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Social Science Journal

Improving Heat Transfer Efficiency by Converting Cylindrical Tubes to Conical Forms in Double Pipe Heat Exchanger

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Abstract

Baffles are becoming more important in twin pipe heat exchangers because they can improve performance. In order to compare designs this study meticulously analyzes the operational efficacy of double-pipe heat exchangers through computational fluid dynamics (CFD) assessments, contrasting configurations both on the shell and tube sides—incorporating configurations with and without helical baffles. Employing the FloEFD software, a three-dimensional computational fluid dynamics (CFD) model was developed within SolidWorks to explore the phenomenon of conjugate heat transfer occurring between the shell and tube segments of the heat exchanger. Key thermal transfer characteristics were scrutinized, including the exit temperatures observed on both the shell and tube sides as well as the pressure differential recorded between these respective domains. Based on the CFD findings, the Type 4 twin pipe heat exchanger—characterized by the presence of helical baffles—has the potential to on both the shell and tube sides, exhibits superior performance relative to alternative configurations. Specifically, Type 4 demonstrates an 8% enhancement in the shell-side outlet temperature and a 5.5% increase in the tube-side outlet temperature compared to previous designs. Nevertheless, this enhanced thermal performance comes with a price. According to these studies, adding helical baffles improves heat transfer efficiency but also causes a significant rise in pressure drop.

Keywords: CFD, Conjugate heat transfer, Double pipe heat exchanger, helical baffles

1. Introduction

Heat exchangers with two pipes are distinguished by their relative ease of design and versatility, frequently utilized in applications characterized by relatively lower flow rates along with elevated temperatures or pressures. Within this classification in the context the working fluids of heat exchangers pass via concentric pipes or tubes; however, they do so in various configurations, notably in parallel and counterflow arrangements. Notwithstanding the relatively reduced diameter of this category of heat exchanger, previous investigations suggest that such devices are equally applicable for high-pressure scenarios. Additionally, they assume a critical function in high-temperature environments, particularly in processes such as pasteurization, preheating, and reheating. (Gabir and Alkhafajiet al., 2021) A multitude of strategies has been developed to augment the efficacy of thermal transfer within double-pipe heat exchangers, irrespective of their dimensions, while simultaneously guaranteeing sufficient pumping power. (Gobinath et tal., 2018) These approaches are delineated into two primary classifications: methods that are both active and passive. The active strategy improves heat transfer through the application of energy to the fluid. Conversely, the passive approach capitalizes on the inherent fluid's energy to optimize the result. The twin pipe heat exchanger can perform better when these two approaches are combined in a synergistic manner, outperforming the effectiveness yielded by each method independently. This integrated approach is referred to as compound enhancement. Nevertheless, the implementation of these methodologies may necessitate greater the augmentation of pumping power is attributable to an associated elevation in pressure drop. Consequently, the selection of an appropriate methodology is imperative for achieving enhanced efficiency. The influence of permeable baffles on theoperational performance of double pipe heat exchangers. have been scrutinized, alongside the assertion that the incorporation of turbulators within the heat exchanger walls contributes to enhanced operational performance. (Hashemian, et al., 2016) It has been demonstrated that under conditions of low Reynolds number, fins with cut and twisted geometries offer superior advantages compared to conventional smooth tubes. A comprehensive array of studies has been executed. (Kapseet al., 2023) The intake pathway within the experimental framework exhibited three distinct cross-sectional geometries: circular, elliptical, and trilobate. Encased within a simplistic cylindrical outer passage. Among these profiles, particular emphasis was directed towards the tri-lobed configuration regarding performance metrics. This analysis was conducted through a mathematical evaluation of various cross-sectional profiles. The pressure drop and the research investigated the thermal transfer characteristics of a dual-pipe heat exchanger integrated with helical twisted strips fabricated from aluminum, possessing a thickness of 1 mm. (Mousaviet al., 2023) The results derived from the experimental investigation indicate that the incorporation of twisted strips into the heat exchanger markedly enhances its capability to conduct heat while simultaneously diminishing pressure losses. (Najafabadiet al., 2022) The outcomes of an experimental analysis employing the implementation of semi-length helical strips within a dual-pipe U-bend heat exchanger demonstrated that these strips significantly surpass the performance of smooth tubes regarding the thermal efficiency of the heat exchanger.

(Naphonet al., 2006) Due to the capacity of baffles in twin pipe heat exchangers to enhance the operational efficiency of the heat exchanger, their significance within this domain has escalated considerably. This study employs computational fluid dynamics (CFD) methodologies to analyze the performance parameters of twin pipe heat exchangers, both with and without helical baffles affixed to the shell tube sidewalls. (Naphon,, 2006) The FloEFD software facilitates the examination of the conjugate heat transfer phenomena that transpire between the shell and tube sides of the heat exchanger. A three-dimensional CFD model is developed utilizing SolidWorks.

(Punniakodiet al., 2022) The exploration of conical tubes as a viable alternative to cylindrical tubes in double-pipe heat exchangers represents a significant advancement in the field of heat transfer technology. Conventional cylindrical tubes, despite their widespread utilization, often face limitations in the optimization of heat transfer due to flow inefficiencies and boundary layer effects. (Senthilkumar, et al., 2020) The shift toward a conical configuration has led to observations of potential improvements in heat exchange effectiveness among researchers. The conical geometry promotes a more gradual variation in flow area, thereby assisting in the mitigation of flow separation and turbulence. This transformation results in a more streamlined fluid pathway and reduced pressure loss, thereby facilitating enhanced heat transfer between the interacting fluids.

(Tanget al., 2015) The current body of literature clarifies that conical tubes offer numerous advantages compared to their cylindrical counterparts. A significant advantage the augmented surface area for thermal transmission may engender a thermal exchange mechanism exhibiting heightened efficiency. (Venkateshet al., 2014) The conical tubes' tapering attribute facilitates enhanced mixing and a more uniform temperature distribution within the heat exchanger. Such design modifications could yield a diminution in thermal resistance alongside an elevation in the overall heat transfer coefficient. Consequently, the implementation of conical tubes might culminate in more proficient heat transmission and potentially allow for a reduction in the dimensions of the heat exchanger necessary to achieve similar thermal performance(Zhang et tal., 2023).

Moreover, the integration of conical tubes within double-pipe heat exchangers aligns with broader engineering design trends focused on enhancing system efficiency and compactness. (Zhang et al., 2012) The enhanced flow characteristics and heat transfer performance linked with conical tubes can substantially contribute to the creation of more compact and economically feasible heat exchanger designs. Such innovations not only improve thermal efficiency but also offer prospects for reductions in material consumption and operational costs. As ongoing research initiatives persist in investigating and validating these benefits, the incorporation of conical tubes into heat exchanger systems may evolve into standard practice for optimizing heat transfer efficiency across a diverse array of industrial applications.

2. The CFD analysis

The Computational Fluid Dynamics (CFD) entails the graphical representation of fluid behavior, encompassing both liquids and gases, within a specified domain or object, and provides a framework for analyzing the interactions of the fluid within said domain, as well as its influence on previously established objects, employing mathematical principles, physical laws, and specialized flow simulation software. The foundation of CFD lies in the Navier-Stokes equations, which outline the connections between a fluid's density, temperature, pressure, and velocity in motion, all of which are pertinent to the defined domain. In the present study, the flow simulation is conducted utilizing FloEFD, a tool that facilitates the resolution of the requisite Navier-Stokes equations. Figure 1 illustrates various configurations of heat exchanger setups that have been employed for the analytical purpose.

A mesh independency analysis was performed for the Type 1 heat exchanger utilizing the cell configuration outlined in the accompanying table 1; it is apparent that there exist minimal discrepancies among the recorded values, which substantiates the selection of the cell configuration (Sl.No. 3) for the analytical assessment of the heat exchanger. The quantity of cells and the results pertaining to mesh independence for each of the four heat exchangers are presented in Tables 1 and 2.

Table 1. Specifications for each type of heat exchanger's cell count

Type of heat exchanger	Type 1	Type 2	Type 3	Type 4
Cell count	54562	55268	56569	58289

Table 2. Mesh autonomy

Sl. No	1	2	3	4
Total number of cells	100040	192292	551810	1352021
Shell outlet temperature	34.01	33.04	35.08	34.34
Tube outlet temperature	55.02	55.37	55.23	55.42
Shell pressure drop	3573.32	3639.63	3698.28	3896.07
Tube pressure drop	5399.49	5292.14	4201.85	4941.89

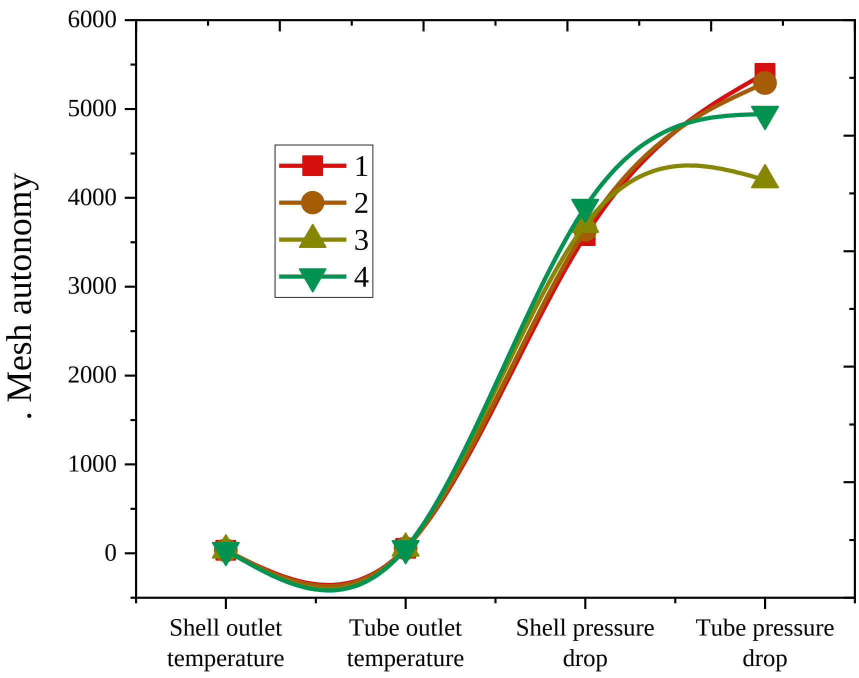

Figure 1. Mesh autonomy for Tube, Shell and Pressure in tube, shell

The preliminary specifications for the working fluid, which is unequivocally designated as water, are delineated below: The subsequent requirements pertain to the inlet temperature for both the sides of the shell and the tube: The following is how the mass flow rate is expressed: relevant to the shell side as well as the tube side. Utilizing FloEFD software on a computational system equipped with a Xeon processor featuring 32 logical cores and 64 GB of RAM, the analysis of conjugate heat transfer is conducted.

3. Result and discussion

The computational fluid dynamics (CFD) simulation was systematically performed for four distinct categories of double-pipe heat exchangers, and the resulting findings and observations are delineated (see Figures 2 and Table 3). According to the analytical outcomes, for the Type 1 heat exchanger's shell side has an entry temperature that is recorded at 0.8 flow rate. Upon reaching the initial bend, the temperature exhibits a significant increase, while at the subsequent second and third bends, it stabilizes around two pre-established values. Another imperative measurement involves the temperature recorded at the shell side exit. Correspondingly, on the tube side, the temperature at the tube side inlet is documented at a specific value at a different flow rate of a distinct magnitude; following the first bend, the temperature declines to 59, and at the second and third bends, the temperature remains within two predetermined ranges. Ultimately, the output temperature on the tube side is also documented.

A pressure differential of 5399 Pa is observed across the tube side, stemming from the recorded pressure of 107531 Pa at the outflow of the tube side, the pressure measures around 102132 Pa, while at the intake of the tube side, it registers a similar value. Simultaneously, the pressure on the shell sideentrance is quantified at 104235.49 Pa, while the exit pressure is documented at 107808.81 Pa, yielding a resultant decrease in pressure of 3573 Pa within the shell side. For the facilitation of optimal conjugate heat transfer, the preferred velocity is identified to span from a specific lower limit to an upper limit, while the velocity across the heat exchanger is estimated to be approximately between another lower limit and 0.95 of a defined value. The inlet velocities at both the egress velocities on both the tube and shell surfaces are documented as specific metrics alongside an additional established parameter, while the tube and shell surfaces are measured at predetermined quantities for each instance.

For the Type 2 heat exchanger, the thermal assessment at the inlet of the shell side is recorded at a specific value that correlates with a particular flow rate exhibiting a range of magnitudes; upon reaching the initial curvature, the temperature increases to a heightened level, which can be ascribed to the existence of helical baffles situated on the shell side. The thermal reading stabilizes during the subsequent second and third bends, oscillating around two defined values. An additional critical parameter is the temperature observed at the shell side exit. At a specified flow rate, the temperature at the entrance of the tube side is documented as 60; at the first bend, it experiences a decline to a lower value, and at the second and third bends, it is noted as observed to lie within two specific ranges. Finally, the temperature at the tube side outlet is recorded.

For Type 3, the thermal condition at the inlet of the shell side, under a predetermined flow rate, undergoes an elevation to a specific value at the initial bend, which can be attributed to the improvement of the shell compartment's flow rate. Following the second and third bends, the thermal condition stabilizes around designated values. Ultimately, the thermal condition recorded at the shell side outlet registers a specific value. Simultaneously, the thermal condition at the tube side inlet, under a specified flow rate, diminishes to a defined value at the first bend as a result of the tube side's helical baffle being implemented. The temperature at the second and third bends varies between 56.5 and a different specified value. Finally, a precise value for the temperature state at the tube side output is recorded. At the tube side input, the pressure is around 115922 Pa, and at the outflow, it is recorded at 107136 Pa, resulting in a pressure differential of 8786 Pa across the tube side. The inlet pressure at the shell side is measured at 107085 Pa, with an outlet pressure of 107131 Pa, culminating in a pressure differential of 1170 Pa within the shell side. For optimal conjugate heat transfer, the preferred velocity is expected to range from one defined value to another, while the velocity across the heat exchanger is anticipated to be within two other specified values. The inlet velocities at both the tube side and shell side are documented at designated values, while the exit velocities on both sides are noted as additional specified values.

In the framework of Type 4 heat exchangers, the thermal state at the ingress on the shell side, assessed under specific thermal conditions and flow velocity, escalates to a defined magnitude at the initial bend, which can be ascribed to the presence of a helical baffle on the shell side. At the subsequent curves two and three, the thermal state remains consistent around predetermined values. Ultimately, the thermal state at the shell side outlet is documented at a specific value. The thermal condition at the tube side inlet, under a defined flow rate, exhibits a reduction to a certain value at the first bend, due to the presence of the helical baffle within the tube side. At the second and third bends, the thermal condition stabilizes around 55.8 and another specified value. In conclusion, the thermal condition at the tube side outlet is measured at a particular value. The pressure at the tube side inlet approximates 135887 Pa, while the pressure at the outlet is recorded at 106887 Pa, resulting in a substantial pressure differential of 29000 Pa. The inlet pressure at the shell side is noted at 116873 Pa, with an outlet pressure of 106472 Pa, leading to a pressure differential of 10400 Pa within the shell side. For effective conjugate heat transfer, the preferred velocity is anticipated to range from one specified value to another, whereas the velocity across the heat exchanger is expected to reside between two additional specified values.

Heat Exchanger Type	Shell Side	Tube Side	Parameter
Type 1	At 0.8 flow rate, significant increase at the first bend	Specific value at a different flow rate; drops to 59 at first bend	Inlet Temperature (°C)
	Stabilizes around two values at the second and third bends	Stabilizes within two predetermined ranges	Exit Temperature (°C)
	3573 (104235.49 Pa inlet, 107808.81 Pa outlet)	5399 (107531 Pa inlet, 102132 Pa outlet)	Pressure Differential (Pa)
	Range from a specific lower limit to an upper limit	Estimated to be between another lower limit and 0.95 of a defined value	Preferred Velocity (m/s)
Type 2	Increases significantly at the first bend due to helical baffles	60 at the inlet; decreases at the first bend	Inlet Temperature (°C)
	Stabilizes around two defined values at the second and third bends	Varies within two specific ranges	Exit Temperature (°C)
	Not specified	Not specified	Pressure Differential (Pa)
Type 3	Specific value at the initial bend due to	Diminishes to a defined value at the first bend	Inlet Temperature (°C)

	flow improvement		
	Stabilizes around designated values at second and third bends	Varies between 56.5 and another specified value	Exit Temperature (°C)
	1170 (107085 Pa inlet, 107131 Pa outlet)	8786 (115922 Pa inlet, 107136 Pa outlet)	Pressure Differential (Pa)
Type 4	Escalates at the first bend due to helical baffles	Reduces to a certain value at the first bend	Inlet Temperature (°C)
	Stabilizes around predetermined values at second and third bends	Stabilizes around 55.8 and another specified value	Exit Temperature (°C)
	10400 (116873 Pa inlet, 106472 Pa outlet)	29000 (135887 Pa inlet, 106887 Pa outlet)	Pressure Differential (Pa)
	Range from one specified value to another	Expected to reside between two additional specified values	Preferred Velocity (m/s)

The thermal distribution representations the characteristics for each of the four types of heat exchangers are depicted in Figure 2. In the instance of Type 1, the central region of the tube exhibits a notable temperature gradient due to the absence of swirl flow; conversely, in Type 3, the presence of swirl flow results in a diminished high-temperature zone at the core.

Regarding Type 4, the fluid exhibits swirling motion on both the shell and tube sides, which effectively reduces the concentration of elevated temperature within the center of the pipe. Figure 3 illustrates the contour representation employed for the comparative analysis of temperature.

Table 3. Total outcomes for every kind of heat exchanger

Type	1	2	3	4
Description	Without baffle	Shell side baffle	Tube side baffle	Both tube side and shell side baffle
Tubeinlet temperature	60	62	64	65
Tubeoutlet temperature	55	54	54	52
Shell inlet temperature	30	32	33	34
Shell outlet temperature	34	35	34	36.5
Pressure drop tube side [Pa]	5399	4959	8786	29000
Pressure drop shell side [Pa]	3573	9050	1170	10400

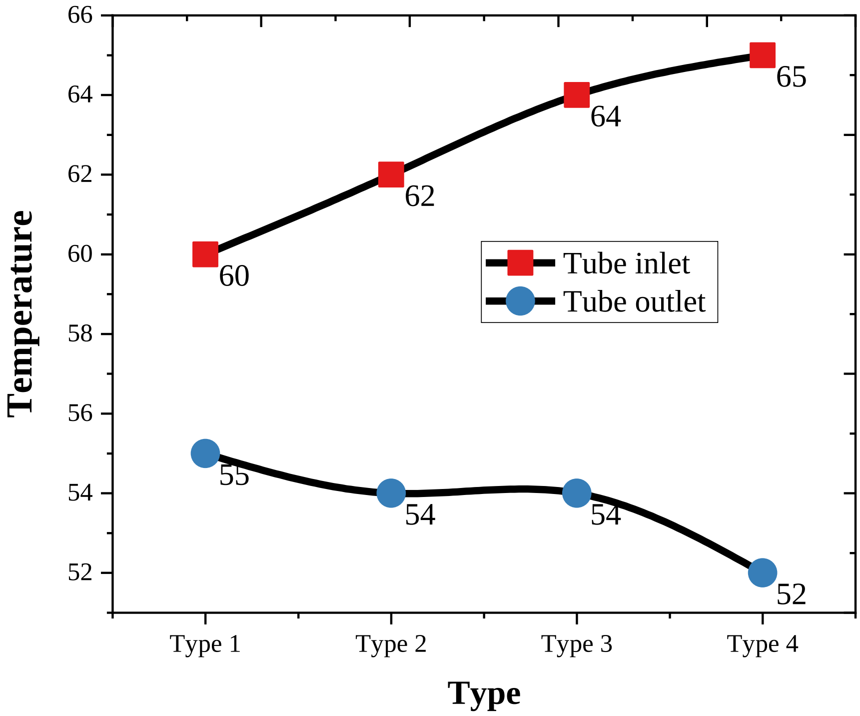

Figure 2. Tube Temperature for inlet and outlet

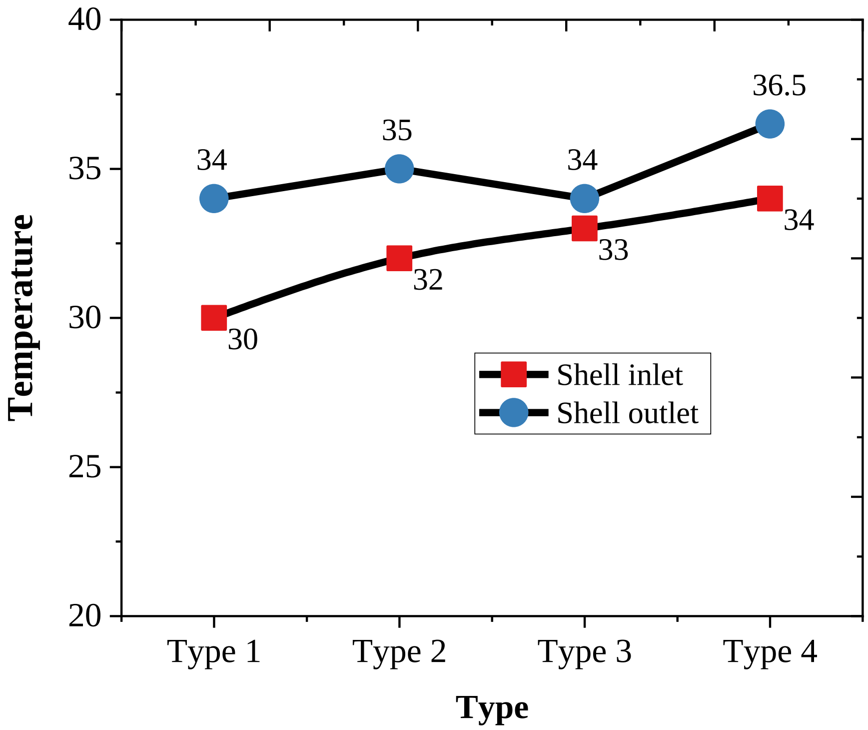

Figure 3. Shell Temperature for inlet and outlet

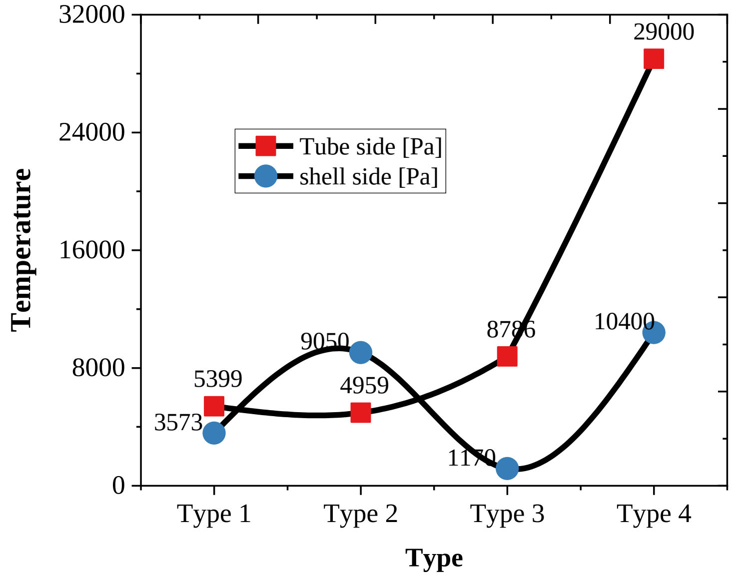

Figure 4. Pressuredrop for inlet and outlet

Figures 4 through 7 present the comparative graphical analyses for Pipe 1 through Pipe 4 on the tube side. As illustrated in Figure 4 about Type 1, a significant elevation in temperature is observed in proximity to the wall, attributable to the absence of swirl flow within the tube.

5. Conclusion

Based on the empirical data, the following conclusions can be deduced:

1. The outlet temperature on the tube side of the Type 1 heat exchanger exhibits a markedly elevated value in comparison to the Type 2 and Type 3 heat exchangers. In juxtaposing the Type 2 heat exchanger with the Type 1 and Type 3 variants, it is evident that the shell side outlet temperature of Type 2 is significantly augmented.
2. Furthermore, a substantial disparity in the tube side outlet temperature is noted between the Type 4 and Type 1 heat exchangers, with the Type 4 configuration reflecting a higher temperature. Additionally, the tube side outlet temperature for the

Type 4 heat exchanger is elevated when compared to both Type 2 and Type 3 configurations.

3. Upon scrutinizing the shell side outlet temperature of the Type 4 heat exchanger about Type 1 and Type 3, an observable increase in temperature is noted for Type 4. Likewise, in the comparison of the shell side outlet temperature of Type 4 with that of Type 2, Type 4 continues to demonstrate a higher temperature. Concerning the pressure drop on the shell side, Type 4 encounters a 12% escalation relative to the other three heat exchangers. The tube side pressure drop for Type 4 is also significantly greater than that observed in the remaining three types.
4. From these analyses, it can be surmised that the Type 4 heat exchanger, which incorporates helical baffles on both the shell and tube sides, exhibits superior heat transfer efficiency and an elevated pressure drop in contrast to the other three types of heat exchangers.

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