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Numerical study of the dynamic and thermal field of a flow in a shell and tube heat exchanger

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Abstract - *In order to gain an understanding of the mechanism of thermal performance increase in the shell side, a shell and tube heat exchanger is proposed in this study. Numerical investigations of the shell side fluid flow and heat transfer are performed using CFD Fluent software based on the k-ε model. The profiles and axial velocity fields, as well as the profiles and temperature distribution at the heat exchanger level were obtained for the whole geometry. The speed increases by 12 % of the reference speed in the shell.*

Résumé - *Afin d'obtenir une compréhension du mécanisme d'augmentation des performances thermiques dans le côté de calandre, un échangeur de chaleur tubes a lisses en pas triangulaire et calandre est proposé dans la présente étude. Les investigations numériques sur le flux du fluide côté de la calandre et le transfert de chaleur sont effectuées en utilisant le logiciel commercial CFD Fluent basé sur le modèle k-ε. Les profils et les champs de vitesse axiale, ainsi que les profils et la distribution de température au niveau d'échangeur de chaleur ont été obtenus pour toute la géométrie. La vitesse augmente de 12 % de la vitesse de référence dans la calandre.*

Mots clés: CFD - Volume fini - Echangeur à faisceau et calandre.

1. INTRODUCTION

The study of fluid flow in annular geometries is a great importance in industrial applications because it is the common design of the shell and tube heat exchangers. Several techniques have been proposed to improve the heat transfer, the inclination or the variation of the geometrical parameters of the walls, the implementation of the baffles in the flow space.

Several studies have been conducted to examine the performance of the system with baffles. Chen *et al.*, [1], proposed a heat exchanger with circumferential trisection helical baffles as a means of limiting leakage. Wang *et al.*, [2], analyzed the characteristics of the fluid flow and heat transfer of three heat exchangers and shown as the heat exchanger with the H-shaped support incorporating the advantages of the other two heat exchangers.

Huadong *et al.*, [3], an experimental study of heat transfer and pressure drop for different baffle spacing, pressure drop and average heat transfer are increased by increasing baffle spacing. Handry *et al.*, [4], presented a numerical study on the liquefied natural gas (LNG) in the heat exchanger, the results show that the heat transfer

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coefficient increases with the increase of the velocity; moreover, the optimization of mass flow fluid in the vaporization process was reported in this study.

Yan *et al.*, [5], have studied the characteristics of fluid flow and heat transfer for gas cooling, the results show, the higher operating pressure improves the heat transfer, and the gas components affect significantly in the pressure drop and heat transfer. Lei *et al.*, [6], have shown the effect of the inclination angle on the helical baffles, this type of baffles increases the heat transfer rate, decrease the pressure drop and vibration.

Dipankar *et al.*, [7], compared the helix angles 10°, 16°, 22°, 28° in a heat exchanger and demonstrated a small decrease in pressure drop with increase in upper inclination of 12°, the heat transfer coefficient in the tubes increases with the increase of velocity. Lutcha *et al.*, [8], experimentally studied various helix angles to increase heat transfer and decrease pressure drop in the shell.

To understand the mechanisms of fluid flow and heat transfer in heat exchanger, a numerical simulation is proposed to study forced convection heat transfer characteristics in this paper.

2. GEOMETRY AND MODELING

2.1 Geometric description

The field of study, shown schematically in figure 1, is a three-dimensional canal of circular horizontal shape of diameter D_s = 90 mm and length L = 600 mm, equipped with seven tubes. Inside which flow the water with a mass flow rate of 01 kg/s

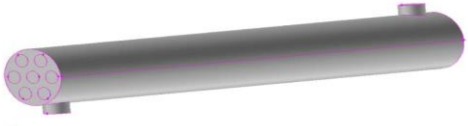

Fig. 1: Geometry of the problem

2.2 Governing Equations

Governing flow equations, continuity, momentum and energy equations are written:

<u>Continuity</u>

$$\frac{\partial(\rho.u_i)}{\partial x_i} = 0 \tag{1}$$

<u>Momentum</u>

$$\rho u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_i}{\partial x_j} - \rho \overline{u'_i u'_j}\right) \tag{2}$$

<u>Energy</u>

$$\rho u_j \frac{\partial(\rho.u_i.T)}{\partial x_j} = +\frac{\partial}{\partial x_j}\left(\left(\frac{\mu}{Pr} - \frac{\mu_t}{Pr_t}\right)\frac{\partial T_i}{\partial x_j}\right) \tag{3}$$

<u>Turbulence kinetic energy</u> k

$$\rho u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right) + G_k - \rho\varepsilon \tag{4}$$

<u>Energy dissipation</u> ε

$$\rho u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{5}$$

<u>Turbulent viscosity</u>

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{6}$$

<u>Turbulence production</u>

$$G_k = -\rho\overline{u_i' u_i'}\frac{\partial u_j}{\partial x_i} \tag{7}$$

The model constants have the following values

$C_{1\varepsilon} = 1.44$; $C_{2\varepsilon} = 1.92$; $C_\mu = 0.09$;

$\sigma_k = 1.0$; $\sigma_\varepsilon = 1.3$ et $Pr_t = 0.09$

2.3 Boundary conditions

A uniform velocity was applied as a boundary condition to the hydraulic input of the computational domain. In addition, as a condition for the thermal limit, a constant temperature of $T_w = 400\,K$ was applied on the walls of the tubes. The temperature of the fluid used was set at $T_{in} = 300\,K$ to the inlet side.

3. RESULTATS

The results obtained by the elaborate calculation code, presented in the form of contours and curves.

3.1 Hydrodynamic behavior

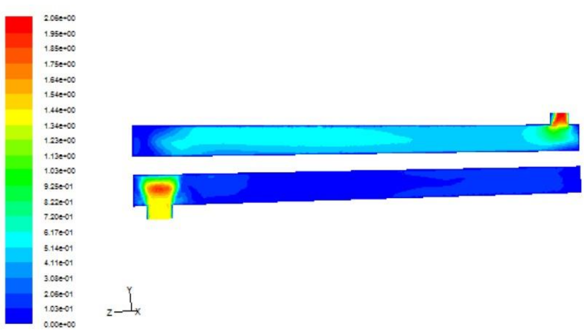

Fig. 2: Velocity contour $x = 0$

Figures 2 and 3 show the velocity contours in the shell and tube heat exchanger. The velocity scale is represented by colors ranging from green (low velocity) to red (high velocity).

From the numerical results it is noted that the values of the velocity are very low in the vicinity of the walls, because of the presence of the strong gradients of friction.

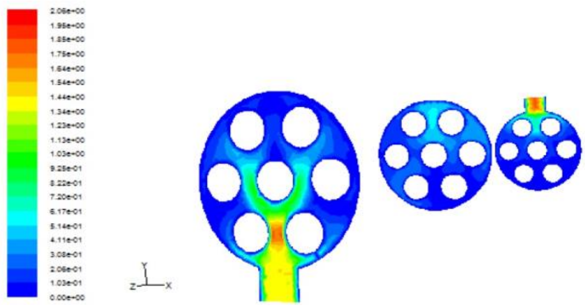

Fig. 3: Velocity contours $z = 0.58\ \mathrm{m}$; $z = 0.3\ \mathrm{m}$; $z = 0.02\ \mathrm{m}$

In the annular passage the velocity in the upper part of the shell is high and the lower part is not used properly.

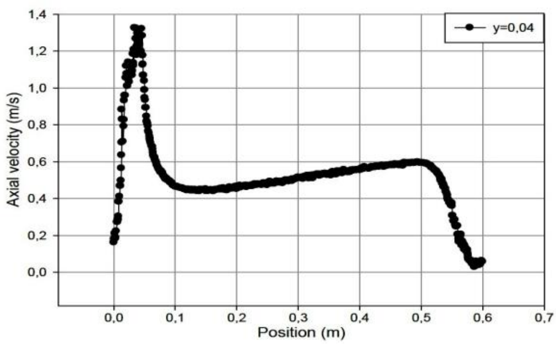

Fig. 4: Velocity distribution $y = 0.04\ \mathrm{m}$

A presentation of the velocity distribution along the heat exchanger has the section $y = 0.04\ \mathrm{m}$ illustrated in the figure 4.

Figure 4 shows a decrease in the velocity along the shell, the velocity of fluid decreases due to the sudden enlargement.

The velocity of the flow in the shell for the case without baffle reaches maximum values of about 12 % of the initial velocity.

3.2 Thermal behavior

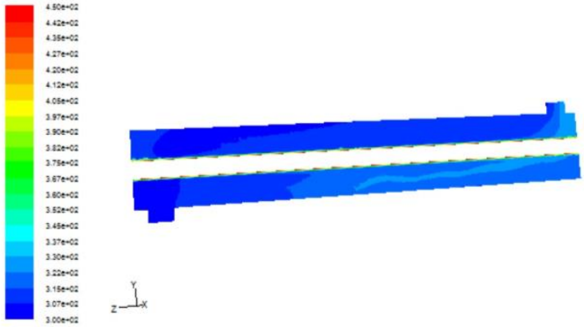

Fig. 5: Temperature contour $x = 0$

Figures 5 and 6 show the isothermals in the heat exchanger. The temperature scale is represented by colours ranging from green (low temperature) to red (high temperature).

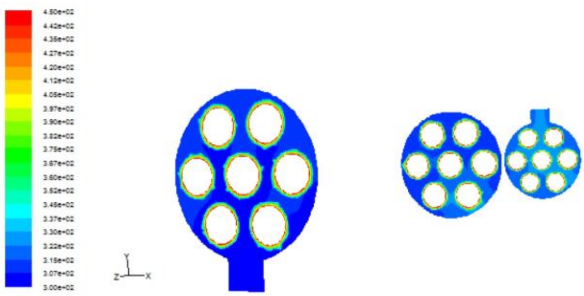

Fig. 6: Temperature contours $z = 0.58\ \text{m}$; $z = 0.3\ \text{m}$; $z = 0.02\ \text{m}$

At the entrance of the exchanger there is a blue outline indicates the temperature of the fluid inlet, begins to vary from center of the lower middle of the shell.

The temperature field presented shows an increase in temperature for the deferent section is important values for areas near the tubes.

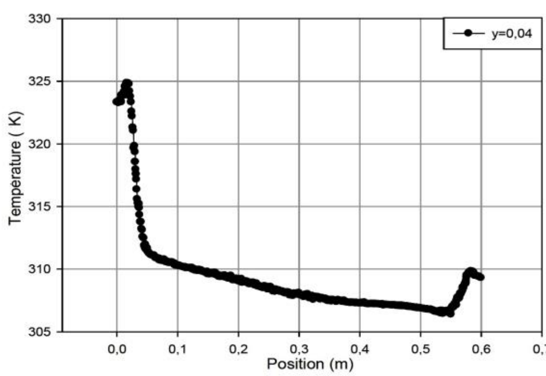

Fig. 7: Temperature distribution $y = 0.04\ \text{m}$

In Figure 7 we see that the fluid temperature increases slightly in the shell.

3.3 Temperature profiles in the shell

The fluid temperature increases in a small way along the flow axis. The flow structure in is characterized by deformations and it will influence considerably the distribution of the temperature field and will allow a better mixing of the fluid which will stimulate the heat transfer

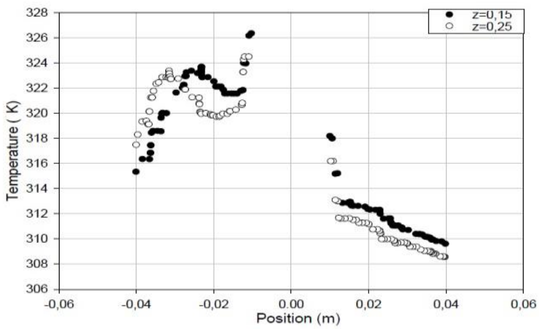

Fig. 9: Temperature profile

The **Table 1** shows the evolution of the temperature, the heat transfer coefficient, the pressure drop, the Nusselt number calculated in the heat exchanger studied.

Table 1: Thermal performance of the exchanger

Parameters	**Value**
T (K)	322.51
h (W/m^2.K)	954.66
ΔP (Pa)	18.79
Nu	26.51
P (W)	0.019

The tubes favours vortices by increasing Reynolds number and greatly increasing the rate of heat transfer.

4. CONCLUSION

A numerical study based on the finite volume method using the Simple algorithm is undertaken in this paper. It consists of stationary turbulent flow in forced convection mode, an incompressible fluid circulating inside a heat exchanger.

The profiles and velocity fields as well as the profiles and temperature distributions in the shell were obtained for the whole geometry.

The simple case assures us a high speed along the shell its measurement more than twelve times the reference speed, the pressure drop is of great importance in the design of the shell and tube heat exchanger because the pumping costs are strongly related to the pressure drop, and thus the decrease of the setting results in lower operating costs.

The flow along the shell is complex and significantly influences the pressure drop due to rapid changes in direction.

NOMENCLATURE

D_s, Shell size, mm	d, Tube diameter, mm
L, Heat exchanger length, mm	N_t, Number of tubes
x, y, z, position coordinates	u, v, w, Velocity components, m/s
λ, Thermal conductivity, W/m.K	T, Temperature, K

ε , Viscous dissipation rate, m^2/s^3	μ , Dynamic viscosity, Pa.S
μ_t , Turbulent viscosity, Pa.S	ρ , Density, kg/m^3
σ_k , Turbulent Prandtl number for k	σ_ε , Turbulent Prandtl number for ε

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