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The Effect of Changing the Coil Wave Amplitude on Improving Heat Transfer for a Natural Gas Heater

Mohamed Amine Belmiloud*, Mohamed Boutkhil Guemmour, Sad Chemloul Nord-eddine

Department of Mechanical Engineering, Ibn Khaldoun University, Tiaret 14000, Algeria

Corresponding Author Email: belmiloud.amine@univ-tiaret.dz

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ABSTRACT

Natural gas is transported from production areas using high-pressure pipelines to pass through pressure reducing stations and then to the consumer. When temperatures drop, the problem of freezing of natural gas and ice appears. To protect against this phenomenon, we use a water bath to improve the temperature of the gas before reducing its pressure. One of the essential elements in the design of a low thermal efficiency water bath heater is the gas coil that runs inside the heater. In this study, the analysis of heat transfers and gas flow velocity inside the corrugated coil was performed with varying wave amplitude (10 mm, 15 mm, and 20 mm). The three-dimensional unsteady Reynolds-Navier-Stokes equations are solved using both the realizable k-ε turbulent model and the finite volume method to obtain the flow field. The results of this study showed that using corrugated coils with a larger capacity leads to an increase in the Nusselt number and friction factor, respectively, by about 6.57% and 114%. In general, the heat exchange inside the corrugated twisted gas coil can be enhanced by increasing the wave amplitude.

1. INTRODUCTION

To enable the heat exchange of heat transfer equipment to be improved, several technologies are used, including active technology, passive technology, and embedded technology. While active technologies require so-called external forces such as an electric field, fluid vibration, or surface vibration. While passive techniques require a special engineered surface, liquid additives or different tube inserts. The combined technology is a combination of the two previous technologies. The coiled tube configuration is widely used in industries such as heat recovery systems, power plants, nuclear reactors and food industries etc. In this study, a type of heat exchanger was chosen, which is a water bath heater with coiled tubes and low thermal efficiency in pressure reducing stations, which is the connection point between high-pressure lines and the consumer where the pressure drops from 790 Psi to 60 Psi. It is used to avoid the phenomenon of ice formation after the process. Extensive research has been conducted in this area, and it is divided into two parts.

The first part dealt with the research carried out on the water bath heater, and among these researches that affect this field, for example: studied [1] the thermal performance of a gas heater using two pipe designs to see which was optimal (vertical or horizontal inclination), where an improvement of about 5% was observed. In the second work [2], a mathematical method was found to enable us to find the Joule-Thomson factor and the lowest temperature of the gas before it is compressed. Where the results of their study helped in determining the minimum values for the temperature to enter the gas at different pressures into the recording device to avoid the formation of hydrates, and this study can be exploited to design appropriate temperature control systems for the water bath heaters and at the same time save the energy consumed to heat the gas. Among the results extracted, heating the gas to the calculated temperatures can save the energy consumption of the heaters by 43%. In a study [3], a ground heat exchanger was used to increase the gas inlet temperature in the pressure reducing station heater, and the results showed that the proposed system saves a large amount of fuel consumption required to heat the gas compared to traditional systems (without a heat exchanger), which confirmed the effectiveness of this method in enhancing economy. In this study [4], in order to save energy in the natural gas expansion station, a vertical ground-coupled heat pump system was proposed. Results showed that the system can save fuel by about 45.80% annually [5, 6]. Different coiled wires were used to increase heat transfer in heater. Where it was discovered that the heat performance of containers with shapes (round, square and cross triangle) increases sections by up to 57%, 62% and 79%, respectively. In the study [7], nanotubes with different concentrations were chosen to test the efficiency of the passive technology of the water heater and to use it as a method of heat transfer. In this study [8], the gases emitted from the chimney were exploited to increase the heating of the water inside the tank. The results showed that fuel consumption decreased by about 45%, while the heater efficiency increased. It was proposed to use alternating current in the tubes to enhance the heat efficiency of the water bath heater [9]. The result obtained showed that the Nusselt number increased by 20%, which leads to a reduction in fuel consumption. In this study [10], the twisted tube shape change was adopted as a means to increase and enhance the heat performance in a water bath heater with a capacity of 1000 m^3/h between the water bath and the heating coil. In this numerical study [11], heat transfer and fluid flow processes in three parts of the heater were studied. Heat is transferred in an indirect process between the combustion zone and the flow of natural gas inside the coil using currents generated in the water bath for natural convection.

In the second section, we highlighted research related to tube shape, including, for example: In this experimental study [12], pressure drop, heat transfer and efficiency were addressed in a corrugated double-tube heat exchanger. The results of the study indicated that the type of arrangement of corrugated pipes and the corrugations of the outer pipe have an effect on the thermal and frictional properties of the fluid passing through the two pipes [13]. In this numerical study, the enhanced flow field for heat transfer in 28 helical and sinusoidal corrugated tubes with different geometries was studied. The results of the study showed that an increase in the wave height will only lead to a slight increase in the Nusselt number, but to a greater loss of pressure. This study [14] provided us with in-depth research on heat and mass transfer inside heat exchangers with corrugated and convex tubes to the outside. While the study presented by Wang et al. [15] focused mainly on improving the double tube heat exchanger based on the outer spiral corrugated tube. This study demonstrated that the improvement of heat transfer is mainly through the effect of the liquid on the wall, while the heat transfers decrease linearly with increasing shell diameter. On the other hand, the pressure drops sharply when the diameter of the casing equals 38 mm, which is the best design in general. In the numerical study presented by Lee and Lee [16], the thermal and hydraulic characteristics of heat transfer inside corrugation heat exchangers of different shapes were analyzed. The results show that strong secondary flow is generated near the contact points, thus improving the heat performance of the thermal exchanger. The study [17] evaluated the effect of heater water temperature and convolution ratio on the heater performance characteristics, comparing the results with those without inclusion condition. The results indicated that the use of the inlet heater increased and improved its economic performance by 16% for both the twist ratio and water temperature of y = 1.05 and 35°C respectively. In this research [18], a heat exchanger with conical corrugated tubes and hard drive barriers was tested to determine the heat transfer efficiency using numerical simulation. The simulation results showed that the thermal transfer efficiency in conical corrugated tubes is much better than smooth tubes and even heat transfer tubes. Other highly efficient. This study [19] examined how heat transfer and gel fuel flow occur in corrugated channels. From the results obtained, it was found that the recirculation zones resulting from the undulations enhance convection and are useful in reducing viscosity, and that the pressure drop increases due to the dissipation of the viscosity resulting from recirculation. On the other hand, for a sine wave, shorter wavelength and deeper wave amplitude are better to reduce the viscosity and increase the pressure drop.

In this work, the corrugated tube is used as the gas coil for a water bath heater with low thermal efficiency. The corrugated tube is designed to comply with the reference model [10] twisted coil configuration under the same working conditions, where the gas enters the coil at a minimum temperature of approximately 283.15 K and a water bath temperature of 313.15 K. While the inlet velocity of natural gas is about 14 m/s, the inlet pressure is 790 psi and the outlet pressure is 60 psi. Moreover, among the aims of this study, we intend to find the relationship between both flow velocity and heat exchange with wave amplitude inside the gas coil.

2. DESCRIPTION AND PRESENTATION OF THE PROBLEM

Natural gas can freeze (due to the Joule-Thomson effect) as it passes pressure reducing stations, interrupting the gas supply and damaging valves and instrumentation. Studies have proven at the present time that the indirect heater (Figure 1) is widely used in natural gas pressure reduction stations, as the water tank contributes significantly to the transfer of heat between the combustion zone and the gas coil that passes through the tank, but with low thermal efficiency. The indirect heater consists of the following elements:

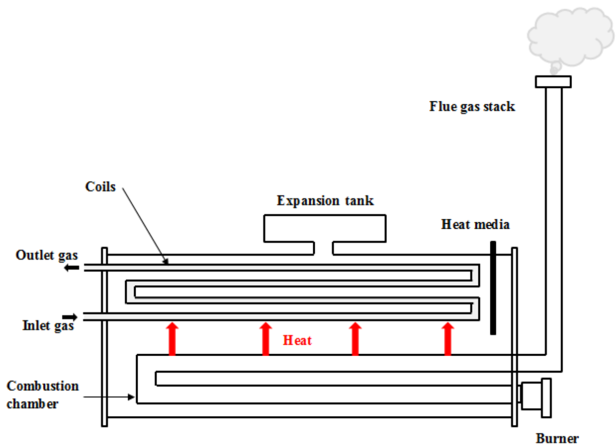


Figure 1. Indirect water bath heater

The natural gas coil consists of four pipe rows with a diameter of 49.22 mm and 20 waves per path. Figure 2 shows the details of the gas profile used.

2.1 Mathematical formulation

To model the turbulent flow, the Navier-Stokes equations of the Reynolds mean with the Reynolds stress equation model were used to solve the process of turbulent flow and heat transfer of the gas in the gas coil of the water bath heater. This mathematical model has been used to predict turbulent flow characteristics and thermal performance in many complex channels. For a constant, compressible fluid, in the present work, the realizable k-ε model was used as given by the mathematical formulas, which are considered in the Cartesian coordinate system as follows:

The instantaneous continuity equation in turbulent regime is an equation which describes the flow of a conserved quantity. Since the flow is permanent and the density is constant, it is expressed as follows:

$$\frac{\partial V_j}{\partial x_j} = 0 \tag{1}$$

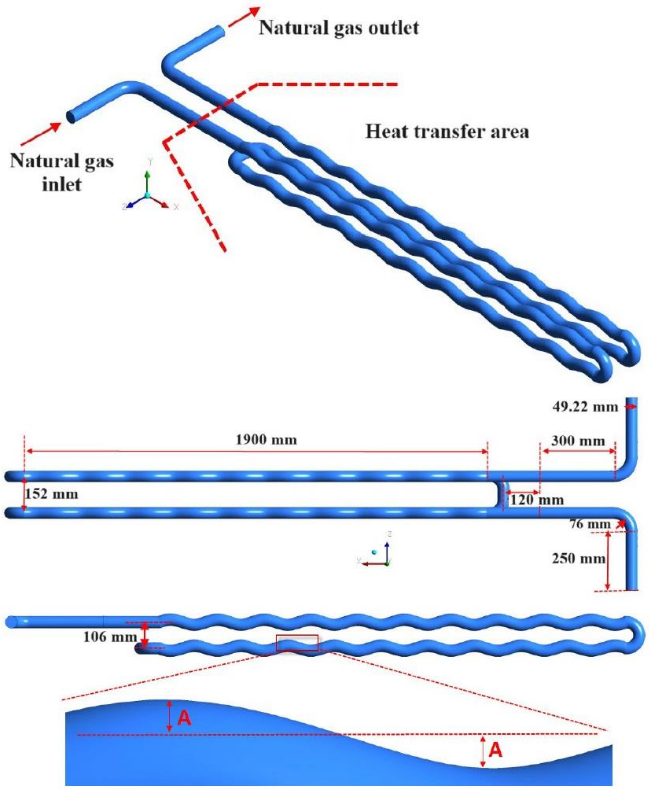
By introducing the Reynolds decomposition:

$$V_j = \overline{V_j} + V_j' \tag{2}$$

In the continuity Eq. (1), and taking its average, we obtain the continuity equation of the mean field

$$\frac{\partial \overline{V}_j}{\partial x_j}=0 \tag{3}$$

This equation arises from the application of Newton's 2nd law to an elementary control volume of fluid. It makes it possible to establish relationships between the characteristics of the fluid, those of its movements and the causes which produce them.



Model 1	Model 2	Model 3
A = 10 mm	A = 15 mm	A = 20 mm

Figure 2. Details of gas coil

$$\frac{\partial V_j}{\partial t}+\rho V_j\frac{\partial V_j}{\partial x_j}=-\frac{\partial P}{\partial x_i}+\frac{\partial \tau_{ij}}{\partial x_j}+\rho g_i \tag{4}$$

By introducing the Reynolds decomposition into the conservation momentum equation and taking the average, we obtain:

$$\frac{\partial \overline{V}_j\overline{V}_i}{\partial x_j}=-\frac{1}{\rho}\frac{\partial \overline{P}}{\partial x_i}+\nu\frac{\partial^2 \overline{V}_i}{\partial x_j^2}+\frac{\partial}{\partial x_j}(-\overline{v_i^{'}v_j^{'}})+\rho g_i \tag{5}$$

Energy equation: We take the equation and we apply the Reynolds decomposition and we average:

$$\frac{\partial}{\partial x_j}(\overline{V}_j\overline{T})=\frac{\lambda}{\rho C_p}\frac{\partial^2 \overline{T}}{\partial x_j^2}+\frac{\partial}{\partial x_j}(-\overline{v_j^{'}t^{'}}) \tag{6}$$

For heat flow

$$-\overline{v_j^{'}t^{'}}=\frac{\nu_t}{\sigma_t}\frac{\partial \overline{T}}{\partial x_j} \tag{7}$$

where the turbulent viscosity is characterized by

$$\nu_t=C_\mu\frac{k^2}{\varepsilon} \tag{8}$$

The dissipation ε is obtained from the transport Eq. (9) and C_μ the parameter of the k - ε model.

$$\frac{\partial}{\partial x_j}(\overline{V}_j\varepsilon)=\frac{\partial}{\partial x_j}\left((\nu+\frac{\nu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}(\frac{\varepsilon}{k})P_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{9}$$

For $C_{1\varepsilon}, C_{2\varepsilon}, C_\mu$ are the constants of the k - ε model, determined experimentally.

2.2 Thermohydraulic performance

The thermal transfer rate, for the corrugated twisted tube, was calculated as follows:

$$\phi=\dot{m}Cp\left(T_{out}-T_{in}\right) \tag{10}$$

While the following relationship shows how to calculate the heat transfer rate between the water and the pipes:

$$\phi=h\cdot S\left(T_s-T_t\right) \tag{11}$$

T_s represents the surface temperature of the tubes, which is assumed to be approximately the same as the heater water temperature, with respect to the average value of the gas temperature inside the tubes T_t, which is obtained from the following equation:

$$T_t=\frac{T_{out}+T_{in}}{2} \tag{12}$$

As shown below, Nusselt number Nu can be expressed by the following Eq. (14):

$$h=\frac{\dot{m}Cp\left(T_{out}-T_{in}\right)}{S\left(T_s-T_t\right)} \tag{13}$$

$$Nu=\frac{h\cdot D}{\lambda} \tag{14}$$

The Fanning friction coefficient can be calculated using the following formula by calculating the pressure loss in the corrugated twisted tube [20]:

$$f=\frac{2D\Delta P}{\rho LV^2} \tag{15}$$

where, ΔP and L are the pressure drop and length of the corrugated twisted tube.

2.3 Boundary conditions of problem

Based on data from previous studies, natural gas enters the coil at a minimum temperature of about 283.15 K, which is generally equivalent to the ground temperature and water bath temperature of 313.15 K. The natural gas inlet velocity is about 14 m/s, inlet pressure is 790 Psi and outlet pressure is 60 Psi. Under these conditions, the natural gas outlet temperature should range between 300.15 and 310.15 K. One of the components of natural gas is methane, which reaches 98.2%. Therefore, we assumed the natural gas flow to be pure methane in the simulations. In this research, the Thermo-physical properties of me-thane were determined as shown in Table 1.

Table 1. Properties of methane

Property	Definition
Density	Ideal gas
Specific heat	2191.40
Dynamic viscosity	1.212e-5
Thermal conductivity	0.0375
Molecular weight	16.043

3. NUMERICAL RESOLUTION

For the numerical validation of the computer code, we compared the results obtained in the present study with the numerical results obtained by Soleimani et al. [10].

Table 2 presents the values of the average gas discharge temperature determined for a volume flow of 1000 m^3/h, the inlet gas temperature in the coil 283.75 K, the water bath temperature 313.15 K, the inlet pressure 5447 KPa and the outlet pressure 413.685 kPa. We notice that the value of the average temperature is almost the same with the relative error being 0.0091%.

Table 2. Comparison between the present study and those of [10]

	Present Study	Soleimani et al. [10]	Error [%]
T_{av}	308.1384	308.1103	0.0091

3.1 Mesh study

The first work to be done to achieve the numerical simulation is to define a mesh that adapts to the flow. The accuracy of the results depends on their quality. To obtain accurate results within acceptable calculation times, the quality of the network must be improved, and a sensitivity analysis of the network is performing. A structural network has been applied to the models used: the network close to the pipe walls is well refined compared to the average network. A mesh study was performed on the Model 1 by obtaining the gas temperature average at the outlet.

Comparing the results of Table 3, mesh 3 was identified as the best compromise between the accuracy of the results and the timeliness of the calculations.

Table 3. Mesh-independent analysis results

	Grid Cell	Outlet Temperature Average
Mesh 1	192785 nodes	309.3504
Mesh 2	329490 nodes	309.4422
Mesh 3	651444 nodes	309.4923

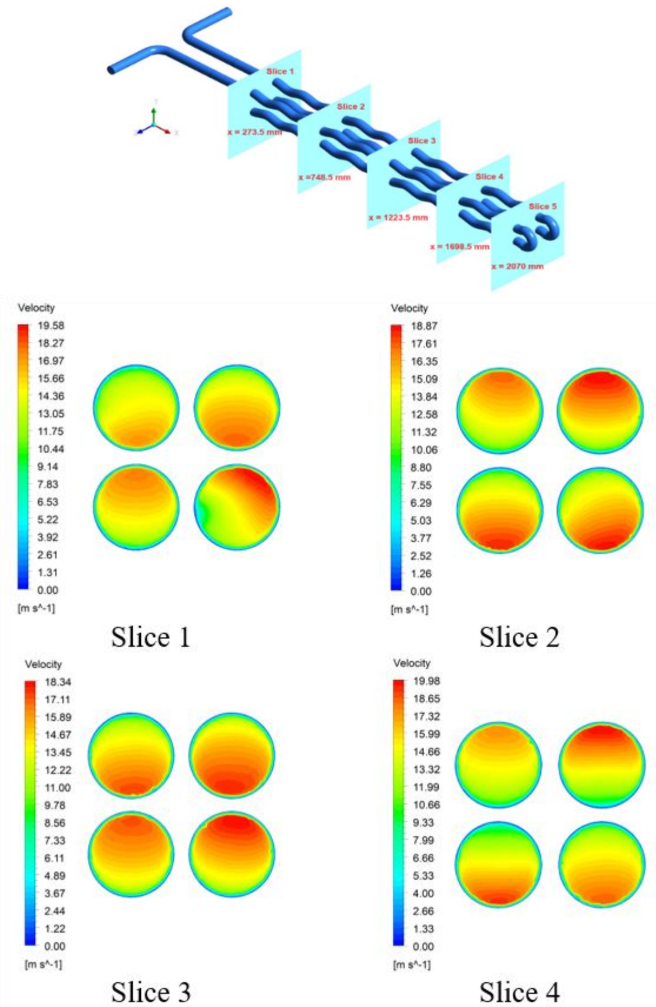
In this work, the coupled algorithm was used in the pressure-velocity pair and a relative error of less than 10^{-5} was used in all flow variables. Gradient terms were also calculated applying the Green-Gauss node-based method.

4. RESULTS AND DISCUSSION

In this study, three different models were used based on changing the wave amplitude. The effects of wave amplitude on fluid flow and heat transfer properties are shown below. Numerical results were obtained for the inlet velocity 14 m/s and the pressure at the inlet and outlet was 790 psi and 60 psi.

4.1 Velocity contours

Figure 3 shows how changing the wave amplitude in Models 1 and 3 affects the flow field for five longitudinal slices for the same operating conditions .In model 1, for wave amplitude A = 10 mm, it can be observed that the horizontal vortices are present in slices (1-4), where the maximum vortex point moves towards the wave head. Similar to slice 1 in the third path, the tilt can be observed in the maximum vortex, and this is due to the centrifugal force experienced by the fluid in the horizontal attachment preceding it at an angle of 180°. As for slice 5, the longitudinal section is the one preceding the vertical elbows, where the point of maximum vortex can be observed at the wave head and on the side of the centrifugal force, while the point of minimum vortex is located on the other side.



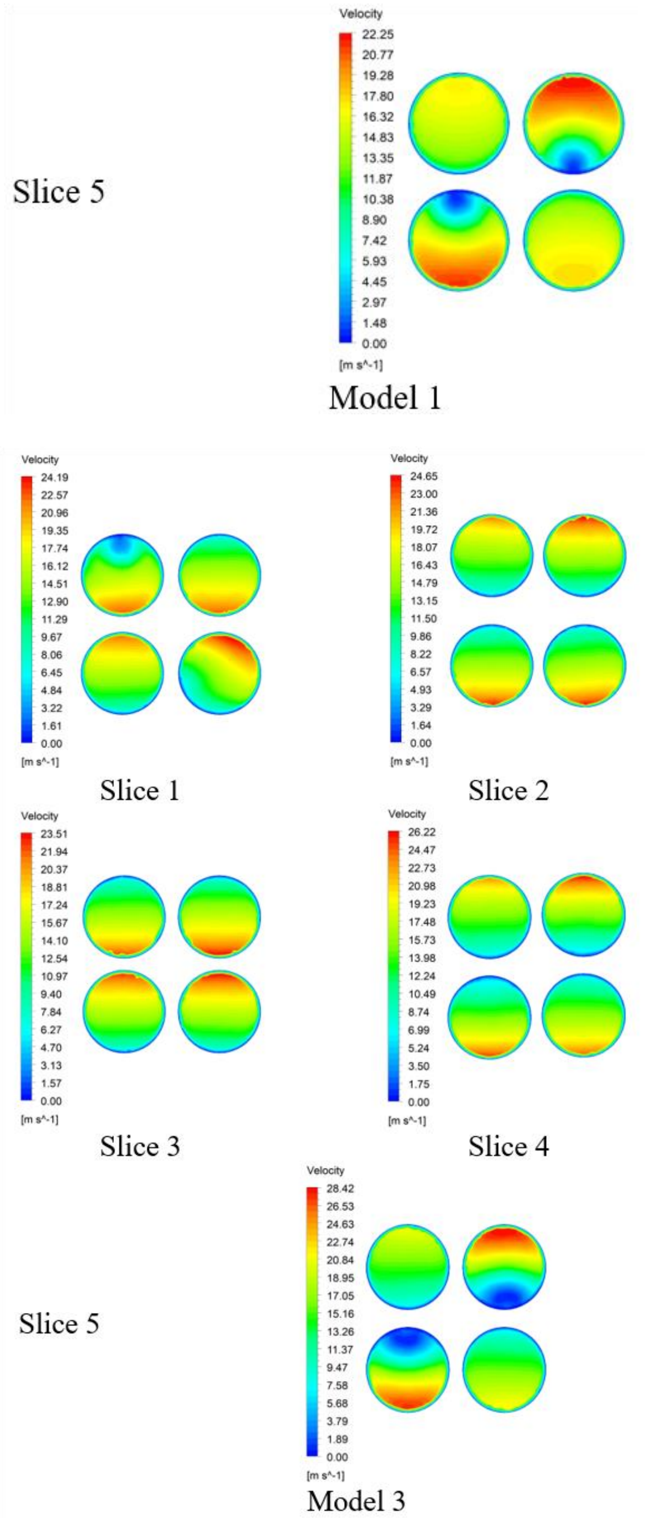


Figure 3. Velocity contour for different longitudinal slices

In Model 3, with respect to the flow distribution in the five slices, the maximum vortex size decreases compared to Model 1, but the velocities increase. While the size of the vortex minimum in slice 5 is increasing, on the other hand, the presence of a vortex minimum in slice 1 can be observed in the first path induced by the first wave after a straight path, which represents a stopping point for the fluid. Moreover, at high wave amplitude, the flow field differs significantly from the flow field in coils at low heights with vortex. It can be said that as the fluid approaches the elbows and because there is not enough space for it to pass, the direction of the flow changes and the normal and normal velocities intensify.

4.2 Isothermal contours

Figures 4 and 5 show the effect of coil amplitude on temperature contours at different longitudinal slices. Natural gas inlet the coils at a temperature of 283.15 K and the water temperature at 313.15 K. It is clear that the amplitude of the coil wave affects the thermal and hydraulic performance of the water heater as the temperature lines differ from each other. As shown in the slides, the wavelength and amplitude of the natural gas coil are not enough to achieve a uniform temperature, and the effect of the generated liquid (gas) turbulence continues to change the temperature. On the other hand, it can be seen that the model 3 shows better heat transfer compared to models 1 and 2. Based on the previous results, the flow velocity increases with the increase coil wave amplitude, this leads to intensification of heat transfer between the hot water in the shell and the gas passing through the coil. Heat exchange has been shown to be effective in areas where fluid recirculation occurs in the coils.

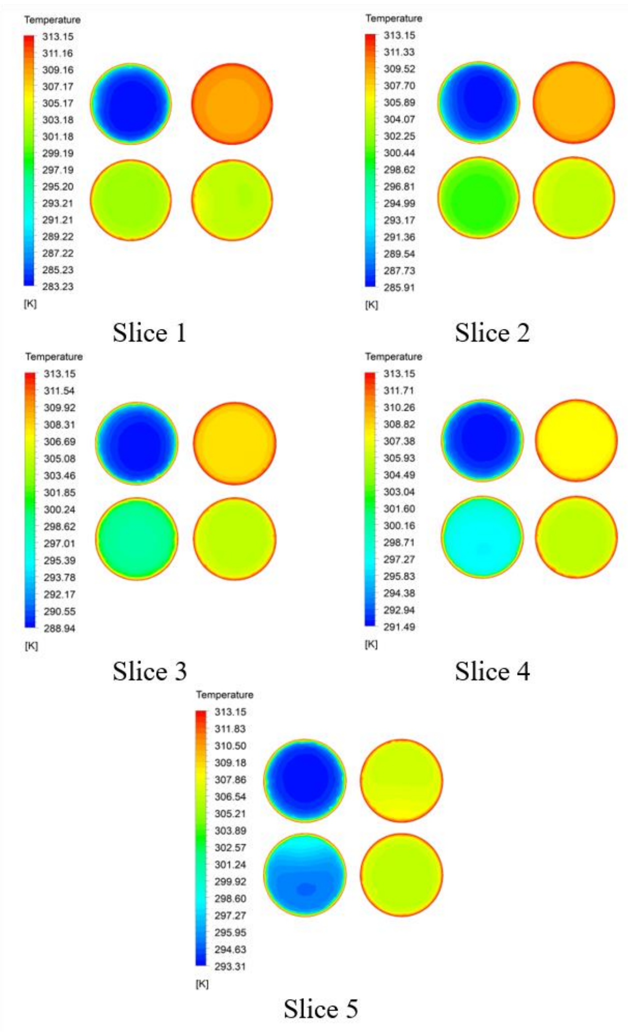


Figure 4. Temperature contour for different longitudinal slices (Model 1)

To validate the previously obtained results and to enable us to make a numerical comparison between the three models used, plots of temperature change in the axial line of the coil are presented, as shown in Figure 6.

The diagrams are drawn in straight sections only without elbows for ease of comparison. What can be noticed at first glance is the high temperature of natural gas in all its paths.

Maintaining the highest altitude in the first path, followed by the second path. Comparing the three models used, increasing the wave amplitude leads to an increase in the size of the recirculation zones in the coil, which leads to increased heat exchange between the hot water and natural gas in the coil. On the other hand, by comparing the three models, a difference was found in the average temperature at the outlet, reaching 6.92‰ between Model 1 and Model 3.

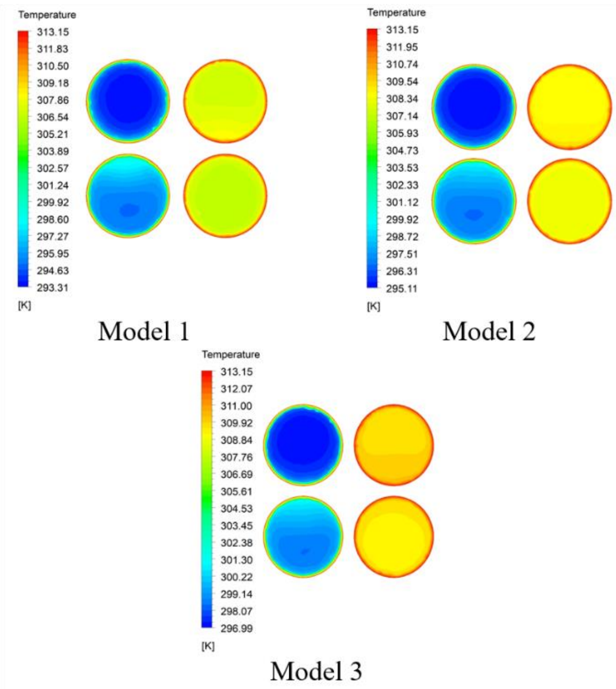
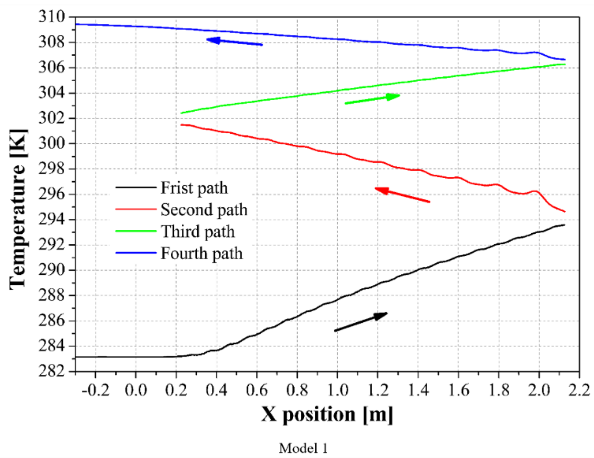
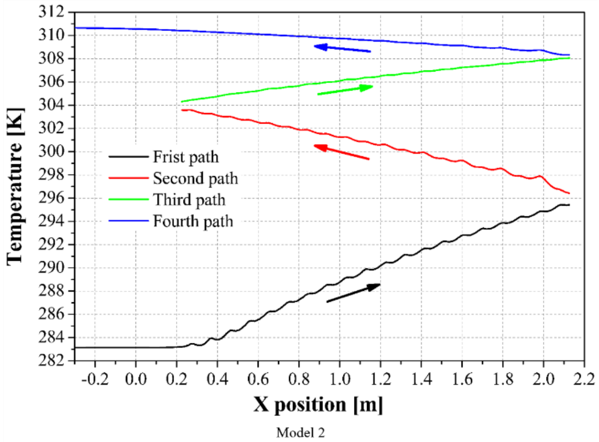


Figure 5. Temperature contour for different models (Slice 5)





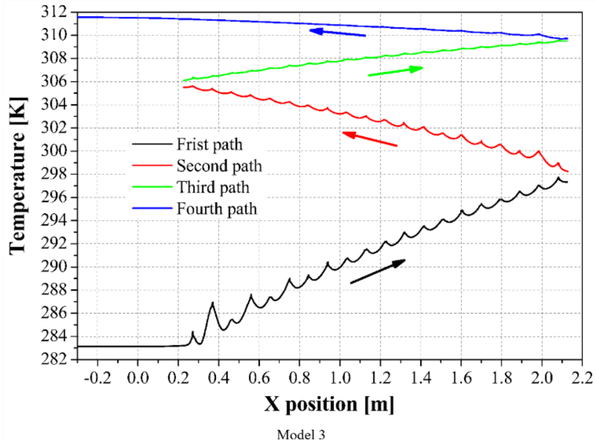


Figure 6. Temperature variation across the center line of the coil

4.3 Variation of the Nusselt number

The relationship between the Nusselt number and the wave amplitude of the corrugated coil is shown in Figure 7. Depending on the change in the wave amplitude, using the twisted corrugated coil as the inner tube of a water bath heater increases the Nusselt number by about 6.57%. The maximum enhancement of the heat exchange coefficient occurred at the largest capacity. The possible reason for this maximum enhancement is described as follows: It can be said that the corrugation of the coil forces the cold fluid (gas) to form rotation points at the level of the thermal boundary layer between the inner and outer tube surfaces. In other words, coil corrugations increase turbulence and increase the rate of heat transfer in the recirculation zones.

Through the results achieved, it was shown that the relationship that predicts the average Nusselt numbers as a function of the wave amplitude along the twisted corrugated coil subjected to heat exchange with the shell, with a maximum error rate of 0.51%, was explained as follows:

$$Nu_{av} = 272.45 + 1911.027A \tag{16}$$

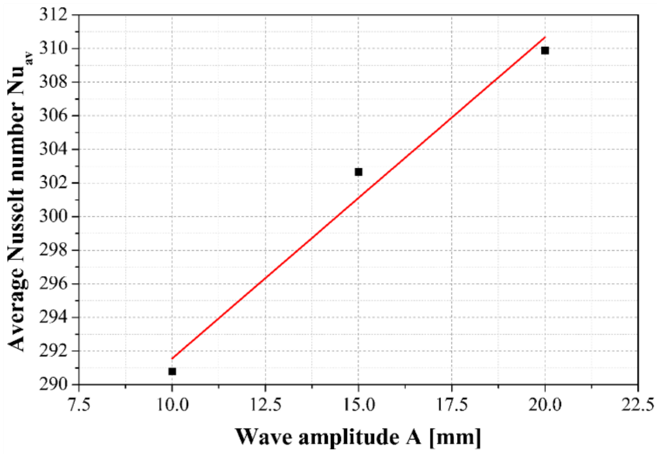


Figure 7. Relationship between average Nusselt number and the wave amplitude of a twisted corrugated coil

4.4 Friction factors

Figure 8 shows the friction factors inside the twisted corrugated coil for different capacities. It can be seen that the greater the amplitude of the wave, the greater the friction factor. Using a larger capacity corrugated coil, such as the inner tube of a water bath heater, increases the friction factor by about 114%. However, the friction factor can be overlooked in the application field since the gas will pass through the pressure reducer after leaving the heater.

The relationship combining the friction factor as a function of the wave amplitude along the twisted coil, with a maximum error rate of 4.96%, is shown as follows:

$$f = -0.0026 + 3.54A \tag{17}$$

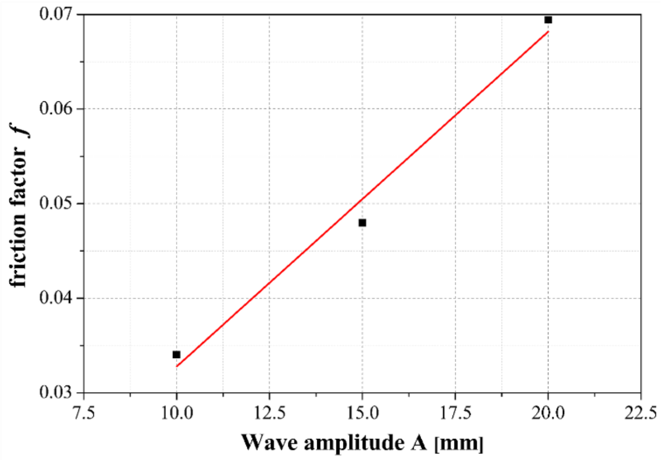


Figure 8. Relationship between friction factor and the wave amplitude of a twisted corrugated coil

5. CONCLUSION

In this research, improving heat transfer in a water bath heater with low thermal efficiency in a gas expansion plant is mentioned. The main scope of this study was to compare three models of corrugated coils with different wave amplitudes. The results of this study were presented in the form of streamlines and isothermal lines, such as the thermal performance coefficient (Nusselt number) and the coefficient of friction. The main findings on how changes in ripple amplitude affect the flow field and thermal performance can be summarized as follows:

- One of the advantages indicated by the results is that the use of high-capacity corrugated tubes helps improve the Nusselt number and heater performance, as the wave amplitude had a significant impact on the thermal and frictional properties.
- Wave height plays a negative role in the friction factor, however, depending on the field of application, this disadvantage can be reduced to a minimum.

In general, the heat exchange inside the corrugated twisted gas coil can be enhanced by increasing the wave amplitude.

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NOMENCLATURE

A	amplitude of wave, mm
Cp	specific heat, J. kg^{-1}. K^{-1}
f	friction factor
g	gravitational acceleration, $m.s^{-2}$
h	average heat transfer rate
k	turbulence kinetic energy, m^2. s^{-2}
L	length of the corrugated twisted tube
$\dot{m}$	mass flow rat
Nu	Nusselt number
P	pressure, Psi
S	exchange surface
T	temperature, K
t	time, s
V	velocity, m/s
$\overline{V}$	average velocity, m/s
V'	fluctuating velocity
x,y	cartesian coordinate, m

Greek symbols

Δ	drop
ε	rate of dissipation of turbulent kinetic energy, m^2. s^{-3}
ϕ	heat flux
λ	conduction heat transfer coefficient, W.m^{-1}. k-^{1}
μ	dynamic viscosity, kg. m^{-1}. s^{-1}
ν	kinematic viscosity, m^2. s^{-1}
ρ	density, Kg.m^{-3}

Subscripts

av	average
i,j	imposed
in	inlet
out	outlet
s	tube surface
t	average temperature of the gas inside the tubes