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# NUMERICAL SIMULATION AND OPTIMIZATION OF THE GEOMETRIC PARAMETERS OF A HELICAL TAPE FOR THE INTENSIFICATION OF HEAT TRANSFER IN SOLAR COLLECTOR

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***Abstract.*** *Nowadays, the thermal energy required for heating water in domestic or industrial application has significantly increased. Therefore, the solar energy has emerged as a suitable renewable energy source for this application, requiring permanent technology development on solar collector regarding its thermal efficiency in order to meet several application requirements. Overall, the main goal of the present research is to provide a numerical simulation and optimization procedure of the heat transfer enhancement through passive device known as Helical Tape, at low Reynolds number (Re = 300 and Re = 600) for a solar water heater with an active system and indirect single-phase flow. Helical Tape geometric parameters submitted to optimization procedure are: Position (translation) of the helical tape relative to the tube inlet, Pitch (P), Length (L) and Radius ($R_1$ and $R_2$). These five input variables were submitted to approach by Direct Optimization which is the integration between numerical model and optimization method (Genetic Algorithm). The results allow concluding that the application of helical tapes to enhance the heat transfer in a solar water heater is a very effective passive technique. The results of the pitch length optimization were 34 mm for Reynolds 600 and 33 mm for Reynolds 300. The intensification of the heat transfer obtained was 200% and 235% for Reynolds 300 and 600, respectively, when analyzing the pitch length optimization. And an asymmetric helical tape produced a significant increase in the heat transfer with moderate pressure drop penalty when compared to the conventional symmetrical helical tape with twist ratio of 4.*
***Keywords:*** *Solar Energy, Solar Collector, Numerical Simulation, Heat Transfer, Helical Tape.*

## 1. INTRODUCTION

Brazil is a country with vast natural resources that can be harnessed for energy purposes (Tiba, 2000). In addition, concerns about the environmental impacts caused by the expansion of the energy supply from non-renewable sources are latent and immediate, causing several important discussions. In this context, supported by the favorable climatic characteristics, Brazil emerges as a protagonist for the use of solar energy for the electric generation and thermal energy (Pereira, 2004).

This research is comprised in the branch of solar thermal, which evaluate the conversion of solar energy to a working fluid for domestic and/or industrial applications by means of a flat plate solar collector. For residential applications, solar collectors can replace electric showers to provide heating water or to minimize their use. This type of demand represents a large percentage of the electric energy consumed in a house (Shukla et al., 2013). Solar collectors are widely used to perform the conversion of solar energy to thermal energy into a working fluid, however, increasing the efficiency its heat transfer is still a great challenge (Shukla et al., 2013). Studies of passive techniques in order to enhance the heat transfer have gained an important space in recent researches (Liu and Sakr, 2013). Zhang et al., 2013 showed that the intensification of heat transfer through helical tapes reached up to 351% with 1020% in the pressure drop penalty due to different pitch. They also concluded that the pitch of the helical tape was the most critical factor for enhancement heat transfer, which was also verified by Kumblar and Sane, 2015 and Wang et al., 2011.

Overall, it is evident that identifying the optimum geometrical parameters configuration of a helical tape in order to optimize the heat transfer with reduced pressure loss penalty is still an important subject for scientific investigation. Thus, the present research proposes the numerical simulation of the heat transfer intensification process in a flat plate solar collector with active system and indirect circulation of single-phase working fluid, using a device known as Helical Tape. ), down to Reynolds number (Re = 300 and Re = 600). The main geometric parameters: length, pitch, radius length and translation (detailed in section 2.2) will be submitted to optimization method, similarly as conducted by (Salviano et al. 2014), (Salviano et al. 2015) and (Salviano et al. 2016), on compact heat exchangers. Understanding the main phenomena makes it possible to extrapolate the optimal geometrical configuration with the aim of increasing heat transfer and minimizing the pressure loss penalty, allowing the technological development of a more efficient solar collector regarding thermal energy conversion. One of the marginal consequences of this work, considering the possible increase of the thermal efficiency of energy conversion, is the possibility of decrease in the solar capture area, reducing, among others, the amount of materials used in the manufacturing process, besides minimizing the complexity of installation, especially for residential applications.

## 2. METHODOLOGY

### 2.1 Governing equations

The numerical modeling of the heat transfer and dynamic flow for a fluid inside a tube with circular cross-section considers the hypotheses of incompressible and tridimensional flow, steady-state and laminar flow (Cheshmeh, 2012), which was performed in software ANSYS Fluent 18.2. For a Newtonian fluid with constant properties, the equations of Continuity, *Momentum* and Energy, can be defined, respectively:

$$\frac{\partial}{\partial x_j}\left(\rho u_j\right)=0 \tag{1}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j u_i-\tau_{ij}\right)=\frac{\partial p}{\partial x_j} \tag{2}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j h_i-k\frac{\partial T}{\partial x_j}\right)=-u_j\frac{\partial p}{\partial x_j}+\tau_{ij}\frac{\partial u_i}{\partial x_j} \tag{3}$$

### 2.2 Contour conditions and computational domain

The numerical model proposed herein considers a commercial flat-plate solar collector used for water heating with 9 elevation tubes. The water flows inside a copper pipes with diameter (D) of 9.52 mm (3/8 ") and longitudinal length of 1000 mm. The average mass flow rate evaluated is 8.35 l/h and 16.7 l/h. These average mass flow rates considered a uniform distribution of water among the elevations tubes, which represents a Reynolds number of 300 and 600, respectively. A constant and uniform velocity is adopted as the boundary condition at inlet of the computational domain, which is known as the *Dirichlet* condition. A constant heat flux is imposed on tube surface and is equal to 750 W/m$^2$. At the exit of the computational domain a fixed pressure is imposed. The helical tape insert is assumed with no thickness and the walls are considered to be no-slip.

The geometrical modeling has been done in software ANSYS Design Modeler 18.2. Two important geometric characteristics of the helical tape insert were considered for the heat transfer intensification process, the first is the half pitch (MP), which consists of the length for a complete revolution of 180°, and the Twist Ratio (Y), or pitch, is defined as the ratio of the half pitch and the diameter of the tube (Hasanpour et al., 2014).

$$Y=\frac{MP}{D} \tag{4}$$

The base helical insert diameter is 8 mm (R1 = 4 mm and R2 = 4 mm). Other parameters are the length of the helical insert equal to 980 mm, which is initially placed at 10 mm after the tube inlet and 10 mm before the end of the tube. The initial pitch is 32 mm, which is equal to twist ratio of 4. The Figure 1 shows the geometrical parameters submitted to optimization procedure.

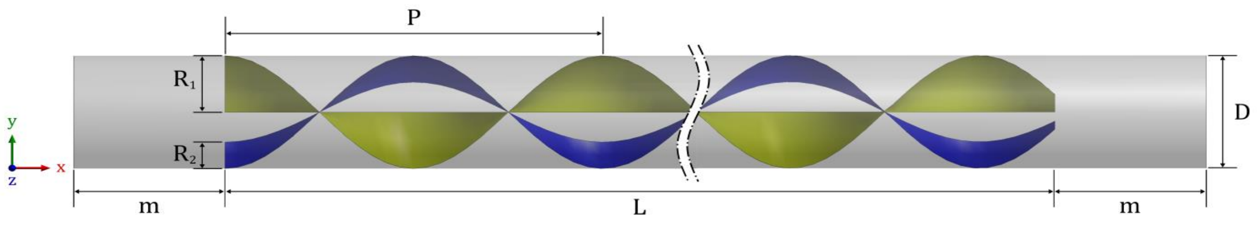



Figure 1. Geometrical parameters of the helical insert for optimization.

### 2.3 Grid Independence and Numerical Validation

Sensitivity analysis was developed based on the GCI (Grid Convergence Index) method (Division et al., 2008), which is based on Richardson's extrapolation method and aims to evaluate the relative error of computational domain discretization. The method estimates numerical uncertainty by analyzing three sets of meshes with different refinements. As recommended, it is desirable that the mesh ratio factor, Eq. 5, be greater than or equal to 1.3:

$$r = \left(\frac{h_{fine}}{h_{course}}\right)^{1/3} \quad (5)$$

Table 1 shows the meshes, element quantities and the ratio factor obtained by equation 5. The GCI results (%) indicate that the numerical uncertainty of the model as a function of mesh refining is significantly small, 0, 28% for the friction factor and 4.92% for the Nusselt number, indicating that the use of Grid 2 is an appropriate option, as it has a small variation of the thermohydraulic parameters and a considerably smaller number of elements in relation to Grid 3, a fact that significantly reduces processing time. Thus, mesh independence was satisfactorily achieved by the GCI method.

Table 1 – mesh characteristics.

| Mesh | Elements | Ratio factor, r |
|---|---|---|
| GRID 1 | 607.874 | * |
| GRID 2 | 1.680.263 | 1,40 |
| GRID 3 | 3.660.236 | 1,30 |

The validation of the numerical approach is conducted by comparison with experimental data. According to (Incropera, 2014), the Nusselt number for a internal laminar flow inside a circular tubes (fully developed flow), under constant heat flux on surface is 4.36, and the friction factor is a function of the Reynold number (f = 64/Re). Considering the higher Reynolds number evaluated in the present work (Re = 600) for a flat tube, the difference between the numerical and experimental results for the Nusselt number and the friction factor are 2.15 % and 2.00 %, respectively. Therefore, considering these previous results, the numerical approach is considered robust and reliable.

### 2.4 Optimization Procedure

In this step, the five input variables: Lenght, Pitch, Radius 1, Radius 2 and Translation, are submitted to the Direct Optimization approach, as recommended by (Salviano et al., 2015) In this approach the numerical model of the flow inside the flat solar collector tubes is assumed to be the "objective function" to be optimized and, therefore, there are no approximation errors as verified when using Response Surface Methods.
Figure 2 shows the outline of the flowchart applied to the Direct Optimization approach that the optimal solution is found directly in a single step by coupling the flat solar collector numerical model and the optimization algorithm.

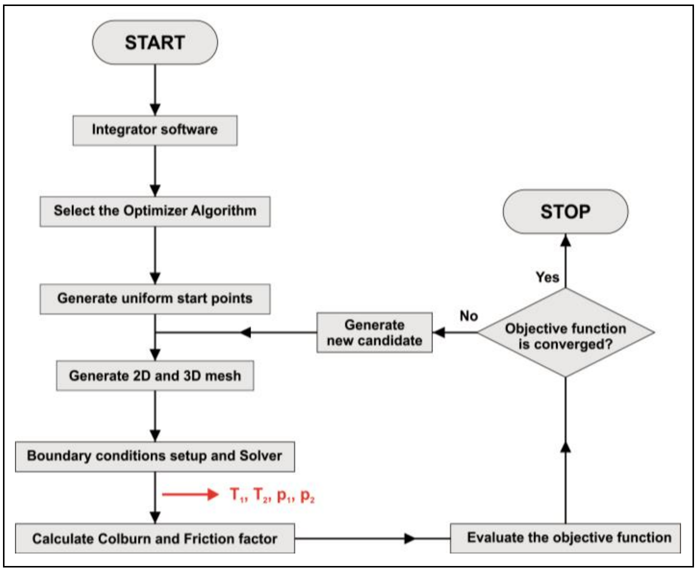



Figura 2 – Flowchart applied to the Direct Optimization approach (Salviano et al., 2015)

For this project we used the optimization method known as the evolutionary method Genetic Algorithm (GA), applied and discussed in detail by (Holland, 1992), (Mitchell, 1997), (Goldberg, 1989), (Michalewicz, 1992) and (Chipperfield, 1997). One of the main advantages of this method is the ability to explore possible global and local optimal points. The search for the optimal solution starts from a set of initial solutions that advance toward the optimal region. Optimal locations can be found quickly, however, global optimum can be difficult to identify especially for multi-variable functions that can have multiple optimum points.

For the optimization process, the software ESTECO ModeFrontier PHD was used. One of the main potentialities of this software is its ability to integrate the numerical simulation steps and the optimization method, also allowing flexibility in the calculation of various thermohydraulic parameters directly in its interface. For the direct optimization approach, operating ranges were defined for the five geometric input parameters. Table 2 indicates the corresponding operating ranges.

Table 2. Operational range of parameters for parametric analysis.

| Parameters | Operating range (mm) |
|---|---|
| Length | 300 – 900 |
| Pitch | 16 – 500 |
| Radius | 0.5 – 4 |
| Translation | 5 – 500 |

In order to evaluate the optimal configurations a preliminary parametric analysis is performed for each geometrical parameters. For this preliminary parametric analysis, a helical twist ratio equal to 4 is used, which is often used for enhance the heat transfer (B. Kumar et al., 2018). The parametric analysis consists in modifying one parameters of the tape for times and keeping others variables fixed and, therefore, it is possible to check the sensitivity of each parameter on enhancement heat transfer.

## 3. RESULTS AND DISCUSSION

The results obtained by the optimization process for Re = 300 and Re = 600 were compared with the twisted tape with twist ratio of 4 (R4). Table 3 shows the configurations of the optimized models and the helical tape R4. Note that the R4 setting is used for both Reynolds numbers. In order to evaluate the enhancement heat transfer for each configuration, the ratio between the Nusselt number (Nu) of the helical tape and the Nusselt number of the smooth tube ($Nu_0$) is performed. Similarly, the ratio between the friction factor ($f$) with helical tape and the friction factor for a smooth pipe ($f_0$) is also used.

Table 3. Geometric parameters of the optimized configurations and R4 helical tape.

| Configurations | Length (mm) | Pitch (mm) | Radius 1 (mm) | Radius 2 (mm) | Translation (mm) |
|---|---|---|---|---|---|
| Re 300 (Opt) | 955 | 33 | 0,7 | 2,9 | 5 |
| Re 600 (Opt) | 900 | 34 | 0,9 | 2,2 | 50 |
| R4 | 980 | 64 | 4 | 4 | 10 |

Figure 3 to Figure 7 show the profiles of the $Nu/Nu_0$ and $f/f_0$ parameters for both Reynolds numbers, compared to the optimal points found by optimization procedure.

### *3.1. Impact of the Length:*

Figure 3 shows the optimal points of the length identified for Reynolds number of 300 and 600. The optimal configuration is found to the largest helical length defined for optimization procedure (955 mm), while the optimal configuration found at Reynolds number of 600 is 900 mm, this difference is due to higher intensity of the secondary flow generated by helical tape for higher Reynolds number. Through this figure, it can be seen that the increase of the length of the helical tape linearly increases the Nusselt number and Friction factor.

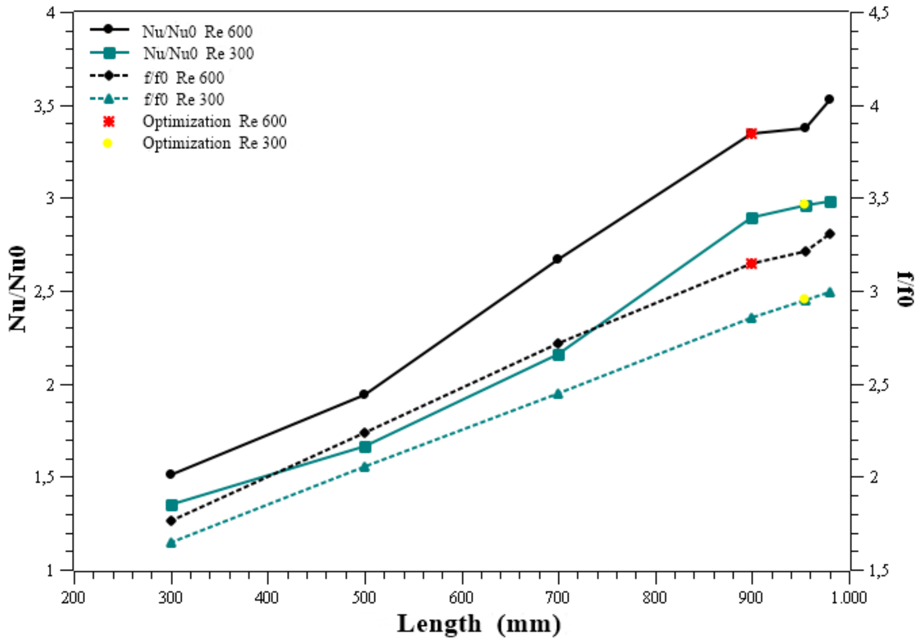



Figure 3. Parametric analysis of the length at Reynolds number of 300 and 600.

### 3.2. *Impact of the Pitch:*

The Figure 4 shows the results for enhancement heat transfer due to helical pitch (P). The behavior of the Nusselt and Friction factor profiles show that for the heat transfer is higher for smaller pitch, because they promote a greater distortion in the flow and decrease the thickness of the boundary layer. This parameter was also evaluated by Kumblar & Sane, 2015 and Wang et al. 2011, who concluded that the pitch is one of the most critical parameters in the heat transfer intensification process. On the other hand, these dynamic effects also cause a significant increase in the pressure loss. This is due to the fact that the shorter helical twist tape provides a longer flow path, resulting in greater tangential contact between the flow stream and the tube surface, as can be noted in Table 4 and Table 5. From Figure 4, it is still possible to observe that for larger pitch, around of 200 mm, independent of the Reynolds number the Nusselt number and Friction factor has a stable behavior. For the pitch between the 16mm to 100mm, the highest variations were observed with a significant reduction of the thermal hydraulic parameters as the pitch is increased. Note that the optimal points are within of this range for both Reynolds numbers. The optimal pitch is 34 mm for Reynolds number of 600 and 33 mm for Reynolds number of 300, which is a similar configuration for a helical tape with twist ratio of 2.

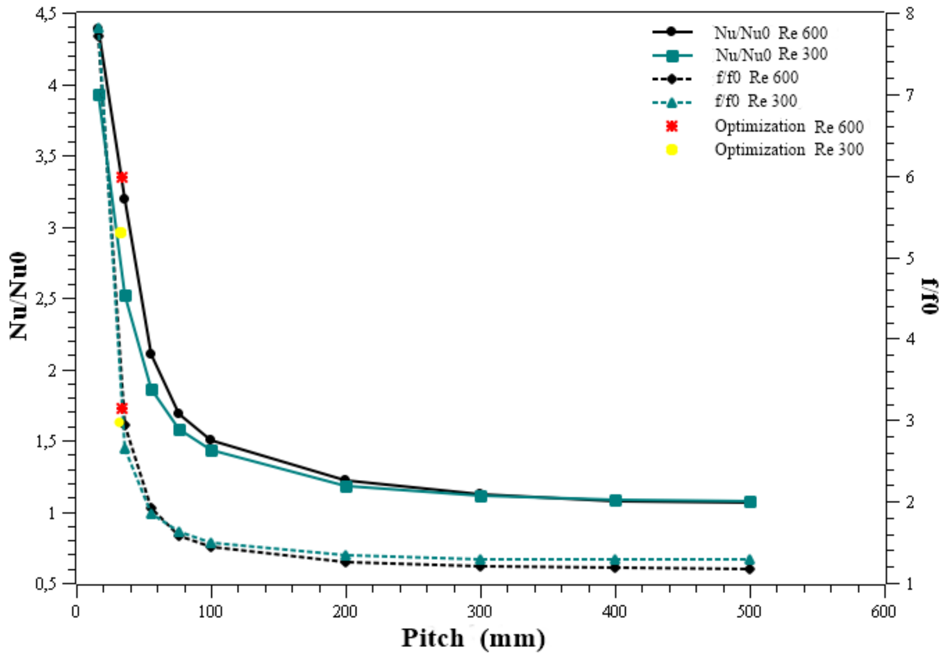



Figure 4. Parametric analysis of the pitch at Reynolds number of 300 and 600.

Tables 4 and Table 5 show distorted current lines produced by the helical tapes, evidenced by radial velocity components which increase the mixtures between the cold and hot flow streams. It is noted that with the difference of the length of the helical tapes the flow distortion can be more intense and more irregular, as well as identified for the optimal asymmetrical configurations. The flow is disturbed due to the blending in the radial direction caused by the vortex which is repeating for each cycle of the pitch length, this movement was identified by Liu S. and Sakr M., 2013. This movement is illustrated by the streamlines in Table 6 and Table 7.

As can be seen from Tables 6 and 7, the current lines show that there are centrifugal forces acting on the fluid and diverting the flow from the central core region to the wall where the high velocity flow collides with the pipe surface, extracting thus a greater amount of thermal energy. Increasing the area in contact with the fluid contributes to increased local pressure loss. In addition, the strong vortex movement causes considerable dynamic pressure drop, according to indicated by Vashistha et al., 2016.

Table 4 - Streamlines in longitudinal direction at Reynolds number 300.

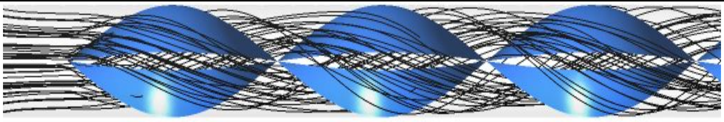
| Re 300 | Position: 0 ↔ 120 mm |
|---|---|
| R4 |  |
| Optimization |  |

Table 5 - Streamlines in longitudinal direction at Reynolds number 600.

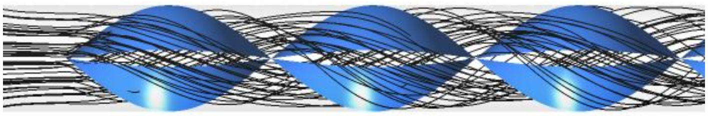
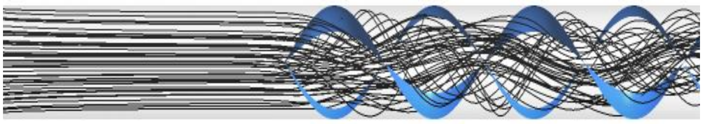
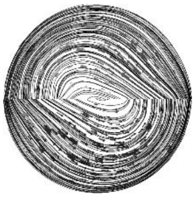
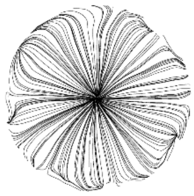
| Re 600 | Position: 0 ↔ 120 mm |
|---|---|
| R4 |  |
| Optimization |  |

Table 6 - Streamlines on transversal planes at Reynolds number 300.

| Re 300 | Position (mm) | | | |
|---|---|---|---|---|
| | 30 | 60 | 500 | 970 |
| R4 |  |  |  |  |
| Optimization |  |  |  |  |

Table 7 - Streamlines on transversal planes at Reynolds number 600.

| Re 600 | Position (mm) | | | |
| --- | --- | --- | --- | --- |
| | 30 | 60 | 500 | 970 |
| R4 |  |  |  |  |
| Optimization |  |  |  |  |

*3.3. Impact of the Radius 1:*

Figure 5 show the impact of the first radius (R1) on Nusselt number and Friction factor, where a single parameter is evaluated by time, so that while the first radius is increased, the second remains constant in its optimal configuration. In this case it is fixed at 2.2 mm and 2.9 mm for Reynolds number 600 and 300, respectively. The friction factor profile has a direct relation with increase of the radius 1. For heat transfer, the Nusselt profile also grows as the size of the rays grow, except twice when there is a decrease in Nusselt; The first is when radius 1 for the Reynolds 600 is 2 mm long and therefore radius 2 held constant at 2.2 mm becomes a substantially symmetrical configuration. Similarly for Reynolds 300, when radius 1 is 3 mm, it becomes an almost symmetrical configuration as radius 2 is kept constant at 2.9 mm, giving two symmetrical configurations and when compared to other points where there is asymmetry explain the falls in the Nusselt profile, since the asymmetry between the rays promotes greater flow distortion, a fact that significantly increases heat transfer with low friction factor value, so the points where there is no symmetry do not reach Nusselt values as high as values in asymmetric cases. However, when radius 1 is 4 mm, and even a decrease in heat transfer is observed, this can be explained by the increase in radius length that causes a decrease in the space between the tape parts in the central region, thus configurations. Asymmetric radius with shorter radius may be more efficient in heat transfer, because the larger space allows fluid in the core and near wall regions to mix evenly under the influence of disturbances and irregular swirls, thereby increasing heat transfer, as identified by He et al.,2018.

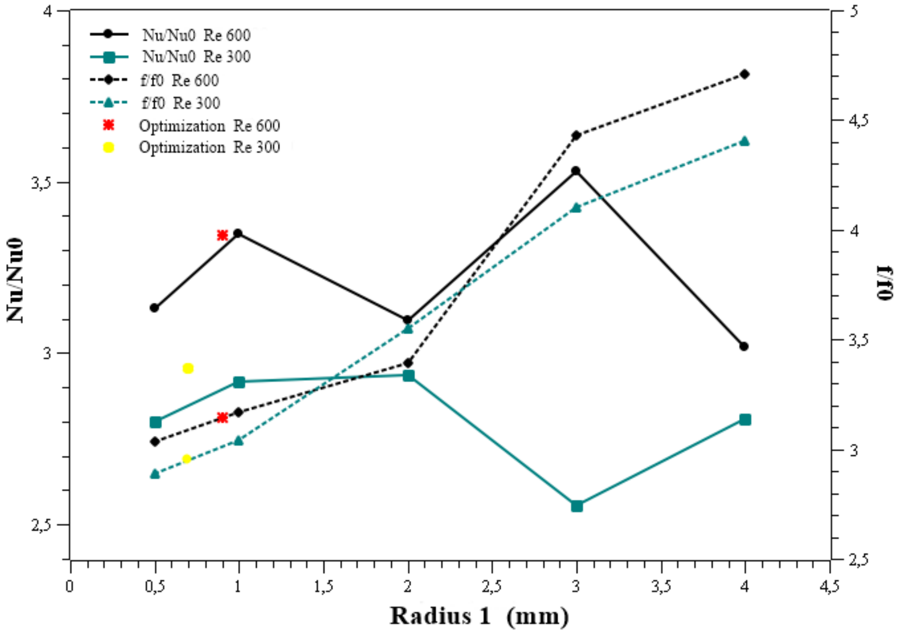



Figure 5. Parametric analysis of the radius 1 at Reynolds number of 300 and 600.

*3.4. Impact of the Radius 2:*

The analysis for the analysis of Radius 2 (R2) is similar that for radius 1 (R1). The optimal configuration identified for the first radius was 0.9 mm for Reynolds number 600 and 0.7 mm for Reynolds number 300, while the first radius remains constant and equal to 0.9 mm and 0.7 mm for Reynolds 600 and 300, respectively. In Figure 6 it is possible to verify that the friction factor always increase as the length of the radius of the helical tape increases. For radius 2 equals to 0.5 mm or 1 mm, it is a configuration almost symmetrical with the radius 1, however, the heat transfer is not decreased, since these radius are very small and thus the dynamic flow is not effect by presence of the helical tape. Also, in Figure 6 it is possible to observe the same conclusion for the radius 1, where the space between the parts of the tape is decreased in the radius 2 configuration with 4 mm, explaining the fall of the heat transfer to both Reynolds with this configuration.

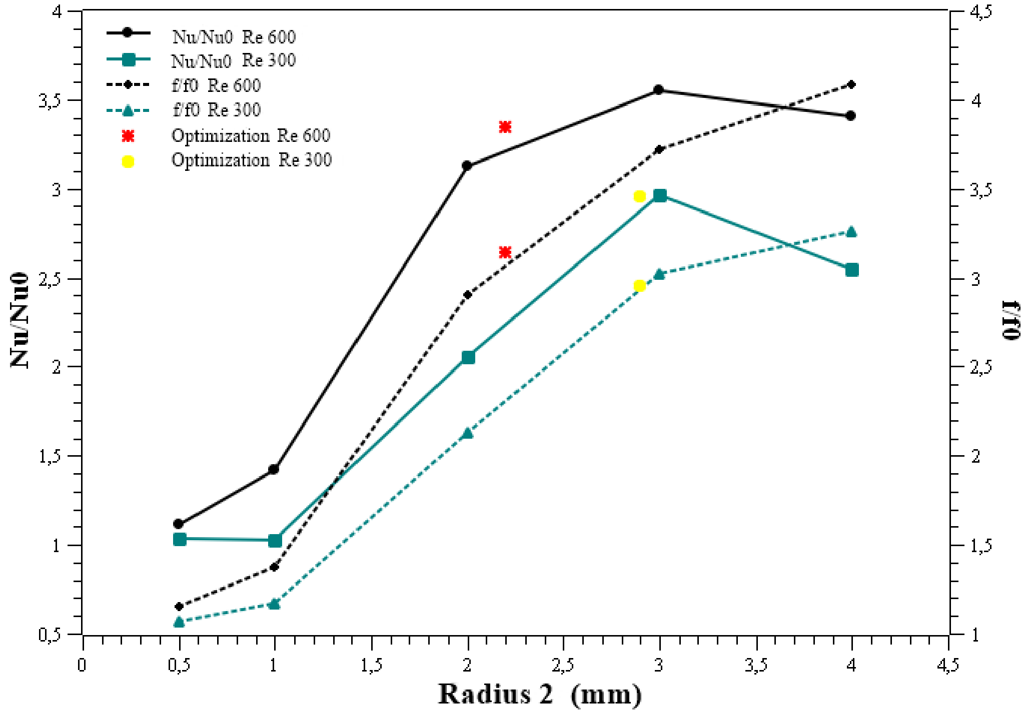



Figure 6. Parametric analysis of the radius 2 at Reynolds number of 300 and 600.

*3.5. Impact of the Translation:*

The analysis of the impact of the translation on Nusselt number and Frictor factor is done considering a geometrical constraint defined by the sum of translation and the Helical length, which must be lower than 990 mm. Thus, for a translation of 500 mm the length of the helical tape is reduced to 490 mm. The results showed that up to 50 mm of translation, insignificant variation is observed on the heat transfer and the pressure drop penalty. The optimum configurations for both Reynolds numbers is reached for the helical tape at beginning of the tube, 5 mm for Reynolds number 300 and 50 mm for Reynolds number 600. Evidently, the Nusselt number and Friction factor decreased as increase the translation due to reduction of the radial velocity component and, consequently, the mixture of the fluid flow. Both profiles of the Nusselt number and Friction facto present a relation quite linear with translation of the helical tape.

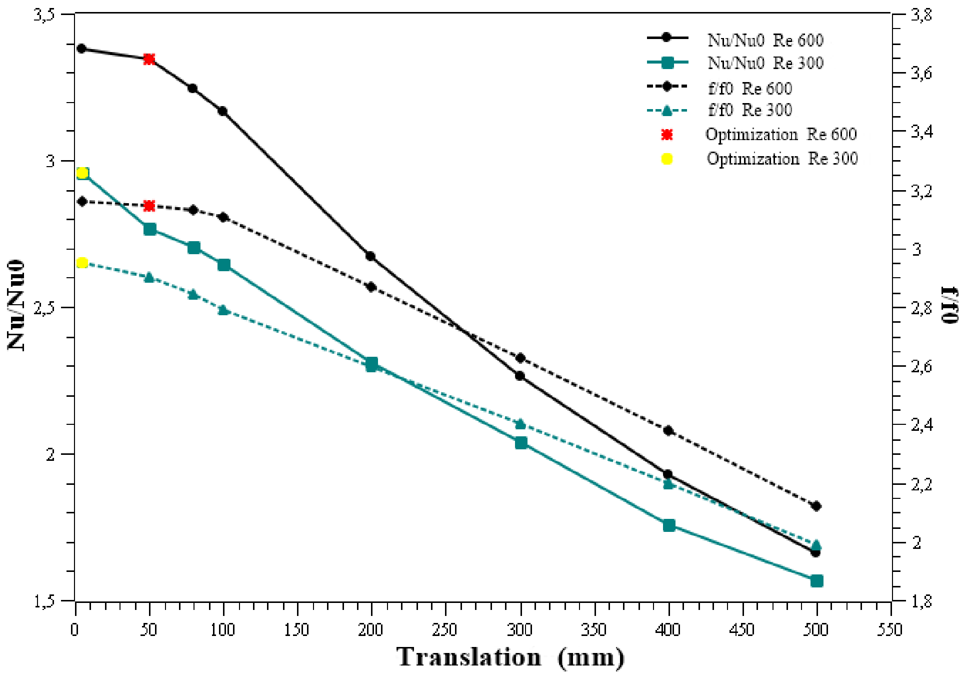



Figure 7. Parametric analysis of the translation at Reynolds number of 300 and 600.

## 4. CONCLUSION

In this work, a parametric analysis of the geometrical parameters of an optimized helical insert at Reynolds number of 300 and 600 is performed, considering a solar water heater. The thermal-hydraulic phenomenons were investigated considering a passive technique to enhance the heat transfer known as helical tape. The software ANSYS 18.2 was used to perform the computational modeling and the software ESTECO modeFRONTIER was applied to design the optimization workflow in order to calculate the Nusselt number and friction factor. The main results found are:

- An asymmetric helical tape produced a significant increase in the heat transfer with moderate pressure drop penalty when compared to the conventional symmetrical helical tape with twist ratio of 4.
- Higher length of the helical insert significantly impact on the Nusselt number and the friction factor.
- The pitch has a great influence on the heat transfer, especially for smaller values, although the increase of the pressure drop penalty is also significant. Moreover, the thermo-hydraulic relationship is also higher.
- The optimal configurations found indicate a helical with high length, small translation, small pitch and different radius to build an asymmetrical helical tape.

## 5. ACKNOWLEDGEMENTS

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