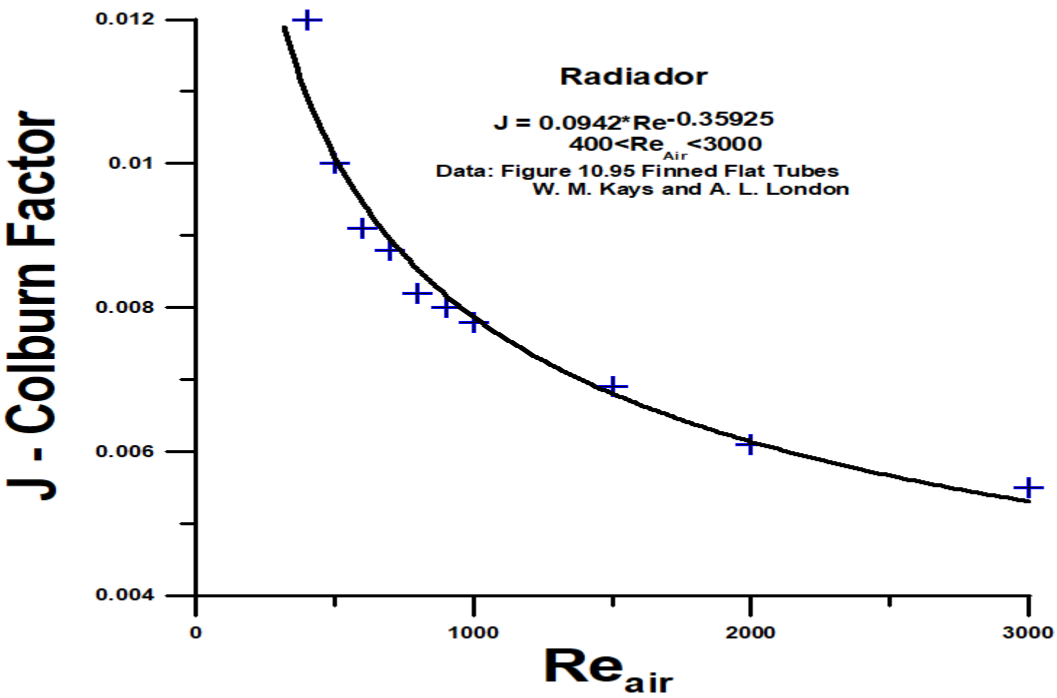

Figure 5 – Colburn factor versus Reynolds number

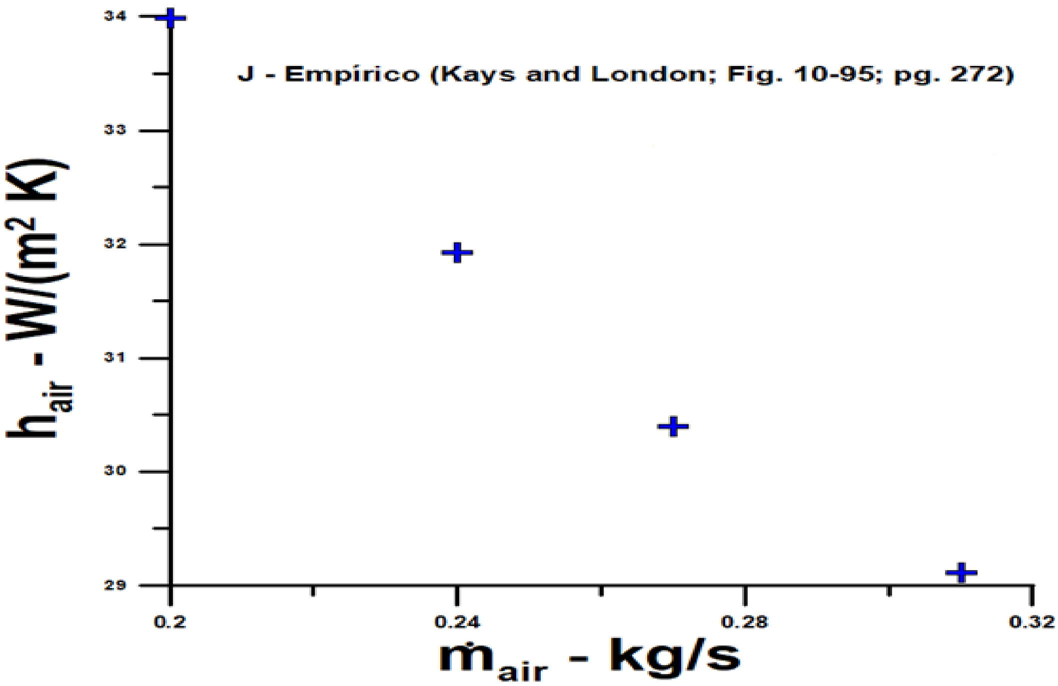

Figure 6 – Convection heat transfer coefficient of theair versus mass airflow

Table 4 – Summary of results for convection coefficient and Nusselt number in water

h_w – Convection heat transfer coefficient (W/m² K)								
	Re_w=377		**Re_w=780**		**Re_w=1425**		**Re_w=2000**	
$\dot{m}_{ar}$	*Exp*	*Teo*	*Exp*	*Teo*	**Exp**	**Teo**	**Exp**	**Teo**
0.31	906	1021	921	1021	934	1021	949	1021
0.27	1318	1381	1381	1379	1420	13811481	1381	
0.24	1494	1663	15441664	1601	16641661	1664		
0.20	1568	1933	1701	1933	1759	19331874	1933	
Nu_w - Nusselt Number								
	Re_w=377		**Rew=780**		**Re_w=1425**		**Re_w=2000**	
$\dot{m}_{ar}$	*Exp*	*Teo*	*Exp*	*Teo*	*Exp*	*Teo*	*Exp*	*Teo*
0.31	4.17	4.70	4.24	4.70	4.30	4.70	4.37	4.70
0.27	6.07	6.36	6.356.36	6.54	6.36	6.82	6.36	
0.24	6.88	7.66	7.11	7.66	7.37	7.66	7.65	7.66
0.20	7.22	8.90	7.83	8.90	8.108.90	8.63	8.90	

A numerical synthesis of the results obtained for convection heat transfer coefficient and Nusselt number in the tubes is shown in Table 4. These same results are partially shown in Figures 8 to 11. The maximum relative error obtained by comparison between experimental and theoretical values is 13.7%.

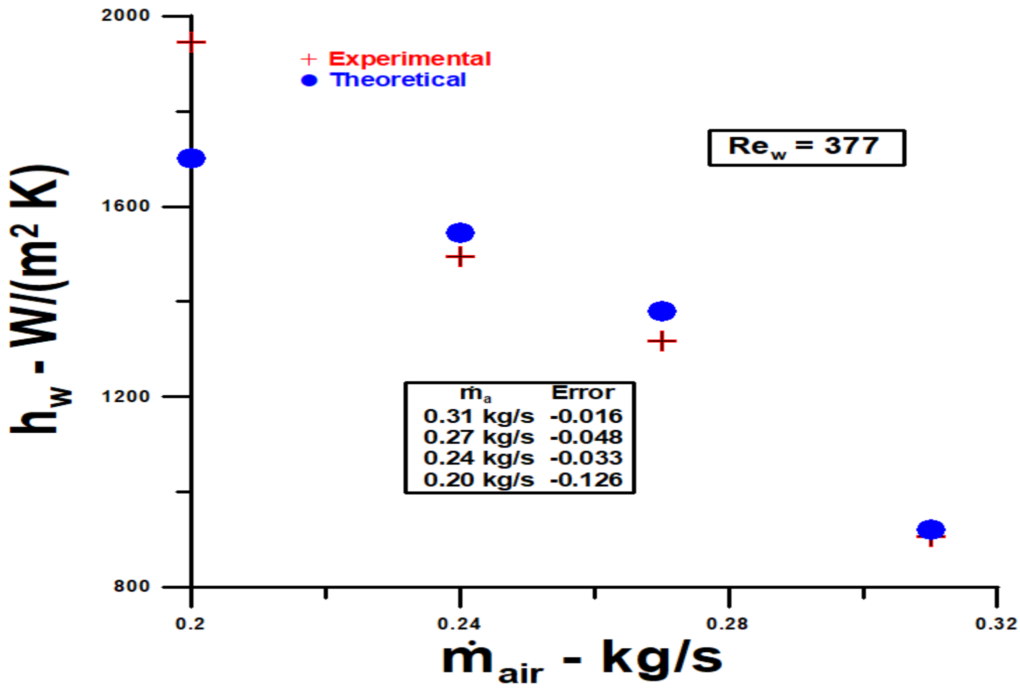

Figure 8 – Convection heat transfer coefficient versus mass airflow – Re_w = 377

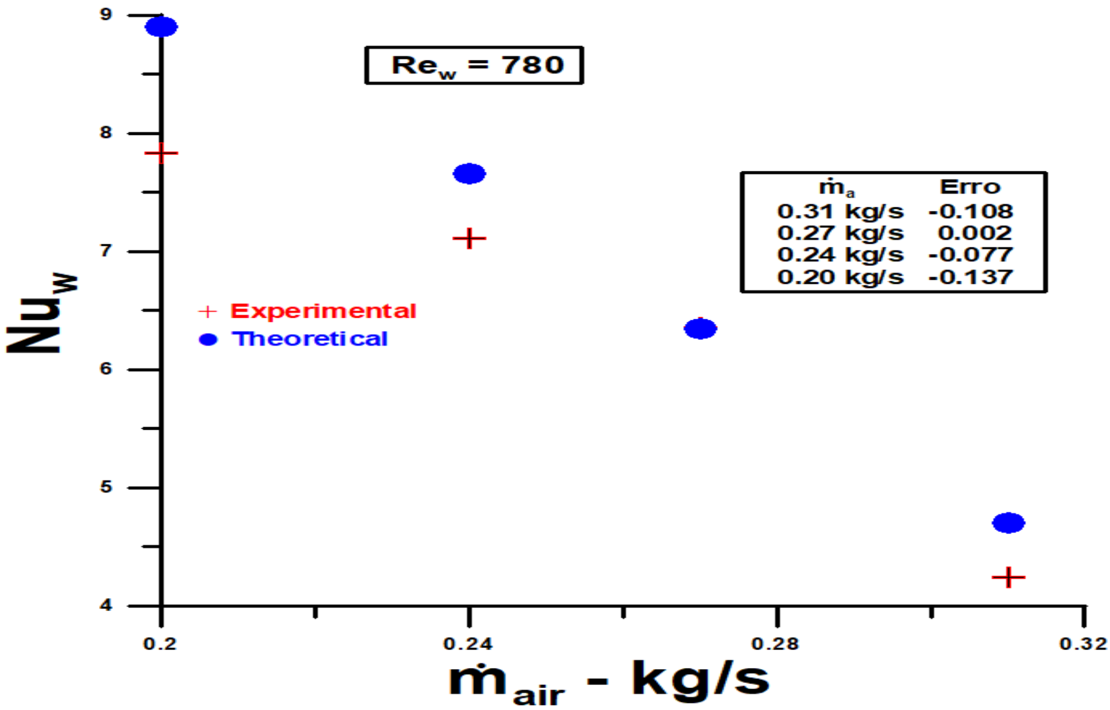

Figure 9 – Nusselt number versus mass airflow – Re_w = 780

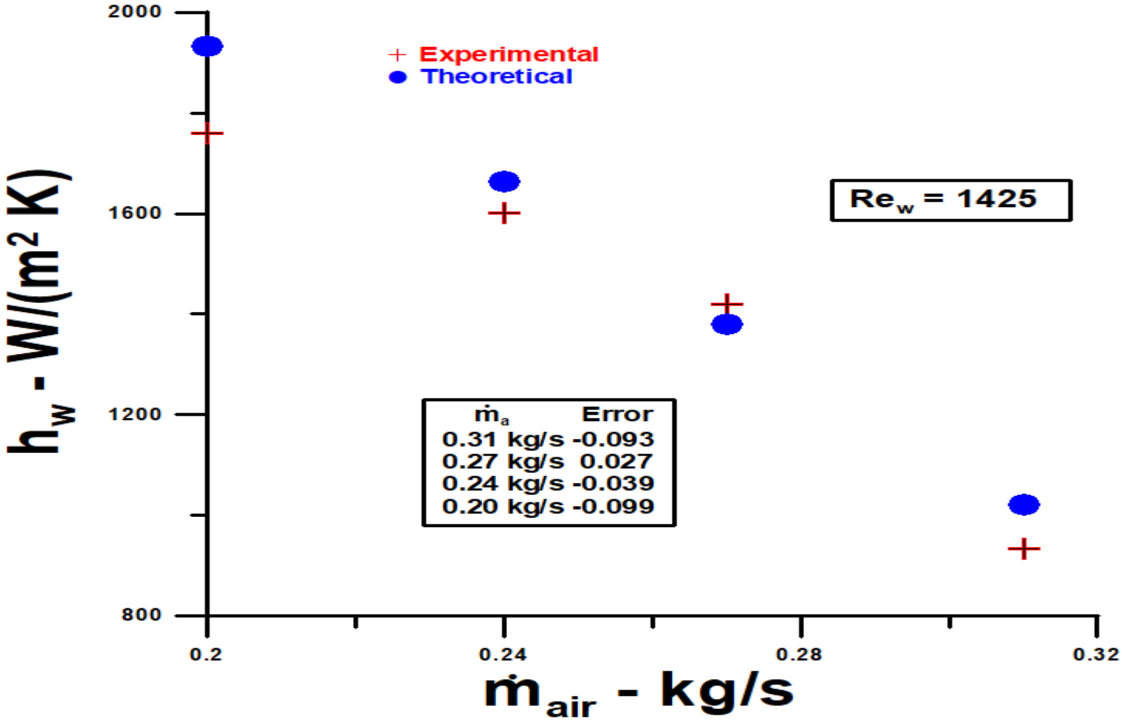

Figure 10 - Convection heat transfer coefficient versus mass airflow – Re_w = 1425

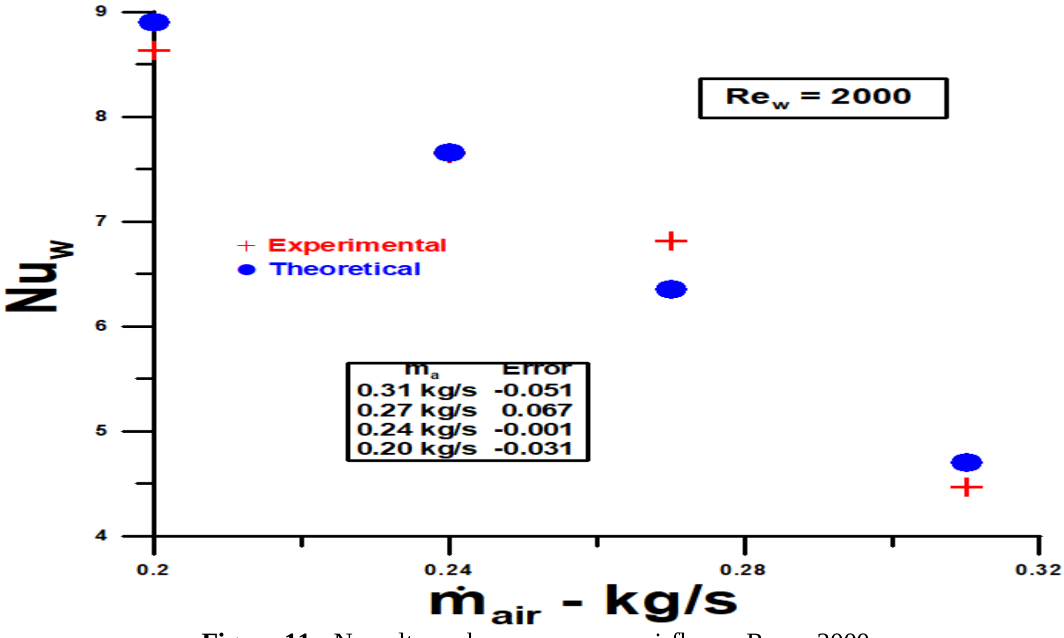

Figure 11 - Nusselt number versus mass airflow – Re_w = 2000

Figures 12 and 13 show the dispersion between theoretical and experimental values for all situations under analysis. The maximum velocity shown in the graphs refers to the velocity in the wind tunnel's straight section. As can be seen, the most considerable deviations are at the lower limit of the water flow, i.e., Rew = 377, with a red highlight.

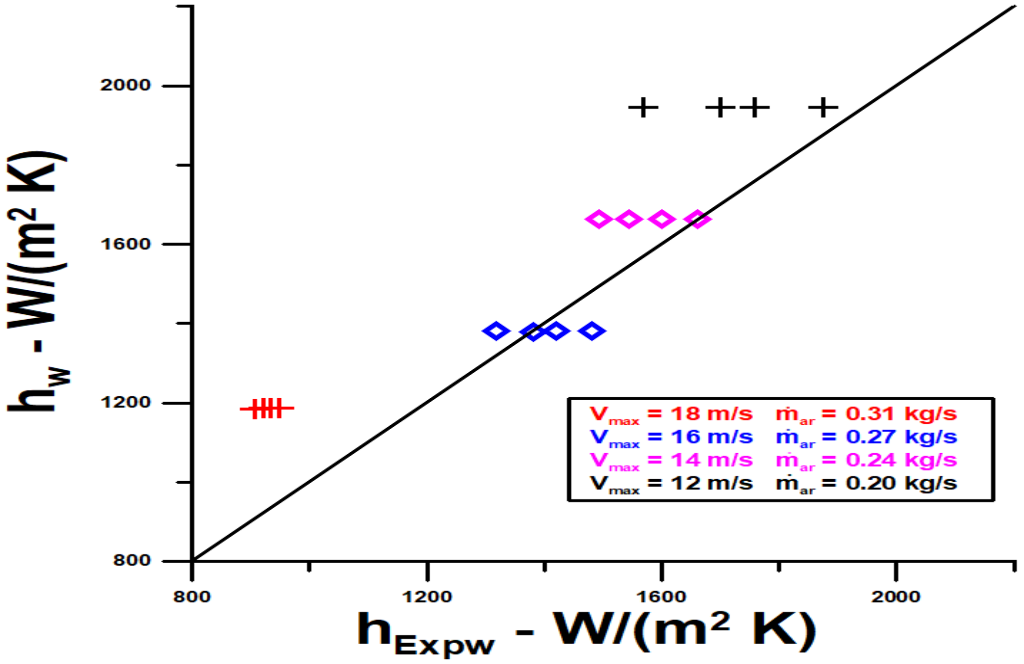

Figure 12 – Theoretical versus experimental convection heat transfer in water

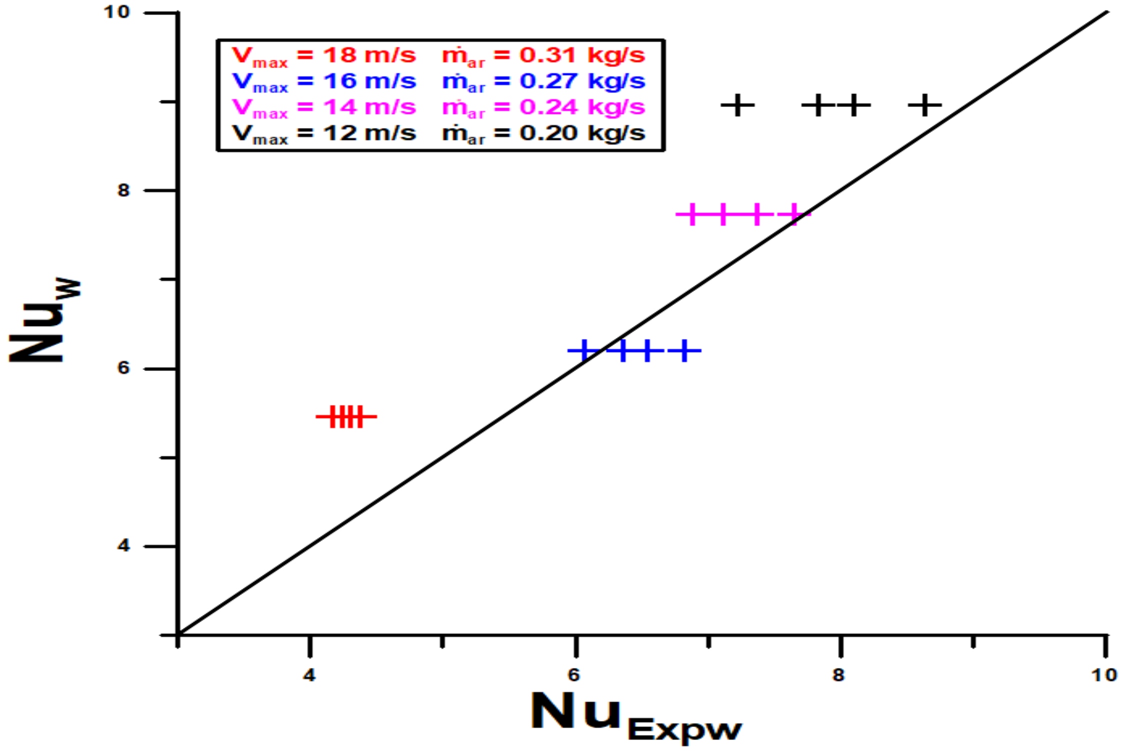

Figure 13 – Theoretical versus experimental Nusselt number in water

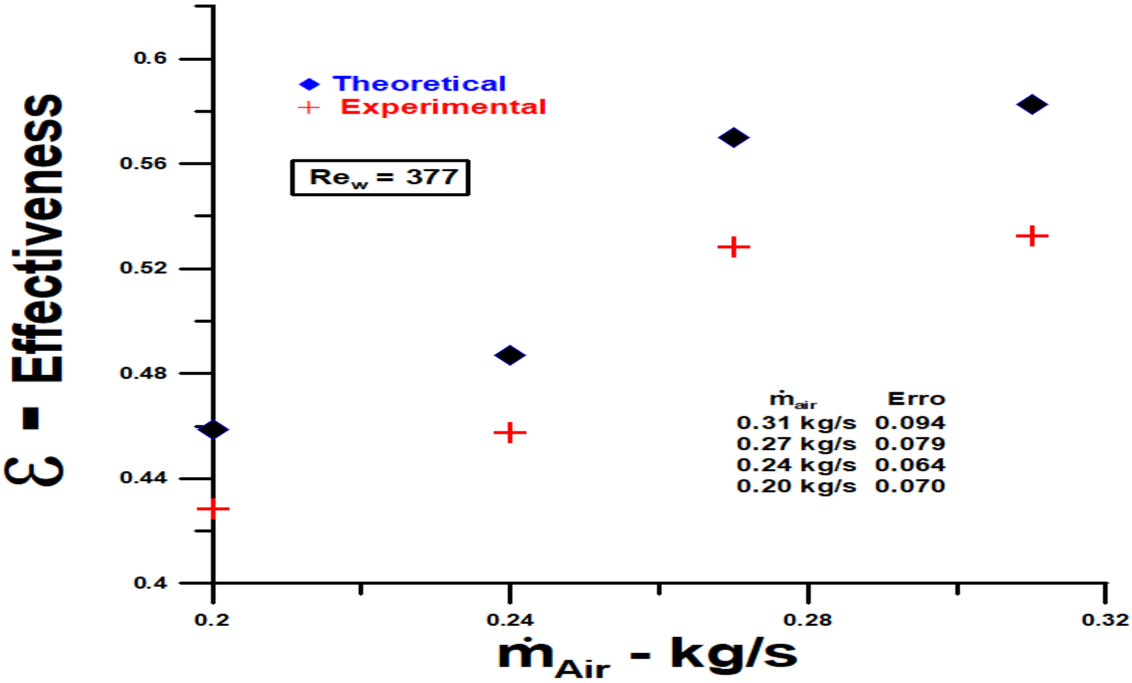

Figure 14 – Theoretical and experimental effectiveness – Re_w = 377

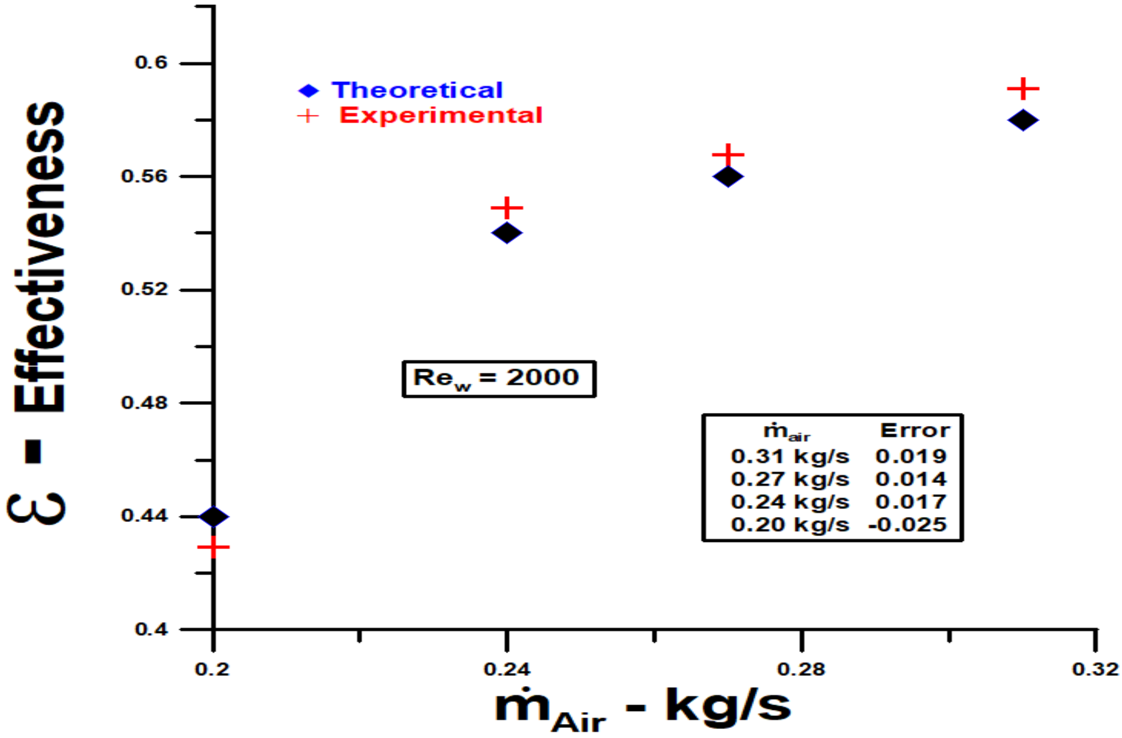

Figure 15 – Theoretical and experimental effectiveness – Re_w = 2000

Figures 14 and 15 show an experimental, theoretical comparison for effectiveness in two situations under analysis, that is, Rew=377 and Rew=2000. The maximum effectiveness obtained, in these cases, is close to 0.6. This result demonstrates that there is ample possibility to increase effectiveness. One of these possibilities is the use of nanoparticles dispersed in the base fluid, as already demonstrated in theoretical [2] and experimental results [3]. The dispersion between theoretical and experimental results already noted above is greater for Reynolds number for water equal to 377.

V. Conclusion

An experimental procedure was developed with the primary objective of analyzing the thermal performance of a flat finned tube type heat exchanger (automotive radiator), with the flow in the laminar regime of water in the tubes. An already consolidated theoretical model was presented, based on thermal effectiveness (ε - NUT), as an instrument to validate the experimental results. The obtained results demonstrate the consistency of the recommended experimental procedure.

The experimental procedure recommended in this work should be used soon, with slight changes in the measure and data acquisition system and the apparatus layout, to analyze the thermal efficiency and effectiveness in laminar regime flow with nanoparticles dispersed in the heat exchanger base fluid.

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