Computational Study of the Turbulent Flow inside a Waste Heat Recovery System with a 25° inclined Angle Diffuser

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Abstract In this paper, we are interested on the study of the turbulent flow inside a waste heat recovery system with a 25° inclined diffuser. For this, we have developed a numerical simulation using a CFD code. Particularly, we are interested to visualize the temperature, the velocity, the total pressure, the dynamic pressure, the vorticity, the turbulent kinetic energy, the turbulent dissipation rate and the turbulent viscosity. The numerical model is based on the resolution of the Navier-Stokes equations in conjunction with the standard k-ε turbulence model. These equations were solved by a finite volume discretization method.

Keywords: waste heat recovery, inclined diffuser, heat exchanger, power generator, water heating


1. Introduction

Exhaust of the engines is a main source that a large amount of energy wastes through it. Researchers confirm that more than 30–40% of fuel energy wastes from the exhaust and just 12–25% of the fuel energy convert to useful work [1,2,3]. The dumped thermal energy can be recovered and a heat exchanger is necessary to transmit the heat from hot gases to working fluid at excellent efficiency. A heat exchanger is thermal equipment, which is built for efficient heat transfer between two fluids of different temperatures. Heat exchangers are used mainly in industrial sectors (chemicals, petrochemicals, steel, food processing, energy production, etc) and transportation (automotive, aeronautics), but also in the residential sector and tertiary (heating, air conditioning, etc). The choice of a heat exchanger, for a given application depends on many parameters such as field temperature and pressure of fluids, and physical properties of these fluids, maintenance, cost and space. In the literature, we can find relevant studies on heat exchanger networks [4-9]. CFD numerical simulations have been used with success to better understand the flow and heat transfer behavior in particular configurations of heat exchangers and to improve heat exchanger designs. Deng et al. [10] designed two thermoelectric exchangers models shown by CFD simulation and used Wilcox k-ω model to discuss on different internal structures, lengths and materials on the exchangers performance. The same study has been performed by Kumar et al. [11] which modeled three exchangers (rectangular, triangular and hexagonal) by FLUENT software and experimentally produced and tested the best model. Zhang and Li [12] proposed a structure of two-stage-distribution and the numerical investigation shows that the flow distribution in plate-fin heat exchanger is more uniform if the ratio of outlet and inlet equivalent diameters for both headers is equal. Wen [13] employed CFD technique to simulate and analyze the performance of fluid flow distribution and pressure drop in the header of plate-fin heat exchanger. Wasewar [14] studied the flow distribution through a plate-fin heat exchanger by using FLUENT. A modified header is proposed and simulated using CFD. The modified header configuration has a more uniform flow distribution than the conventional header configuration. Hence, the efficiency of the modified heat exchanger is seen to be higher than that of the conventional heat exchanger. Gan et al. [15] performed a CFD simulation on the tubes of a heat exchanger used in closed-wet cooling towers. Pressure drop was found to depend on the tube configurations and water to air ratio. Predicted pressure loss coefficient was found inversely proportional to transverse pitch but was in direct relationship with water to air mass flow rate. Sheikh Ismail et al. [16] numerically investigated the effects of nozzle and header orientation on the hydrodynamic performance of a plate-fin heat exchanger, their studies showed that the orientation of the header and nozzle plays a major role in the exchanger performance. In this context, we are interested in studying the turbulent flow inside a waste heat recovery system with a 25° inclined diffuser.
2. Numerical Model

The computational domain is shown in Figure 1. It is defined by the interior volume of the gas flow and by the two diffusers. It is limited by the inlet and the outlet of the water. In the inlet of the gas and the water, the mass flows are equals to 0.0335 kg.s\(^{-1}\) and 0.07 kg.s\(^{-1}\) respectively, and the temperature values are defined by T=526°C and T=10°C respectively. In the outlet, we consider the atmospheric pressure of 101325 Pa.

In this study, we have used the Software “Solid Works Flow Simulation” for calculation. The k-ε turbulence model has been considered for the analysis of turbulent flow. In fact, this model has been used in different anterior works and satisfactory results were obtained [17-23]. We have used the Local Initial Mesh option. This mesh option allows us to specify an initial mesh in a local region of the computational domain to better resolve the model geometry and flow particularities in this region. Local mesh settings are applied to all cells intersected by a component, face, edge, or a cell enclosing the selected vertex. We have used two local meshing for the entry diffuser and the heat exchanger and a basic meshing for the outlet meshing (Figure 2). In these conditions, the cells size for the local mesh is equal to 0.001 m and for the basic mesh is equal to 0.01 m.

3. Numerical Results

Two longitudinal planes defined by z=0 m, y=0 m and three transverse planes defined by x=-0.2 m, x=0 m, and x=0.2 m are considered. Particularly, we are interested to visualize the temperature, the velocity, the total pressure, the dynamic pressure, the vorticity, the turbulent kinetic energy, the turbulent dissipation rate and the turbulent viscosity.

3.1. Temperature

Figure 3 presents the distribution of the temperature in the longitudinal planes defined by z=0 m and y=0 m. According to these results, it is clear that the temperature is at its maximum at the entry of the diffuser, which is the value of the inlet boundary condition. The temperature shows a decrease at the sides of the first diffuser and through the heat exchanger. The decreased value of the temperatures is the quantity of heat transported to the water in the heat exchanger. After going out of the heat exchanger, it is clear that the temperature of the gas decreases but still represents important values.

Figure 3. Distribution of the temperature in the longitudinal planes

Figure 4 presents the distribution of the temperature in the transverse planes defined by x=-0.2 m, x=0 m and x=0.2 m. In the first plane x=-0.2 m, the temperature distribution is located in the upstream of the heat...
exchanger. However, in the third plane $x=0.2\ m$, the temperature distribution is located in the downstream. According to these results, the high temperatures are concentrated in the middle of the diffusers and there is a decrease of the temperature along the sides of the diffusers.

The row defined by the plane $x=0\ m$ presents the evolution of the water in the middle of the heat exchanger. In these conditions, the water reaches a temperature equal to $T=50^\circ\text{C}$. The last transverse plane defined by $x=0.2\ m$ presents only the gas through the second diffuser. All these figures show a decrease in the value of the gas temperature. This fact is due to the heat loss through the heat exchanger. All the temperatures are between $200^\circ\text{C}$ and $300^\circ\text{C}$. In the diffuser sides, it has been observed a heat loss more superior.

3.2. Magnitude Velocity

Figure 5 presents the distribution of the magnitude velocity in the longitudinal planes defined by $z=0\ m$ and $y=0\ m$. According to these results, the maximum value of the velocity appears in the gas inlet which is imposed by the boundary conditions. The magnitude velocity shows a decrease in the first diffuser. Also, a decrease has been noted on the sides of the first diffuser which there is an important drop of the magnitude velocity. Out of the heat exchanger, the magnitude velocity value decreases in the second diffuser. At the end of the diffuser, an increase of the magnitude velocity value has been noted due to the reduction of the diffuser section. Indeed, it is clear that through the reduction of the size of the diffuser, the magnitude velocity value increases at the end of the diffuser.

3.3. Total Pressure

Figure 7 presents the distribution of the total pressure in the longitudinal planes defined by $z=0\ m$ and $y=0\ m$. According to these results, a compression zone characteristic of the maximum value of the total pressure has been observed on the heat exchanger upstream and the middle of the diffuser. A progressive decrease has been observed on the sides of the first diffuser. A decrease of the total pressure has also been noted at the second
diffuser and the heat exchanger downstream. The total pressure decreases out of the heat exchanger. In fact, the total pressure has approximately the same value and do not have a great value change in all the length of the diffuser.

Figure 8 presents the distribution of the total pressure in the transverse planes defined by \(x=-0.2\) m, \(x=0\) m and \(x=0.2\) m. While examining these results, it can easily be noted that the total pressure is on its maximum in the intake and is globally uniform in the diffuser for the first plane defined by \(x=-0.2\) m. A pressure drop of the total pressure has been noted in the sides of the diffuser. In the plane defined by \(x=0.2\) m, the distribution of the total pressure shows that the maximum values of the total pressure are located in the middle of the diffuser, and decreases in the sides. In the transverse plane defined by \(x=0.2\) m and \(x=0\) m, results confirm that the distribution of the total pressure is uniform in the tube and the difference is located in the heat exchanger. An increase of the total pressure has been observed in the middle. Along the sides, it has been noted a decrease of the total pressure.

3.4. Dynamic Pressure

Figure 9 presents the distribution of the dynamic pressure in the longitudinal planes defined by \(z=0\) m and \(y=0\) m. These results show that the compression zone characteristics of the maximum values of the dynamic pressure are localized in the heat exchanger and the middle of the diffuser. A decrease of the dynamic pressures values appear in the sides of the diffuser and in the second diffuser. Indeed, the dynamic pressure decreases out of the heat exchanger. In the second diffuser and through the gas flow, the dynamic pressure decreases also. This fact is due to the difference in the section and the size of the diffuser. In the end of the second diffuser, a progressive increase of the dynamic pressure has been observed.

Figure 9. Distribution of the dynamic pressure in the longitudinal planes

Figure 10. Distribution of the dynamic pressure in the transverse planes
Figure 10 presents the distribution of the dynamic pressure in the transverse planes defined by \(x=-0.2\) m, \(x=0\) m and \(x=0.2\) m. According to these results, it has been noted that the dynamic pressure is on its maximum in the intake and is globally uniform in the diffuser. A pressure drop has been noted in the sides of the diffuser. In the plane defined by \(x=0.2\) m, it is clear that the depression zones are located in the middle and the superior sides of the diffuser. In the transverse planes defined by \(x=-0.2\) m and \(x=0\) m, the distribution of the dynamic pressure is uniform in the tube and there is a difference in the heat exchanger. Also, an increase of the dynamic pressure has been observed in the middle of the heat exchanger. However, along the sides, it has been noted a decrease of the dynamic pressure.

3.5. Vorticity

Figure 11 presents the distribution of the vorticity in the longitudinal planes defined by \(z=0\) m and \(y=0\) m. According to these results, the vorticity has been observed in the entry of the gas inlet. In the middle of the diffuser, the vorticity is low and does not present an important value. The vorticity presents an increase in the middle of the heat exchanger near the obstacles like the fin plates. In the heat exchanger downstream, the vorticity importantly decreases. With the second diffuser an increase of the vorticity has been noted in the outlet of the diffuser.

3.6. Turbulent Kinetic Energy

Figure 13 presents the distribution of the turbulent kinetic energy in the longitudinal planes defined by \(z=0\) m and \(y=0\) m. The results show a wake characteristic of the maximum values of the turbulent kinetic energy in the diffuser. The results show also a decrease in the value of the turbulent kinetic energy almost at the end of the heat exchanger and at the entry of the second diffuser. After that, the turbulent kinetic energy decreases progressively.
Figure 14 presents the distribution of the turbulent kinetic energy in the transverse planes defined by \(x=0.2\ m\), \(x=0\ m\) and \(x=0.2\ m\). The results, presented in the plane \(x=0.2\ m\), show a wake characteristic of the maximum value of the turbulent kinetic energy in the sides of the diffuser. In the middle of the diffuser, the turbulent kinetic energy decreases. Also, a progressive decrease in the far sides of the diffusers has been noted. The plane \(x=0\ m\) presents a wake characteristic of the maximum value of the turbulent kinetic energy. In the sides, a progressive decrease of the turbulent kinetic energy has been observed. The turbulent kinetic energy in the tubes presents low values. The plane \(x=0.2\ m\) presents the turbulent kinetic energy in the second diffuser where the gas goes out of the heat exchanger. In this zone, the turbulent kinetic energy values are very low.

3.7. Dissipation Rate of the Turbulent Kinetic Energy

Figure 15 presents the distribution of the dissipation rate of the turbulent kinetic energy in the longitudinal planes defined by \(z=0\ m\) and \(y=0\ m\). The results show a wake characteristic of the maximum values of the dissipation rate of the turbulent kinetic energy in the sides of the diffuser. In the middle of the diffuser and in the gas inlet, the dissipation rate of the turbulent kinetic energy decreases. The wake extension has been observed until the gas leaves the heat exchanger where there is a great drop on the dissipation rate of the turbulent kinetic energy values.

Figure 16 presents the dissipation rate of the turbulent kinetic energy in the transverse planes \(x=0.2\ m\), \(x=0\ m\) and \(x=0.2\ m\). In the plane \(x=0.2\ m\), the dissipation rate of the turbulent kinetic energy has been observed in the sides of the diffusers. After that, the dissipation rate of the turbulent kinetic energy decreases progressively until they reach the limit of the diffusers. In the middle of the diffuser, the values of the dissipation rate of the turbulent kinetic energy are low. The plane \(x=0\ m\) shows a wake characteristic of the maximum values of the dissipation rate of the turbulent kinetic energy.
rate of the turbulent kinetic energy in the middle of the planes. The dissipation rate of the turbulent kinetic energy decreases in the other sides of the diffuser. The dissipation rate of the turbulent kinetic energy in the tubes is very low and does not presents a significant effect. The plane $x=0.2$ m presents the dissipation rate of the turbulent kinetic energy in the second diffuser in the exit of the gas from the heat exchanger. These figures present very low values of dissipation rate of the turbulent kinetic energy. According to these results, it is clear that the values of the dissipation rate of the turbulent kinetic energy increases in the middle of the heat exchanger and then it starts to decrease progressively until the sides of the diffuser.

### 3.8. Turbulent Viscosity

Figure 17 presents the distribution of the turbulent viscosity in the longitudinal planes defined by $z=0$ m and $y=0$ m. According to these results, a wake characteristic of the maximum values of the turbulent viscosity has been observed in the second diffuser in the outlet of the heat exchanger. A decrease in the value of the turbulent viscosity has been noted in the gas inlet for the first diffuser. Indeed, the turbulent viscosity values decrease importantly in the heat exchanger. The results show an increase in the middle of the diffuser and a decrease in the sides of the diffuser.

Figure 17. Distribution of the turbulent viscosity in the longitudinal planes

Figure 18 presents the distribution of the turbulent viscosity in the transverse planes defined by $x=-0.2$ m, $x=0$ m and $x=0.2$ m. While examining these results, a wake characteristic of the maximum values of the turbulent viscosity has been observed in the middle of the diffuser. In the sides of the diffuser, the turbulent viscosity decreases. According to the plane defined by $x=0$ m, an increase in the middle of the heat exchanger has been noted and a progressive decrease has been observed in the sides. The plane defined by $x=0.2$ m presents the distribution of the turbulent viscosity in the second diffuser after the heat exchanger outlet. The results show an increase of the values of the turbulent viscosity in the middle of the heat exchanger and present a progressive decrease in the sides.

Figure 18. Distribution of the turbulent viscosity in the transverse planes

### 4. Comparison with Analytical Results

Figure 19 presents the evolution of the water temperature profiles through the tube for the inclined diffuser. These profiles present the same evolution of the curve. In fact these profiles present a progressive increase of the water temperature through the tube. The comparison with the analytic method shows a good agreement and confirms the validity of the numerical method.

Figure 19. Water temperatures profile through the tube for the inclined diffuser
5. Conclusions

In this work, numerical simulations have been developed to study the turbulent flow inside a heat exchanger using an inclined diffuser. We present all the results from simulation, such as temperature, velocity, total pressure, dynamic pressure, vorticity, turbulent kinetic energy, turbulent dissipation rate and turbulent viscosity. According to the numerical results, fluid flow characteristics decrease in the sides of the first diffuser and in the second diffuser out of the heat exchanger. The evolution of the water temperature profile through the tube results for the inclined diffuser is compared to analytical profile. A good agreement was obtained and confirms the validity of the numerical method. In the future, we propose to study the shape effect on the turbulent flow inside the heat exchanger.

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References