



Article

Performance enhancement of water bath heater at natural gas city gate station using twisted tubes

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ARTICLE INFO

Article history:

Received 12 November 2018

Received in revised form 1 February 2019

Accepted 11 March 2019

Available online 4 April 2019

Keywords:

Natural gas city gate station

Water bath heater

Heat transfer enhancement

Twisted tube

ABSTRACT

Natural gas is transported from producing regions to consumption regions by using transmission pipelines at high pressures. At consumption regions, the pressure of natural gas is reduced in city gate stations (CGSs). Before the pressure reduction process, the temperature of natural gas is increased usually by using a water bath heater, which burns natural gas as fuel, to protect against freezing of natural gas. These types of heat exchangers have a low efficiency and consume a lot of fuel to generate the required heat. In the current study, the twisted configuration of the heating coil is proposed and investigated to enhance the heat transfer through a water bath heater with a nominal capacity of $1000 \text{ m}^3 \cdot \text{h}^{-1}$. Firstly, the implementation procedure is validated with data collected from the CGS of Qaleh-Jiq (located in Golestan province of Iran). A very good agreement is achieved between the obtained results and the real data. Then, three different twist ratios are considered to examine the twisting effects. The proposed technique is evaluated in the terms of velocity, temperature, and pressure variations, and the results are compared with the conventional case, *i.e.* straight configuration. It is found that both the heat transfer rate and the pressure drop augment as the twist ratio is raised. Finally, it is concluded that the twisted tubes can reduce the length of the gas coil by about 12.5% for the model with low twist ratio, 18.75% for the model with medium twist ratio, and 25% for the model with high twist ratio as compared to the straight configuration.

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1. Introduction

The consumption of natural gas in the world is rising sharply. According to the report provided by BP Statistical Review of World Energy [1], Iran is the first largest consumer in Eurasia & Europe countries with the total amount of 200.8 billion cubic meters. It clarifies that reducing natural gas consumption in such countries is so serious, and it must be implemented from producing wells to end consumers.

A city gate station (CGS) is one of the most important parts in the natural gas transmission system, where the processes of filtering, heating, pressure dropping, flow rate measuring, and odorizing (addition of an odorant to natural gas) occur, respectively. From the consumption point of view, the major issue in the CGSs is extreme consumption of natural gas as fuel during the heating process which is generally performed in a water bath heater. This equipment usually has a low thermal efficiency, so it is very important to look for appropriate heat transfer enhancement schemes in order to reduce natural gas consumption and save energy. As mentioned by Ghezelbash *et al.* [2], investigations on energy saving of CGSs could be classified into two areas;

(I) energy recovery: most studies focus on this area, such as [3–9] and (II) reducing energy consumption: few studies focus on this area, such as [10–12].

Akhlaghi *et al.* [13] proposed the use of a solar collector array at a CGS to provide a part of heat demand and reduce the amount of fuel consumption. It was reported that a system with an array of 430 collector modules can save the annual fuel about 41765 USD. The effect of the number of collector on the performance of a CGS was investigated by Farzaneh-Gord *et al.* [14]. It was found that as the number of collector was enhanced, the fuel cost decreased, but the capital cost increased. They detected the economic feasibility using Simple Payback Ratio and Net Present Value methods. The payback ratios of 6.9 and 11 years were reported by the Simple Payback Ratio and Net Present Value methods, respectively. Ashouri *et al.* [15] examined the influences of the tube bundle arrangement on the thermal performance of a water bath heater. It was found that the horizontal and vertical pitches had noticeable effects, and the optimum design can enhance the thermal performance of the heater by about 5%. In the other work [16], the same team performed a computational technique to calculate the Joule-Thomson coefficient and the minimum possible temperature of natural gas before the pressure regulator. These values can be used to design an appropriate heating system, *i.e.* water bath heater. For the considered

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case, it was concluded that natural gas heating at the calculated minimum temperature can save energy consumption of heater up to 43%. Although employing solar energy in CGSs was previously proposed, the automatic control system and exergy destruction rate were not considered. Hence, Farzaneh-Gord *et al.* [17] studied the ability of solar heating equipped with controllable systems in the terms of fuel providence and exergy destruction rate.

Heat loss from the exhaust flue gas system is the main factor to enhance the efficiency of the water bath heaters. In order to preheat the inlet natural gas to the water bath heater, a shell-tube heat exchanger working with the exhaust gas as hot fluid was suggested by Azizi *et al.* [18]. The results showed that the proposed technique can increase the efficiency of the water bath heater up to 11%. The possibility of using a geothermal heat exchanger was investigated by Farzaneh-Gord *et al.*

[19]. It was reported that a system with 8 boreholes with 150 m depth was the best configuration for the considered case. In order to decrease the fuel consumption in the water bath heater, a new temperature controller system was also proposed by Zabihi and Taghizadeh [20]. The results disclosed that the proposed system can decrease the fuel consumption more than 431000 SCM and save the cost over 43000 USD annually. Adding a solar heating system as well as replacing the throttling valve by a turbo expander were studied by Arabkoohsar *et al.* [21], and 3.5 years of payback ratio was obtained by the Net Present Value method. Further, the use of vortex tube and vertical ground heat exchanger was proposed Ghezlbash *et al.* [2]. In this work, a vortex tube, which can divide the mainstream into two cold and hot parts, was applied instead of the throttle valve. The separated cold stream was directed to a shell-tube heat exchanger and received the

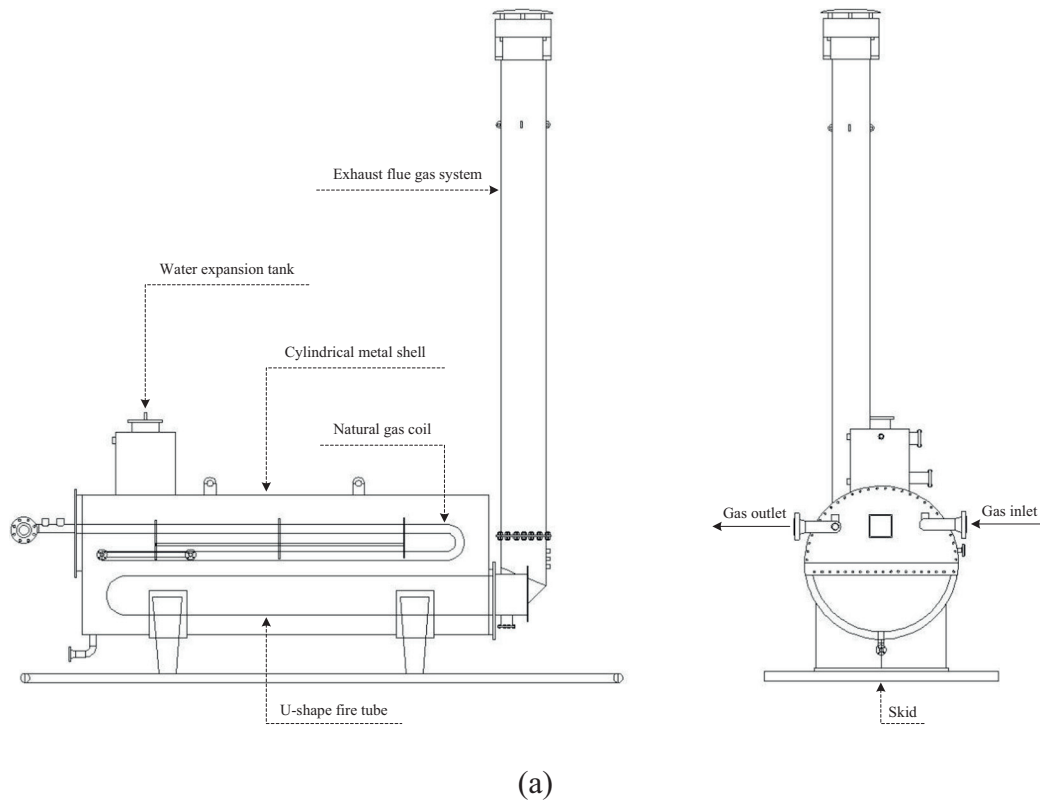


Fig. 1. A typical indirect water bath heater: (a) schematic diagram and (b) real photograph [28].

geothermal heat. It was reported that this technique can reduce the energy consumption up to 88% with a payback period less than 4.5 years. In the other study [22], they used a vertical ground-coupled heat pump system with a reduction potential up to 65% in the fuel consumption. Also, a combined heat/power technology was proposed and investigated by the same team [23]. The feasibility of employing an evacuated solar system to supply the required heat for a hybrid power plant was tested by Arabkoohsa *et al.*[24]. In order to evaluate this technique compared with the previous candidates, the Internal Rate of Return method was used, and the criterion of 11.1% was reported. An analysis was carried out on different seasonal strategies to explore energy/exergy losses as well as exergy destructions by Olfati *et al.*[25]. In all strategies, the maximum losses were obtained in the stack, and the exergy efficiency values of 77% and 69% were reported in the winter and the summer, respectively. Finally, the possibility of using a novel plant configuration consisting of a two-stage expansion system was presented by Borelli *et al.*[26].

Based upon the above review of the literature, the most important problems in the CGSs are high natural gas consumption and low efficiency of water bath heaters. Thus, substantial modification in the design of water bath heaters is an essential issue. However, studies in this field are very scarce. For instance, Salari and Goudarzi [27,28] used different coiled wire inserts to intensify the heat transfer in a water bath heater. It was detected that the thermal performance of the cases with circular, square, and triangular cross-sections can

increase up to 57%, 62% and 79%, respectively. Rahmati and Reiszadeh [29] proposed the use of multi-walled carbon nanotubes in ethylene-glycol/water as an advanced heat transfer media in a water bath heater. The efficiency of the applied passive technique was tested at different concentrations of nanotubes. The results showed the increase of 34.5 °C in the outlet temperature of natural gas and the enhancement of 48% in the thermal efficiency at the volume fraction of 0.05. In the current study, the possibility of improving hydrothermal performance of a water bath heater, which is used in a CGS with the capacity of $1000 \text{ m}^3 \cdot \text{h}^{-1}$, is investigated by employing the twisted tubes as an enhanced passive technique. During the last years, the twisted tubes are employed in various heat exchange devices for the heat transfer enhancement of both laminar [30–33] and turbulent [34–36] flows. However, this is the first time that these geometries are proposed and tested for the application in the water bath heater of the CGSs.

2. Details of Considered Cases

2.1. Description of a real system

As mentioned early, heating of the natural gas is one of the steps in a CGS. The natural gas must be heated before the pressure dropping step to ensure that it remains above the hydrate-formation zone and dew point. It is usually performed using an indirect water bath heater to preserve safety. The thermal performance or heating duty of this heat

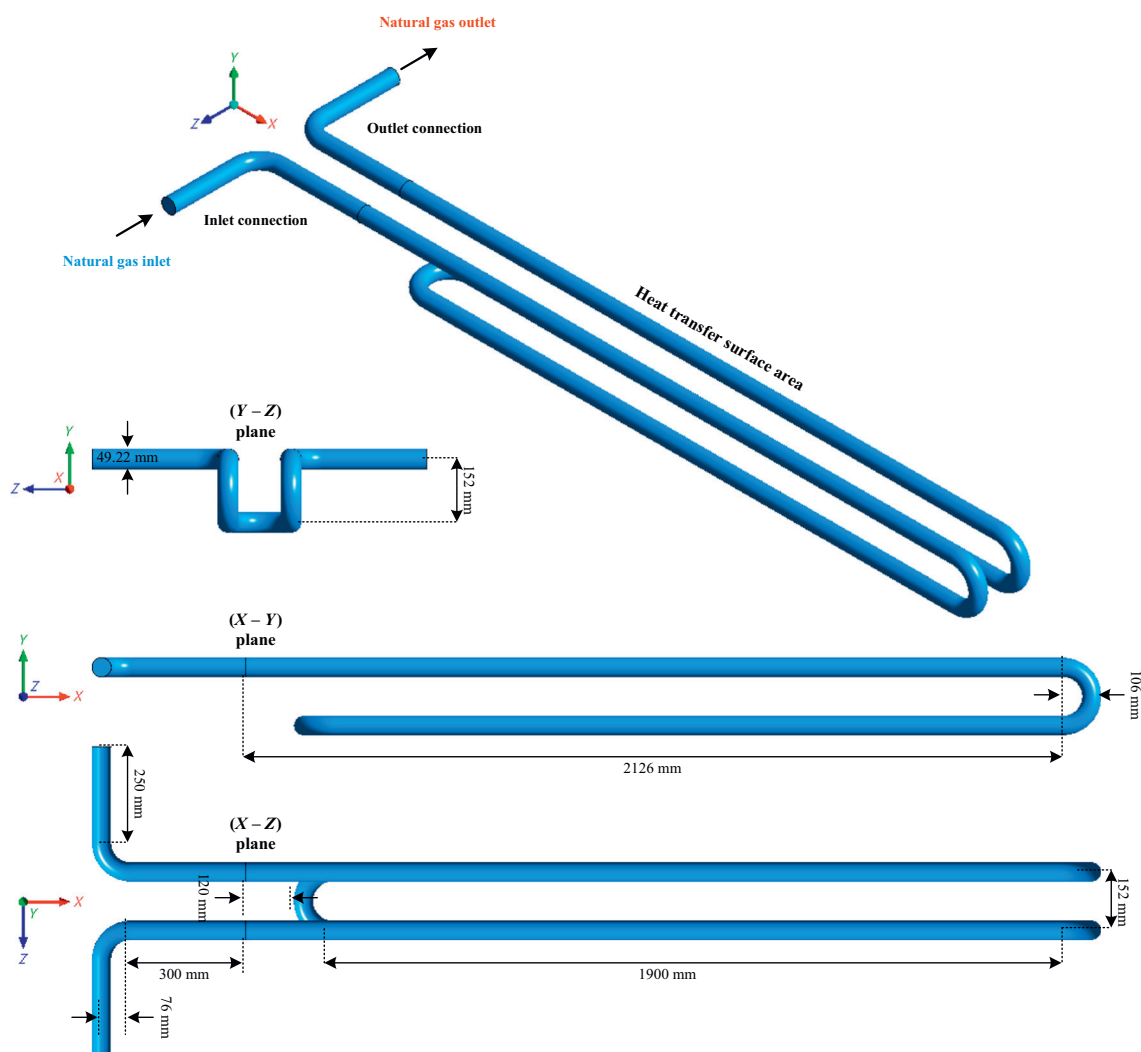


Fig. 2. Details of gas coil.

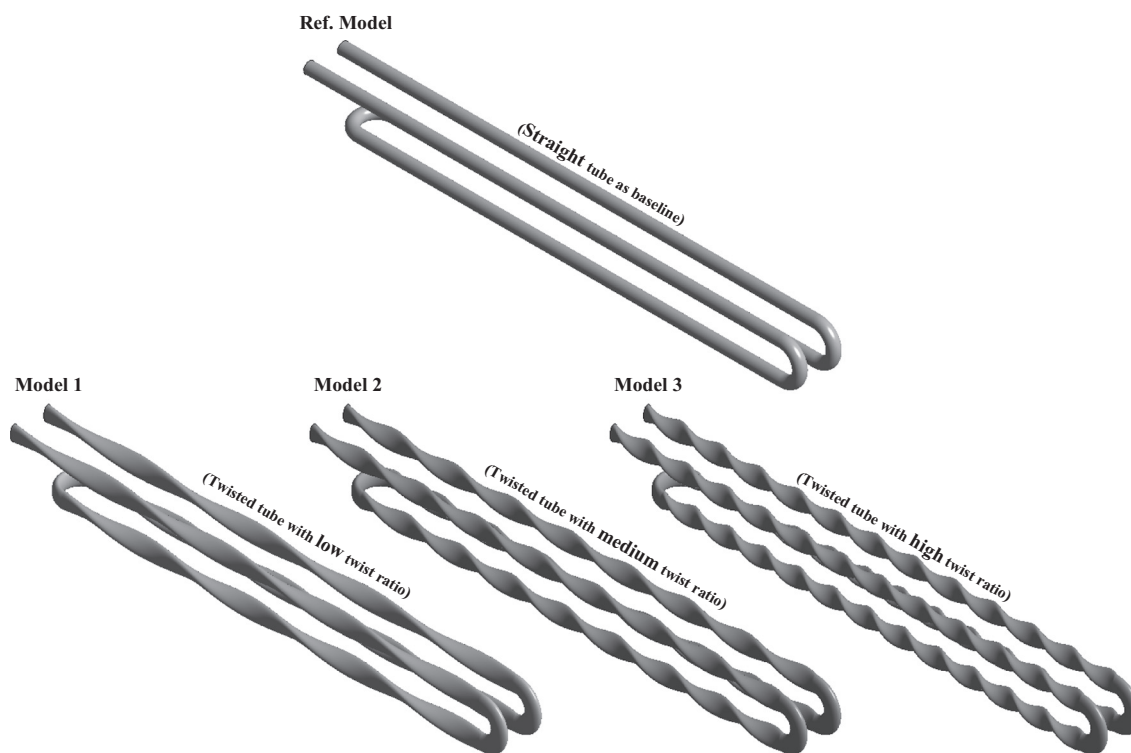


Fig. 3. Considered models.

exchanger depends on operating factors, such as inlet temperature, pressure, and flow rate of natural gas. Therefore, it should be designed based on desirable values of operating factors and environmental conditions of the area.

In this study, the considered case is an indirect water bath heater with a nominal capacity of $1000 \text{ m}^3 \cdot \text{h}^{-1}$. It is generally applied in the CGSs working at the inlet pressure and temperature lower than 1000 kPa and 288.15 K, respectively. Fig. 1(a–b) illustrates a schematic

Table 1
Operating condition of Qaleh-Jiq CGS at first month of year

Average daily values					
Day	Station inlet pressure/kPa	Station outlet pressure/kPa	Natural gas temperature before water bath heater/K	Natural gas temperature after water bath heater/K	Flow rate/ $\text{m}^3 \cdot \text{d}^{-1}$
1	4895.277	413.685	283.15	305.15	8415
2	5515.805	413.685	284.15	307.15	9264
3	5550.279	413.685	283.15	305.15	8695
4	5515.805	413.685	284.15	304.15	9751
5	5308.963	413.685	283.15	303.15	9069
6	5446.858	413.685	281.15	303.15	8467
7	5033.172	413.685	282.15	305.15	6798
8	5826.069	413.685	284.15	304.15	4761
9	5446.858	413.685	284.15	307.15	6627
10	5102.120	413.685	284.15	307.15	6742
11	5102.120	413.685	283.15	306.15	6393
12	5619.227	413.685	284.15	306.15	8264
13	5550.279	413.685	284.15	306.15	8613
14	5550.279	413.685	284.15	306.15	9794
15	4895.277	413.685	283.15	299.15	10,685
16	5377.910	413.685	283.15	303.15	9204
17	4516.066	413.685	282.15	305.15	7086
18	5653.700	413.685	282.15	305.15	5667
19	5653.700	413.685	284.15	294.15	5519
20	5619.227	413.685	285.15	305.15	3690
21	5722.648	413.685	281.15	307.15	3798
22	5653.700	413.685	285.15	303.15	6267
23	5550.279	413.685	285.15	306.15	6261
24	5481.332	413.685	283.15	306.15	7159
25	5619.227	413.685	284.15	306.15	4543
26	5757.122	413.685	286.15	306.15	3900
27	5584.753	413.685	281.15	306.15	4913
28	5550.279	413.685	285.15	307.15	4753
29	5653.700	413.685	285.15	307.15	3688
30	5653.700	413.685	287.15	303.15	2702

Table 2
Thermo-physical properties of Methane

Property	Definition
1. Heat capacity	$c_p/R = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4$ $a_1 = 5.14987613$, $a_2 = -1.36709788 \text{ e}^{-2} \text{ K}^{-1}$, $a_3 = 4.918005599 \text{ e}^{-5} \text{ K}^{-2}$, $a_4 = -4.84743026 \text{ e}^{-8} \text{ K}^{-3}$, $a_5 = 1.66693956 \text{ e}^{-11} \text{ K}^{-4}$
2. Dynamic viscosity	$\mu = \mu_0 (T/T_0)^{3/2} (T_0 + S/T + S)$ $T_0 = 273.15 \text{ K}$, $S = 197.8 \text{ K}$, $\mu_0 = 1.172592 \text{ e}^{-5} \text{ Pa} \cdot \text{s}$
3. Thermal conductivity	$k = k_0 (T/T_0)^{3/2} (T_0 + S/T + S)$ $T_0 = 273.15 \text{ K}$, $S = 197.8 \text{ K}$, $k_0 = 0.0362914 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$

Table 3
Effects of grid number on outlet temperature of fluid

Test No.	Number of grids	Outlet temperature/K or °C	Deviation/%
1	1051440	307.6316 or 34.4816	–
2	1422720	307.8568 or 34.7068	0.6531
3	1778400	307.9655 or 34.8155	0.3131
4	2134080	308.0266 or 34.8766	0.1755
5	2489760	308.0643 or 34.9143	0.1081
6	2803840	308.0953 or 34.9453	0.0887
7	3201120	308.1211 or 34.9711	0.0738
8	3504800	308.1274 or 34.9774	0.0180
9	3912480	308.1308 or 34.9808	0.0097
10	7680560	308.1310 or 34.9810	0.0006

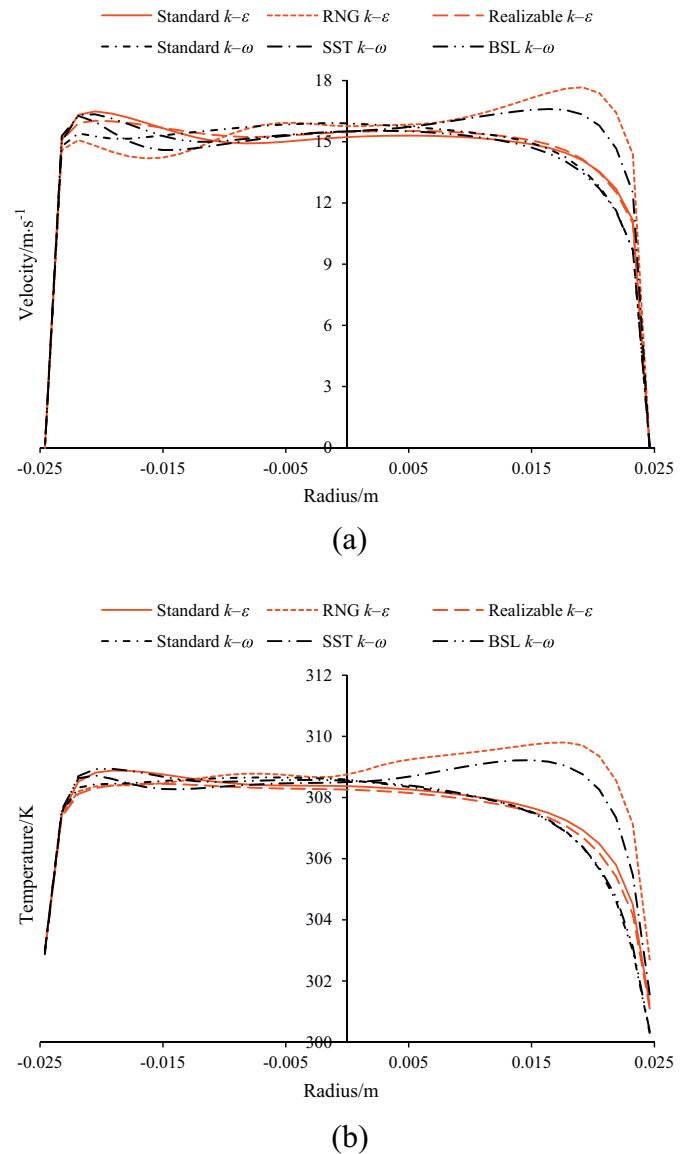
diagram and the real photograph of a typical indirect water bath heater. It consists of a cylindrical metal shell, a U-shape fire tube as the heat source with an exhaust flue gas system, a natural gas coil, and a water expansion tank. The cylindrical metal shell is filled with water, which is the heat transfer media. Both the U-shape fire tube and the gas coil are located inside the shell and immersed in water. The heat generated by natural gas combustion through the fire tube is transferred to water and then to the natural gas flowing through the gas coil. The expanded volume of water during the heating process is collected in the expansion tank.

The gas coil, which is the main part under consideration, has four rows of tubes, three 180° elbows, and two 90° elbows with a circular cross-section. The configuration and dimensions of the gas coil are presented in Fig. 2.

2.2. Proposed geometries

As described above, water is heated by the flue gas combustion in the U-shape fire tube and transfers the heat to the natural gas flowing into the gas coil. It seems that at the same heating duty, the flue gas consumption of the system strongly depends on the thermal performance of the gas coil. In the current study, the idea of replacing the straight gas coil with the twisted gas coil is investigated for the first time as a passive heat transfer enhancement technique in the indirect water bath heaters. A twisted tube, which is generated from twisting non-circular ducts, is a modified and capable flow pass introduced in the 1980s of the last century [37]. It is usually made into an elliptical

cross-section [38], however, other configurations of cross-section have been studied [39]. Engineering applications can be found in heat exchange devices encountered in chemical and food industries. Actually, the use of the twisted tubes can increase the thermal performance of heat exchange devices by creating some rotational flows in the flowing fluid along with a certain penalty in the pressure drop. It is worth noting that unlike other heat exchange devices, the pressure drop augmentation is not a negative and costly factor in the indirect water bath heaters

**Fig. 4.** Effects of different turbulent models on (a) velocity profile and (b) temperature profile at outlet of Ref. Model.**Table 4**
Comparison between real data and results predicted by different turbulent models

Turbulent model	Number of iterations	Predicted value of outlet temperature/K	Deviation with real value/%
Standard $k-\epsilon$	10250	308.1103	2.824
RNG $k-\epsilon$	8100	308.3763	3.606
Realizable $k-\epsilon$	9000	307.5121	1.065
Standard $k-\omega$	8900	308.2112	3.121
SST $k-\omega$	7700	308.2482	3.230
BSL $k-\omega$	8700	308.2303	3.177

applied in the CGS, since the pressure drop occurred in these heat exchangers reduces the performance load of the pressure reduction process, which is done by using the throttle-valves. On the other hand, the gas temperature drop in the pressure reduction process is also reduced, which is a positive point to prevent the natural gas from freezing.

Fig. 3 shows the view of the considered models. Note that, only the main part, where the heat transfer occurs, is shown in this figure, and the inlet and outlet connections are eliminated to save the space. As depicted, all models have an elliptical cross-section. It was proven that the best overall performance of the elliptical twisted tube can be achieved in the aspect ratio (a/b) of 1/2 [40], so this value is adopted

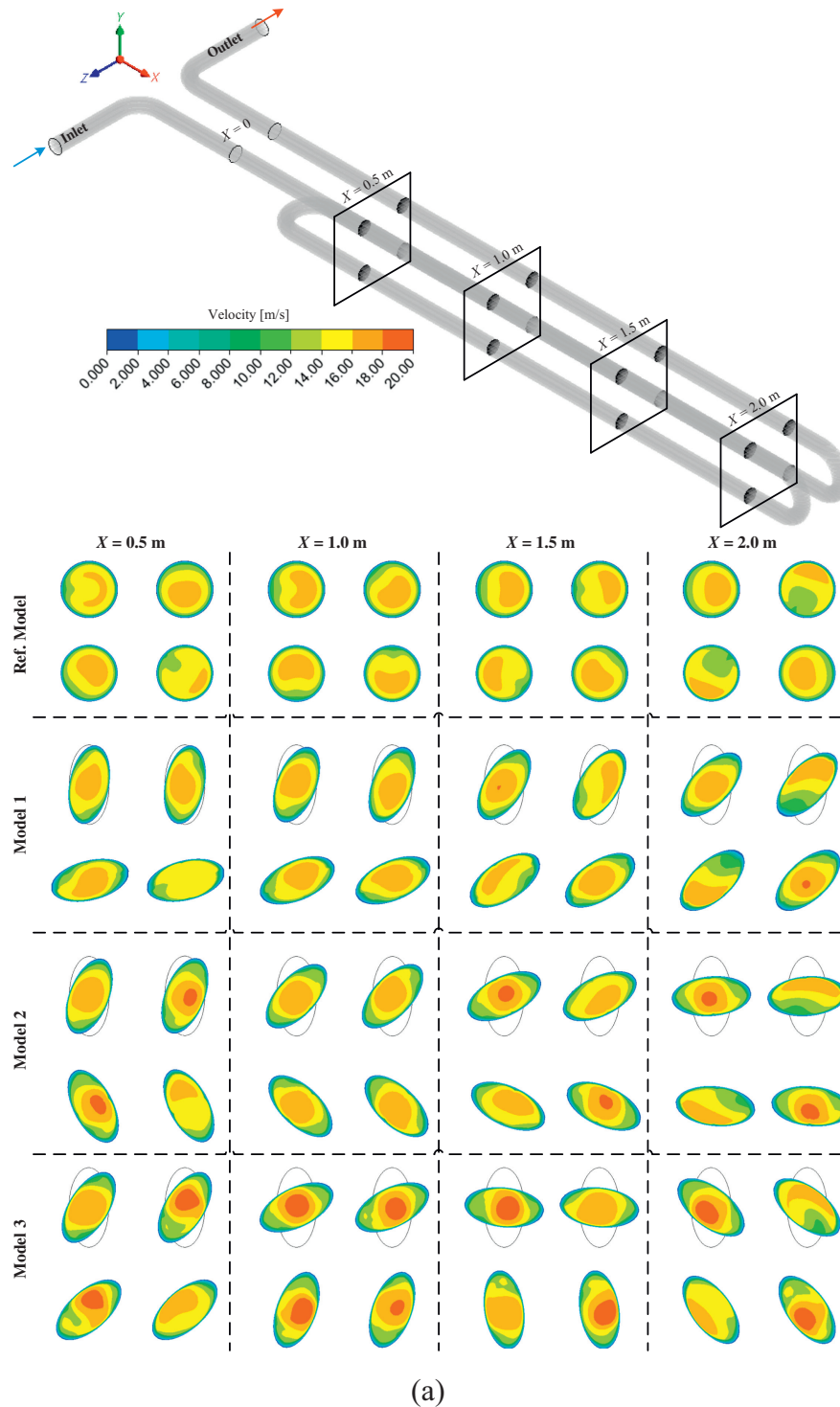


Fig. 5. Effects of twist ratio on (a) velocity, (b) temperature, and (c) pressure contours on four normal planes with $x = 0.5, 1.0, 1.5$, and 2.0 m.

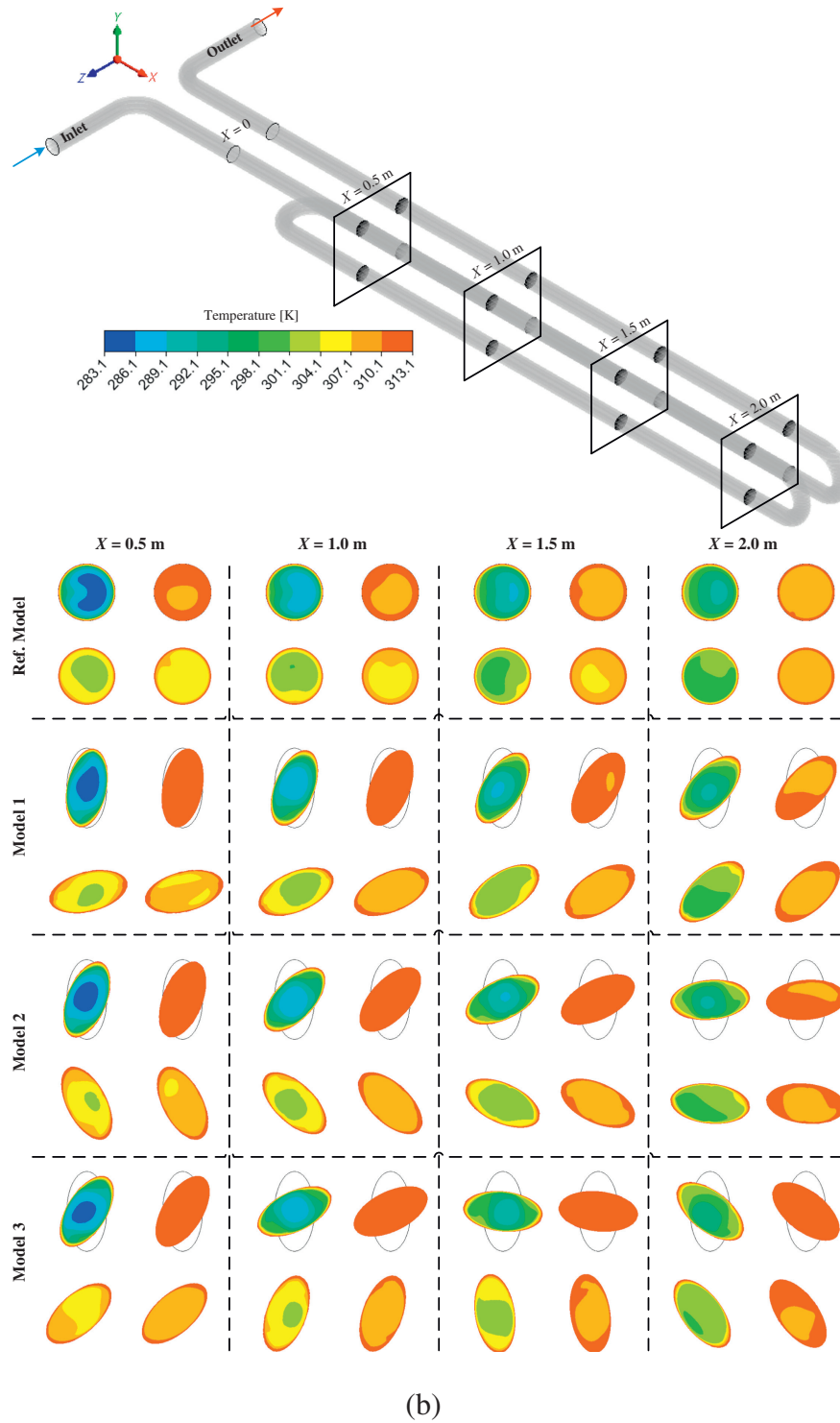


Fig. 5 (continued).

for the studied models. Also, it should be noted that three different 360° twist lengths are selected in order to evaluate the effects of twist ratio. In the present study, the twist ratio is defined as the ratio of the tube diameter to the length of one full twist 360° . Hence, the considered models are named for short as Model 1 (twisted tube with low twist ratio), Model 2 (twisted tube with medium twist ratio), and Model 3 (twisted tube with high twist ratio). Also, the straight model is named as the Ref. Model.

3. Implementation Procedure

3.1. Mathematical formulation

In the present work, to detect the suitable turbulent model, a comparative study is carried out on the common turbulent models, i.e. Standard $k-\epsilon$, RNG $k-\epsilon$ [41,42], Realizable $k-\epsilon$ [43], Standard $k-\omega$ [44], SST $k-\omega$ [45], and BSL $k-\omega$. As cited, these models

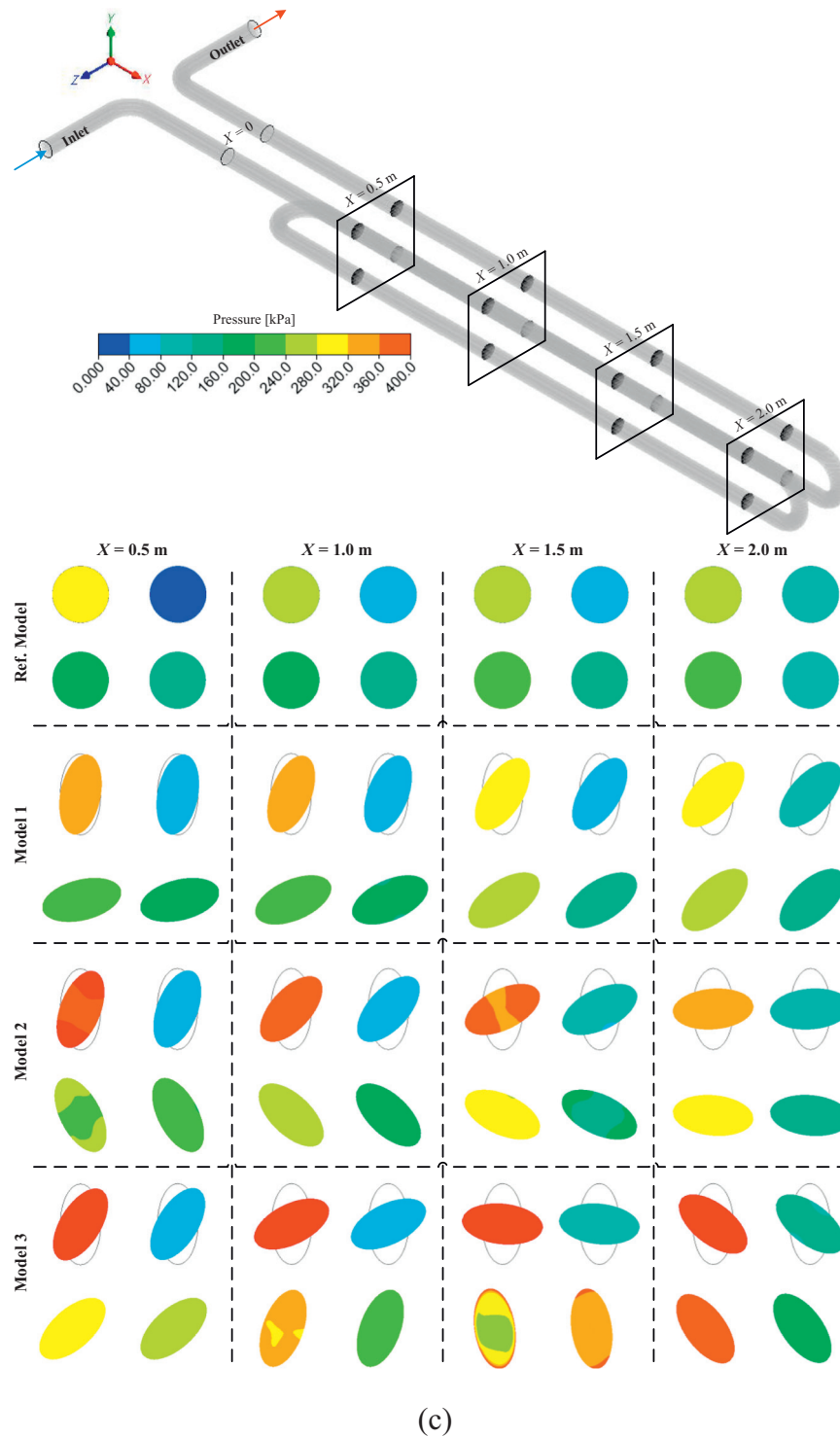


Fig. 5 (continued).

were used for the same geometries in other studies [41–45]. The results of the comparative study, which is presented in the next section, show that the Realizable k - ε model is the most suitable model for the considered case, so the mathematical formulation of this model is presented here. Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0$$

(1)

• Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}(\mu + \mu_t) \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \quad (2)$$

• Energy equation:

$$\frac{\partial}{\partial x_i}(u_i T) = -\frac{\partial}{\partial x_i} \left[\left(\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial x_i} \right] \quad (3)$$

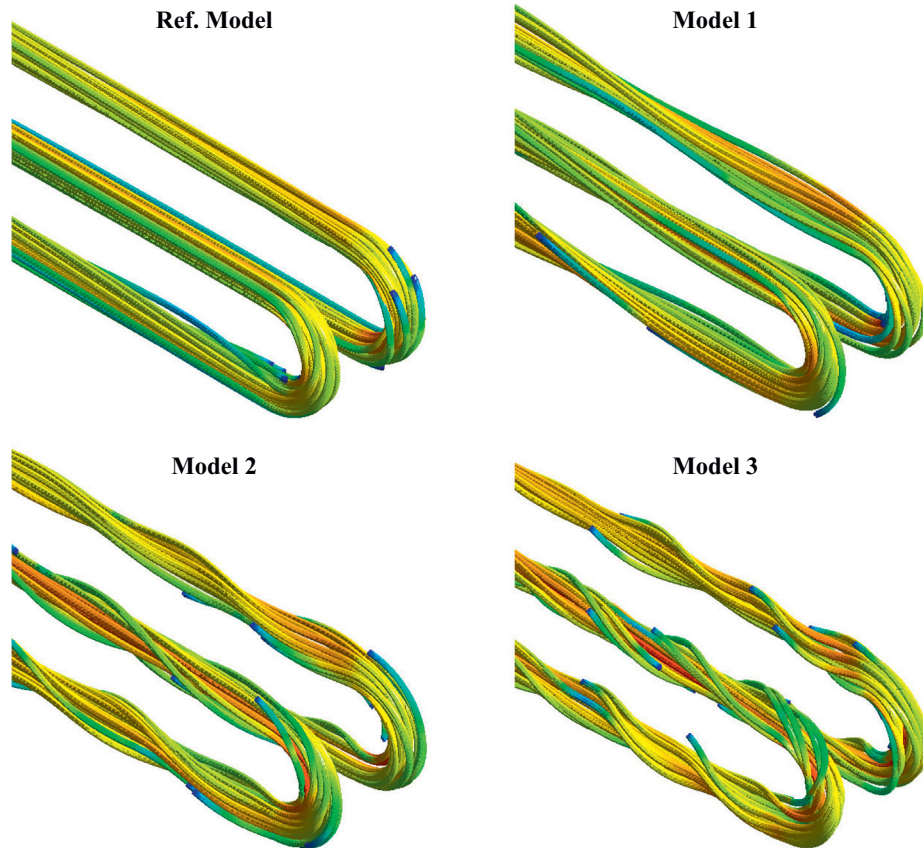


Fig. 6. Effects of twist ratio on a zoom part of stream lines.

where ρ is the natural gas density, u is the velocity, p is the pressure, μ is the dynamic viscosity, μ_t is the turbulent viscosity, T is the temperature, and Pr is the Prandtl number.

Equation for the turbulence kinetic energy k equation and specific dissipation rate ε :

$$\frac{\partial}{\partial x_i}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \Gamma - \rho \varepsilon \quad (4)$$

$$\frac{\partial}{\partial x_i}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \Gamma \varepsilon - C_2 \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}} \quad (5)$$

where Γ can be expressed as follows,

$$\Gamma = -u_i u_j \frac{\partial u_i}{\partial u_j} = \frac{\mu_t}{\rho} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \quad (6)$$

and

$$C_1 = \max \left[0.43 \frac{\mu_t}{\mu_t + 5} \right], C_2 = 1.0, \sigma_k = 1.0, \sigma_\varepsilon = 1.2, \mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (7)$$

It should be explained that the Realizable k - ε model is a relatively recent development widely used for engineering applications, which yields better performance in many industrial turbulent flows, and also

likely to provide superior performance for flows involving rotation, boundary layers under strong adverse pressure gradients, separation, and recirculation to other k - ε models [46].

3.2. Boundary conditions

Based on data collected from the CGS of Qaleh-Jiq (located in Golestan province of Iran) at the first month of a year, the natural gas enters the gas coil with the temperature around 283.75 K, which is mostly equal to soil temperature because the transmission lines are buried underground before the CGS. In order to remain above the hydrate-formation zone and dew point, the outlet temperature of the natural gas from the water bath heater depends on the inlet pressure. As depicted in Table 1, in the real case, the inlet pressure of the natural gas is around 5447 kPa (790 psi). At this condition, the outlet temperature of the natural gas should be in the range of 303.15–313.15 K. Note that, the maximum consumption of natural gas happens in the winter season as well as the first month of spring. Hence, it is better to check the performance of the water bath heater in this period of the year. Therefore, the data recorded in the first month of the year is selected for the current study, and it is used to validate the obtained results.

In this investigation, the simulated fluid is Methane with the thermo-physical properties as presented in Table 2.

The inlet velocity condition is adopted at the entrance of the considered models with the temperature of 283.75 K, which is the average value during 30 days of real data. At the external walls, a constant temperature condition is used based on real data of the water bath heater, whereas at the internal walls, the conjugate condition is imposed. At the exit of the considered models, the outlet boundary condition is used.

3.3. Numerical method

The following assumptions are considered,

- The simulation is three-dimensional,
- The flow is continuous, turbulent, and steady-state,
- The simulated fluid is Newtonian and single phase,
- The viscous dissipation, radiation, and natural heat transfer are negligible.

The governing equations along with the defined boundary conditions are solved by means of the finite volume method. The semi-implicit method for pressure linked equation (SIMPLE) algorithm is utilized to solve the differential equations for the velocity, pressure, and temperature fields. The diffusive and convective terms are discretized by means of the QUICK scheme. The convergence criterion is obtained when the normalized residuals of governing equations are less than 10^{-5} for all variables. As an indication of the computational time, between 6500 and 8900 iterations are needed to achieve the convergence criteria. This takes 22–34 h with Intel Xeon E3-1225v5 (3.2 GHz/3.6 GHz)-(4c/4t) and 32 GB DDR3 1600 MHz of RAM.

3.4. Mesh study

In order to produce accurate results in acceptable computational times, a sensitivity analysis of grids is performed. For all the models, a structured grid with a non-uniform trend is applied: zones near the tube walls are well meshed to ensure that intermediary grids between the fluid and the solid are small enough to resolve the boundary layers. The mesh independence study is also performed on all the models by obtaining the outlet temperature of the fluid. For instance, the grids employed for the Ref. Model are 1051440, 1422720, 1778400, 2134080, 2489760, 2803840, 3201120, 3504800, 3912480, and 7680560. The obtained results at the considered conditions are shown in Table 3. It is clear that the outlet temperature of the fluid becomes independent of grid number after 3201120 and further increment will not have noticeable effects. As the best trade-off between the results accuracy and the computations time, a grid number around 3200000 is used for all the models.

4. Results and Discussions

The comparison between the results of different turbulent models and the recorded data from the CGS of Qaleh-Jiq is provided in Table 4. It is detected that the Realizable $k-\varepsilon$ model gives the best results. The mean deviation between this model and the real data is about 1%. Therefore in the present study, the Realizable $k-\varepsilon$ model is selected to simulate the Methane flow through the considered models. Furthermore, the profiles of velocity and temperature obtained by different models at the outlet of the Ref. Model are presented in Fig. 4(a–b). These figures disclose the effects of turbulent model on the fluid flow and heat transfer characteristics of the considered case.

Fig. 5 shows the velocity, temperature, and pressure contours of natural gas through different models on four normal planes with $x = 0.5, 1.0, 1.5,$ and 2.0 m. It is clear that twisting the natural gas flow path affects both hydraulic and thermal performances of the water bath heater, and the velocity, temperature, and pressure contours are different from each other. As shown in the figures, the straight length of the natural gas coil is not sufficient to reach a uniform velocity and temperature condition, and the effect of fluid mixing generated in an elbow remains until the next bend.

As the natural gas flows through the twisted tubes, its velocity plays an important role. As depicted in Fig. 5(a), at the same operating condition, the natural gas in the twisted tubes, i.e. Model 1, Model 2, and Model 3, has higher velocities compared to the Ref. Model, and the

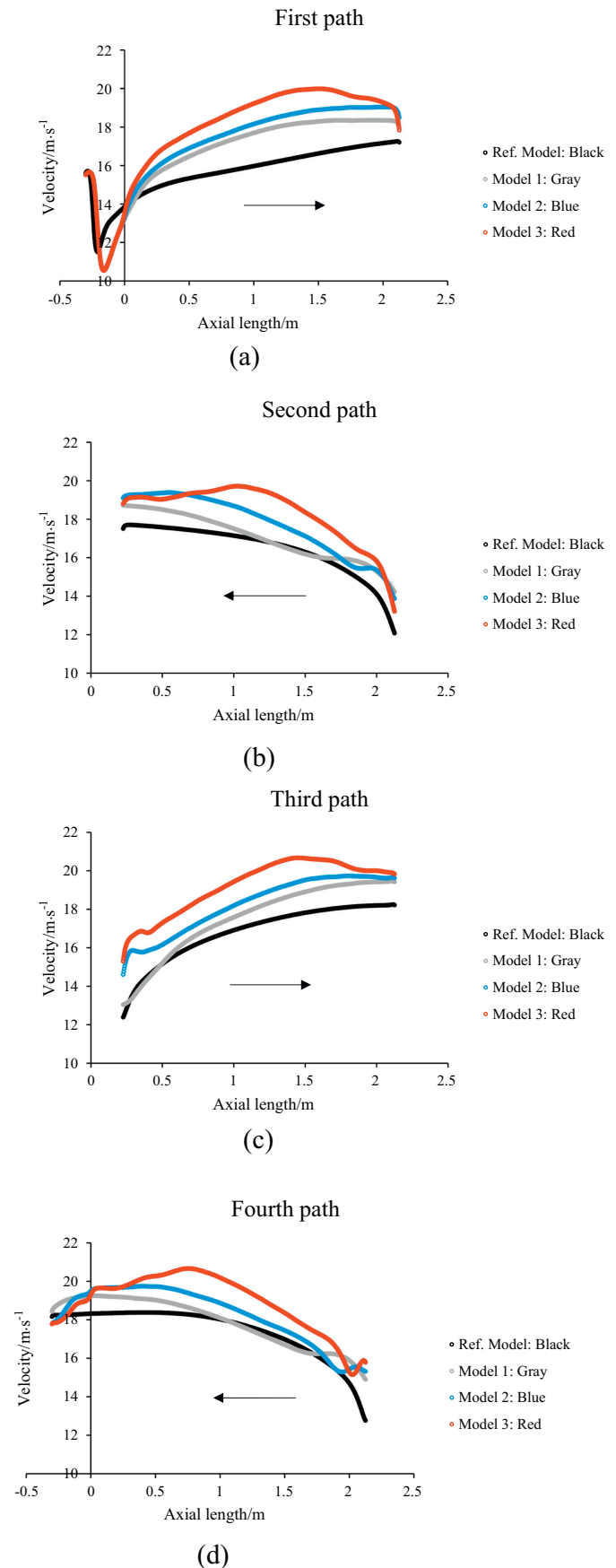


Fig. 7. Variations of velocity on a center line across of different models: (a) first path or row, (b) second path or row, (c) third path or row and (d) fourth path or row.

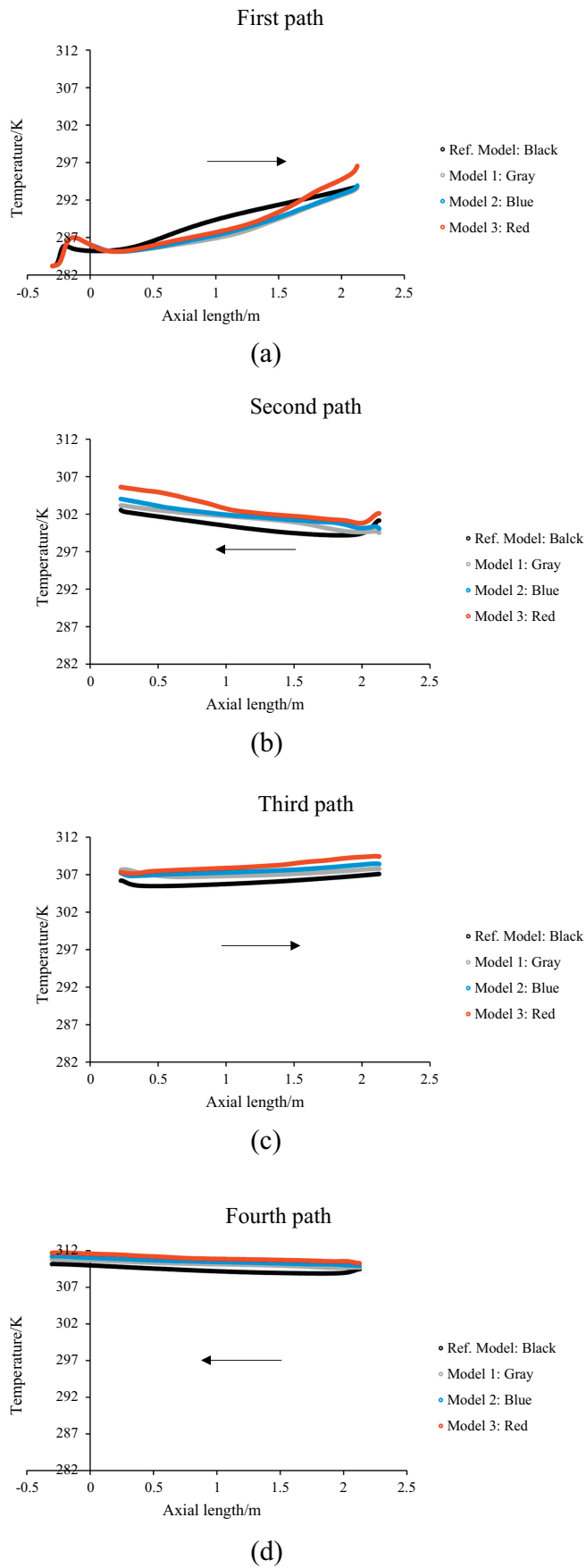


Fig. 8. Variations of temperature on a center line across of different models: (a) first path or row, (b) second path or row, (c) third path or row and (d) Fourth path or row.

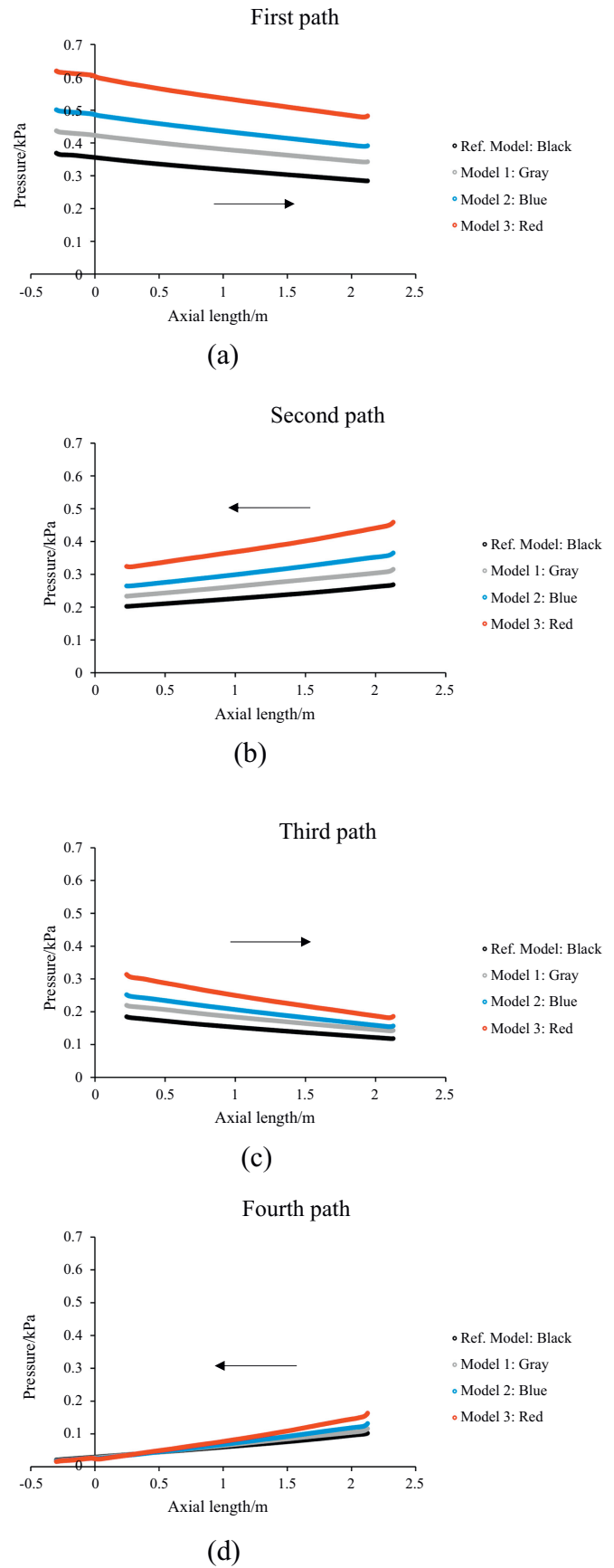


Fig. 9. Variations of pressure on a center line across of different models: (a) first path or row, (b) second path or row, (c) third path or row, (d) fourth path or row.

difference enhances as the twist ratio is increased. This intensifies the fluid mixing and heat transfer between the hot gas near the wall and the cold gas in the core region. On the other hand, when the natural gas flows through the twisted tubes, swirl flows may be produced and thus enhance the heat exchange inside the coil. In order to create a better insight, the streamlines of natural gas at a zoom part of different models are depicted in Fig. 6. It can be discussed that as the natural gas passes through a twisted tube, the main direction of flow is changed and both the transverse and the normal velocities are intensified leading to a better fluid mixing. The streamlines in the Ref. Model are almost uniform, so the natural gas near the wall remains in the thermal layer regime and accumulates the heat. The streamlines in the twisted tubes turn by centrifugal forces and have more complex configurations, so high temperature fluid parts do not stay in the thermal layer regime and move to the core region of fluid. This mixing disturbs the thermal boundary layer. It is clear that the mixing increases as the twist ratio is

increased. Therefore, Model 3 has the highest thermal performance, and Model 2 and Model 1 come in the second and third, respectively.

Considering the top-right contours in Fig. 5(b) as a comparison criterion: although the temperature of natural gas at the output of Ref. Model has not reached the maximum value, it gives the maximum value at $x = 1.0$ m and $x = 1.5$ m inside Model 1 and Model 2, respectively. It is noticeable to illustrate that in the fourth row of the natural gas coil in Model 3, the natural gas has the maximum temperature. As a rule of thumb, it can be concluded that Model 1, Model 2, and Model 3 can reduce the length of the natural gas coil by about 12.5%, 18.75%, and 25%, respectively, as compared to the Ref. Model.

As anticipated, the pressure drop increases as the Ref. Model is replaced with a twisted tube. Referring to Fig. 5(c), one can observe that all the twisted tubes have higher pressure variations than the Ref. Model. However, in the studied case, the pressure drop is not a limiting factor, because the pressure of the natural gas will be reduced after the

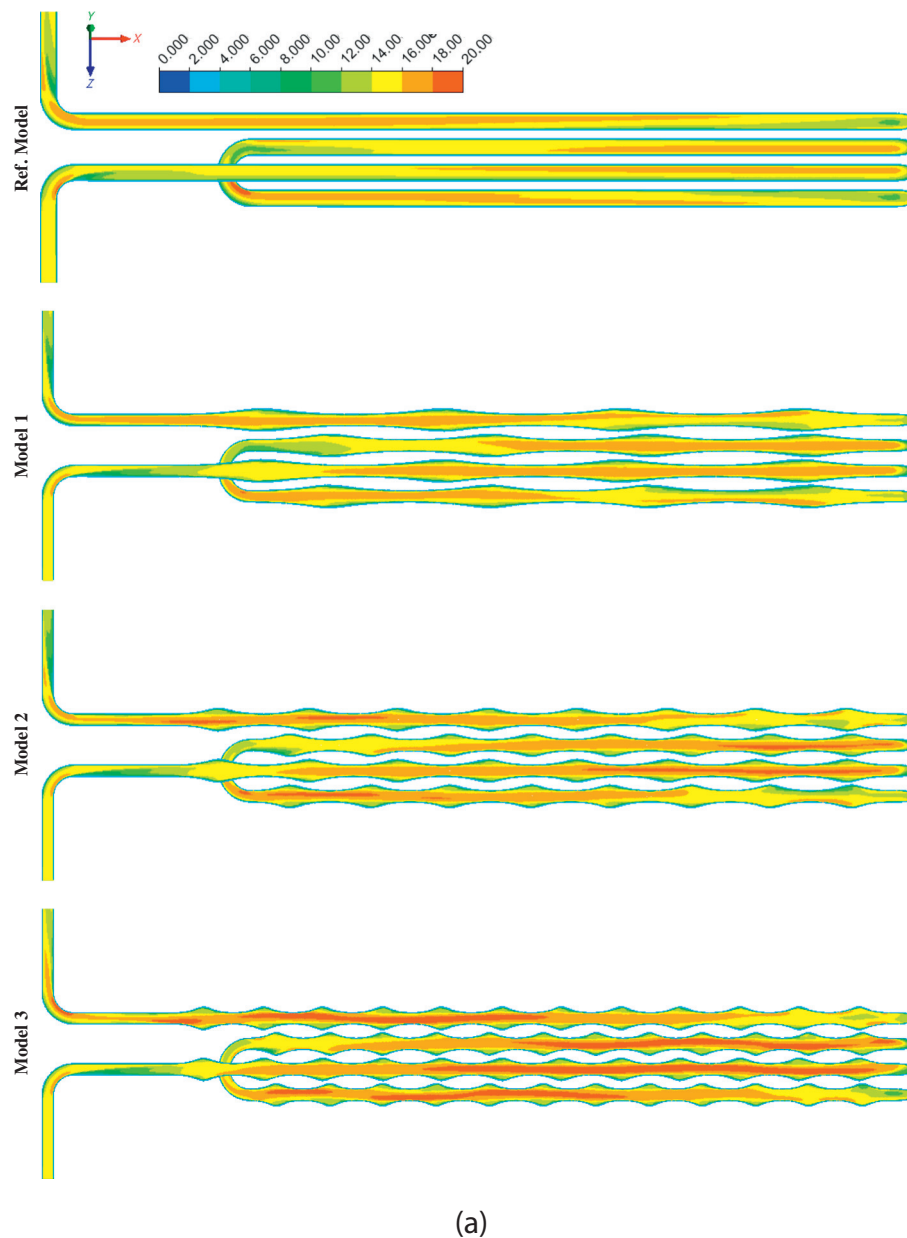


Fig. 10. Effects of twist ratio on (a) velocity, (b) temperature, and (c) pressure contours on four transvers planes with $y = 0$ and 0.152 m.

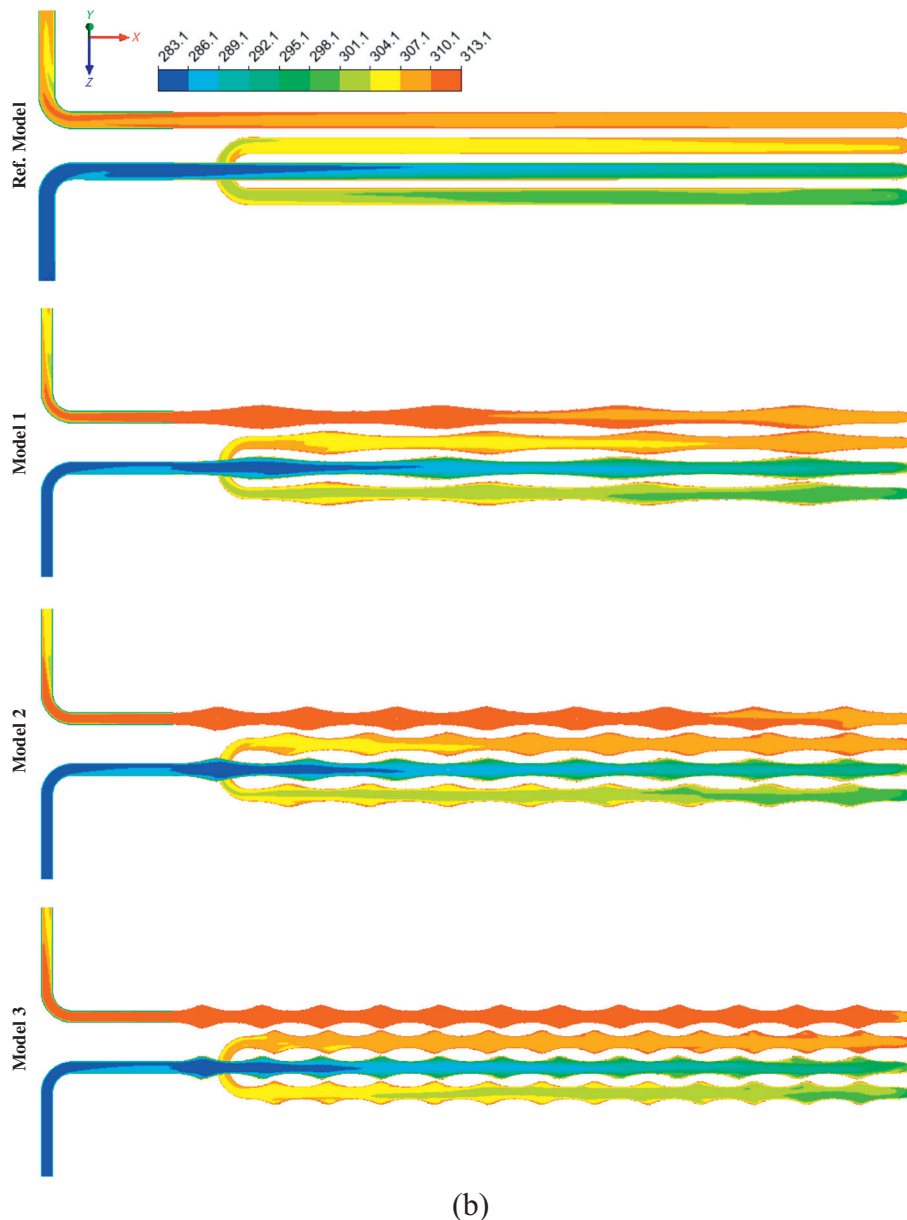


Fig. 10 (continued).

water bath heater using the pressure reduction system, i.e. throttle-valves. Similar to the velocity and temperature, the variations of pressure enhance as the twist ratio is increased.

Furthermore, to create a quantitative comparison, the plots of velocity, temperature, and pressure of different models are presented in Figs. 7–9. Note that, they are plotted only for the straight sections of each model, and the flow direction is specified by a black arrow. As depicted in Fig. 7(a–d), after the inlet 90° elbow, the velocity of natural gas begins to increase until the first 180° elbow in all the models. Fig. 6 shows that as the natural gas enters a 180° elbow, it twists by centrifugal forces, so high temperature parts of natural gas do not stay in the thermal layer regime and move to the core region of fluid. As previously mentioned, this mixing disturbs the thermal boundary layer and increases the temperature of the natural gas in this section, see Fig. 8(a–d). A similar discussion can be made for the next elbows. However, it is clear that the curvature effects on both the velocity and the temperature enhance as the twist ratio is increased, and these parameters are greater for the models with higher twist ratio. Fig. 9(a–d) discloses

that the pressure of natural gas is reduced by an almost linear trend through all the models, and the pressure variations are more significant for the models with higher twist ratio. Furthermore, the velocity, temperature, and pressure contours of the natural gas through different models on two transverse planes with $y = 0$ and 0.152 m are presented in Fig. 10(a–c).

5. Conclusions

The current study investigates the hydrothermal performance of the natural gas coil inside the water bath heater of a CGS with the nominal capacity of $1000 \text{ m}^3 \cdot \text{h}^{-1}$. Utilizing twisted tubes instead of straight ones is proposed to enhance the thermal performance of the water bath heater and decrease the flue gas consumption of the CGS. In order to investigate this idea, the velocity, temperature, and pressure variations in the gas coils are calculated by a numerical method. The corresponding equations are iteratively solved by the finite volume method in Intel Xeon 32 GB DDR3 and 1600 MHz of RAM. Also, the real data for the

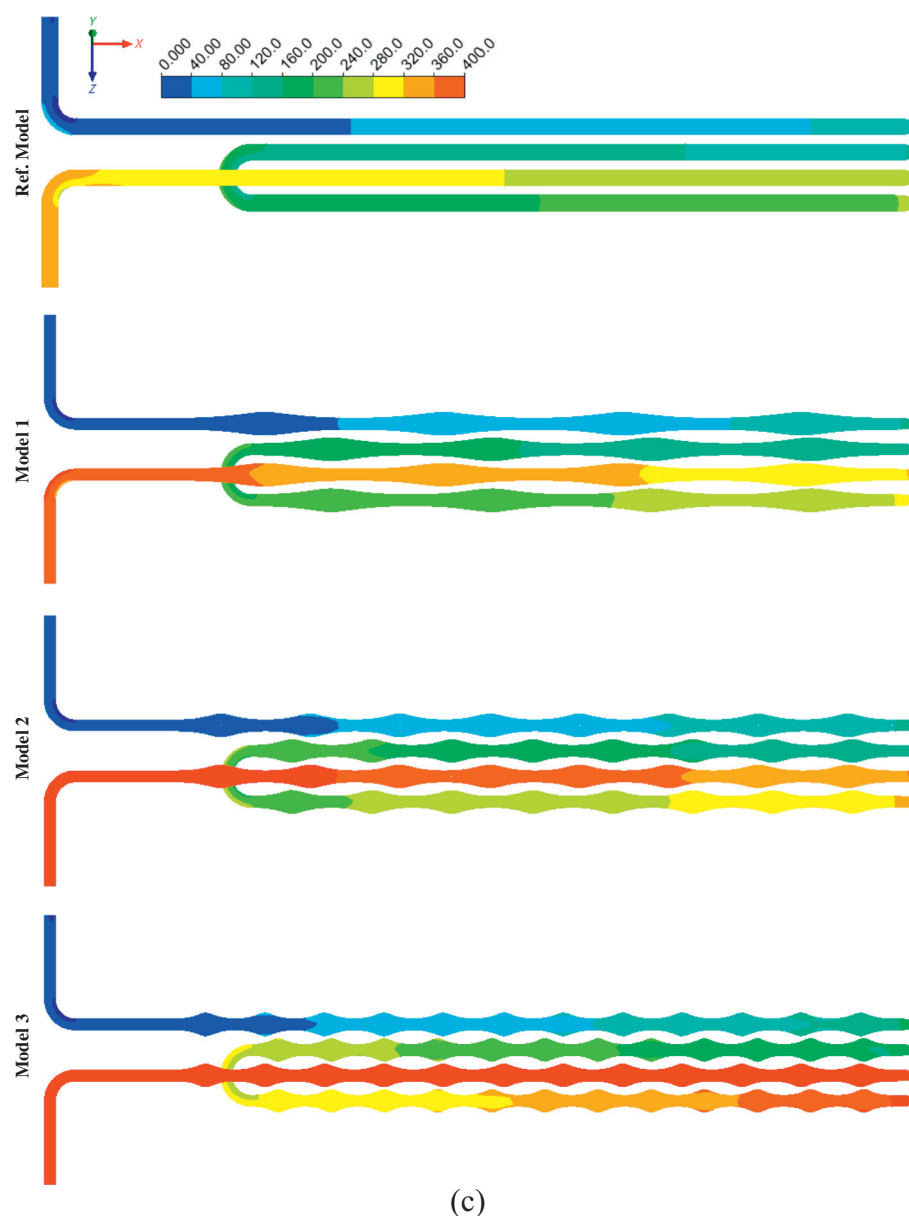


Fig. 10 (continued).

validation of the computational results is obtained from the CGS of Qaleh-Jiq (located in Golestan province of Iran) within 30 days of the first month of the year. The real data and the current results are in a good agreement within $\pm 1\%$, and the most important results are concluded here. Firstly, the velocity, temperature, and pressure contours of straight and twisted gas coils are compared. In this case, it is found that the twisted tubes can achieve better thermal performance compared with the straight one. In general, any variation in the hydrothermal performance of the gas coil depends on the twist ratio of the twisted tubes. The thermal performance of the gas coil can be improved by increasing the twist ratio. For a given length of the gas coil, the maximum outlet temperature of natural gas, which is one of the most important parameters, is achieved for the twisted tube with the highest twist ratio. In other words, when the straight or conventional gas coil is replaced with a twisted one, heating the natural gas up to the maximum outlet temperature can reduce the length of the gas coil up to 25%. Also, the results show that the pressure drop across the gas coil increases as the twist ratio is raised. It can be concluded that these positive

specifications, i.e. better thermal performance and higher pressure drop, reduce the size of the water bath heater at the same operating conditions and can make it very cost-effective.

Acknowledgment

The authors would like to express their thanks to Islamic Azad University (IAU) Shahrood Branch, Shahrood, Iran and Golestan Province Gas Company, Gorgan, Iran for their sponsorships.

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