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Optimisation of an industrial high-temperature thermal shell-and-tube heat exchanger: a petrochemical case study

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Ntunde, D. I[1] and Akpan, P. A[2]
[1,2]Department of Mechanical Engineering, Michael Okpara University of Agriculture, Umudike, Abia State, Nigeria.
Correspondence email: ntunde.dilibe@mouau.edu.ng

Abstract: This study presents computational fluid dynamics (CFD) evaluation of Syltherm 800 and ethylene glycol as shell-side working fluids in a shell-and-tube heat exchanger (STHE) used for feedstock preheating in a petrochemical facility. The major objective was to enhance thermal efficiency while minimizing pressure drop and pumping power in energy-intensive operations at Indorama Eleme Petrochemicals Company Limited, Nigeria as a case study. A three-dimensional steady-state CFD model was developed in COMSOL multiphysics using the Navier–Stokes and energy equations under turbulent flow conditions. Ethylene glycol at 120°C was used as the baseline case and compared with Syltherm 800 at 120°C, 300°C, and 350°C. Results showed that Syltherm 800 significantly improved the thermal performance at elevated temperatures; while ethylene glycol at 120°C produced a total heat flux of approximately 9.1kW with an inlet pressure of 3,197Pa, Syltherm 800 at 350°C achieved a heat flux of 30.996kW and reduced inlet pressure to 1,676Pa. Further, the tube-side outlet water temperature increased from 35.28°C (ethylene glycol) to 44.03°C (Syltherm 800 at 350°C). The findings revealed that Syltherm 800 enhances heat transfer while reducing hydraulic resistance. The research work established the required fluid material to be used for optimal performance of the case studied industrial heat exchangers for high-temperature petrochemical applications.

Keywords: Syltherm 800, Pressure Drop, High Temperature, Ethylene Glycol, Petrochemical

1. Introduction

The Shell-and-tube heat exchangers (STHEs) are among the most ubiquitous and critical components in process industries, particularly in large-scale petrochemical facilities, oil refineries, power plants, and chemical reactors. Their widespread adoption stems from their mechanical robustness, straightforward design, ability to withstand high pressures and temperatures, and adaptability to various flow configurations (Zhu, 2016; Li, Wang, and Wang, 2017). In petrochemical complexes such as Indorama Eleme Petrochemicals Company Limited one of the leading feedstock producers in Port Harcourt, Nigeria heat exchangers are essential for preheating crude streams, waste heat recovery, and conditioning process fluids for downstream reactions. In continuous operations where energy cost and thermal efficiency directly influence profitability, even modest improvements in exchanger performance can yield substantial operational savings (Alomair and Al-Hamidi, 2021; Patel and Shah, 2022). A typical STHE operates by conveying a hot fluid through the shell and a cold

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fluid through the tubes, enabling heat exchange through the solid tube wall. The effectiveness of this heat transfer process is governed by the temperature differential between the fluids, the convective heat transfer coefficients on both sides, and the geometric configuration of the exchanger, including baffle spacing and tube layout (Kakaç and Liu, 2002; Ahmed and Ghojel, 2021). For shell-side flow, the formation of secondary flows and turbulence induced by baffles significantly enhances mixing and accelerates heat transfer, provided the fluid properties and flow regime support such behaviour (Yang, Luo, and Li, 2020).

Despite their widespread use, STHEs in industrial settings often operate below their theoretical performance potential. In the case study plant, ethylene glycol has historically served as the shell-side working fluid due to its favourable freeze protection, chemical compatibility, and extensive industrial availability. However, ethylene glycol exhibits notable limitations when used in high-temperature processes. At temperatures above about 150-200°C, ethylene glycol's thermal stability declines, and its dynamic viscosity remains relatively high compared to advanced heat transfer fluids, resulting in increased frictional resistance, higher pressure drop, and reduced convective heat transfer (Shah and Sekulić, 2003; Karagiannis et al., 2010). These properties not only limit the thermal driving force but also increase a pumping power requirement, which in large facilities directly translates to elevated energy costs and maintenance burdens (Gandomi and Shirazi, 2017). The observed performance limitations of the existing STHE system at the case study plant manifested in two primary ways: sub-optimal temperature rise in the tube-side fluid and comparatively high pressure drop across the shell side. Under the baseline operational condition with ethylene glycol at 120℃, simulation and field data indicated relatively modest wall heat flux values (=9.1kW) and a tube outlet temperature of 35.28°C. Concurrently, the shell-side pressure drop was measured at approximately 3,197Pa, indicating significant frictional resistance. These findings highlighted a dual challenge that is common in industrial heat exchangers: achieving a balance between thermal performance and hydraulic efficiency (Wang, Xu, and Tao, 2019; Cao et al., 2021). Consequently, there arose a clear need for optimisation not through costly redesign of the heat exchanger geometry, but through intelligent selection and evaluation of working fluids capable of enhancing thermal performance while reducing hydraulic penalties.

A promising solution involves the introduction of high-temperature synthetic thermal fluids, such as Syltherm 800. Syltherm 800 is silicone-based thermal oil with a high flash point, broad operating temperature range, and relatively low viscosity, even at elevated temperatures (Dow Chemical Technical Data, 2023). These characteristics make it well suited for applications requiring extended thermal stability and robust heat transfer performance. In contrast to ethylene glycol, Syltherm 800 maintains thermal stability at temperatures well above 300°C, enabling larger temperature differentials across the heat exchanger without approaching decomposition limits. Lower viscosity at higher temperatures also translates to reduced friction factors, which in turn reduces shell-side pressure drop and associated pumping energy consumption (Farid et al., 2018; Pandey and Verma, 2020). The expected behaviour of the improved system when using Syltherm 800 can be anticipated from classical heat transfer principles. First, an increase in the shell-side fluid temperature directly increases the log-mean temperature difference, thus raising the overall heat transfer rate according to (Incropera and DeWitt, 2006; Çengel and Ghajar, 2020). Secondly, a reduction in fluid viscosity with increasing temperature promotes higher Reynolds numbers and enhances convective mixing, which increases local Nusselt number and reduces thermal boundary layer thickness. The combined effect gives an improved convective heat transfer coefficient, on the shell side, further contributing to increased overall heat transfer. (Lienhard and Lienhard, 2019; Bejan and Kraus, 2003).

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Despite the clear thermo-hydraulic benefits anticipated with Syltherm 800, only a limited number of studies have conducted direct, controlled comparisons between conventional glycol fluids and high-temperature thermal oils using high-fidelity CFD under identical geometric and boundary conditions. Previous works have focused on geometric enhancements, baffle optimisation, or nano-fluid applications to improve exchanger performance (Bahiraei and Monavari, 2021; Khosravi and Mohseni, 2021), but a fluid-centric optimisation approach remains relatively under-explored in the literature. This study therefore provides a comprehensive CFD-based analysis comparing the thermal and hydraulic performance of ethylene glycol and Syltherm 800 under conditions representative of industrial operation. By maintaining identical shell-and-tube geometry and flow constraints, differences in performance can be directly attributed to the working fluid and operating temperature. The objectives of this research are to quantify the heat transfer enhancement achievable through high-temperature operation, evaluate the impact of fluid property variation on pressure drop, and establish a performance ranking that supports fluid selection and design optimisation in industrial heat exchangers. The findings of this analysis are expected to guide engineering decisions in petrochemical facilities, enabling improved energy efficiency, reduced operational cost, and enhanced process reliability without the need for costly mechanical retrofits. In doing so, the study contributes both to academic understanding and practical process optimisation in energy-intensive industries.

1.1 Research Problem

In petrochemical operations such as Indorama Eleme Petrochemicals in Port Harcourt, heat exchanger performance directly affects energy consumption, process continuity and operating costs. Excessive pressure drop and non-uniform flow reduce process efficiency and may force premature cleanings or equipment replacement. Although a 100% efficiency machine is said not to be possible, but optimization of the system can improve on its efficiency by 5%. This study seeks to identify areas that need improvement for optimal efficiency through CFD model analysis and optimization of the reference shell and tube heat exchanger. A single shell and a single tube passing with seamless tubing will be used. Descriptively, the flow pattern will be parallel.

1.2 Purpose of the Study

The purpose of this research is to carry out analysis and optimisation of the thermal and hydraulic performance of an industrial shell-and-tube heat exchanger operating under high-temperature conditions using computational fluid dynamics (CFD) taking Indorama Eleme Petrochemical as a case study.

1.3 Research Questions

- How operating parameters such as temperature, flow rate and pressure affect the performance of the shell-and-tube heat exchanger?
- What is the relationship between heat transfer enhancement and pressure drop in the optimized heat exchanger system?
- How can optimization techniques improve the heat transfer rate and overall thermal efficiency of the heat exchanger?

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1.4 Research Hypotheses

H_{o1}: Optimisation of the shell-and-tube heat exchanger significantly improves the thermal performance of the heat exchanger in the petrochemical process.

H_{o2}: Optimisation techniques significantly reduce pressure drop within the shell-and-tube heat exchanger.

2. Research Methods

2.1. Flow Properties of the Introduced Thermal Fluid

To overcome the thermal and hydraulic limitations observed with ethylene glycol, Syltherm 800 was introduced as the shell-side working fluid in this case study. Syltherm 800 is a silicone-based synthetic thermal fluid designed for high-temperature industrial heat transfer applications and is characterized by excellent thermal stability over a wide operating range. This property allows the fluid to operate safely at elevated temperatures where glycol-based fluids are no longer suitable. In this study, Syltherm 800 was examined at inlet temperatures of 120°C, 300°C, and 350°C to evaluate the combined influence of fluid properties and temperature on exchanger performance. The fluid has a relatively low density of approximately 936 kg/m³ and, more importantly, a significantly lower dynamic viscosity than ethylene glycol. As the operating temperature increases, the viscosity of Syltherm 800 decreases markedly, leading to higher Reynolds numbers and improved turbulence within the shell. This enhanced flow behaviour strengthens convective heat transfer and improves heat distribution across the tube bundle.



Plate 1: Picture of shell and tube heat exchanger

(*Source: Indorama Eleme Petrochemicals, Port Harcourt*)

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Table 1: Key Thermo-physical Properties at 25°C

Parameter	Ethylene Glycol (Typical)	Syltherm 800 (Typical)
Dynamic Viscosity @25°C	16.1×10^{-3} Pa·s	9.1×10^{-3} Pa·s
Flash Point (Closed Cup)	111 °C	160 °C
Flash Point (Open Cup)	155 °C	177 °C
Autoignition Temperature	398 °C	385 °C
Freeze Point	−12.9 °C	−60 °C
Density @25°C	1113 kg/m³	936 kg/m³
Specific Heat Capacity @25°C	2420 J/kg·K	2300 J/kg·K
Specific Gravity @25°C	1.11	0.93
Heat of Combustion	28,659 kJ/kg	28,659 kJ/kg
Estimated Critical Temperature	445 °C	367 °C
Estimated Critical Pressure	10.9 bar	10.9 bar
Estimated Critical Volume	1.8 L/kg	3.22 L/kg

Sources: National Institute of Standards and Technology. (2023), Dow Chemical Company (2023), Perry & Green (2019)

2.2 Model Development

The thermo-hydraulic behaviour of the shell-and-tube heat exchanger was modelled using a three-dimensional steady-state computational fluid dynamics (CFD) framework in COMSOL multi-physics. The model captures the coupled interaction between fluid flow and heat transfer within the shell domain Syltherm 800 fluid material and tube water domain that included the conjugate heat transfer across the tube wall. The developed models and simulation performed were done under a steady-state incompressible flow condition of the system that experienced no phase change of the fluid at a turbulent flow regime. The Governing principles of the developed model of the system works under the fundamental conservation equations for mass, momentum, and energy, where the velocity vector of the continuity equation for the incompressible flow was expressed as in equation (1):

$$\nabla \cdot u = 0 \tag{1}$$

The momentum equation that governs the fluid motion and determines the velocity and pressure distribution within the shell and tube domains as expressed in equation (2) was factored into the computational analysis, where ρ, p, μ and F are the respective density, pressure, dynamic viscosity and body force flow parameters of the fluid.

$$\rho(u \cdot \nabla)u = -\nabla p + \mu\nabla^2 u + F \tag{2}$$

The energy equation as expressed in equation (3) was to describe by analysis the heat of convection and conduction within the fluid domains as:

$$\rho C_p(u \cdot \nabla T) = k\nabla^2 T \tag{3}$$

Where C_p , k and T and are the specific heat capacity, thermal conductivity and temperature respectively. Given the operating Reynolds numbers in industrial heat exchangers, the flow was assumed turbulent. Hence; the realizable k–ε turbulence model was employed due to its robustness in predicting recirculation zones and industrial shell-side flows. The transport equations for turbulent

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kinetic energy k and its dissipation rate ε expressed in equation (4) and equation (5) was adopted too in the study for turbulence modelling:

$$\rho(u \cdot \nabla k) = \nabla \cdot [(\mu + \frac{\mu_t}{\sigma_k})\nabla k] + G_k - \rho\epsilon \quad (4)$$

$$\rho(u \cdot \nabla \epsilon) = \nabla \cdot [(\mu + \frac{\mu_t}{\sigma_\epsilon})\nabla \epsilon] + C_1 \frac{\epsilon}{k} G_k - C_2 \rho \frac{\epsilon^2}{k} \quad (5)$$

Where G_k is production of turbulent kinetic energy, μ_t is turbulent viscosity, C_1, C_2, σ_k, and σ_ϵ are model constants. The turbulence model was developed to predict the shell-side mixing and boundary layer behaviour that was directly influencing heat transfer coefficient. Thus, the turbulent viscosity as expressed in equation (6) was used:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (6)$$

The Heat transfer between shell fluid and tube fluid occurred through the tube wall. This ensured energy conservation across the boundary. Thus, conduction equation in the solid tube wall was computed as in equation (7):

$$\nabla \cdot (k_s \nabla T) = 0 \quad (7)$$

Where k_s is the thermal conductivity of tube material and k_f thermal conductivity of fluid material while ∂T is temperature difference over n areas across the system as captured in equation (8). Thus, the Continuity of heat flux at the fluid-solid interface as expressed below in equation (8):

$$-k_f \frac{\partial T}{\partial n} = -k_s \frac{\partial T}{\partial n} \quad (8)$$

The expression to analyse the exchanger performance were computed in equation (9), where $\dot{m}$ T_{out} and T_{in} are the mass flow rate inlet and outlet temperatures respectively.

$$Q = \dot{m} C_p (T_{out} - T_{in}) \quad (9)$$

Furthermore, the Log Mean Temperature Difference (LMTD) for the counter flow configuration of the studied system was analysed using equation (10).

$$\Delta Tlm = \frac{(Th,in - Tc,out) - (Th,out - Tc,in)}{ln\frac{(Th,in - Tc,out)}{(Th,out - Tc,in)}} \quad (10)$$

Consequently, the Overall Heat Transfer Coefficient and Pressure Drop across the shell were obtained using equation (11) and equation (12)

$$U = \frac{Q}{A\Delta Tlm} \quad (11)$$

$$\Delta P = Pin - Pout \quad (12)$$

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Where A, Pin and $Pout$ are the heat transfer area, inlet and out pressure of the shell of the heat exchanger.

2.3. Mesh and Numerical Stability

The computational domain was discretized using physics-controlled fine mesh. A mesh independence study was conducted by refining the mesh until variations in heat flux were less than 2%. Convergence criteria:

- Residual tolerance: 10^{-6}
- Energy balance error < 1%

2.4 Model Simulation and Validation

The CFD simulation results of the developed flow models using Syltherm 800 fluid material will present a comparative analysis of the heat transfer rate, outlet temperature and pressure drop with the operating data of ethylene glycol fluid presently used in the case studied system.

3. Results

Table 2 presents the comparative thermal performance of the shell-and-tube heat exchanger using ethylene glycol and Syltherm 800 under different shell-side operating temperatures. Under the baseline case of ethylene glycol at 120 °C, the total heat flux was 9,100 W, while the tube-side outlet temperature reached 35.28 °C. This result indicates moderate thermal transfer under conventional plant operating conditions. When Syltherm 800 was introduced at the same temperature of 120 °C, the heat flux increased to 15,198 W and the tube outlet temperature rose to 36.42 °C. This represents a clear increase in energy transfer despite identical temperature input conditions. At 300 °C, Syltherm 800 further increased heat flux to 27,115 W with a tube outlet temperature of 37.03 °C, showing substantial thermal improvement. The highest result was recorded at 350 °C, where Syltherm 800 produced 30,996 W heat flux and increased the tube outlet temperature to 44.03 °C. This condition delivered the maximum thermal energy transfer among all evaluated cases. Overall, the results show a progressive increase in heat transfer as Syltherm 800 operating temperature increased.

Table 2: Heat Transfer Performance for Different Working Fluids

Case	Heat Flux (W)	Tube Outlet Temp (°C)
Ethylene Glycol (120°C)	9,100	35.28
Syltherm 800 (120°C)	15,198	36.42
Syltherm 800 (300°C)	27,115	37.03
Syltherm 800 (350°C)	30,996	44.03

These results indicate that working fluid substitution and temperature elevation significantly enhanced heat exchanger thermal performance. The increase from 9,100 W to 30,996 W demonstrates that Syltherm 800, particularly at elevated temperatures, provided stronger thermal driving force and superior heat recovery capacity than ethylene

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Table 3: Thermal properties of heat exchanger of syltherm 800 temperature at 350°C

Description	Value	units
Inlet pressure, syltherm 800	1676.2	Pa
Outlet pressure, syltherm 800	-0.55478	Pa
Inlet pressure, water	72.816	Pa
Outlet pressure, water	-0.59858	Pa
Inlet temperature water	25	degC
Outlet temperature water	44.033	degC
Heat transfer coefficient for syltherm 800	281.91	W/m^2K
Inlet temperature syltherm 800	350	degC
Outlet temperature syltherm 800	332.18	degC
Normal total heat flux (W)	2654.4	W
Total heat flux magnitude (W)	10884	W
Total heat flux magnitude water domain	20487	W
Normal total heat flux exteroir syltherm domain	-84856	W
Normal total heat flux exteroir water domain	6503.9	W
Total heat flux magnitude syltherm domain wall	30996	W
Temperature gradient magnitude water domain wall	15327	W.K

Table 3 presents the thermal properties of the heat exchanger when Syltherm 800 was used as the shell-side fluid at an inlet temperature of 350°C. The inlet pressure of Syltherm 800 is 1676.2Pa, while the outlet pressure approaches zero (-0.55Pa), indicating a moderate pressure drop across the shell side. This relatively low pressure drop reflects favourable flow characteristics and reduced resistance, which is expected at high operating temperatures where the viscosity of Syltherm 800 is significantly reduced. On the tube side, water enters at 25°C and exits at 44.03°C, representing a temperature rise of approximately 19°C. This substantial increase confirms strong thermal interaction between the shell-side Syltherm 800 and the tube-side water. Correspondingly, Syltherm 800 experienced a temperature drop from 350°C to 332.18°C, indicating effective heat release to the water. The shell-side heat transfer coefficient is 281.91W/m²K, which, although lower than that of ethylene glycol, is compensated for by the large temperature difference between the two fluids. This is further supported by the normal total heat flux of 2654.4W and a total heat flux magnitude of 10,884 W. The high wall heat flux magnitude of 30,996W confirmed intense heat transfer at the tube-fluid interface. In addition, the temperature gradient magnitude at the water-domain wall (15,327W/K) indicated a steep thermal gradient across the tube wall, which is a key driver of heat transfer. Overall, these results confirmed that Syltherm 800 at 350°C provides excellent thermal performance with manageable hydraulic losses. According to Fourier's law of heat conduction, a higher temperature gradient directly increases conductive heat transfer across the tube wall, hence; the enhanced turbulence at elevated temperature increases the convective heat transfer coefficient,

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creating a combined effect that maximizes overall heat exchanger performance. From an industrial perspective, these results have direct implications for petrochemical operations such as those used in the case study. The CFD simulation revealed that a higher outlet temperature achieved with the proposed Syltherm 800 at 350°C; revealed an improved thermal recovery and reduced auxiliary heating across the case studied system. Enhanced energy transfer translates into improved process efficiency and potential reductions in operational energy costs. Additionally, the ability of Syltherm 800 to operate effectively at higher temperatures without significant degradation provides an operational advantage over ethylene glycol in high-temperature process environments. These findings align with earlier research demonstrating that higher temperature differentials enhance Nusselt number and heat transfer coefficient (Amarjit et al., 2011).

Table 4: Shell-Side Pressure Performance at different Input Temperature

Case	Inlet Pressure (Pa)	Observed Hydraulic Behaviour
Ethylene Glycol (120°C)	3,197	Higher resistance and steeper pressure gradient
Syltherm 800 (120°C)	2,850	Improved flow distribution
Syltherm 800 (300°C)	2,050	Reduced frictional losses
Syltherm 800 (350°C)	1,676	Smooth pressure variation and lowest resistance

Table 4 shows the shell-side pressure performance for all cases. Ethylene glycol at 120 °C recorded the highest inlet pressure of 3,197 Pa, indicating the greatest flow resistance. Syltherm 800 at 120 °C reduced inlet pressure to 2,850 Pa. At 300 °C, inlet pressure further reduced to 2,050 Pa, while the lowest pressure was observed at 350 °C with 1,676 Pa. This represents a 47.5% reduction in pressure drop relative to ethylene glycol.

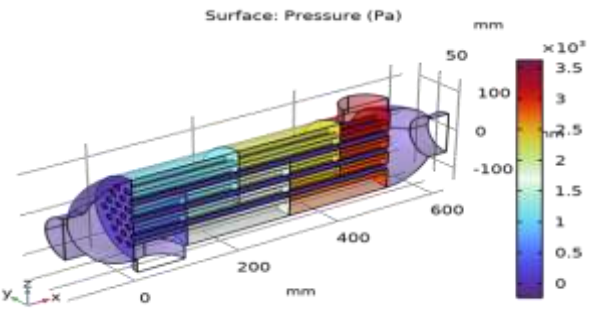


Fig 1: Tube and shell Surface of Pressure (Pa) for ethylene glycol temperature at 120°C

The CFD pressure contour plots for the ethylene glycol case in Fig. 1 confirmed this behaviour, showing more pronounced pressure accumulation near the inlet region and sharper pressure decline along the axial direction. When Syltherm 800 was introduced at the same temperature of 120°C, the inlet pressure reduced to approximately 2,850Pa. Although the temperature was identical, the lower viscosity of Syltherm 800 improved flow mobility within the shell.

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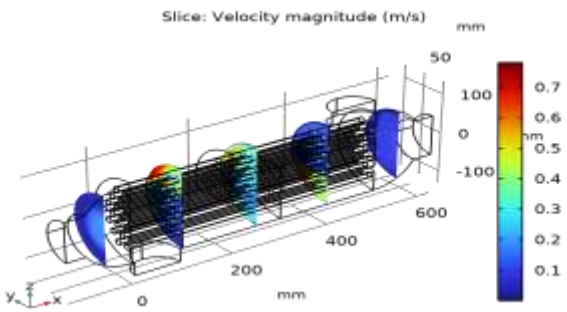


Fig 2: Velocity magnitude (m/s) of fluid for ethylene glycol temperature at 120^{o}C

The velocity contour for ethylene glycol at 120 °C as in Figure 2was dominated by blue and light-green zones, indicating low shell-side velocities generally below 0.4 m/s, with stagnation zones behind tube bundles. In contrast, Syltherm 800 at elevated temperatures showed stronger yellow-red zones with velocity values reaching approximately 1.1–1.3 m/s, especially around tube bundles. These results indicate higher turbulence intensity and improved flow mixing.

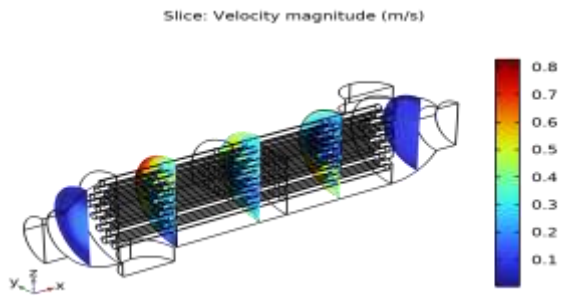


Fig 3: Velocity magnitude (m/s) of fluid for syltherm 800 temperature at 300^{o}C

The velocity magnitude contours for Syltherm 800 at 300 °C are represented using a colour scale ranging from dark blue (low velocity) to red (high velocity) as shown in fig. 5. From the colour legend in the velocity plot, the shell-side velocity varies approximately from below 0.1m/s in the blue regions to about 1.1-1.3m/s in the red regions. The predominance of yellow and red colour bands around the tube bundle indicates regions of high flow momentum and enhanced mixing. These zones corresponded to areas where Syltherm 800 was forced to accelerate as it flowed across the tubes, promoting turbulence and improving convective heat transfer. Only limited blue regions were observed downstream of the tubes, indicating minimal stagnation and effective shell-side flow distribution. This velocity behaviour is consistent with the reduced viscosity of Syltherm 800 at

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elevated temperature and confirms that the shell-side fluid flow is favourable for heat transfer enhancement.

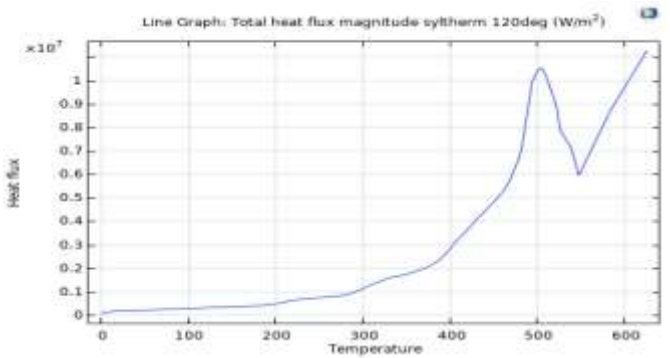


Fig 4: Heat flux against temperature graph for Syltherm 800 at 120degC

The heat flux versus temperature graph for Syltherm 800 at 120 ℃ showed a positive correlation between temperature and heat flux, indicating that heat transfer increased with rising temperature. Similarly, the heat flux versus axial position graph for ethylene glycol showed that heat flux was highest near the inlet region and gradually declined toward the outlet as temperature difference reduced.

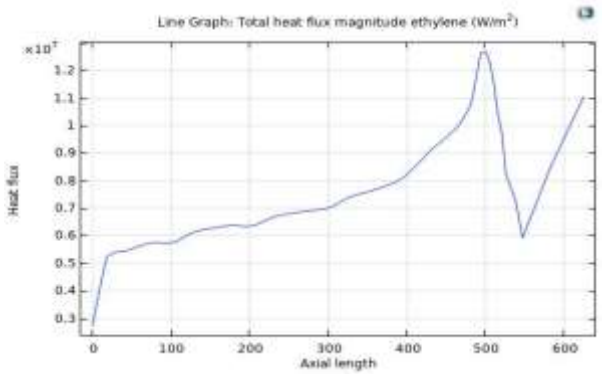


Fig 5: Heat flux against axial position graph for ethylene glycol at 120 degC

Similarly, the heat flux versus axial position graph in figure 5 indicates that heat transfer is not uniform along the tube length, with higher heat flux values observed near the inlet region and a gradual reduction toward the outlet as the temperature difference decreases.

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4. Discussion

The results obtained from this study provide a detailed thermo-hydraulic evaluation of shell-and-tube heat exchanger performance under four distinct operating conditions: ethylene glycol at 120 ℃ (baseline industrial case), Syltherm 800 at 120 °C, Syltherm 800 at 300 °C, and Syltherm 800 at 350 °C. The comparative analysis clearly demonstrates that both fluid selection and shell-side operating temperature exert major influence on thermal efficiency, pressure behaviour, velocity distribution, and overall system optimisation. Under the baseline case using ethylene glycol at 120 °C, the shell-and-tube heat exchanger produced a total heat flux of 9,100 W with a tube-side outlet temperature of 35.28 °C. Considering that the water inlet temperature was approximately 25 °C, this represents a temperature rise of about 10.28 °C across the tube domain. Although this confirms functional heat transfer, the thermal output remained the lowest among all tested cases. Ethylene glycol also recorded the highest shell-side inlet pressure of 3,197 Pa, indicating the greatest hydraulic resistance. The velocity contour further showed predominantly blue and light-green zones with shell-side velocities generally below 0.4 m/s, particularly behind tube bundles where recirculation and stagnation zones were observed. These low-velocity regions are important because they reduce turbulence intensity, lower Reynolds number, and weaken convective heat transfer. From a thermal-hydraulic perspective, this means that ethylene glycol, while operationally acceptable, imposed greater frictional resistance and lower thermal productivity under the studied geometry. This behaviour is strongly linked to its comparatively higher dynamic viscosity, which restricts shell-side flow mobility and increases wall shear stress. When Syltherm 800 was introduced at the same shell-side temperature of 120 ℃, significant improvements were immediately observed even without increasing thermal input. Heat flux increased from 9,100 W to 15,198 W, representing approximately 67% improvement over ethylene glycol. Simultaneously, tube outlet temperature increased from 35.28 ℃ to 36.42 ℃, confirming stronger energy transfer to the water domain.

This is particularly significant because the operating temperature remained unchanged, meaning the improvement was primarily due to superior thermophysical properties of Syltherm 800 rather than increased temperature gradient alone. The shell-side inlet pressure reduced from 3,197 Pa to 2,850 Pa, reflecting approximately 10.9% reduction in pressure loss. This demonstrates that Syltherm 800's lower viscosity improved flow uniformity, reduced internal friction, and enhanced turbulence. In practical terms, this result proves that fluid substitution alone can substantially optimise exchanger performance without any geometric redesign, making it a highly attractive industrial strategy. At Syltherm 800 operating temperature of 300 °C, the heat exchanger entered a significantly enhanced thermal regime. Total heat flux increased to 27,115 W, representing approximately 197.9% improvement over the ethylene glycol baseline and about 78.4% improvement over Syltherm 800 at 120 °C. This dramatic increase confirms that rising shell-side temperature substantially strengthens thermal driving force. The shell-side pressure simultaneously reduced to 2,050 Pa, indicating approximately 35.9% reduction relative to ethylene glycol. Velocity contours for this condition showed shell-side velocity values ranging from below 0.1 m/s in limited stagnant zones to approximately 1.1–1.3 m/s in high-flow regions around the tube bundle. The predominance of yellow and red colour bands indicated stronger momentum transfer, improved turbulence, and more effective shell-side mixing. These results confirm that elevated Syltherm 800 temperature improved both thermal energy exchange and hydraulic efficiency simultaneously, thereby overcoming the common engineering trade-off where higher heat transfer often causes excessive pressure penalties.

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The highest-performing case was Syltherm 800 at 350 °C, which delivered the optimum thermo-hydraulic condition. At this operating state, total wall heat flux magnitude reached 30,996 W, while total heat flux magnitude was 10,884 W and water-domain heat flux magnitude reached 20,487 W. Compared to ethylene glycol, this represents approximately 240.6% improvement in heat transfer capacity. Tube-side outlet temperature increased to 44.03 °C, representing a 19.03 °C rise above water inlet and an 8.75 °C increase beyond the ethylene glycol case. This substantial increase confirms far more effective thermal recovery. Simultaneously, shell-side inlet pressure dropped to 1,676.2 Pa, corresponding to approximately 47.5% reduction in pressure drop compared to the baseline case. This dual benefit is highly significant because it indicates that the system achieved maximum thermal enhancement while also minimizing pumping power requirements. Further insight is provided by the detailed thermal parameters at 350 °C. Syltherm 800 cooled from 350 °C to 332.18 °C, meaning approximately 17.82 °C of thermal energy was successfully transferred across the exchanger. The shell-side heat transfer coefficient was 281.91 W/m²K, and although this coefficient alone may appear moderate, the very large temperature difference compensated strongly, resulting in exceptional total heat transfer. The temperature gradient magnitude at the water-domain wall reached 15,327 W/K, confirming an intense conductive gradient through the tube wall according to Fourier's law. The normal total heat flux exterior Syltherm domain was −84,856 W, indicating substantial outward heat release from the shell-side fluid, while the total heat flux magnitude at the Syltherm wall reached the highest recorded value of 30,996 W.

These values collectively confirm that the dominant optimisation mechanism was the combined effect of elevated thermal gradient and reduced viscosity-induced hydraulic resistance. The heat flux versus temperature graph for Syltherm 800 further reinforced this interpretation by showing a progressive increase in heat flux with temperature, particularly at higher temperature zones where the curve steepened. This suggests that as thermal energy increased, both convective and conductive heat transfer intensified. Similarly, the axial heat flux profile for ethylene glycol revealed strong inlet-region heat transfer followed by gradual decline toward the outlet due to thermal depletion. This demonstrates that ethylene glycol's lower thermal reserve and higher resistance restricted sustained energy delivery along the exchanger length. From an industrial case-study perspective, these findings are highly relevant to petrochemical systems such as Indorama Eleme Petrochemicals Company Limited, where feedstock preheating and thermal recovery are critical. Replacing ethylene glycol with Syltherm 800, particularly at elevated temperatures, could significantly improve process heat recovery, reduce auxiliary fuel demand, lower circulation energy costs, and improve thermal system stability. The reduction from 3,197 Pa to 1,676 Pa implies lower pumping power and reduced mechanical stress on pumps, seals, and piping networks. Simultaneously, the increase from 9,100 W to 30,996 W demonstrates a major gain in energy utilisation efficiency.

Overall, the detailed values confirm that this study successfully optimized shell-and-tube heat exchanger performance by identifying Syltherm 800 at 350 °C as the most effective operating condition. The optimisation was not geometric but thermo-hydraulic, achieved through advanced fluid selection and operating temperature control. This contribution is particularly important because it demonstrates that industrial heat exchanger performance can be substantially improved through fluid engineering alone, providing a practical pathway for enhanced energy efficiency in high-temperature petrochemical operations.

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5. Conclusion

The results presented in Table 2 clearly validate the objective of thermal optimisation established in this study. A progressive improvement in heat transfer performance was observed as the working fluid changed from ethylene glycol to Syltherm 800 and as the operating temperature increased. While the baseline case using ethylene glycol at 120°C produced a total heat transfer rate of 9,100W, the use of Syltherm 800 at 350°C resulted in a heat flu2wex of 30,996W approximately a 240% increase. This substantial enhancement confirmed the strong influence of both fluid thermophysical properties and temperature driving force on heat exchanger effectiveness. The superior performance of Syltherm 800 is primarily attributed to several interacting factors. First, the elevated inlet temperature significantly increased the temperature gradient between the shell-side and tube-side fluids, thereby enhancing the logarithmic mean temperature difference and overall heat transfer rate. Second, the reduction in dynamic viscosity at higher temperatures improved flow mobility within the shell, leading to higher Reynolds numbers and stronger turbulent mixing. Increased turbulence enhances the convective heat transfer coefficient, which directly increases energy transfer across the tube wall. These combined thermal and hydrodynamic effects produced a substantial rise in heat flux without introducing excessive hydraulic penalties. The findings therefore confirmed that high-temperature silicone-based thermal fluids such as Syltherm 800 provide markedly superior thermo-hydraulic performance compared to conventional ethylene glycol in shell-and-tube heat exchangers. For high-temperature petrochemical applications, this improvement will translate into enhanced energy recovery, improved process efficiency, and more effective utilisation of thermal resources.

6. Recommendations

Syltherm 800 is recommended as a more effective heat transfer fluid for high-temperature shell-and-tube heat exchanger applications due to its superior thermal performance and reduced hydraulic resistance compared to ethylene glycol. Industries operating similar systems should consider fluid substitution and operating temperature optimization as practical strategies for improving exchanger efficiency without structural modification. However, experimental validation, economic feasibility assessment, and further investigation into geometric optimization and material compatibility are necessary before large-scale industrial implementation.

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