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Evaluating U/PD Ratio and Energy Efficiency in Heat Exchangers with Novel Baffles

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Abstract
This research investigates a novel baffle configuration specifically designed to significantly reduce hydraulic resistance without sacrificing thermal capabilities. While traditional shell-and-tube heat exchangers (STHeX) are widely utilized, conventional segmental baffles frequently result in high pressure drops and the formation of thermal dead zones, directly compromising system efficiency. Existing studies have explored various modifications, yet a significant gap remains in achieving a streamlined flow that optimizes the trade-off between hydraulic resistance and thermal performance at higher mass flow rates. This research addresses this limitation by introducing trapezoidal baffles to fundamentally alter fluid dynamics, inducing a torsional flow pattern that eliminates stagnant zones and enhances mixing. Consequently, the proposed designs were numerically tested and optimized using three-dimensional Computational Fluid Dynamics (CFD) simulations to maximize the performance of the STHeX. The simulation results demonstrated that the Trapezoidal design offered superior hydraulic stability, maintaining pressure drops below 3200 Pa compared to over 4000 Pa in conventional segmental designs at peak flow rates (1.4 kg s^{-1}). Furthermore, this geometric modification resulted in a substantial improvement in the overall heat transfer coefficient per pressure drop (U/PD) ratio, achieving a peak efficiency of 0.9 W $m^{-2}K^{-1}Pa^{-1}$ compared to 0.7 W $m^{-2}K^{-1}Pa^{-1}$ for the conventional STHeX. This study concludes that modifying the baffle geometry into a trapezoidal profile is a vital strategy for drastically reducing hydraulic resistance while maintaining thermal performance, thereby significantly enhancing overall energy efficiency.

Keywords: Computational Fluid Dynamic; Heat Exchanger; Overall Heat Transfer Coefficient; Pressure Drop;

1 Introduction

Shell-and-tube heat exchangers (STHeX) have firmly established themselves as the industry standard across metallurgy, petrochemicals, and light industry, largely due to their adaptability and efficient fluid heat transfer [1][2][3]. These systems are valued for their simplicity and economic viability, playing a vital role in energy conservation and heat recovery. However, they are not without flaws. The use of traditional segmental baffles often introduces structural risks from vibration and makes the system susceptible to fouling [4]. Furthermore, managing pressure drop remains a critical engineering challenge; high flow resistance inevitably drives up pumping costs and drags down system efficiency [5][6]. Consequently, a true assessment of STHeX performance must look beyond simple thermal metrics to prioritize pressure drop minimization, overall reliability, and lifecycle maintenance costs [7]. In response to these issues, researchers worldwide have focused on structural modifications to enhance heat transfer capabilities [8][9][10]. A notable study by Abbasian Arani Moradi investigated the synergy between ribbed tubes (both triangular and circular) and segmental baffles. Their findings highlighted the superiority of the DB-TR (disk baffle with triangular ribbed tubes) configuration, which achieved a 39% efficiency gain compared to standard STHeX designs. This confirms that optimizing the tube-baffle interface is essential for managing thermal output and pressure drop [11]. Furthermore, Liu et al. [12] proposed the STHeXFBTB—a novel design utilizing modified folded helical baffles and bent oblate tubes to eliminate central shell leakage. Numerical simulations indicate that this geometry offers a

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comprehensive upgrade over traditional units, yielding higher heat transfer coefficients with lower pressure loss. The study further identified that performance is highly sensitive to length-width and cutting ratios, whereas parameters like screw pitch ratio and tube diameter remain largely inconsequential [12]. The primary goal of this investigation is to push the performance boundaries of shell-and-tube heat exchangers by introducing a novel baffle configuration. Specifically, we propose the implementation of swirl baffles. Swirl baffles is a design concept that has received surprisingly little attention in current research. By inducing a rotational flow, these baffles create a more complex fluid path and ensure superior mixing on the shell side, directly boosting the convective heat transfer coefficient. We analyzed four key factors affecting system performance to identify the optimal parameter combination. To ensure accuracy, we validated our Computational Fluid Dynamics (CFD) results against mathematical models for conventional exchangers found in existing literature; the simulation data showed excellent agreement with established theory. Ultimately, our findings demonstrate that this new swirl baffle design significantly outperforms conventional models, delivering a superior overall heat transfer coefficient per unit pressure drop (U/PD).

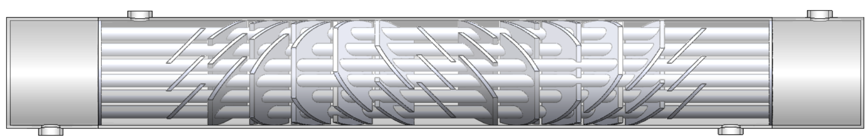


Figure 1. Geometry of computational STHeX

2 Methods

2.1 Geometry Model

The study employs a physical model of a heat exchanger defined by its baffle arrangement, which consists of inclined plates twisted at set angles. This computational geometry is shown in Fig 1, and the corresponding geometric parameters are listed in Table 1.

Table 1. Geometric detail sizes

No.	Parameter	Quantity/size
1.	Inner Dia. Of Shell (mm)	160
2.	Tube length (mm)	1034
3.	Dia. of tube (mm)	19
4.	Tube pitch (mm)	23.81
5.	tube arrangement	Quadratic
6.	Baffle thickness (mm)	4
7.	The spacing between baffles	64

2.2 Turbulence Scheme and Governing Equation

In numerical simulations it is very important to fulfill the governing equation, with the following equations: Continuity equation:

$$\frac{\partial \rho}{\partial t} + div(\rho u) = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial \rho u}{\partial t} + div(\rho u \mathbf{u}) = -\frac{\partial p}{\partial x} + div(\mu grad u) + S_M x \tag{2}$$

$$\frac{\partial \rho v}{\partial t} + div(\rho v \mathbf{u}) = -\frac{\partial p}{\partial y} + div(\mu grad v) + S_M y \tag{3}$$

$$\frac{\partial \rho w}{\partial t} + div(\rho w \mathbf{u}) = -\frac{\partial p}{\partial z} + div(\mu grad w) + S_M z \tag{4}$$

Energy equation:

$$\frac{\partial \rho i}{\partial t} + div(\rho i \mathbf{u}) = -p div \mathbf{u} + div(k grad T) + \Phi + S_i \tag{5}$$

Turbulent kinetic energy equation:

$$\frac{\partial \rho k}{\partial t} + div(\rho k \mathbf{u}) = div(-\overline{p'\mathbf{u}'} + 2\mu\overline{\mathbf{u}' s'_{ij}} - \rho\frac{1}{2}\overline{u'_i \cdot u'_i u'_j}) - 2\mu\overline{s'_{ij} \cdot s'_{ij}} - \rho\overline{u'_i u'_j} \cdot S_{ij} \tag{6}$$

The complex shell-side flow, characterized by significant streamline curvature, demands a high-precision turbulence model. Guided by Liu et al. [12], who established the superior accuracy of the RNG k- model over standard alternatives, we adopted it for this analysis. We paired this with improved wall functions to resolve near-wall fluid dynamics. For the discretization scheme, the first-order upwind method was utilized for all variables—spanning momentum and energy to turbulent kinetic energy and dissipation rates.

2.3 Limiting Scenarios and Fundamental Premises

Water was selected as the working fluid for both the shell and tube sides. At the inlets, we defined specific mass flow rates, setting the temperatures to 50°C for the tube side and 28°C for the shell side. The outlet allows for free outflow, maintaining a pressure of 101,325 Pa. To streamline the simulation, we applied several key assumptions: the turbulent flow is treated as steady-state, fluid properties remain constant, and the shell walls and baffles are thermally insulated (adiabatic). Furthermore, to simplify the geometry without sacrificing critical data, we neglected both buoyancy effects and the clearance gaps between the baffles and tubes.

2.4 Validation of Numerical Simulation

To validate the accuracy and efficiency of our numerical model, we benchmarked the simulation results against mathematical calculations using the Bell-Delaware method. The study focuses on a 1-meter long Shell and Tube Heat Exchanger (STHeX) featuring 19 mm tubes arranged in a quadratic layout with a 23.8 mm pitch, housed within a 160 mm shell. Operationally, hot water enters the tubes at 50°C, while cold water flows through the shell side at 28°C with rates ranging from 0.30 to 0.85 kg/sec. We determined the heat transfer coefficient and pressure drop using Equation 7 – Equation 17, which are derived from established literature [8][13]. For a deeper dive into the specific experimental details, please refer to [12].

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{7}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{8}$$

$$\Delta T_l mtd = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)} \tag{9}$$

$$A = N_t \pi d l \tag{10}$$

The variable A corresponds to the heat transfer surface area, comprising the number of installed tubes (Nt), tube diameter (d), and length (l). Additionally, we use Tlmtd to signify the logarithmic mean temperature difference. To determine the Nusselt number, we tailored our calculations to the specific flow conditions, treating the tube side as laminar forced convection and the shell side as turbulent. It is worth noting, however, that the Reynolds and Prandtl numbers were calculated using the same standard formulas for both fluid streams.

$$Pr = \frac{G_s d_i}{\mu_i} \tag{11}$$

$$Re = \frac{\mu_s C_s}{k_s} \tag{12}$$

$$Nu_t = 1.86(Re_t Pr_t d/L)^{0.33}(\frac{\mu_b}{\mu_w})^{0.152} \tag{13}$$

$$h_t = \frac{Nu_t k}{d_i} \tag{14}$$

$$Nu_s = 0.36({Re_s}^{0.55})({Pr_s}^{0.33})(\frac{\mu_b}{\mu_w})^{0.14} \tag{15}$$

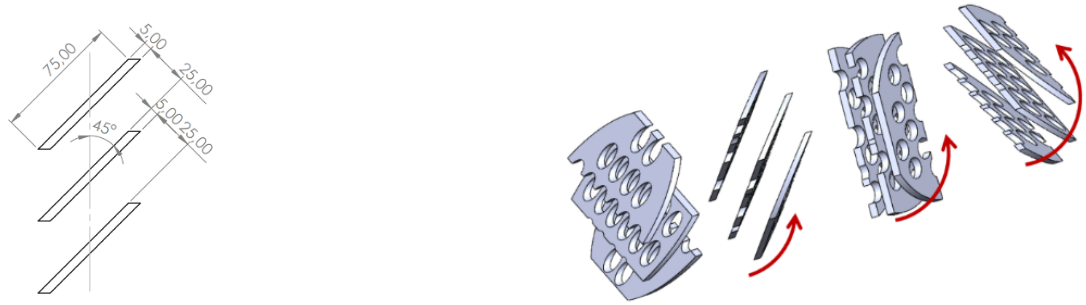
For $2x103 < Res < 1x106$

$$h_s = h_o J_c J_i J_s J_b J_r \tag{16}$$

To ensure accuracy, we utilized the Bell-Delaware method, which integrates flow loss values into the heat transfer calculation. Unlike simpler models, this approach accounts for the intricate leakages and flow variations within the heat exchanger, providing a more robust prediction for the shell side [14]. The method employs a set of correction factors Jc, Ji, Js, Jb, and Jr, which act as enhancement multipliers for the shell-side coefficient [15]. These variables address specific geometric impacts: Jc reflects the baffle window effect, while Ji accounts for leakage across baffle interfaces. Both Jr and Jb address bypass flows near the shell wall and partition passes, while Js corrects for variations in inlet and outlet baffle spacing [16].

The overall heat transfer coefficient on the shell side, U, can be expressed as

$$U = \frac{1}{(\frac{d_o}{d_i}\frac{1}{h_t}) + (\frac{d_o ln(d_o/d_i)}{2k}) + (\frac{1}{h_s})} \tag{17}$$



(a) (b)

Figure 2. (a) Studied geometry (b)Swivel angle every baffle arrangement

Here, di refers to the tube's internal diameter, k to the wall's thermal conductivity, and hs and ht to the heat transfer coefficients for the shell and tube sides.

Using these parameters, we compared the simulation outputs with our mathematical baseline. This deviation analysis supported by the visual breakdown in Figure 2, helps us interpret the consistency of the results and identify underlying behavior patterns in the numerical model. Furthermore, by applying sensitivity analysis, we were able to pinpoint which input parameters most significantly impact the results. The final validation is compelling: Figure 3 shows an average deviation of just 3.51%, and 4.01% for pressure drop. Since these values are within a reasonable range, they serve as strong justification for the accuracy and reliability of our simulation.

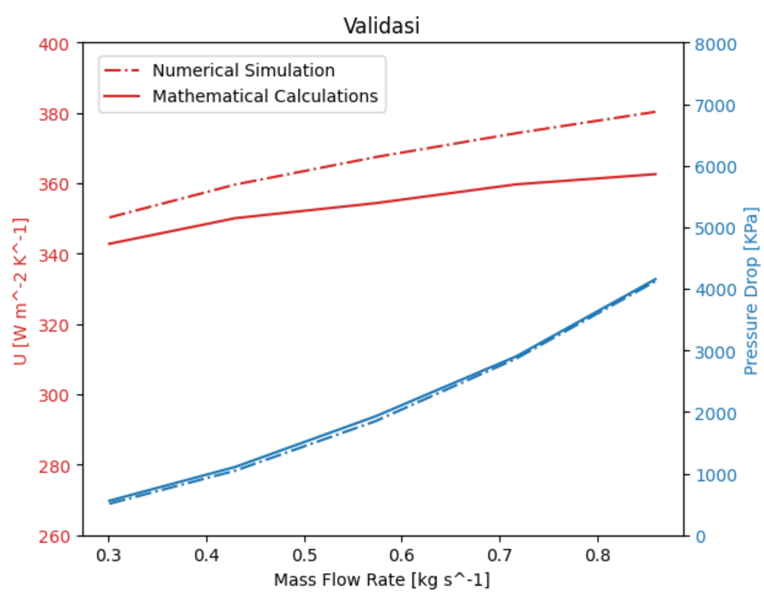


Figure 3. Validation between numerical simulation and experimental

2.5 Characterization Selection

Defining the overall performance of a heat exchanger requires a dual focus on heat transfer and pressure drop. This balance is best captured by the U/PD ratio, a dimensionless factor extensively used to quantify efficiency. This metric enables us to target a greater heat transfer rate for a given pressure drop, making it an ideal candidate for optimization. Therefore, we have adopted U/PD as the governing output characteristic, with U factors defined as [17]:

$$\varepsilon = \frac{c_c(T_{c2} - T_{c2})}{c_{min}(T_{h1} - T_{c1})} \tag{18}$$

$$Q = \varepsilon(\dot{m}c_p)_{min}(T_{h1} - T_{c1}) \tag{19}$$

$$U = \frac{Q}{AF\Delta T_{lmtd}} \tag{20}$$

Simultaneously, the numerical simulation yields the pressure drop data. This value serves as the denominator for the U value calculation.

3 Result and Discussion

Simulation results highlight a defining characteristic of Shell and Tube Heat Exchangers (STHeX) equipped with trapezoidal baffles: the generation of a distinct 'torsional flow.' Unlike the predictable patterns seen in conventional designs, this unique mechanism forces the fluid to move in a spiraling, twisting path through the shell. This rotational movement is far from passive; it significantly amplifies turbulent mixing across the tube bundle, ensuring more active contact with the heat transfer surfaces. As the fluid swirls, it effectively sweeps away the stagnant 'dead zones' that typically hinder performance in the baffle corners. Consequently, this continuous disruption of the thermal boundary layer leads to a marked improvement in heat transfer rates. Ultimately, the induction of torsional flow represents a strategic design advantage, maximizing thermal efficiency without disproportionately increasing pressure penalties. The velocity distribution within the shell reveals that fluid reaches its peak speed as it squeezes through the narrow gaps between the baffles and the shell wall. Conversely, the lowest velocities are found tucked behind the baffle plates, signaling the presence of stagnation zones, though it is worth noting that these zones are considerably smaller than those found in conventional segmental designs. This behavior is clearly visualized in Figure 6.

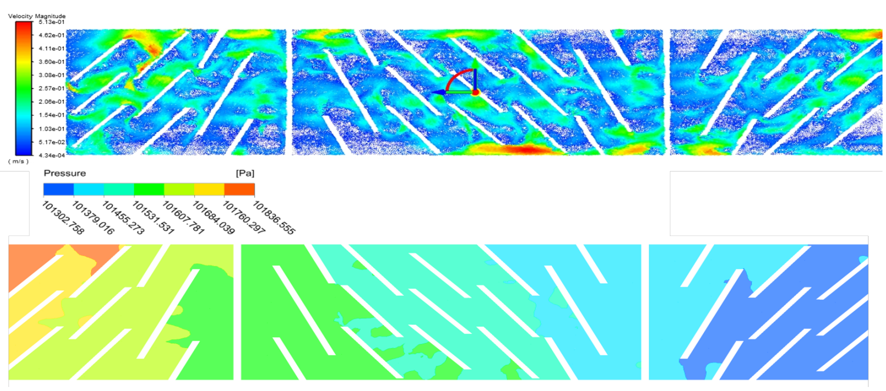


Figure 4. SHTeX longitudinal section on fluid velocity vector and fluid pressure on the shell

Our simulations reveal significant velocity gradients that trigger the formation of numerous eddies throughout the shell, particularly in the wake of the baffle plates. These vortices are highly advantageous, as they vigorously enhance fluid mixing and drive more frequent contact between the water and the tube surfaces. This intensified interaction ultimately boosts the convective heat transfer coefficient, transforming what would be simple flow into a highly efficient thermal exchange.

Our experimental data comparing the pressure drop in Shell and Tube Heat Exchangers (STHeX) reveals a strong linear correlation between mass flow rate and hydraulic resistance for both the segmental and trapezoidal baffle designs (Figure 6). Across the tested range of 0.4 to 1.4 kg s^{-1}, the segmental design consistently exhibits a significantly higher pressure drop than its trapezoidal counterpart. This disparity is clearly reflected in their respective regression equations:

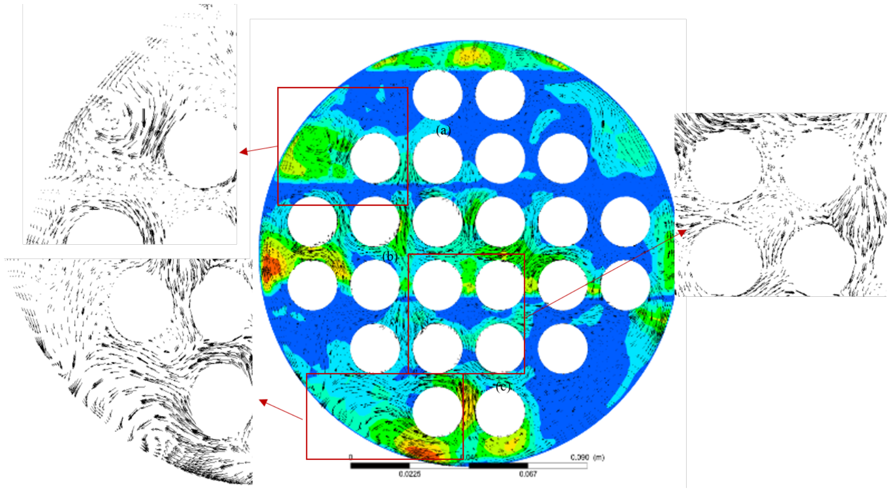


Figure 5. Velocity vector on the surface of a trapezoidal baffle

the segmental type follows y = 3854.956x - 1538.701, while the trapezoidal type maintains a much flatter trajectory at y = 2955.077x - 1157.647. Essentially, the segmental design suffers from a much steeper rise in pressure loss as the flow rate increases. Further analysis highlights the clear hydraulic superiority of the trapezoidal baffle in minimizing flow resistance. At the peak mass flow rate of approximately 1.4 kg s^{-1}, the pressure drop in the segmental design surges past 4000 Pa, whereas the trapezoidal design remains comfortably below 3200 Pa. This confirms that the trapezoidal geometry is far more effective at reducing energy losses caused by friction and shell-side turbulence compared to conventional segmental baffles. In an industrial context, this significant reduction in pressure drop is vital, as it translates directly into a lower pump workload and reduced overall energy consumption for the heat exchanger system.

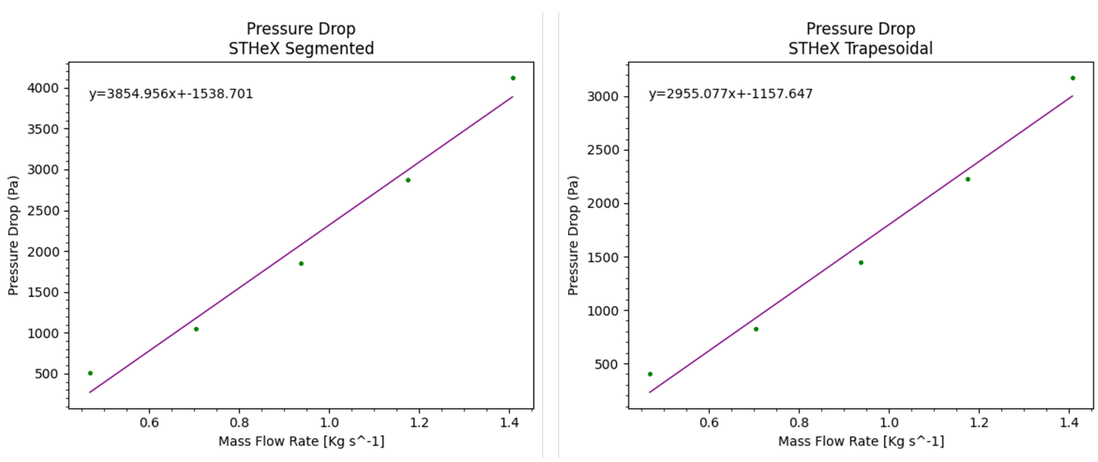


Figure 6. Comparative pressure drop analysis across different STHeX configurations.

Our analysis of the overall heat transfer coefficient (U) reveals a striking difference in thermal performance between the segmental and trapezoidal baffle designs (Figure 7). While both configurations show a steady rise in U as the mass flow rate climbs from 0.4 to 1.4 kg s^{-1}, the trapezoidal design consistently outperforms its segmental counterpart across the entire testing range. This thermal edge is mathematically underscored by the linear regression equations: the trapezoidal design boasts a steeper gradient (y = 36.351x + 352.169) compared to the segmental model (y = 31.825x + 336.528), signaling superior heat transfer effectiveness, particularly at higher flow rates. The performance gap becomes most apparent at the final operational set point. Here (Figure 7), the U value for the trapezoidal design surges past 400 W $m^{-2}K^{-1}$, while the segmental design trails behind at approximately 380 W $m^{-2}K^{-1}$. This measurable improvement proves that trapezoidal geometry excels at inducing turbulence and eliminating stagnant "dead zones" on the shell side, leading to a much more optimized exchange process. Ultimately, adopting trapezoidal baffles offers a highly efficient path for boosting heat transfer capacity while keeping operational energy consumption firmly under control.

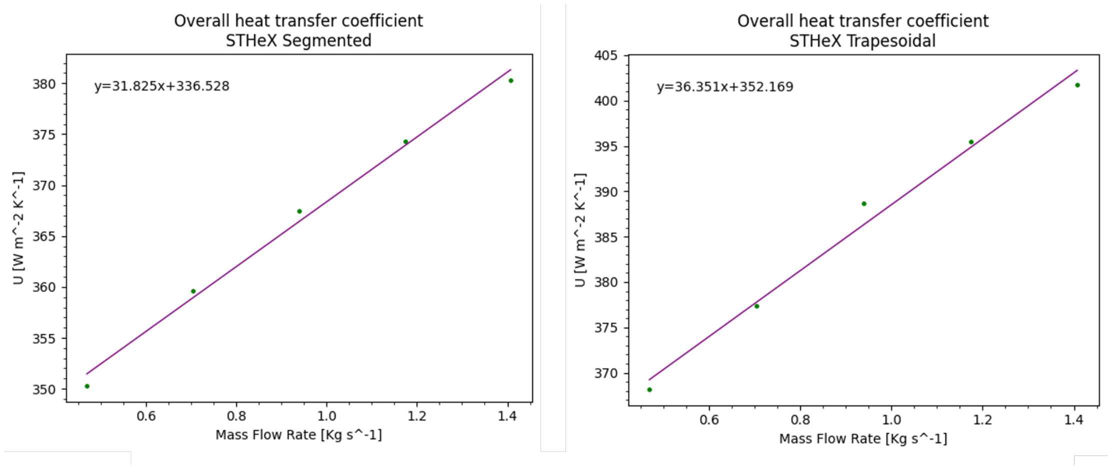


Figure 7. Comparative U analysis across different STHeX configurations.

A comparative analysis of the thermo-hydraulic efficiency (U/PD) across different Shell and Tube Heat Exchanger (STHeX) designs reveals a clear and consistent performance advantage for the trapezoidal baffle configuration (Figure 8). As we scale the mass flow rate from 0.3 to 0.85 kg s^{-1}, both baffle types exhibit an exponential decline in their U/PD values. This trend highlights a critical engineering trade-off: while higher flow rates do boost heat transfer, the resulting surge in pressure drop is much more aggressive, ultimately pulling down the system's overall efficiency ratio. The data confirms that the trapezoidal STHeX maintains a superior U/PD profile throughout the entire operational range. At the lower end of the spectrum (0.3 kg s^1), the trapezoidal design hits a peak efficiency of 0.9 W $m^{-2}K^{-1}Pa^{-1}$, comfortably outperforming the segmental design, which lingers around 0.7 W $m^{-2}K^{-1}Pa^{-1}$. This competitive edge persists even at the highest flow rates, proving that trapezoidal geometry is exceptionally effective at optimizing the overall heat transfer coefficient (U) without incurring the heavy pressure drop (PD) penalties seen in conventional designs. Ultimately, choosing trapezoidal baffles emerges as a much more sophisticated design strategy, one that elevates thermal performance while keeping pump energy consumption remarkably lean.

4 Conclusion

This study confirms that the implementation of trapezoidal baffles in a Shell and Tube Heat Exchanger (STHeX) offers a dual advantage over conventional segmental designs, excelling in both thermal and hydraulic performance. From a hydraulic standpoint, the trapezoidal configuration significantly mitigates pressure drop, exhibiting a much shallower increase rate (y = 2955.077x - 1157.647) than its segmental counterpart. Thermally, the trapezoidal geometry consistently yields a higher overall heat transfer coefficient (U), surpassing 400 W $m^{-2}K^{-1}$ at peak flow rates. The synergy of these parameters is best captured by the U/PD ratio, which identifies the trapezoidal design as the most efficient solution; it maximizes heat transfer while simultaneously reducing the energy penalties associated with flow resistance. Consequently, transitioning to trape Heat exchanger design handbook zoidal baffle geometries represents a viable and superior design strategy for developing high-performance, energy-efficient heat exchangers in industrial applications.

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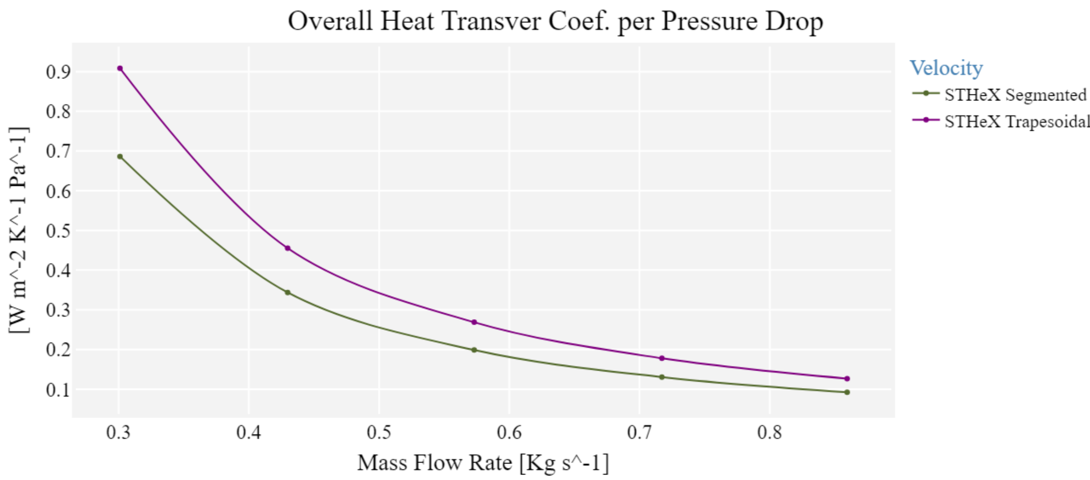

Figure 8. Comparative thermo-hydraulic efficiency (U/PD) for each STHeX design

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