Ansys CFD Analysis of the Thermal Behavior of Coolant 134a in a Condenser within a Refrigeration Cycle

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Abstract

This research document shows the thermal analysis and simulation design by CFD (Computational Fluid Dynamics) of ANSYS of the thermal behavior of the refrigerant 134a inside a helical condenser in the refrigeration system. It was initiated with definition of boundary conditions, to then delimit the geometric design and evaluation of the thermal behavior of the 134a refrigerant and its condenser, in addition it determines suitable models for each regime such as: Euler-Euler model, turbulence models, the Lee tool, which handles a heat transfer and phase change equation. The results obtained are reflected in three exposed cases in which the so-called ideal model is compared with others obtained from changes such as: speed of entry, number of turns, temperature of entry, among others, which presented a variation in pressure in 2.5 %, for system losses and speed increases of less than 40 % in general, also a capacitor efficiency greater than 50 % heat transfer and an effective phase change.

Keywords *Helical condenser, ANSYS, phase change, cooling, R134a*

I. INTRODUCTION

Refrigeration and the desire to conserve food have caused humanity to develop cooling processes and environmentally friendly refrigerants. Ferdinand Philippe and Edmond Carré are considered as the pioneers of the absorption system, with the use of the ammonia refrigerant in a gaseous state.

The progress in the field of refrigeration was primarily in the direction of refrigerant gases that do not destroy the ozone layer, since Latin American countries eliminated the use of chlorofluorocarbons (CFCs). In Ecuador the technology has not yet been developed to meet all the requirements of the cold lines, it continues to import 60% of components; However, this country is part of the Montreal Treaty, which is why it is looking for replacement by clean technologies for the protection of the environment. On the other hand, several authors such as Kang et al. [1] studied the heat transfer and pressure changes of R134a in pipelines that are helical in shape and gave heat transfer correlations from the experimental data.

Al-Hajeri et al. [2] claim that; the high efficiency of heat transfer of the helical tube and its compact volume make it important in many engineering areas such as petrochemical, biomedical, power generation and refrigeration and air conditioning. Yu et al. [3] experimentally investigated the heat transfer of condensation of R134a in a helical pipe and reported that the orientation of the pipe has a significant effect on both the refrigerant and the global heat transfer coefficients.

Wongwises and Polsongkram [4] have experimentally investigated the heat transfer of biphasic condensation and the pressure drop of R-134a in a tube heat exchanger in helical concentric tube. The test trials were carried out at saturation temperatures of 40 and 50 ° C.

The helical and straight tubes were compared by Prabhanjan et al. [5] the results showed that a helical spiral heat exchanger increases the heat transfer coefficient and the temperature increase of the fluid depends on the geometry of the spiral and the volumetric flow that flows through it. On the other hand, Awhadi [6] claim that, the phenomenon of condensation of two phases in the helical tube is more complex than in the straight tube attributable to the centrifugal force due to its curvature. The vapor phase of the fluid flowing at high velocity in the tube experiences a high centrifugal force than the liquid phase.

Through the simulation that will be carried out in this investigation it will be possible to observe that each of the R134a refrigerant phases are placed in different parts of the tube, for this, the Eulerian and Lee method was used for the phase change, in the same way we used the models of two equations κ-ε for turbulent flow by realizable method. On the other hand, a base case was determined, where the geometry and the heat flow were calculated so that the phase change inside the helical tube is produced, in addition this base case was compared with variations in velocity, number of turns and coolant inlet temperature.

II. MATERIALS AND METHODS

The design evolution of a helical condenser, with refrigerant 134a, starting with the selection of dimensional parameters, followed by the analysis of the fluid system, until reaching the heat transfer delineation necessary for condensation to result. In this sense, we seek to detail equations, perform dimensional analysis and briefly synthesize each segment of the thermal design.

There are two fluids that is considered in the design of the helical condenser, the first is refrigerant 134a, subject to phase change and another fluid is water, which is going to reduce the temperature to the system and being responsible for that there is a transfer of heat to achieve the condensation process.

A. Properties of 134a Refrigerant

The working fluid that is used in the steam compression refrigeration processes in Ecuador is R134a, because the condensation process will be carried out, it is necessary to present the characteristics of this refrigerant fluid, these are described in table 1.

Table 1: Physicochemical properties of refrigerant 134a [7].

Properties R134a	Dimension
Molecular mass (g/mol)	102.03
Boiling temperature (°C)	-26.06
Critical temperature (°C)	101.08
Boiling temperature at 1,013 (Bar), (°C)	-26,3
Ozone depletion potential (ODP)	0
Global warming potential (GWP)	1430

The 134a refrigerant has an ozone layer destruction index of zero, which reflects that it is environmentally friendly in specific with the recovery of the ozone layer, which was affected between the 70s and 90s however, the greenhouse effect index is still large, so it is advisable not to throw it outwards.

B. Selection of Pipe for the Condenser

Cold-drawn extrusion copper pipes can achieve single-piece, smooth-walled seamless pipes, ensuring pressure resistance and minimal friction losses in fluid handling, In addition to having a high degree of thermal conductivity facilitating the transfer of heat from or to the fluid, applications range from the domestic level to large industrial facilities [8].

The condenser is a helical spiral of extruded copper material special for heat transfer processes. The characteristics of this are displayed in table 2.

Table 2: Condenser physical dimensions and characteristics

Coil Condenser Measurements	Dimensions
Inner diameter (Di)	8 [mm]
Outer diameter (Do)	9,525 [mm]
Thickness (e)	1,525 [mm]
Helical coil Diameter (Ds)	15 [cm]
Material	Copper
Thermal conductivity of the material (k)	386 [W/m K]
Coil Area (A)	$5.027x10^{-5}[m^2]$

C. Parameters of Water

For the heat transfer to cause the phase change of the refrigerant fluid, it is necessary that one of the two fluids be at a lower temperature than the R134a, the water is the fluid used for this purpose, which is under pressure and ambient temperature, approximately 74 kPa and 19 °C respectively. On the other hand, the water will enter the process of phase change at a lower temperature T_{Ai} of 15 °C and after carrying out this process it will go out to T_{Ao} of 22 °C, another indispensable aspect in the fluid is its heat capacity that for this temperature it is taken as Cp_{liquid} of 4.22 kJ/kg·K.

D. Calculation of LMTD

As a previous step to find the heat necessary to perform the heat transfer in the condensation process, it is necessary to determine the average logarithmic temperature difference LMTD, by means of
equation 1.

$$LMTD = \frac{(T_{Ri} - T_{AO}) - (T_{RO} - T_{Ai})}{ln\left[\frac{(T_{Ri} - T_{AO})}{(T_{RO} - T_{Ai})}\right]} \quad (1)$$

The temperature TRi, TRO are the temperatures at the inlet and outlet of the refrigerant, respectively, in °C, TAi, TAO are the water inlet temperatures, in °C.

For the condensation process to be effective, it is imperative that refrigerant 134a has temperatures higher than water, therefore, the refrigerant is considered a pure substance; superheated to the condenser inlet and subcooled to the outlet, on the other hand, the pressure drop in the device will be minimal, so, it is taken as a constant pressure process of 1 MPa, due to these considerations the two temperatures of the refrigerant are found outside the saturation lines in the Pressure vs. Enthalpy, see figure 1, for this reason the qualities of the refrigerant fluid will not be used.

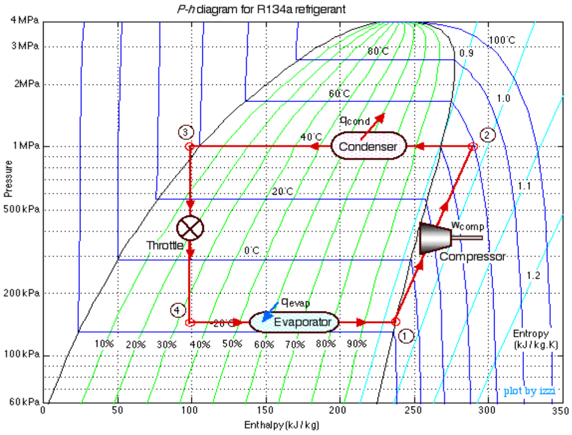

Figure 1. Condensation with R134a [9].

E. Heat Transfer

Heat is known as the form of energy associated with the movement of atoms in all directions, it is also closely linked to the first principle of thermodynamics according to this, two bodies in contact exchange heat between them until reaching thermal equilibrium [10].

In this way Incropera et al. [11] argue that heat is proportional to mass, specific heat, and temperature variation, as it is linked in equation 2.

$$Q = \dot{m} \cdot Cp \cdot \Delta T \quad (2)$$

To find that the refrigerant rejects the mass flow $\dot{m}$ is considered, in kg/s, Cp heat specific of the fluid, kJ/kg·K, and ΔT is the variation of temperatures, in K, they are considered, for the total heat, the three states to be evaluated: gaseous state, liquid and the latent heat transition.

Another important aspect in this process is the amount of water required, for this is taken into account another principle of thermodynamics, the energy is not created or destroyed is only transformed [12], for this reason the heat lost by a substance will be gained by the other, as seen in equation 3.

$$Q_{ganado} = -Q_{perdido} \quad (3)$$

In the same way, it is essential to find the area of heat transfer, using equation 4.

$$A = \frac{Q}{U \cdot LMTD} \quad (4)$$

Where A is the area of heat transfer, in m^2 and U is the global coefficient of heat transfer, W/m^2·K. The effective length of heat transfer is determined by equation 5, where Di is the internal diameter of the pipe.

$$LE = \frac{A}{\pi \cdot D_i} \quad (5)$$

To establish the number of spirals that the helical capacitor will have, equation 6 is used.

$$n = \frac{LE}{\pi \cdot D_s} \quad (6)$$

Where n is the number of turns of the condenser and D_s is the outside diameter of the coil.

Table 3 shows all the values found by the equations described above for the condensation process with refrigerant 134a.

Table 3: Values found in the helical condenser

Characteristic	Dimension
LMTD	29.973 [°C]
Heat transfer, Q	0.720 [kJ/s]
Heat Transfer Area, A	0.1895 [m^2]
Global heat transfer coefficient, U	181.180 [W/ m^2·K]
Effective length, LE	7.540 [m]
Number of whorls, n	16 [Coils]
Coil height, Hs	0.304 [m]

F. Computational Models

The computational mesh is a description of the spatial domain in which the numerical simulation is performed, in the regions of interest a greater refinement is used, however, in these days the meshing process has become the bottleneck of all the numerical simulations, to solve finer meshes, strong computational requirements are needed in the field of processing.

Figure 2 shows the meshing of the helical condenser, which is analyzed in this research work.

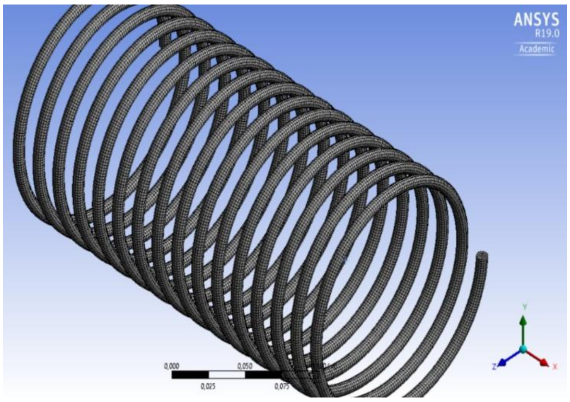

Figure 2. Helical condenser meshing.

It is recurrent to elaborate a mesh convergence, for this the Skewness tool is used, this stresses that if the values are close to zero the mesh is good and on the contrary if these values tend to one the mesh degenerates, in figure 3 the mesh convergence is visualized in this numerical study, the majority of elements are between 0 - 0.25, this guarantees a good mesh.

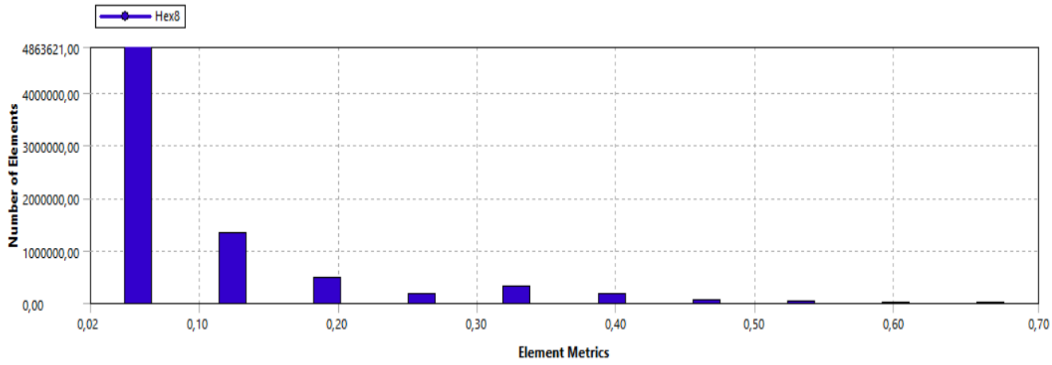

Figure 3. Mesh convergence

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_j}(\rho k u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{u_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon-Y_M+S_k \quad (7)$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_j}(\rho\varepsilon u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{u_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+\rho C_1 S\varepsilon-\rho C_2\frac{\varepsilon^2}{k+\sqrt{v\varepsilon}}+C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b+S_\varepsilon \quad (8)$$

Because the process of condensation is a phenomenon of phase change, it is imperative to use equations that govern the numerical simulation process. These equations are those of conservation of mass and energy, movement, turbulent flow and change of state. However, because it is a multiphase fluid, the turbulence and phase change models are emphasized.

Equations 7 and 8 represent the additional transport equations; one associated to the kinetic energy of turbulence, k, and the other to the speed of dissipation of the turbulence, ε, specific for the model k-ε realizable, respectively.

The transport model for the phase change is given by Lee Model, which allows the condensing process of refrigerant 134a, this model is represented in equation 9.

$$\frac{\partial}{\partial t}(\propto_v \rho_v)+\nabla\cdot(\propto_v \rho_v \vec{V_v})=\dot{m}_{lv}-\dot{m}_{vl} \quad (9)$$

Inside the simulation software defines the positive mass transfer from the liquid phase to the vapor phase for the evaporation-condensation processes. For this it is necessary to use the coefficient of adjustment and relaxation, expressed in equations 10 and 11 for condensation and evaporation respectively.

$$\dot{m}_{vl}=coeff*\propto_v \rho_v\frac{(T_{sat}-T_v)}{T_{sat}} \quad (10)$$

$$\dot{m}_{lv}=coeff*\propto_l \rho_l\frac{(T_l-sat)}{T_{sat}} \quad (11)$$

III.RESULTS AND DISCUSSION

In the first case of the simulation carried out in this study, it is the condensation process of R134a in a helical exchanger with the variables obtained by analytical techniques, detailed in table 3 and observed in figure 4.

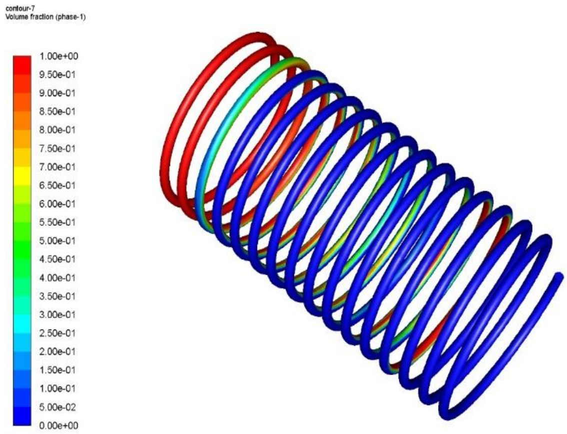

Figure 4. R134a Ideal condensation.

Figure 4 shows that the condensation was performed correctly, because the quality at the output of the helical is zero, which means that there is no fluid in the liquid phase.

One of the objectives of this simulation is to improve the conditions of the exchanger, for these certain variables were modified with reference to the ideal analytical process, such as: temperature and speed of refrigerant entry, and number of turns.

Case 1: Comparison of the ideal model condenser versus the model with turns reduction from 16 to 15 turns, visualized in figure 5.

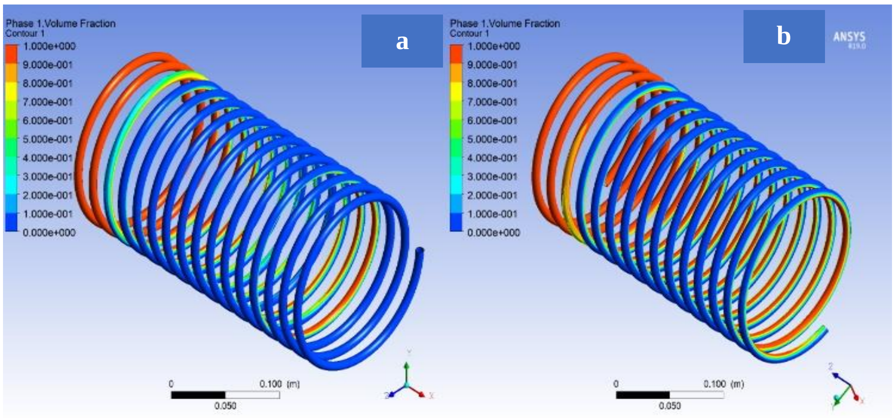

Figure 5. Condensation with reduction of turns, case 1.

In figure 5a, the ideal case is observed, and in case 5b case 1 it is possible to appreciate that the refrigerant does not completely condense.

Case 2: Decrease in the speed of entry of refrigerant 134a in half, from a value of V = 1.5 m/s to V = 0.75 m/s, observed in figure 6.

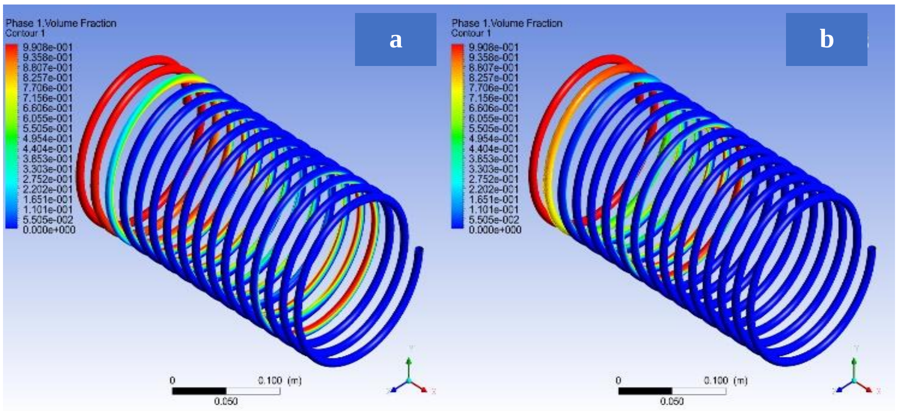

Figure 6. Condensation with decrease in velocity, case 2.

By reducing the velocity of entry of the R134a and maintaining the same heat output, condensation is achieved in advance, several turns before.

Case 3: Variation of coolant inlet temperature from 333 K to 325 K, as indicated in figure 7.

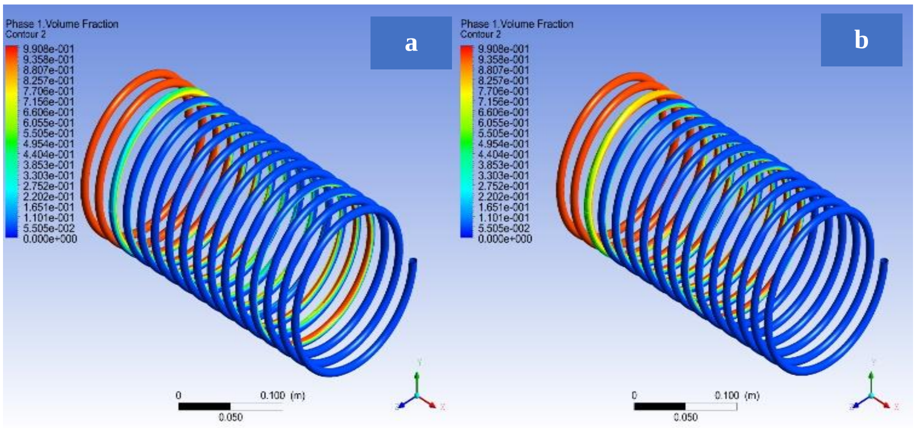

Figure 7. Condensation with temperature decrease, case 3

In the case of reducing the intake temperature by 8 K, the condensation result is satisfactory since the vapor phase is changed to liquid, in addition the condensation time is reduced by 5.5%, the ideal being figure 8a and the variation 8b.

IV. CONCLUSIONS

As a culmination to the investigation, it is emphasized that the use of refrigerant 134a in refrigeration processes, in Latin America has been established over the years, being the most used in domestic refrigeration, for these reasons the use of helical heat exchangers was analyzed. who have this fluid for their heat transfer processes, in addition certain conclusions of the investigation are emphasized:

- The numerical simulation by means of CFD helps to better understand the internal behavior of the fluid, as well as to interpret the phase change processes and specifically the condensation.
- The changes that were made to the geometry or the process variables caused the condensation to occur early or retarded this process, so it is argued that these aspects should be taken into account when applying this type of capacitor. to cooling systems.
- The flow regime inside the pipe corresponds to a turbulent flow, for this reason a resolution process was implemented using the κ-ε model, which shows a behavior very similar to the real effect that occurs on the internal pipeline with the diameter analyzed.

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REFERENCES

[1] H. J. Kang, C. X. Lin, and M. A. Ebadian, "Condensation of R134a flowing inside helicoidal pipe," *Int. J. Heat Mass Transf.*, vol. 43, no. 14, pp. 2553–2564, 2000.

[2] M. H. Al-Hajeri, A. M. Koluib, M. Mosaad, and S. Al-Kulaib, "Heat transfer performance during condensation of R-134a inside helicoidal tubes," *Energy Convers. Manag.*, vol. 48, no. 8, pp. 2309–2315, 2007.

[3] B. Yu, J. T. Han, H. J. Kang, C. X. Lin, A. Awwad, and M. A. Ebadian, "Condensation heat transfer of R-134A flow inside helical pipes at different orientations," *Int. Commun. Heat Mass Transf.*, vol. 30, no. 6, pp. 745–754, 2003.

[4] S. Wongwises and M. Polsongkram, "Condensation heat transfer and pressure drop of HFC-134a in a helically coiled concentric tube-in-tube heat exchanger," *Int. J. Heat Mass Transf.*, vol. 49, no. 23–24, pp. 4386–4398, 2006.

[5] D. G. Prabhanjan, G. S. V Raghavan, and T. J. Rennie, "Comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger," *Int. Commun. Heat Mass Transf.*, vol. 29, no. 2, pp. 185–191, 2002.

[6] A. L. I. Owhadi, "Forced Convection Boiling Tubes Helically-Coiled," *October*, vol. II, pp. 1779–1793, 1968.

[7] A. H. Tarrad, F. A. Saleh, and D. M. Mahmood, "A Thermal Assessment for Vertical Helical Immersion Coil Evaporator in a Water Chiller," vol. 2, no. 5, pp. 62–73, 2015.

[8] S. A. Productos Nacobre, "Manual Técnico Nacobre," p. 121, 2010.

[9] E. Perrin, "Implementation of a Carbon Dioxide Refrigeration System as a Cogeneration Appliance and Replacement for Halocarbon-based Refrigeration Systems," *Tesla STEM Journal*, 2014. [Online]. Available: http://www.ethanperr.in/co2-refrigeration/. [Accessed: 05-May-2018].

[10] J. Andrade,Laura- Mejía, Abraham- Silva, "Diseño De Una Cámara Frigorífica Para La Preservación De Carne De Res En La Cuidad De Puebla, Puebla," p. 120, 2008.

[11] F. Incropera, A. Lavine, and D. DeWitt, "Fundamentals of heat and mass transfer," *US Pat. 5,328,671*, p. 1048, 2011.

[12] R. Pintó, "Algunos conceptos implícitos en la 1a y la 2da Leyes de laTermodinámica: una aportación al estudio de las dificultades de su aprendizaje," Universidad Autónoma de Barcelona, 1991.

APPENDIX

Appendix A1 represents the nomenclature used for the equations of this project.

	Nomenclature
ρ	Fluid density
τ	Reynolds Stress Tensor
k	Kinetic energy of Turbulence
ε	Turbulence dissipation rate
v_T	Kinetic Swirl Viscosity
S_{ij}	Warp Speed tensor
G_k	Generation of turbulence kinetic energy due to speed
G_b	Generation of turbulence kinetic energy due to speed
σ_k	Number of Prandtl turbulent K
σ_ε	Number of Prandtl turbulent ε
Y_M	Fluctuating dilation contribution
u_t	Turbulent viscosity computed
$\propto_v$	Vapor Volume fraction
ρ_v	Steam density
$\vec{V}_v$	Steam phase speed
$\dot{m}_{vl}$	Mass flow of condensation
T_{sat}	Saturation temperature
T_v	Steam temperature
$C_1, C_2, C_{1\varepsilon}, C_{3\varepsilon}$	Constants