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Performance of Shell and Tube Heat Exchangers with Varying Tube Layouts

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Authors' contributions

This work was carried out in collaboration between both authors. Both authors read and approved the final manuscript.

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Original Research Article

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ABSTRACT

Shell and tube heat exchangers (STHEs) are the most common type of heat exchangers and are applicable for wide range of operating temperatures and pressures. Numerical analyses on thermal-hydraulic performance of three sets of shell and tube heat exchangers (STHEs) with different geometrical tube layout patterns variations namely; triangular (30º, STHE_T), rotated triangular (60º, STHE_RT) and the combined (STHE_C) patterns were carried out in this study. The results from solving the governing continuity, momentum and energy equations showed that bulk of the heat transfer and pressure drop occur during the cross-flow of shell-fluid through the tube bundles. Evaluation of the performances of the heat exchangers showed that the STHE_T is more desirable followed by the STHE_C as they exhibit higher heat transfer coefficient than the STHE_RT for the same pressure drop in the shell-side.

Keywords: Tube layout; thermal-hydraulic; shell-side; performance factor; baffle window; cross-flow.

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1. INTRODUCTION

A heat exchanger (HE) is a device that is used to transfer thermal energy between two or more fluids, at different temperatures and in thermal contact [1]. Heat exchangers are used in a wide variety of engineering applications like power generation, waste heat recovery, manufacturing industry, air-conditioning, refrigeration, space applications, and petrochemical industries [2]. Heat exchangers may be classified according to transfer process, construction, flow arrangement, surface compactness, number of fluids and heat transfer mechanisms [3]. Shell and Tube Heat Exchangers (STHEs) are the most common type of heat exchangers applicable for a wide range of operating temperatures and pressures. This widespread use can be justified by its versatility, robustness and reliability [4].

As its name implies, STHE consists of a shell (a large pressure vessel) with a bundle of hollow tubes fitted inside the shell. The basic principle of operation is very simple as flows of two fluids with different temperature brought into close contact for heat exchange, but prevented from mixing by some sort of physical or metal barriers (bundle of tubes) [5]. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers. The selection of the tube layout pattern depends on fundamental issues that may which influence the shell-side performance and hence the overall performance. Such include compactness, heat transfer, pressure drop, accessibility for mechanical cleaning, and phase change if any on the shell-side.

Using Tubular Exchangers Manufacturers Association (TEMA) standard for tube layouts as shown in Fig. 1, triangular (30º) and rotated triangular (60º) patterns will accommodate more tubes than square (90º) and rotated square (45º) patterns for the same tube pitch. While a triangular pattern produces high turbulence and therefore a high heat-transfer rates but at the expense of a higher pressure drop, a square, or rotated square arrangement, is used for heavily fouling fluids, where it is necessary to mechanically clean the outside of the tubes [6,7,8].

According to [9], the flow characteristic around some tube rows in tube bundle is strongly influenced by the tube layout pattern and has direct influence on heat exchange between fluids. Other factors influencing the shell-side performance of STHEs are not limited to the number of tubes and baffles, twisted and fin tubes, baffle cut, baffle spacing, baffle orientation, size of inlet and outlet zones. Different research works have been carried out by researchers on each of these factors to establish facts for design improvement of STHEs. Raj and Ganne [10] investigated the impact of various baffle inclination angles on fluid flow and the heat transfer characteristics of a shell-and-tube heat exchanger for three different baffle inclination angles: 0º, 10º, and 20º with baffle cut of 36%. From this work, the 20º baffle inclination angle showed a better performance than the remaining. Kumar and Jhinge [11] conducted experimental work on STHEs containing segmental baffles at different baffle angular orientations: 0º, 15º, 30º and 45º to the horizontal. It was discovered that the heat transfer rate increased up to 30º angular orientation of the baffles and dropped at 45º but the shell-side pressure drop decreases continuously from 0º to 45º. Sivarajan et al. [12] carried out a 3D numerical simulation of a Shell and tube heat exchanger with continuous helical baffle, their results indicated that the STHE with helical baffles have better flow and heat transfer performance than the STHE with conventional baffles. Also, Zhang et al. [13] performed 3D simulation of STHEs with non-continuous middle-overlapped (30º, 40˚ and 50º helix angles) helical baffles. Having found out that STHE with 40^0 helix angles gave a better performance than others, it was compared with continuous helical baffle STHE and the obtained results obtained indicated that the heat transfer coefficient per unit pressure drop of the former is appreciably larger than that of the later.

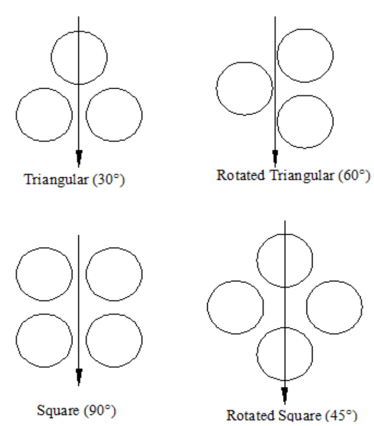


Fig. 1. Tube layout patterns

This study is however designed to numerically investigate the performance of STHEs with three different set of tube bundles, namely triangular (STHE_T), rotated triangular (STHE_RT) and the combined (STHE_C) patterns.

2. GEOMETRICAL MODELLING

The geometric parameters of the STHEs model are as listed in Table 1. The three layout patterns are as shown in Fig. 2. It is assumed that there are no clearances between the shell and the baffles except for the baffle window, and between the baffles and tubes. Also, the tube material is of negligible thickness. As the working fluids temperature change, the thermal properties of the fluids; water and engine oil in tube-side and shell-side respectively vary. This variations with temperature were found in [14,15].

3. GOVERNING EQUATIONS AND NUMERICAL METHODS

The governing equation for the analysis is the k - ε turbulence RANS model given in equations (1) to (5) [16] below:

Continuity Equation

$$\nabla.(\rho u) = 0 \tag{1}$$

Momentum Equation

$$\rho u.\nabla u = \nabla.\left[-pI + (\mu+\mu_T)\left(\left(\nabla u + (\nabla u)^T\right) - \left(\tfrac{2}{3}\right)(\nabla.u)I\right) - \left(\tfrac{2}{3}\right)pkI\right] + F \tag{2}$$

Turbulent Kinetic Energy Equation

$$\rho u.\nabla k = \nabla.\left[\left(\mu+\tfrac{\mu_T}{\sigma_k}\right)\nabla k\right] + \mu_T P(u) - \left(\tfrac{2\rho k}{3}\right)\nabla.u - \rho\varepsilon \tag{3}$$

Turbulent Energy Dissipation Equation

$$\rho u.\nabla \varepsilon = \nabla.\left[\left(\mu+\tfrac{\mu_T}{\sigma_\varepsilon}\right)\nabla \varepsilon\right] + \left(C_{\varepsilon 1}\tfrac{\varepsilon}{k}\right)\left[\mu_T P(u) - \left(\tfrac{2\rho k}{3}\right)\nabla.u\right] - C_{\varepsilon 2}\tfrac{\varepsilon^2}{k} \tag{4}$$

Energy Equation

$$\rho C_p u.\nabla T = \nabla.(k\nabla T) + Q \tag{5}$$

where

$P(u) = \nabla u : \left(\nabla u + (\nabla u)^T\right) - \tfrac{2}{3}(\nabla.u)^2$, and $\mu_T = \rho C_\mu \tfrac{k^2}{\varepsilon}$

The constants of this model are given as:

$C_\mu = 0.09$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_k = 1.0$, $\sigma_k = 1.3$

Table 1. The geometric parameters of the shell and tube heat exchanger

Shell-side parameter	Shell Diameter	108.06 mm
	Inlet and Outlet Diameter	30 mm
Tube parameter	Tube Diameter, *d*	15.88 mm
	Layout Pattern	Triangular (30º), Rotated Triangular and The Combined Layout
	Pitch	1.25*d*
	Number of Tubes	19
Baffle parameter	Number of Baffles	6
	Baffle Cut	25%
	Baffle Spacing	43.26 mm

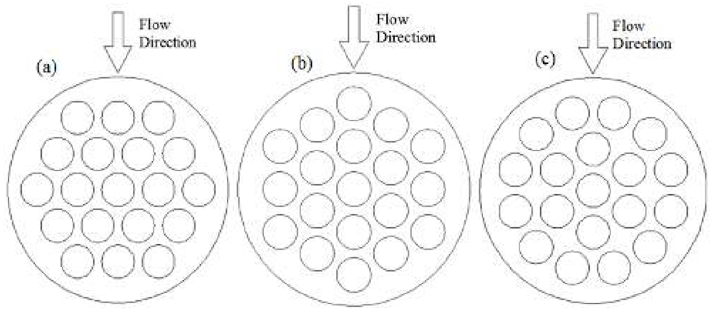


Fig. 2. Tube bundle arrangement for (a) triangular (STHE_T), (b) rotated triangular (STHE_RT), and (c) combined (STHE_C) patterns

At the computational domain boundaries for the tubes and shell; velocity-inlet and outflow boundary conditions were applied at the inlets and outlets respectively. The standard wall function condition is applied to all the tubes and shell walls including the baffles. The inlet temperature for the tube-side was fixed as 303.15 K while the corresponding shell-side inlet temperature was taken as 373.15 K. Heat loss to the environment is totally neglected because shell wall is assumed to be insulated.

The finite element mesh of the computational domain contained tetrahedral elements. Solutions to the discretized domain were obtained using segregated solvers: two iterative solvers, GMRES with Incomplete LU as preconditioner for velocity and pressure, and temperature respectively; and one direct solver for the turbulent kinetic energy and turbulent energy dissipation [17].

A shell gain factor [18], to evaluate the performance of the three STHEs is defined as

$$\Gamma = \frac{h}{\Delta p} \tag{6}$$

Where h is the coefficient of heat transfer and Δp is the pressure drop.

The performance factor of each STHE is determined as the ratio of the shell gain factor of STHE_T (chosen as standard) to each of the other layouts. When this ratio is greater than one, it indicates that the STHE with such tube-layout pattern is more desirable.

4. RESULTS AND DISCUSSION

The vertically cut-plane sections for the velocity distributions of tube-side fluid of all the heat exchangers are as shown in Fig. 3. In each of the STHEs, the fluid velocity increases from tubes inlet (left) to outlet (right) and this may be attributed to reduction in fluid density as the shell-side is heated up while its velocity increases. It is also observed that active zones, portions with higher velocities, occurred between baffles and this is due to the cross flow heating of the tubes from the shell.

Fig. 4 shows vertically cut-plane section for the pressure distributions of the shell-side fluid in each of the heat exchangers. It is also observed with the same trend in each of the STHEs that the pressure drop is more pronounced in the shell zones than in the baffle window and this is due to the cross-flow obstructions caused by the tube bundles.

As could be noticed in the region where the Reynolds number is less than 10000 in Fig. 5, the performance factors of STHE_RT and STHE_C decrease sharply but this trend slowed down in the higher Reynolds number region. STHE_T has higher performance than the other two.

In the lower pressure drop region, the overall heat transfer coefficient increases at a faster rate than the increase observed in the higher pressure drop region as shown in Fig. 6. This substantiate some facts found in literature that as the mass flow rate or Reynolds number increases, the increase in heat transfer rate drops with increase in pressure drop [8]. The STHE_T gives higher heat transfer coefficient for the same pressure drop than the other two, while the heat transfer coefficient for STHE_C is also higher than that of STHE_RT for the same pressure drop.

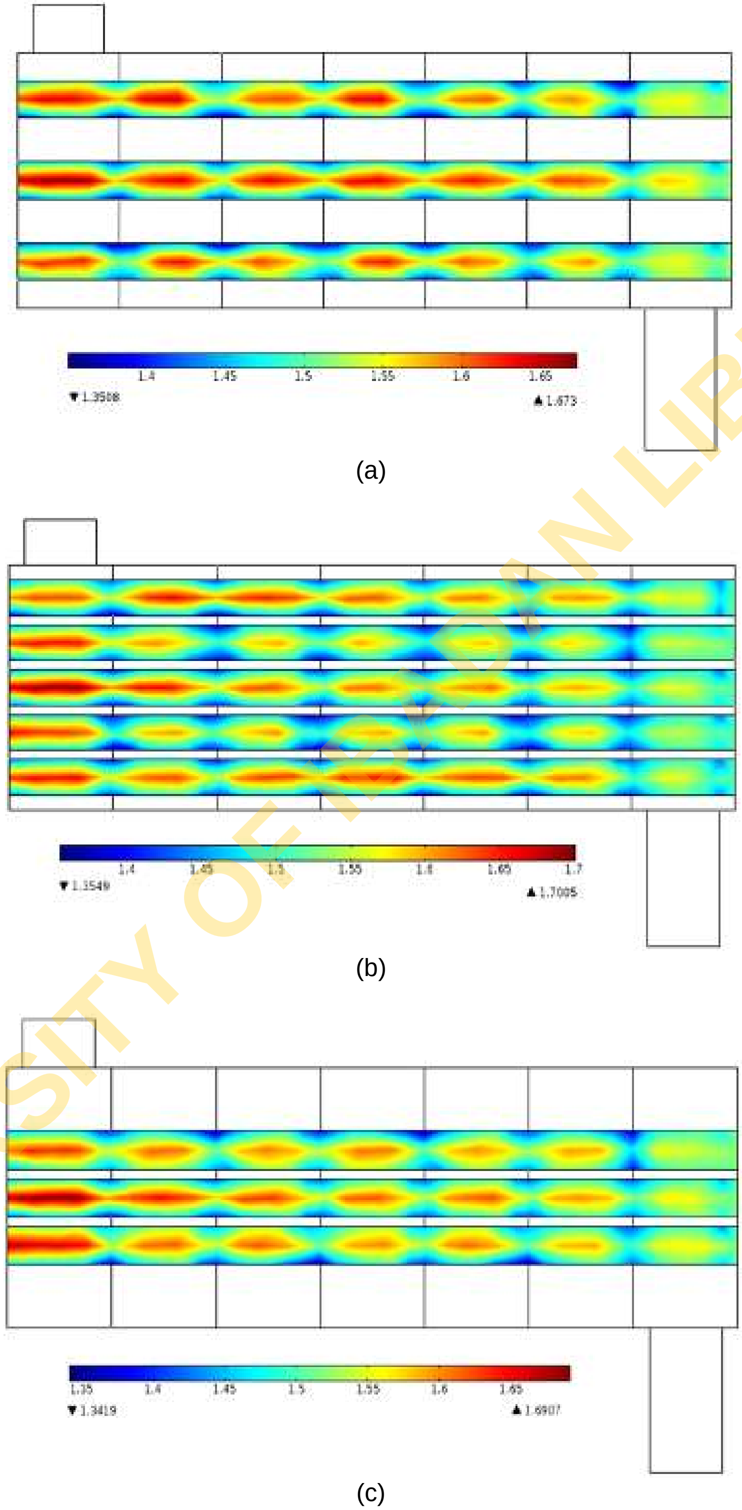
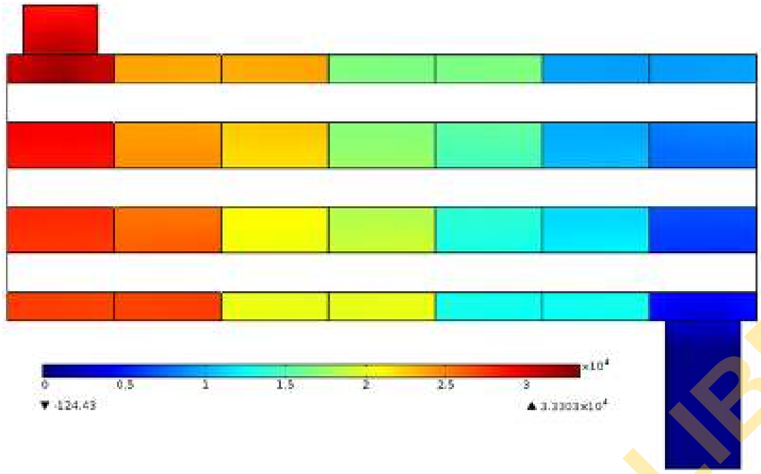
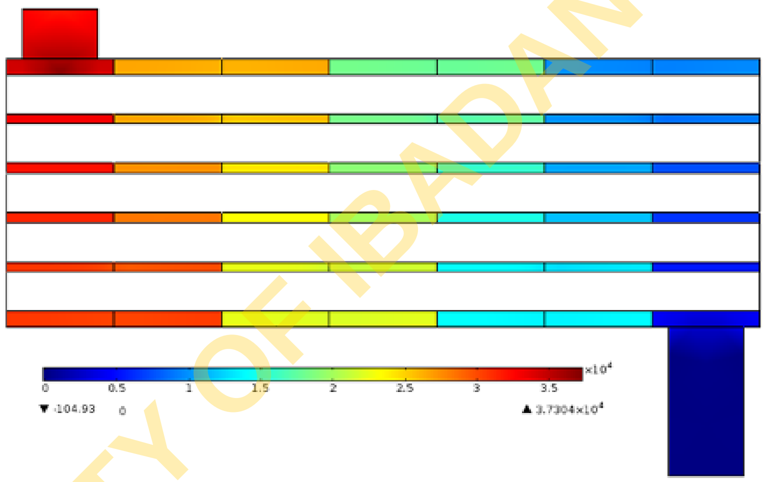


Fig. 3. Velocity distributions (m/s) in the tube-side through the cut-plane section at 0.3 kg/s (Re≈30000) for (a) the STHE_T, (b) the STHE_RT, and (c) the STHE_C



(a)



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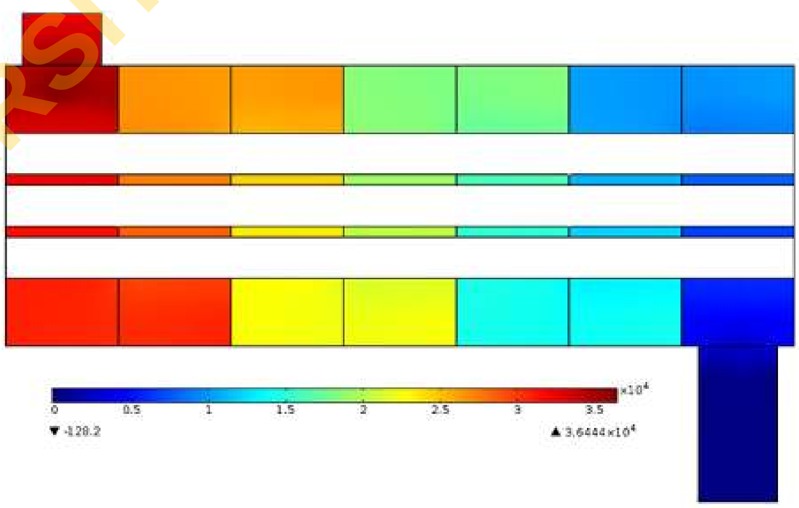

(c)

Fig. 4. Pressure distributions (Pa) in the shell-side through the cut-plane section for (a) the STHE_T, (b) the STHE_RT, and (c) the STHE_C

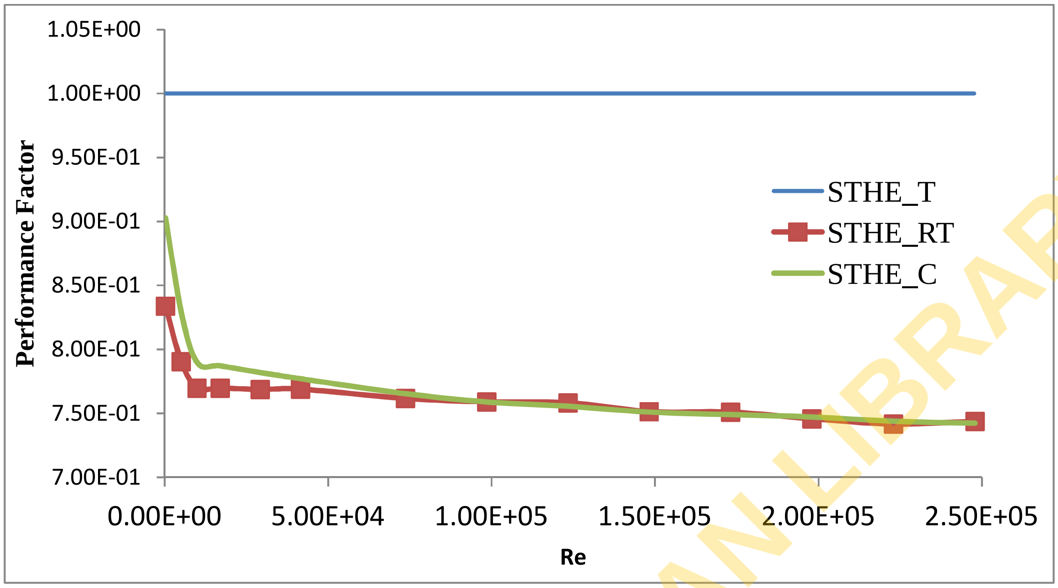

Fig. 5. The performance factor as a function of Reynolds Number

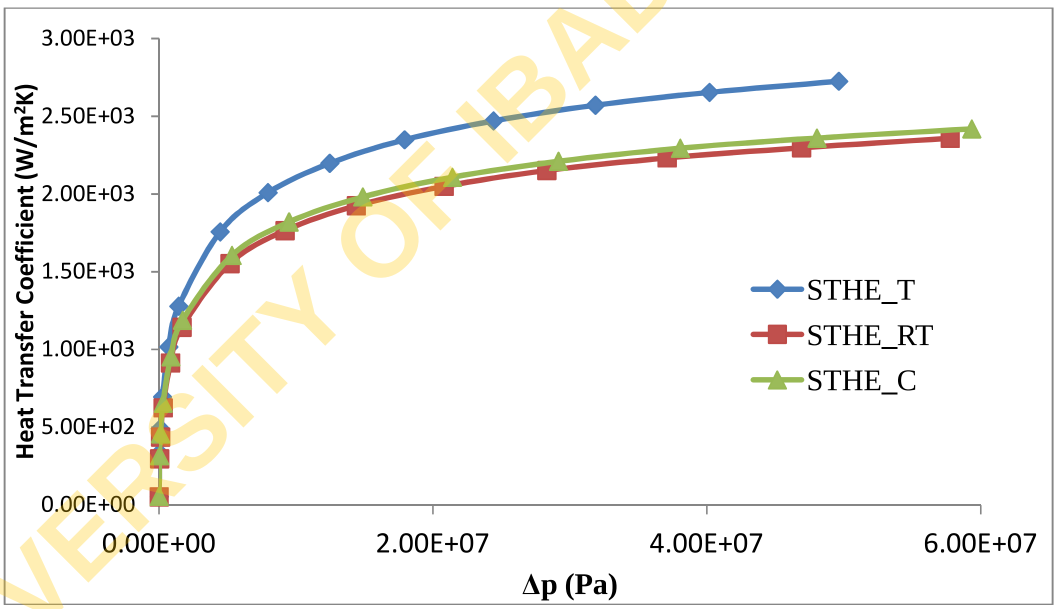

Fig. 6. Heat transfer coefficient as a function of pressure drop

5. CONCLUSION

In this study, numerical investigation has been carried out for predicting the performance of shell and tube heat exchangers with three different tube layout patterns. The results showed that much of the heat transfer and pressure drop occur during the cross-flow of shell-fluid through the tube bundles. In comparison of STHE_T with others, the average deviations of heat transfer coefficients are 11.2% and 8.3% for STHE_RT and STHE_C respectively while the pressure drops 16.0% and 18.8% for STHE_RT and STHE_C respectively. From the two criteria selected to evaluate the performances of the heat exchangers, the STHE_T is more desirable follow by the STHE_C as they exhibit higher heat transfer coefficient than the STHE_RT for the same pressure drop in the shell-side.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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