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CFD Investigation of Shell and Tube Heat Exchanger: Impact of Various Tube Bundle Combinations on Heat Transfer Coefficient and Pressure Drop

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Abstract: Shell and tube heat exchangers play a crucial role in various industries, including chemical processing, oil and gas, power generation, and HVAC systems. These exchangers are widely used for transferring heat between two fluids while maintaining their separation. The design of a shell and tube heat exchanger consists of a bundle of tubes enclosed within a larger shell. The hot fluid flows through the tubes, while the cold fluid circulates around them in the shell, facilitating efficient heat transfer and pressure drop of the system. The performance of a shell and tube heat exchanger is influenced by various factors, including the configuration of the tube bundle. The choice of different tube bundle combinations can significantly impact the overall performance of the heat exchanger. In this paper, a numerical study is conducted to examine the effects of two combination of circular and square tube bundle with the simple arrangement: i) circular, ii) square, on a shell and tube heat exchanger. COMSOL Multiphysics is employed to model and simulate this heat exchanger under various mass flow rates. The result shown that the combined geometry heat exchanger exhibits the highest overall heat transfer coefficient compared to heat exchangers with single arrangements. The pressure drop on the tube and shell side was also studied for all cases of heat exchangers. The placement of circular tubes in the center and near the shell has a significant effect on heat transfer and pressure losses. It has been demonstrated that the tubes located at the ends of the shell have a much greater impact on heat transfer compared to the tubes positioned in the center.

Keywords: Shell and tube heat exchanger, Tube bundle, Square tube, Circular tube, Heat transfer Coefficient.

Introduction

Heat exchangers are pivotal components in a wide array of industries and engineering disciplines, including chemical, petroleum, food processing, nuclear power plants, thermal power plants, and electric power plants. They play a vital role in the transmission and recovery of energy(Master et al., 2003).

Among various heat exchanger types, shell and tube heat exchangers are the most widely employed, accounting for 35% to 40% of the usage (Zhang et al., 2009), This popularity can be attributed to their numerous advantages, including ease of maintenance, high efficiency, compact design, and robustness. Shell and tube heat exchangers consist of two primary components: tubes and a shell. Tubes are responsible for conveying aggressive fluids, whether hot or cold, compressible or incompressible, and especially those at high pressures. The shell facilitates the circulation of fluids at lower pressures and typically incorporates longitudinal and/or transverse baffles. The inclusion of baffles serves to extend the fluid's path within the shell, increase turbulence, and enhance heat transfer. Nonetheless, baffles introduce certain drawbacks, such as increased flow resistance, induced vibrations, and areas of stagnant flow along the shell's periphery (Unverdi, 2022) .

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To enhance the performance of shell and tube heat exchangers, many researchers have conducted extensive investigations. Some recent studies have focused on the shell side, particularly examining baffles' characteristics such as material type, quantity, shape, and cutting angles. For instance, El-Said et al. (2021) delved into the impact of varying baffle geometries segmental curved baffles (convex core baffles with convex peripheral baffles , convex peripheral baffles with concave core baffles,) A conventional STHE with straight baffles (SB) is also studied for comparison on exchanger performance. Abbasian Arani and Moradi (2019) explored the combination of disk baffles with other types of drawn baffles. Another avenue of research has concentrated on the tube side, which involves fewer parameters.

The arrangement of tubes significantly influences the energy efficiency of tube and shell heat exchangers. Kallannavar et al. (2020) investigated the effects of four tube arrangements (30°, 90°, 60°, and 45°) in the context of a tube and shell heat exchanger with circular tubes.Meloukı et al. (2013) presented a Comparative numerical study of the influence of square and cylindrical tubes in an aligned bundle on the thermal performance of a tube-and-shell heat exchanger. Ibrahim and Gomaa (2009) and Matos et al. (2004) conducted a comparative analysis of 12 elliptical and circular tubes, demonstrating that elliptical tubes enhance heat transfer by 20% compared to circular tubes, especially within a Reynolds number range of 300 to 800. Tao et al. (2007) performed a numerical investigation on a tube and shell heat exchanger with elliptical tubes, revealing a remarkable 30% increase in heat transfer efficiency compared to circular tubes.

This manuscript's primary objective is to assess the impact of varying tube cross-sections (two combination of circular and square tube bundle with the simple arrangement circular and square) on the hydro-thermal performance of a tube and shell heat exchanger. This assessment is conducted using numerical software for design, modeling, and simulation, which combines heat transfer and fluid mechanics based on the Navier-Stokes equations and the k-ε turbulence model.

Method

Geometrical Model

The exchanger under investigation in this study is categorized as a multi-tube heat exchanger and features a bundle containing 37 tubes. It operates with a single pass on the tube side and a single pass on the shell side, with a counterflow configuration. The shell of the exchanger is equipped with four vertical single segment baffles, each having a 25% cut. The 3D representation of this exchanger's geometry was created using COMSOL Multiphysics 5.5 software, as illustrated in Figure 1.

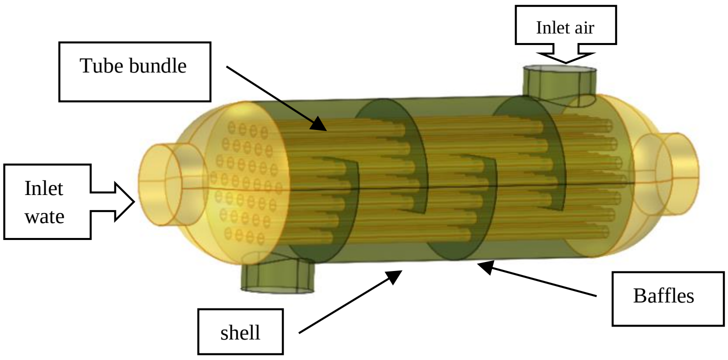

Figure 1. Numerical model of the shell and tube exchanger

The hot fluid (water) enters the shell at of 80°C while the cold fluid (air) enters the tubes with a temperature of 5°C. Table 1 shows the thermo-physical properties of the two fluids.

Table 1. Thermophysical properties of shell and tube heat exchanger

Properties	Fluid		
	water	air	unit
T_{inlet}	80	5	°C
Thermal conductivity	0.6562	0.02401	(W/m.K)
Density	971.8	1006	(kg/m^3)
Specifc heat capacity	4194	1006	(J/Kg K)
Dynamic viscosity	3.54×10^{-4}	1.75×10^{-5}	(Pa s)

The exchanger's overall geometry remained consistent across the cases under examination, with variations introduced solely to the tube bundles, as depicted in Figure 2. Detailed dimensional specifications for these heat exchangers can be found in Table 2.

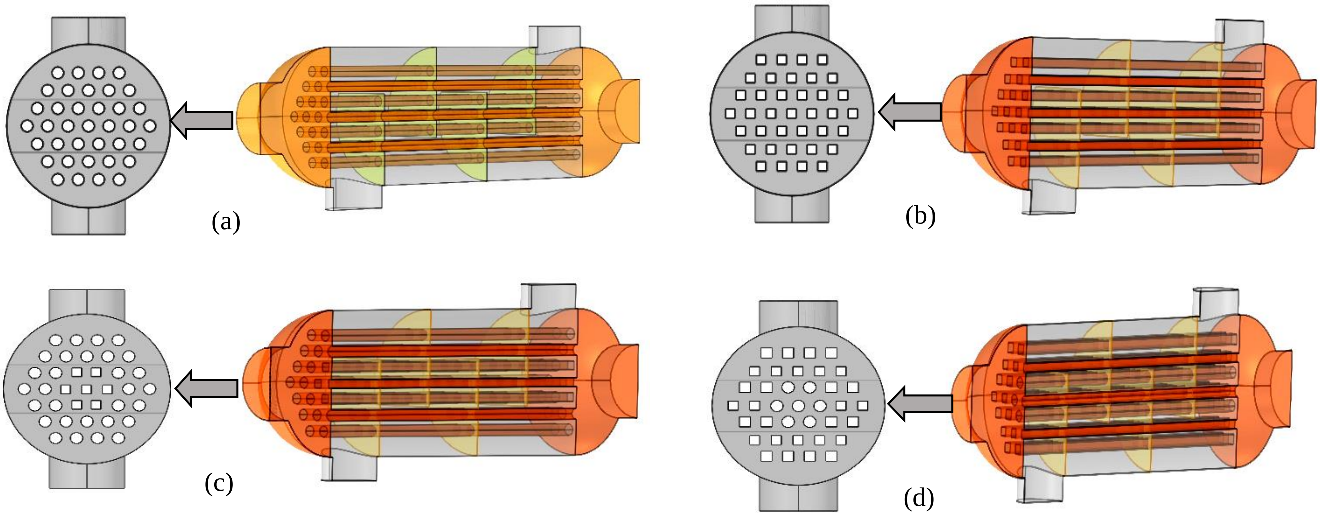

Figure 2. Cross sectional models of shell and tube exchanger : (a) circular , (b) square , (c) combined cir-sqr ,(d) combined sqr-cir

Table 2. Size of heat exchanger

Propertie	Value	Unit
Shell diameter	200	mm
Tube diameter	15	mm
Length shell /tube	500	mm
Number of tubes	37	
Number of baffles	4	

Mathematical Model

Heat Transfer Coefficient

The overall heat transfer coefcient is calculated by the following equation (He et al., 2016) :

$$U = \frac{1}{\frac{1}{hi} + \frac{diln(\frac{de}{di})}{2\lambda} + \frac{1}{he}} \qquad (1)$$

hi : convective transfer coefficients for the tube side, he : convective transfer coefficients for the shell side, k : thermal conducivity of the tube, d_i : inner diameters of the tube, d_e: outer diameters of the tube.

Pressure Drop in Shell Side

The pressure drop in the shell side can be obtained using the following equation (2)(Sarkar & Bhattacharyya, 2012) :

$$\Delta P = f\frac{D_S}{D_e}(N_b + 1)\frac{1}{2}\rho V^2$$

$$f = \exp(0.576 - 0.19lnR_{es})$$

$$D_e = \frac{4\left(\frac{\sqrt{3}P_t^2}{4} - \frac{\pi d_0^2}{8}\right)}{\frac{\pi d_0}{2}}$$

$$Re_s = \frac{\rho u_m D_e}{\mu} \qquad (2)$$

ΔP: pressure drop in shell side **Re_s**: Reynold's number in shell, **f** = friction factor, **D_s** : Sell diameter, **D_e**: equivalent diameter for the triangular pitch,, **N_b** : number of bafe, **ρ**= density of the fluid, **μ**= dynamic viscosity of the fuid, **u_m**: velocity of the fuid,,**V** : mean fow velocity, **P_t**: tube pith ,d_0: outer diameters of the tube.

Turbulent Fow k–ε Model:

The turbulent flow model (k - ε) is widely used in the literature to the study the turbulent flows with very high Reynolds numbers [12]. It is a model with two partial differential equations: i) turbulent kinetic energy (k) and ii) dissipation (ε). COMSOL uses the Navier - Stokes equations as the basic equations to solve the fluid flow models (3) :

$$\rho(\mu.\nabla)\mu = \nabla.\left[-PI + (\mu+\mu_T)(\nabla\mu + (\nabla\mu)^T - \frac{2}{3}(\mu+\mu_T)(\nabla.\mu)I - \frac{2}{3}\rho k I\right] \qquad (3)$$
$$\nabla.(\rho\mu) = 0$$

ρ : density (kg/m^3) ; μ : Dynamic viscosity (kg/m/s) ; P : fluid pressure (Pa)
k : Turbulent kinetic energy (m^2/s^2) et μ_T : turbulent viscosity (Pa.s).
The turbulent kinetic energy is defined as follows (4):

$$\rho(\mu.\nabla)k = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\nabla k\right] + p_k - \rho\varepsilon \qquad (4)$$

σ_k : the turbulent Prandtl number; ε : Turbulent dissipation(m^2/s^2)
Turbulent dissipation is the rate at which velocity fluctuations dissipate. It is defined by the equation (5) :

$$\rho(u.\nabla)\varepsilon = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\nabla\varepsilon\right] + C_{e1}\frac{\varepsilon}{k}p_k - C_{e2}\,\rho\frac{\varepsilon^2}{k} \qquad (5)$$

u : fuid flow velocity,.

For a turbulent flow, the viscosity is defined by the equation (6) :

$$\mu_T = \rho C_\mu \frac{k^2}{\varepsilon} \qquad (6)$$

The production of turbulent kinetic energy may be expressed by equation (7):

$$P_k = \mu_T\left[\nabla u\!:(\nabla u + (\nabla u)^T) - \frac{2}{3}(\nabla.u)^2\right] - \frac{2}{3}\rho k \nabla.u \qquad (7)$$

The constants used in the equations for turbulent kinetic energy, turbulent dissipation and turbulent viscosity are: C_{e1}=1.44, C_{e2}=1.92, C_μ=0.99, σ_k = 1, σ_ε =1.3

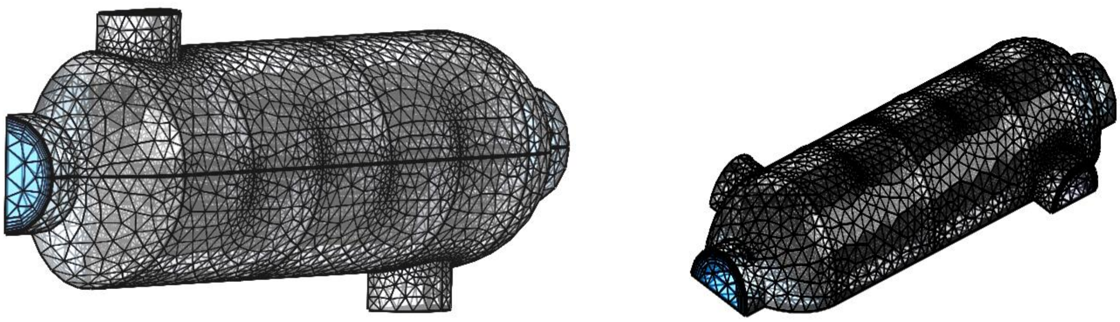

Figure 3**.** The meshed model used in the numerical calculations of heat exchangers

Mesh Control

The computational grid was created with the aid of COMSOL software. To discretize the volume of the shell and tube assembly, a free unstructured tetrahedral mesh was employed, as illustrated in Figure 3. A mesh sensitivity analysis was conducted to determine the optimal quantity and size of mesh elements, aiming to minimize computational expenses.

Results and Discussion

Temperature Evolution

In Figure 4, we observe the 3D temperature contours within the heat exchanger for the different geometries. Heat transfer within the shell and tube exchanger occurs through two primary modes: conduction and convection. Conduction is facilitated by the separating walls between the two fluid streams, which include the tube plate inside the exchanger, the shell wall, and the baffles. On the other hand, convection is responsible for heat transfer between the cold fluid (air) and the inner surfaces of the tubes, as well as between the hot fluid and the external surfaces of the tubes and the interior of the shell.

Temperature distribution is remarkably uniform throughout the exchanger. The lowest temperature within the tube bundle corresponds to the outlet temperature of the cold fluid, which is 320.68, 320.75,322.33 and 322.6K for the combined square- circular, square, circular and combined circular-square geometries and, respectively. Notably, the heat transfer performance is higher in the exchanger with a combined geometry with square - circular geometry compared to the other geometries. This observation can be attributed to the specific characteristics of the combination between square and square geometry.

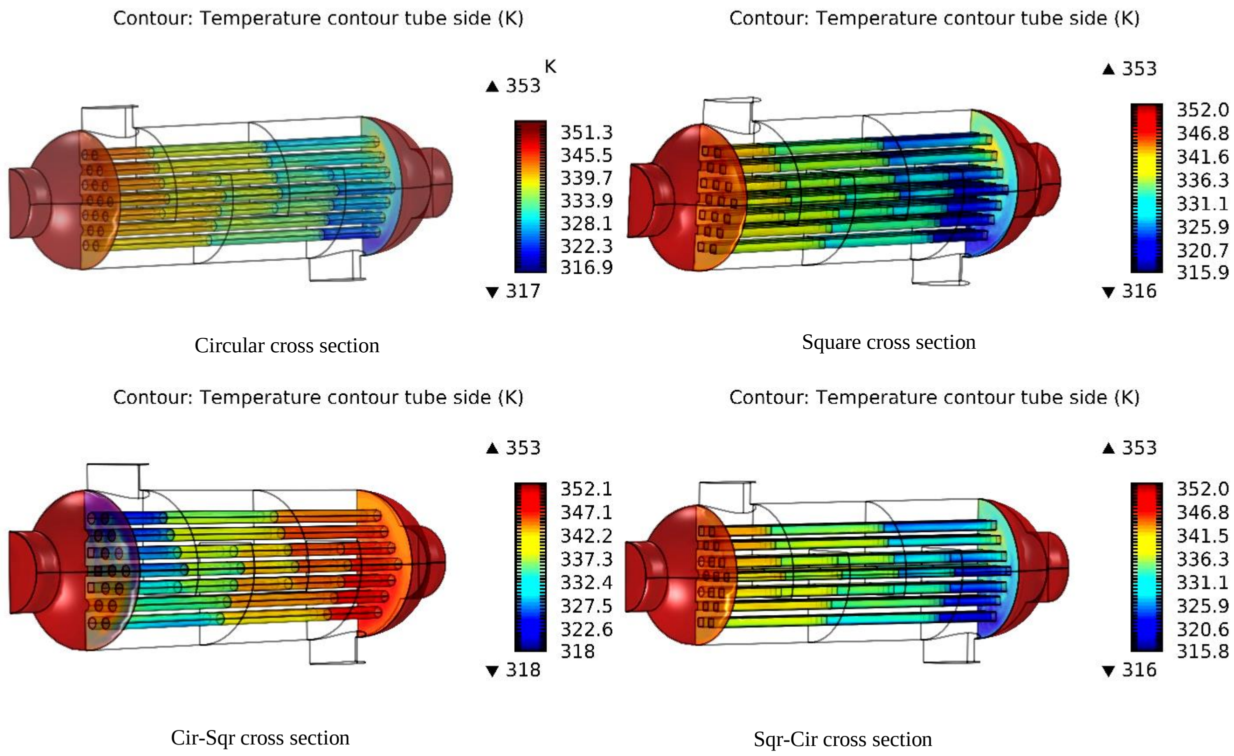

Figure 4. Temperature contour tube side of different heat exchangers

Velocity Evolution

In Figure 5, velocity contours are depicted within the cross-sectional areas of various geometries, namely: square, circular, and the combined circular-square configurations. Notably, the highest velocity values within these heat exchanger geometries were observed proximate to the edges of the baffles. Specifically, for the square section, the velocity reached 3.48 m/s, whereas for the circular section, it measured 3.04 m/s. In the case of combined geometries, the velocities were noted at 2.97 m/s for the circular-square arrangement (cir-sqr) and 3.07 m/s for the square-circular arrangement (sqr-cir).

This accelerated flow pattern on the shell side can be primarily attributed to the restricted flow area between the shell and the baffle plates. Importantly, it is worth noting that this phenomenon predominantly impacts the shell side and does not significantly affect the tube side of the heat exchanger

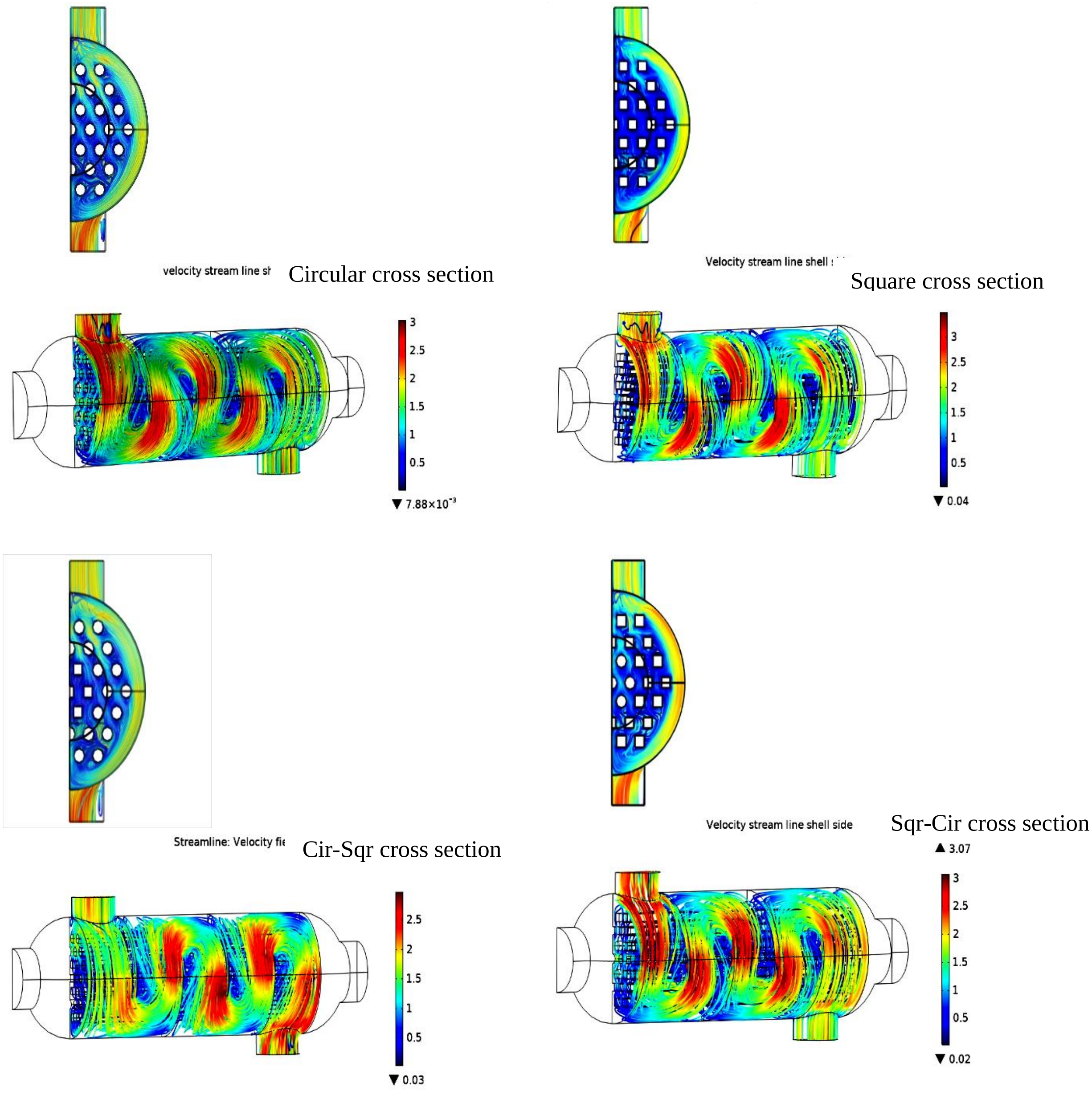

Figure 5. Streamline velocities on the shell side of various heat exchangers

Heat Transfer Coefficient

Figure 6 provides a graphical representation illustrating the correlation between the global heat transfer coefficient and the Reynolds number for four distinct heat exchanger geometries: square, circular, and two combined circular-square configurations. The relationship observed demonstrates a direct proportionality between the global heat transfer coefficient and the Reynolds number. Notably, the point of maximum Reynolds number signifies the juncture of the highest mass flow rate, consequently resulting in the most substantial heat transfer coefficient.

Among the investigated geometries, the heat exchangers featuring a circular section exhibit the most remarkable heat transfer performance, boasting a coefficient of 10.458 W/m²·K. This surpasses the performance of the other geometries. This superiority in heat transfer efficiency can be attributed to the circular geometry's reduction in cross-sectional area on the shell side. This reduction leads to an augmented flow rate, enabling a more rapid and efficient transfer of heat to the cold fluid.

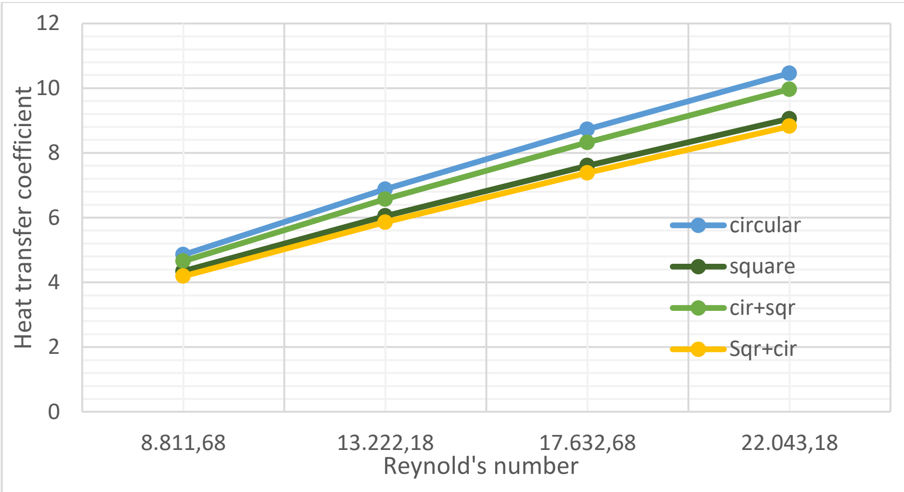

Figure 6. The overall heat transfer coeffcient in the different geometries:

Pressure Drop in Shell Side

According to the findings presented in Figure 8 which delineates the relationship between pressure drop on the shell side and the Reynolds number across various cross-sectional shapes (square, circular, and a fusion of circular and square geometries), provide compelling evidence of a direct proportionality between pressure drop and Reynolds number. Notably, the heat exchanger employing the combined circular-square cross-section exhibits the highest pressure drop, signifying a substantial impact on the system's hydraulic resistance.

In contrast, both the circular geometry and the combined circular-square configuration demonstrate significantly lower pressure drops, registering at 20.434 Pa and 21.512 Pa, respectively. This divergence in pressure drop performance is attributed to the nuanced flow dynamics induced by distinct cross-sectional shapes. The introduction of a square cross-section geometry is observed to augment the formation of recirculation zones, thereby amplifying frictional effects on the shell side. Consequently, this phenomenon leads to an appreciable elevation in pressure drop, emphasizing the influence of cross-sectional geometry on the heat exchanger's hydraulic behavior

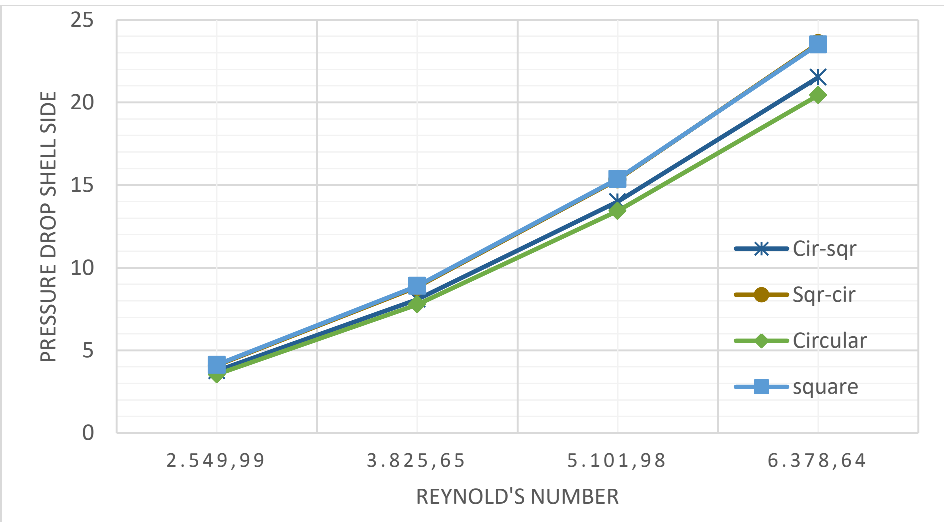

Figure 7. Pressur drop shell side in the different geometries

Conclusion

The performance of the heat exchanger was assessed through simulations conducted using Comsol Multiphysics 5.5 software, employing the (k, ε) model. The results of the simulations for heat exchangers with single arrangements unequivocally favored the circular geometry, showcasing the highest heat transfer coefficient. The combination of square and circular tubes, strategically positioned at the center and ends of the heat exchanger, was the chosen configuration. This combined arrangement was comprehensively evaluated for its energy-related characteristics, including the overall heat transfer coefficient and the pressure drop on the shell side. The study's findings can be distilled into the following key points:

- Geometric variations in the tube configuration directly impact both the overall heat transfer coefficient and pressure drop on the tube and shell sides.
- Parameters such as flow velocity, tube cross-sectional area, flow rate, and Reynolds number exert significant influence on these heat transfer characteristics.
- Notably, minimizing pressure drop translates to reduced pump power requirements, consequently augmenting the overall system efficiency.
- The strategic placement of tubes, particularly with circular tubes at the ends and square tubes at the center, emerged as the superior configuration, outperforming the reverse arrangement (square at the ends and circular at the center)

Scientific Ethics Declaration

The authors declare that the scientific ethical and legal responsibility of this article published in EPSTEM journal belongs to the authors.

Acknowledgements or Notes

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