



Techno-economic viability of using twisted tubes in shell-and-tube heat exchangers for the petrochemical industry

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Dissertation accepted in fulfilment of the requirements for the degree *Master of Engineering in Mechanical Engineering* at the North-West University

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Graduation: : July 2026

**TECHNO-ECONOMIC VIABILITY OF USING TWISTED TUBES IN SHELL-AND-TUBE HEAT
EXCHANGERS FOR THE PETROCHEMICAL INDUSTRY**

A Thesis Presented

By

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Submitted to the Graduate Engineering School of the North-West University in partial fulfilment
of the requirements for the degree of

MASTER OF ENGINEERING IN MECHANICAL ENGINEERING

November 2025

Mechanical Engineering

ABSTRACT

The petrochemical industry of South Africa is constantly developing and improving the necessary heat exchanger designs. Heat exchangers play a crucial role within the petrochemical industry. This equipment is necessary for heating or cooling fluids from one process before it can be used for the following process.

Twisted tubes within shell-and-tube heat exchangers have been proven, within experimental studies, to provide significant benefits to the potential of the heat exchanger's design. While twisted tubes do not only provide a self-reinforcing design, they also provide a change in tube side & shell side flow. This mentioned change in flow increases the ability of the heat exchanger to provide better heat exchange to occur during an operating process.

While this study is only focused on shell-and-tube heat exchangers (STHE's) and fixed tube sheets within the heat exchangers, the study outcomes for the selected heat exchanger designs, presented to be significant. Comparing the change in a single heat exchanger output, while only changing the tube design from plain straight tubes to twisted tubes, provided significant information for the petrochemical industry of South Africa.

This thesis has been completed using mathematical and thermal physics solutions for each heat exchanger design. The calculations for the completion of the thesis have been done using Engineering Equation Solver (EES). EES was also used to generate the graphed outputs used for the validating comparisons and the comparisons between the two heat exchanger designs.

The EES calculation setups used within this study was validated using the data of other completed studies. The validations done on each aspect of interest, presented to all be below a maximum percentage error of 8%, which certifies that the EES calculation setups are accurately setup compared to the outputs reached within the other completed studies.

Comparing the different outcomes found within this study, it was found that the twisted tube heat exchanger generated a much higher tube side heat transfer coefficient than that of the plain tube heat exchanger, however it was also concluded that the twisted tube heat exchanger generated a higher tube side & shell side pressure drop than the plain tube heat exchanger.

Another conclusion found during the study analysis was that the twisted tube heat exchanger generated a higher operating effectiveness. It was also clearly concluded by means of using the data of a fully operational petrochemical plant in Mpumalanga, South Africa, that the twisted tube

is roughly 350 million rand more beneficial than the plain tube heat exchanger over a yearly operational period, based on its generated effectiveness and heat transfer capabilities.

The findings of this thesis provide insightful outcomes and reasoning perspectives for engineers and plant managers regarding the best selection of heat exchanger design for each petrochemical process within Mpumalanga, South Africa. The thesis clearly stated the findings of performance between the two heat exchanger designs, and the thesis motivates the viability for implementation in petrochemical plants located in Mpumalanga, South Africa.

KEYWORDS

- Twisted tubes
- Plain tubes
- Shell-and-tube heat exchangers
- Techno economic analysis
- Petrochemical industry
- Viability analysis
- Heat transfer coefficient
- Pressure drop
- effectiveness

DECLARATION

I, Michael Ernest Crumplin, declare that this research report is my own work except as indicated in the references and acknowledgements. It is submitted in partial fulfilment of the requirements for the degree of Master of Engineering in Mechanical Engineering to the North-West University, Potchefstroom.

This document has not been submitted before for any degree or examination in this or any other university. I acknowledge that I am aware of the content of the Faculty of Engineering Honour Code document and the subsequent sanctions proposed for academic dishonesty.

2026/06/19

Date

Michael Ernest Crumplin

ACKNOWLEDGEMENTS

Firstly, I would like to thank God for the ability that He has given me to complete this thesis. I am able to do anything through Him that strengthens me. It is an absolute privilege to be able to complete this study and know that it was Him leading me to my conclusions.

I would also like to thank my parents and my brother for all their support and motivation during this journey. I would not have been able to complete this study without their support and encouragement.

Lastly, I would especially like to thank my study leader Dr. Werner Kaiser. His understanding, support, and guidance lead me to my conclusions and findings within this study.

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NOMENCLATURE

Symbol	Description	Unit
A	Total Heat Transfer Area	mm^2
A_c	Free Flow Area	mm^2
$A_{c,t}$	Cross Sectional Tube Area	mm^2
A_s	Shell Side Flow Area	mm^2
$A_{s,sec}$	Shell Sectional Area	mm^2
A_t	Tube Heat Transfer Area	mm^2
$A_{t,sec}$	Tube Sectional Area	mm^2
a	Longitudinal Tube Spacing	mm
B	Annual Benefit	[-]
$B_{s,t}$	Surface Width of the Tubes	mm
b	Transverse Tube Spacing	mm
C_{min}	Minimum Heat Capacity Rate	kJ/K
C_{max}	Maximum Heat Capacity Rate	kJ/K
C_{if}	Installation Cost	R
C_C	Specific Heat Capacity Cold Fluid	kJ/kgK
C_H	Specific Heat Capacity Hot Fluid	kJ/kgK
c_i	Cost per Unit Transfer Area	R
C_p	Specific Heat Capacity	kJ/kgK
c_{pump}	Pumping Power Cost	R/Wh
C_s	Salvage Price of the Heat Exchanger	R
d	Flow depth	m
D_s	Shell Diameter	mm
D_e	Equivalent Diameter	mm
D_h	Hydraulic Diameter	mm
d_i	Inside Diameter	mm

$d_{i,tube}$	Inside Tube Diameter	mm
d_o	Outside Diameter	mm
$d_{h,t}$	Hydraulic Tube Diameter	mm
$d_{h,s}$	Shell Side Hydraulic Diameter	mm
ε	Heat Exchanger Effectiveness	[-]
ε_1	Heat Exchanger Effectiveness with a Single Shell Side Pass	[-]
Fr_m	Modified Froude Number	[-]
f	Friction Factor	[-]
f_t	Tube Side Friction Factor	[-]
f_s	Shell Side Friction Factor	[-]
f_A	Arrangement Factor	[-]
$f_{A,aligned}$	Aligned Tubes Arrangement Factor	[-]
$f_{A,stag}$	Staggered Tubes Arrangement Factor	[-]
g	Gravitational Acceleration Constant	m/s^2
G_s	Mass Flow Velocity on the Shell Side	kg/m^2s
h	Heat Transfer Coefficient	W/m^2K
h_e	Fluid Enthalpy	kJ
H	Enthalpy	kJ
i	Interest Rate	%
j_f	Tube Side Friction Factor	[-]
k	Specific Heat Conductivity	W/mK
Ke	Fluid Kinetic Energy	kJ
k_{s3}	Defined Parameter	[-]
L	Tube Length	mm
L_1	Total Length of One Twisted Tube Turn	mm
L_t	Total Length of Flow Inside the Tubes	mm
L_s	Length of Shell	mm

L_{tt}	Twisted Tube Length	mm
L_B	Baffle Spacing	mm
m	Operating Time Period	[-]
m_1	Fa Coefficient	[-]
m_2	Fluid Mass	kg
$\dot{m}$	Mass Flow Rate	kg.s
n	Number of Payments	[-]
n_1	Fa Coefficient	[-]
n_2	Number of Shell Side Passes	[-]
N_B	Number of Baffles	[-]
N_{tu}	Number of Transfer Units	[-]
Nu	Nusselt Number	[-]
$Nu_{1,lam}$	Laminar Flow Nusselt Number	[-]
$Nu_{1,turb}$	Turbulent Flow Nusselt Number	[-]
Nu_1	The Nusselt number on the shell side	[-]
N_{tp}	Number of Tubes per Pass	[-]
N_t	Number of Tubes	[-]
N_{tt}	Number of Turns Within the Twisted Tube	[-]
n_t	Number of Tube Passes	[-]
Pr	Prandtl Number	[-]
Pr_t	Tube Side Prandtl Number	[-]
Pr_s	Shell Side Prandtl Number	[-]
ΔP_{tube}	Total Pressure Drop on the Tube Side Within the Heat Exchanger	MPa
ΔP_{shell}	Total Pressure Drop on the Shell Side Within the Heat Exchanger	MPa
ΔP_t	Tube Side Pressure Drop	MPa
ΔP_s	Shell Side Pressure Drop	MPa
ΔP	Pressure Drop	MPa

$\mathcal{P}$	Perimeter	mm
Q	Heat Energy	kJ
q	Heat Transfer Rate	kJ/s
R_c	Capacity Ratio	[-]
Re	Reynolds Number	[-]
Re_t	Tube Side Reynolds Number	[-]
Re_s	Shell Side Reynolds Number	[-]
R_w	Thermal Resistance Due to Conduction Through the Wall	$^{\circ}\text{Cm}^2/W$
R_f	Thermal Resistance Due to Fouling	$^{\circ}\text{Cm}^2/W$
r	Annual Charge/Annuity of the Heat Exchanger	R
S_m	Crossflow Area	mm^2
S_1	Horizontal Tube Pitch	mm
S_2	Vertical Tube Pitch	mm
ΔT_m	Log Mean Temperature Difference	$^{\circ}\text{C}$
T_i	Fluid Inlet Temperature	$^{\circ}\text{C}$
T_o	Fluid Outlet Temperature	$^{\circ}\text{C}$
U	Overall Heat Transfer Coefficient	$\text{W}/\text{m}^2\text{K}$
V_t	Tube Side Volumetric Flow Rate	m^3/s
V_s	Shell Side Volumetric Flow Rate	m^3/s
ν	Fluid Kinematic Viscosity	Pa-s
ν_D	Dynamic Viscosity	Pa-s
ν_K	Kinematic Viscosity	Pa-s
w	Fluid Flow Velocity	m/s
w_t	Tube side Fluid Flow Velocity	m/s
w_s	Shell side Fluid Flow Velocity	m/s
W	Work Energy	kJ
μ	Viscosity	Pa-s
μ_w	Fluid Viscosity at Tube Wall Temperature	Pa-s

μ_s	Viscosity of Shell Side Fluid	Pa-s
μ_t	Viscosity of Tube Side Fluid	Pa-s
ρ	Density	kg/m^3
ρ_t	Tube Side Fluid Density	kg/m^3
ρ_s	Shell Side Fluid Density	kg/m^3
ϕ	Diameter	[-]
δ	Thickness	[-]
ψ	Void Fraction	[-]
π	PI	[-]
ξ_i	Annual Initial Cost	R
ξ_p	Pump Operating Cost	R
τ	Operating Time Per Year	h/yr
η_p	Pump Efficiency	%
ϵ	Heat Value	R/Wh

ABBREVIATIONS

EES	Engineering Equation Solver
LMTD	Log Mean Temperature Difference
NTU	Number of Transfer Units
PTSTHE	Plain Tube Shell-and-Tube Heat Exchanger
SST	Shear Stress Transport
STHE	Shell-and-Tube Heat Exchanger
TTSTHE	Twisted Tube Shell-and-Tube Heat Exchanger
TEMA	Tubular Exchanger Manufacturer's Association
UA	Overall Heat Transfer Coefficient

CHAPTER 1 INTRODUCTION

1.1 Background

Plain tube shell-and-tube heat exchangers (PTSTHE's) are currently dominating the South African petrochemical industry with a global market share value of \$23 billion (Market, Heat Exchanger, 2024). The local market value for plain-tube STHE's within South Africa is currently standing at a value of \$82.7 million since revenue calculations of 2024 (Horizon, 2025).

The value of STHE's within the global petrochemical industry grows at a 6% rate yearly, and has reached a top market value of \$18.94 billion within 2021 (HVAC, Chemicals and Petrochemicals, Power Generation, 2021). The value of STHE's within the petrochemical industry is estimated to increase to a market value of \$34.24 billion by 2031 (HVAC, Chemicals and Petrochemicals, Power Generation, 2021). The value of shell-and tube heat exchangers currently within South Africa is estimated to grow to a total value of \$110.2 million at 2030 (Horizon, 2025).

Heat exchangers are commonly used within the petrochemical industry to heat or cool substances or materials for required operating processes (Dusan & Ramesh, 2024). According to Alva Laval (2002), a large petrochemical industry in South Africa has approximately 11 000 heat exchangers. Heat exchangers play a vital role within the South African petrochemical industry. According to Arnold Benade, "More efficient heat exchange means higher production". Being more efficient and effective will result in a better performance of the heat exchanger by means of generating a better heat transfer coefficient. The overall efficiency of heat exchangers is thus a key factor in considering heat exchanger design and heat transfer within the petrochemical industry. Being able to change the output temperature of two different mediums or materials in shorter time periods will be more efficient and effective.

1.1.1 Schematic Overview of Shell-and-Tube Heat Exchanger

The operation behind heat transfer in shell-and-tube heat exchanger (STHE) consists basically of one medium that travels through the tube bundle, while the other passes through the shell within the STHE. For the best heat transfer, a counter current flow pattern is usually established. The mediums move forward and exchange heat through the tube walls of the tube bundle. Heat transfers from the heated medium to the cold medium according to thermodynamic principles such as conduction and convection, which in this case occurs simultaneously, allows for accurate heating or cooling of the process mediums. Total surface area, temperature differential, thermal conductivity of the tube material, wall thickness, medium characteristics, and flow rate are some of the variables that affect the rate of heat transfer within STHE (Samah, et al., 2018).

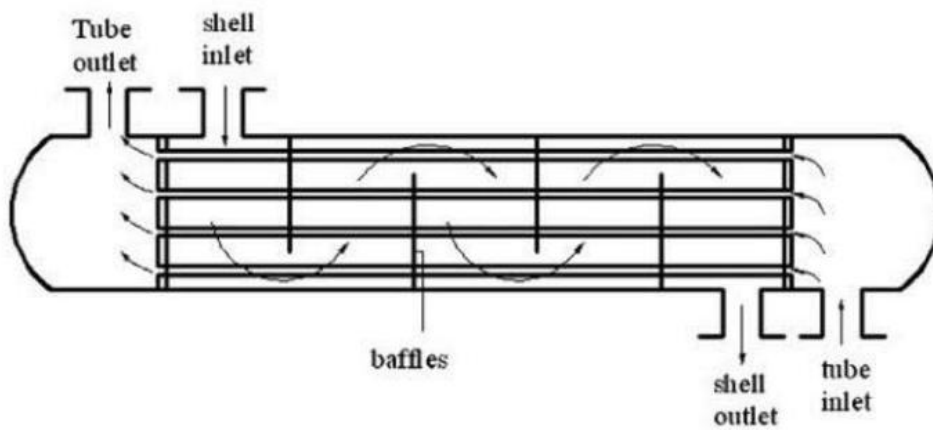


Figure 1: Visual illustration of the STHE counter-flow process (Abdullah, et al., 2019).

Heat exchangers are also very important in the petrochemical industry when considering energy efficiency. They enable the recovery and reuse of heat, reducing energy costs, and improving the overall efficiency of petrochemical processes. By exchanging heat transfer between different streams, heat exchangers help optimise energy usage and minimise energy waste (Ahmed, 2024).

When optimising petrochemical processes, heat exchangers play a vital role. Heat exchangers help maintain consistent temperatures within petrochemical equipment and systems, which is critical to ensuring the proper operation of chemical reactions and other processes. They also play a key role in maintaining the stability and efficiency of petrochemical plants.

In general, heat exchangers are a vital component of the petrochemical industry, playing a crucial role in energy efficiency, process optimisation, and equipment maintenance. Without heat exchangers, petrochemical processes would be less efficient, more costly, and more prone to equipment failure.

A technique to increase the efficiency of STHE is the use of twisted tube bundles as seen in Figure 2. Twisted tubes are passive heat-transfer enhancement devices that belong to the swirl-flow device category. Through a variety of geometric flow designs, swirl-flow devices create forced vortex fluid motion in constrained flow regimes. In each instance, enhanced heat transfer is primarily caused by fluid agitation and mixing brought on by swirl flow. One attractive feature of the twisted tube-induced swirl-flow is that, in contrast to previous techniques, it is produced by the tube geometry rather than by an extra device that is fastened or affixed to the tube (Alsulaiei, et al., 2021). They therefore don't need to be assembled, maintained, inspected, or cleaned with any extra caution (Ljubicic, 2004).

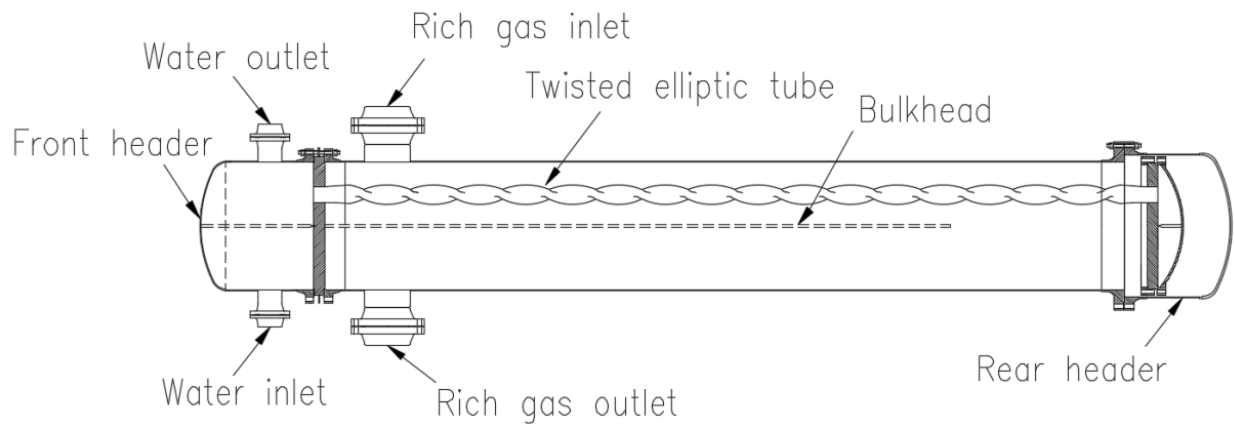


Figure 2: Schematic outline of twisted tube bundle within STHE (Liu, et al., 2023).

Another interesting aspect of the twisted tube bundle design is that the shell-side flow over the tubes creates circulation and mixing on the shell side. The centrifugal force of spinning creates a secondary flow circulation that impacts in-bundle flow, or flow between tubes. This affects the occurring heat transfer, which causes an increase in heat transfer of the heat exchanger.

However, twisted tubes within STHE's are not widely nor commonly used with the petrochemical industry. Twisted tubes in STHE's present different beneficial factors of consideration. Twisted tube heat exchangers provide enormous opportunity for improved heat recovery and heat transfer enhancement.

It is important to compare the performance between plain tube and twisted tube, and shell-and-tube heat exchangers in the sense that it will provide a comparison between the two heat exchanger designs within South Africa. The detailed research study will provide a comprehensive comparison of their performance in terms of heat transfer efficiency, pressure drop, and overall effectiveness. This information is crucial for engineers and designers in the industry to make informed decisions when selecting or designing the most suitable heat exchanger for a specific application.

1.2 Problem Statement

Currently, TTSTHE's are not widely adopted within the petrochemical industry of South Africa. A possible reason is that designers and engineers are not aware of the benefits that the twisted tube design provides within the industry. However, the most common STHE design in South Africa consists of the PTSTHE design.

Thus, a detailed research comparison between TTSTHE's and PTSTHE's is required within the petrochemical industry of South Africa in order to determine which technology provides superior

heat transfer efficiency, cost-effectiveness, and overall performance within the petrochemical industry and applications of South Africa.

1.3 Goal

The goal of this research study is to contribute in detail to the available data on twisted tube configurations and plain tube configurations within STHE, considering the petrochemical industry of South Africa. The study aims to elaborate in detail on the benefits and disadvantages considering the heat exchangers' physical performance, as well as financial implications when comparing twisted tube shell-and-tube heat exchangers (TTSTHE's) and plain tube STHE's within the petrochemical industry of South Africa.

1.4 Research Objective

The effect of heat transfer within twisted tube and plain tube heat exchangers has not yet been reached to its fullest potential. Heat transfer and heat exchanger efficiency are issues of growing concern among the petrochemical industry in South Africa. Thus, a detailed research comparison on the performance of PTSTHE's and TTSTHE's will contribute to widening the field of selection availabilities for plant operators when selecting the correct and accurate heat exchanger design for their plant.

The purpose of the research study is to evaluate and utilise the operating performance of PTSTHE's and TTSTHE's. Calculating optimising availabilities for the heat exchanger industry by means of evaluating the heat transfer capabilities, flow optimisation, cost effectiveness, manufacturing availability, and measuring the calculated heat transfer operating efficiency between TTSTHE's and PTSTHE's. The comparison of the researched and calculated data with the previously concluded and conducted study results will contribute to an accurate result and conclusion of this study.

Analysing a new field of study regarding the two different heat exchanger designs will contribute to generating an understanding of the performance capabilities of a TTSTHE and a PTSTHE in terms of a few different aspects.

- Understanding the cost effectiveness of each STHE design. This additional understanding can help companies optimise their investment and operational costs. By comparing factors such as the initial purchase price, maintenance requirements, and energy consumption, organisations can make informed decisions to maximise their return on investment.
- Providing a wider range of knowledge regarding the energy efficiency of each STHE design. A detailed research study between TTSTHE's and PTSTHE's can help identify

which design offers better energy efficiency, thereby allowing companies to reduce their energy consumption and environmental impact.

- Understanding the reliability and durability of each STHE design. By conducting a detailed research study, engineers and designers can assess the performance of each design under different operating conditions and determine which design offers better long-term durability and reliability with respect to their specific operational needs.
- Comparing and analysing the operating performance of a TTSTHE versus a traditional PTSTHE in terms of heat transfer efficiency, pressure drop, and overall performance.
- Providing insights into the feasibility and potential benefits of the use of twisted tube technology in STHE's in petrochemical applications and processes.

With respect to the above-mentioned reasoning, it can be seen that conducting a detailed research study on TTSTHE's and PTSTHE's is important to gain a deeper understanding of their performance and capabilities. This knowledge is essential for engineers, designers and organisations to optimise their heat transfer processes, reduce costs, improve energy efficiency, and drive innovation.

1.5 Method of Investigation

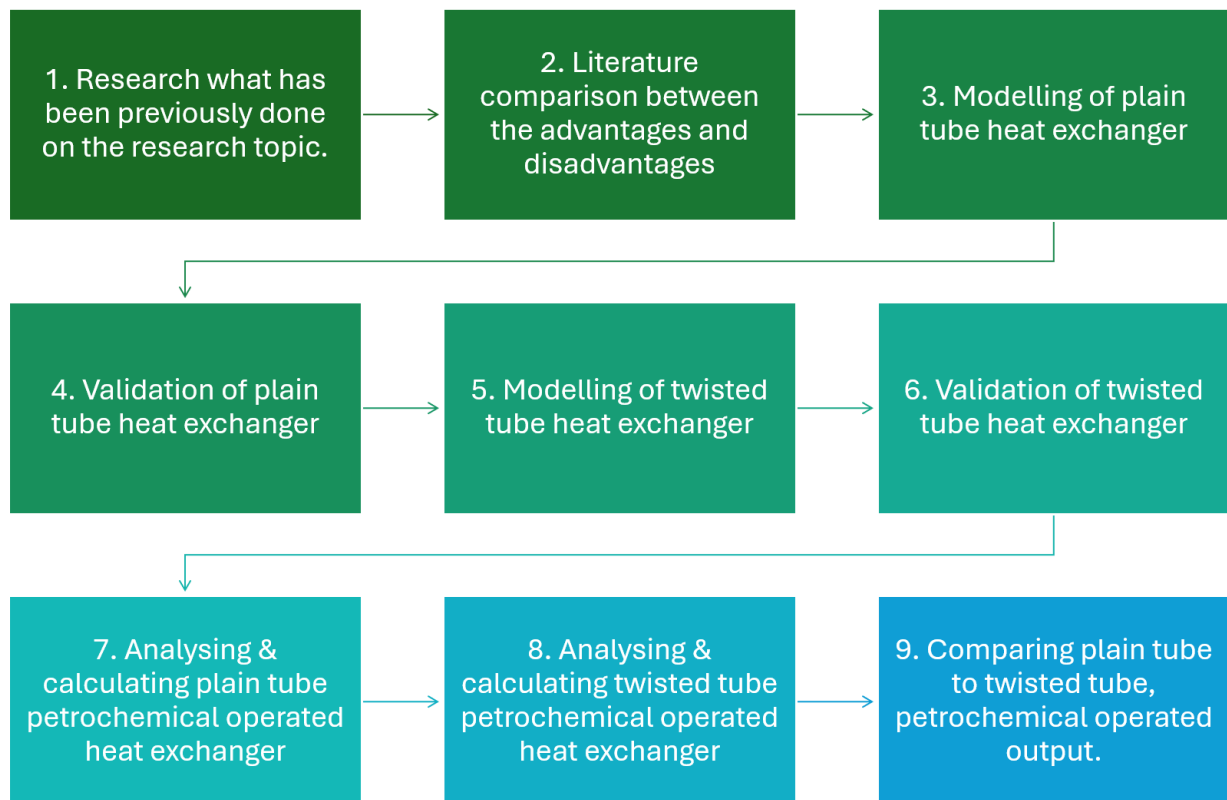


Figure 3: Method of investigation

During the method of investigation, Figure 3, a thorough evaluation will be conducted between the two heat exchanger designs, the plain tube heat exchanger design and the twisted tube heat exchanger design, to evaluate the ability and performance capabilities of each design under selected operating and design conditions to determine which design will be a better heat exchanger design choice for certain petrochemical applications and processes.

1.6 Scope and Constraints

1.6.1 Scope

- **Definition of the research problem and objectives.**

The scope of the research study is to provide a detailed conclusion on the performance evaluation of twisted tube and PTSTHE's within the petrochemical industry of South Africa. The industry requires a more elaborate, detailed proposal on the abilities and potential that the two different STHE designs can provide. A detailed evaluation between the two heat exchanger designs will contribute to increasing the ability to make decisions when designing a STHE for a certain petrochemical process.

The research study will be conducted using the following method. A detailed step-by -step presentation on the key process elements of investigation can be seen in the table below. The table below contains the method of investigation as shown in the previous subsection.

- **Description of the research methodology and approach.**

Table 1: Method of investigation

Investigation Element	Methods
Research	Research previously conducted studies related to the topic of interest of the research study.
Obtain & identify	Obtain all relevant information to be used and mention the details of each. Identify an unanalysed field within each study to investigate and improve upon.
Evaluate	Evaluate the information obtained regarding the benefits and disadvantages of each heat exchanger design.
Determine	Determine the performance abilities of each heat exchanger design under different operating and design conditions.
Generate	Generate a comparison field to accurately identify the performance of each design under the operating and design conditions evaluated.

Investigation Element	Methods
Compare	Compare the results obtained with previous existing studies and/or data, as well as with practically measured and determined results.
Conclusion	Provide a detailed conclusion on each heat exchanger design and performance according to their performance data evaluated.

- **Identification of specific research questions or hypotheses.**

In order to fully complete the aim of the detailed research study between the advantages and benefits that twisted tube and PTSTHE's provide within the petrochemical industry of South Africa, the following questions have to be answered.

1. How and when will twisted tubes and/or plain tubes within a STHE prove to be beneficial and the better design choice within the industry of interest?
2. How does the performance between the twisted tube design and the plain tube design compare within the industry of interest?
3. Which design is more affordable for manufacture and maintenance, and how does the relationship between cost and performance compare between the two heat exchanger designs?
4. What are the design benefits of the TTSTHE design?
5. What are the design benefits of the PTSTHE design?
6. How does the efficiency between the two heat exchanger designs compare within the industry of interest?
7. What design constraints does each heat exchanger design have in terms of operability when considering limited space within the industry of interest?
8. Which heat exchanger design overall provides better energy savings within the industry of interest?

- **Explanation of the significance and contribution of the research study to the existing body of knowledge.**

This study will be beneficial to the industry by adding to the currently active information on twisted and PTSTHE's. The study will also be beneficial to the industry in the sense that the study will be elaborating on the advantages that both these heat exchanger designs contribute to their operating performance within the petrochemical industry. The research study can serve as a valuable resource for knowledge sharing and collaboration within the industry. By sharing the findings and generated results from the study with the industry, industry professionals and operational energy plants can benefit from each evaluated heat exchanger design and learn from

experiences and best practices, leading to continuous improvement and innovation in the field of heat exchange technology within the petrochemical industry, when considering twisted tube and plain tube STHE design.

Overall, a detailed research study comparing the advantages of twisted tube and PTSTHE's will provide valuable information and insights that can benefit the industry in terms of performance optimisation, cost savings, innovation, regulatory compliance, knowledge sharing, and many other aspects.

- **Details of any limitations or assumptions made in the study.**

The limitations and assumptions made within this study are:

- The study is limited to STHE's with fixed tube bundles.
- The only mechanical tube designs of interest that will be evaluated are plain tubes and twisted tubes.
- The research study will only be conducted and based upon the petrochemical industry of South Africa.
- The tubes within the heat exchangers are smooth tubes.
- Due to a limitation on research availability within South Africa on the topic of interest, studies done in other countries considering the same industry have been used for the literature review and validations done within this study.

1.6.2 Constraints

The study is calculated and focused on the following few constraints and conditions.

- The study is conducted only on the petrochemical industry of South Africa, while focused on the area of Mpumalanga.
- The study is only focused and conducted for the tube side fluid Reynolds number range of $4000 \leq Re \leq 15000$.
- The study is only focused on fixed tube STHE's.
- The study was conducted only for water as the tube side fluid, and steam as the shell side fluid.
- The study only focused on a single STHE, using the specified conditions and parameters presented for this heat exchanger, while operating within the industry.
- The twist pitch length and aspect ratio of the twisted tubes used within the analysis were selected on a brief evaluation for each, and only the best performing parameter was selected for the evaluation.

- The study is only analysed for industrial manufactured materials and supplies, which included standardized lengths and manufactured components, considering the plain tube and twisted tube designs.

1.7 Deliverables

This research study aims to enhance the existing data on the performance of twisted tube and plain tube heat exchangers, particularly their respective advantages in the South African petrochemical industry. The study aims to provide a comprehensive evaluation of the petrochemical performance of twisted tube and plain tube heat exchanger designs. This research will enhance the available data on these two types within the petrochemical industry by offering detailed findings relevant to the current field of interest.

CHAPTER 2 LITERATURE REVIEW

Considering the literature review and previous research conducted on the topic of interest, it can be seen that the aspects of importance have been previously studied and investigated, although the studies that have been previously conducted on the aspects of relevance to this study only aimed on the information of other countries. It is important to note that this study only focuses on the petrochemical industry of South Africa and is limited to the information of Mpumalanga.

Twisted tube heat exchangers are developed apparatus that allow for the improvement of a more efficient and economical industry. The design of twisted tube heat exchangers allows for heat transfer to be conducted between two process mediums during a chemical or mechanical process. This improves the recovery of heat during petrochemical processes and allows for efficient temperature control on products during petrochemical operations.

2.1 Overview of Twisted Tube Shell-and-Tube Heat Exchangers

Within this section, a detailed overview is presented on twisted tubes in STHE's. This section will focus on elaborating on the performance techniques and design features that the twisted tube design presents in STHE's, considering the petrochemical industry.

The twisted tube heat exchanger is a design specific to STHE's and is utilized in the petrochemical industry. Although this technology is not yet fully developed in South Africa's petrochemical sector, it has been successfully developed and installed in other countries.

Twisted tubes, Figure 4, fall under the swirl-flow device type and are passive heat-transfer enhancement devices. Swirl-flow devices generate forced vortex fluid motion in confined flow regimes through a range of geometric flow design (Figure 4). In every case, swirl-flow-induced fluid agitation and mixing are the main causes of increasing heat transfer. One appealing aspect of the twisted tube induced swirl-flow is that, unlike other methods, it is not created by an additional device that is fastened or attached to the tube, but by the tube geometry (Alsulaiei, et al., 2021). As a result, they do not require any special care when it comes to assembly, maintenance, inspection, or cleaning (Ljubicic, 2004).

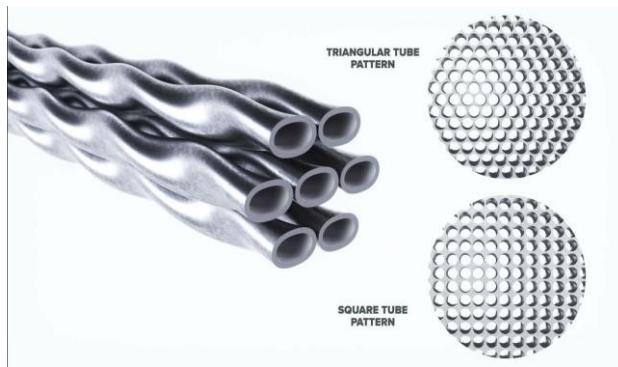


Figure 4: Twisted tube swirl-flow enhancement with triangular and square tube pattern (Transfer, Koch Heat, 2024).

Heat transfer from the shell side and from the tube side is improved by means of the twisted tube bundle. The fluid on the tube side is agitated and affected by varying fluid layer velocities, as well as wall turbulence, causing a swirling effect (Liu, et al., 2023).

On the shell side, circulation and mixing are produced during the shell-side flow over the tubes. In-bundle flow, flow in between tubes, is affected by a secondary flow circulation that is produced by the centrifugal force of swirling.

According to (Ljubicic, 2004), when considering the twisted tube geometry on the in- and outside of the tubes, the tubes are referred to as helically shaped. Ljubicic found that the helical tubes on the inside and outside increase the overall heat transfer rate by two to three times while doubling the hydraulic resistance. When these tubes are used, the heat exchanger size can be reduced by 30% while maintaining a similar heat exchange duty (Ljubicic, 2004).

When considering the twisted tube design, Geckointech (2024) found that there are a few similarities that can be established between the material used for its manufacture. The manufacturing material of the tubes is crucially important. The material properties will definitely be considered when designing the STH for the perfect application. The material for the twisted tube assembly is not limited to certain materials, but can be manufactured from various materials such as carbon steel, stainless steel, monel, brass, nickel, etc. (Geckointech, 2024).

Taking into account the mechanical characteristics of the twisted tube design, it can be established that these tubes are firmly supported when considering their design and geometry, while simultaneously allowing for well-distributed longitudinal flow along the length of the tube. The twisted tube bundle offers total vibration resistance, making them more robust and industry sustainable (Morgan, 1980).

The process of repairing twisted tubes in heat exchangers is the same as that for other conventional plain tubes. The twisted tube assembly can be cleaned on the tube and shell sides.

Twisted tube heat exchangers do not foul as often when considering their higher turbulence and wall-distributed flow without anchored zones, which is typically the case in baffled heat exchangers. It was also concluded that twisted tubes are suitable for chemical cleaning, in which the heat exchanger is cleaned using chemicals such as acids, alkalis, oxidants and solvents (Morgan, 1980).

Analysing the swirl-like flow (Figure 5) that results from the twisted tube design, it is found that there is a secondary flow in the twisted helical channels and an uneven cross-section flow in the straight channels because of the anisotropy of turbulent stresses (Alsulaiei, et al., 2021). Compared to slower particles near the wall, fluid particles near the tube axis have a higher velocity and are subject to a greater centrifugal force, generating a large amount of kinetic energy. The secondary flow that results is directed inwards close to the wall and outwards in the tube's centre. Compared to laminar flow, this effect is stronger in turbulent flow regimes (Eiamsa-ard, 2023).

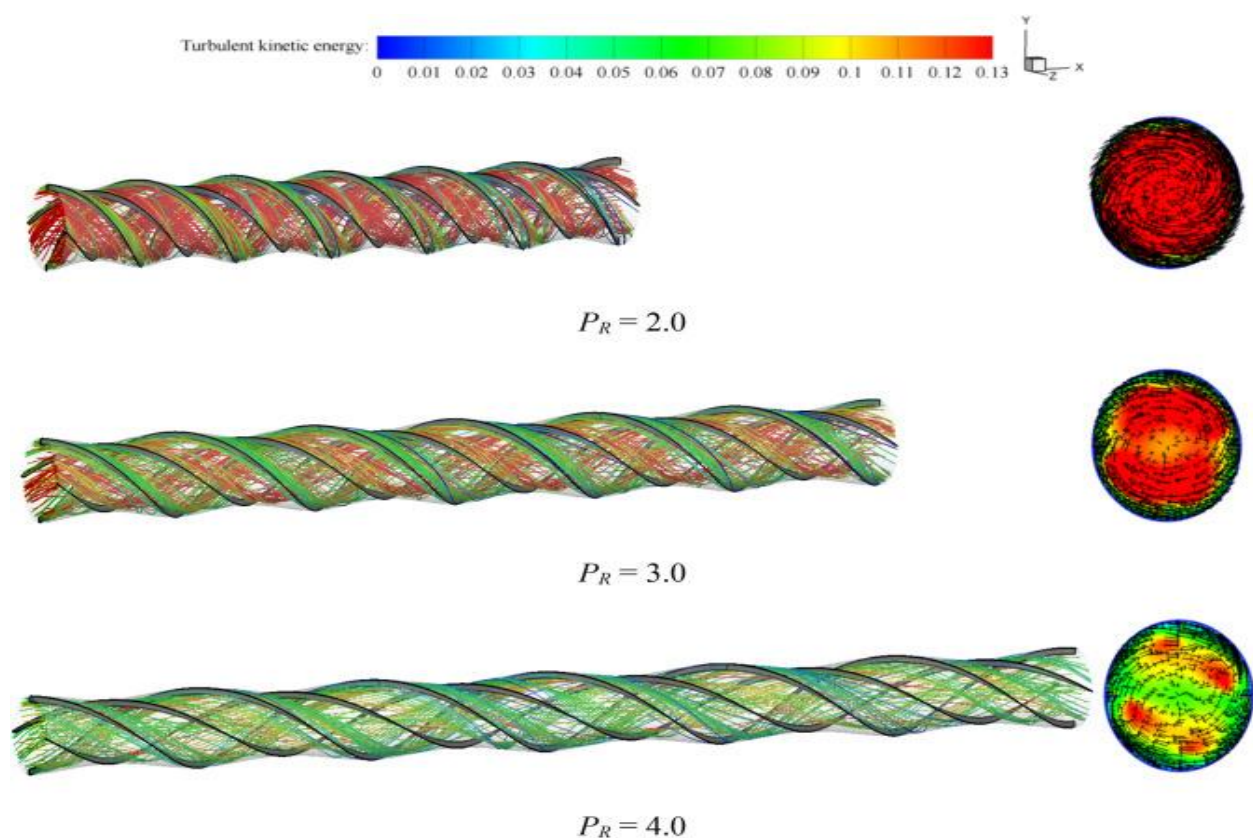


Figure 5: Illustration of swirl-like flow in twisted tubes regarding the difference in tube pitch length (P) and the kinetic energy generated by the turbulent flow (Eiamsa-ard, 2023).

Through better mixing in the bulk and a decrease in the thickness of the boundary layer, an increase in turbulence of the shell side fluid over the twisted tube bundles is generated, which enhances the heat transfer of the heat exchanger. Heat transmission is more uniform in tubular heat exchangers because of the good local velocity field and flow distribution, which also allow for the best feasible heat transfer coefficient. Furthermore, by adjusting the temperature

difference, cross flow is always penalised in comparison to pure counter-current flow. In twisted tube exchangers, pure longitudinal flow facilitates more efficient utilisation of the available temperature differential (Eiamsa-ard, 2023).

In order to explain how twisted tubes accomplish their high heat transfer coefficients, Eiamsa-ard explained that one should consider the helical channels as a sequence of brief, consecutive portions where the flow constantly changes direction, disrupting the steady velocity profile's accumulation in between. These disruptions provide good transverse mixing, and even at relatively low Reynolds numbers, the flow remains turbulent as a result of the multiple disturbances. No influence from the entry is discernible, since the design of twisted tubes constantly disrupts the build-up to a fully defined profile.

In both a single-twisted tube and a bundle, heat transfer is influenced by a number of equally significant factors. Generally speaking, creating a swirl in every single tube will improve the turbulence mixing process and will either intensify or weaken other mechanisms.

Within the twisted tube design, the centrifugal force and the radial pressure gradient will push heavier, colder fluid outwards if the cold fluid is on the tube side. This will cause the hot fluid to climb towards the centre of the pipe. As a result, the local heat transfer coefficient decreases. When it comes to the heated fluid on the shell side, the opposite will occur. The bundle's flow distribution, on the other hand, is another component that influences the overall heat transfer coefficient by producing a less favourable flow distribution with a favourable density gradient (Ljubicic, 2004).

2.1.1 The Twisted Tube Configuration

As proven by Yang, Zhang and Hong (2011), the flow of the fluid within the twisted tube design is generally referred to as turbulent flow. The flow type is expressed in many research studies as a swirl-like motion. The swirl-like motion generated, causes an increase in turbulence as a result of the twisted tube design structure. The turbulent flow leads to an increase in mixing and heat transfer because of the fluid's whirling, swirl-like motion caused by the twisted geometry. The heat exchanger performs better thermally as a result of the turbulence created. Additionally, the fluid's contact with the tubes' heat transfer surface is improved by the enhanced turbulence. This improves the heat transfer coefficient, increasing the overall efficiency of the heat exchanger.

The turbulent flow generates a higher pressure drop that occurs within the twisted tubes of the heat exchanger. As a result of the twisted geometry that the twisted tubes provide, the fluid experiences impediments and variations in flow direction as it passes through the twisted tubes. These factors contribute to an increase in resistance to flow, resulting in a higher pressure drop.

The twisted tube configuration also helps mitigate fouling and scaling issues. The turbulent flow generated by the twisted tubes minimises the build-up of deposits and improves self-cleaning. This property of the twisted tube design is advantageous, as it reduces maintenance requirements and extends the overall operating life of the heat exchanger.

2.1.2 Fluid Flow in Heat Exchangers

In a STHE, two fluids at different temperatures flow through the heat exchanger. One fluid flows through the shell and in between the tubes on the shell-side of the heat exchanger. The other fluid flows within and through the tubes on the tube-side of the heat exchanger. The flow of each fluid on each side of the heat exchanger is different, and will thus be analysed as such.

2.1.3 Flow Effectiveness in Shell-and-Tube Heat Exchangers

The impact of the generated flow, turbulent and/or laminar shown in Figure 6, within STHE has a very important effect on the overall heat transfer rate and efficiency of the heat exchanger. The flow type generated within the heat exchangers is crucial for heat transfer and pressure drop control. By means of controlling the flow type within the heat exchangers, the process output temperatures can be controlled, and in such manner the process deliverable temperatures can be accurately determined. It is thus very important to understand the flow type generated with the heat exchanger design, as well as how it is generated to control the heat exchanger process.

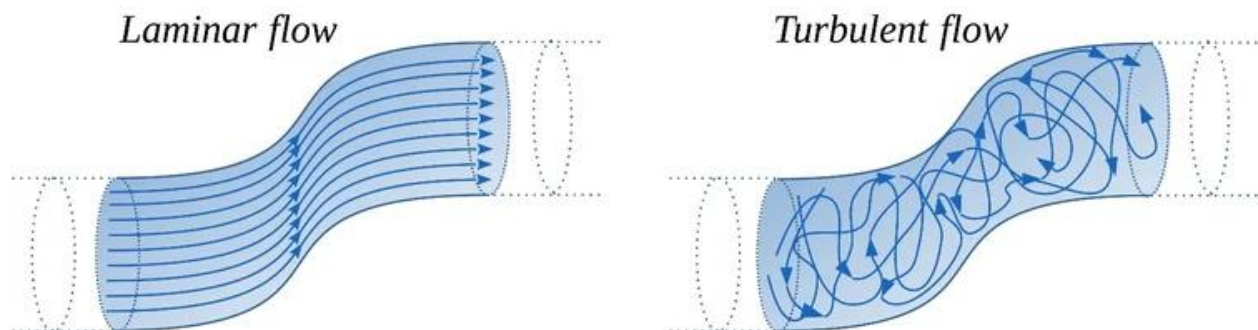


Figure 6: Turbulent & laminar flow (Services, Fossil Consulting, 2024).

2.1.4 Turbulent Flow Type

Turbulent flow is generated in twisted tubes by the introduction of swirl or rotation of the fluid within the tubes (Figure 6). When fluid flows through the twisted tube configuration, the irregular geometry of the tube causes the fluid to rotate or swirl as it flows through the tube. This swirling motion generated disrupts the flow patterns of the fluid and causes turbulence to develop. The twisting geometry of the tubes creates vortices and eddies within the fluid during flow, which increases the mixing and chaotic motion of the particles within the fluid (Doran, 2013). This disruption in the flow patterns results in higher levels of turbulence occurring compared to plain

conventional tubes. In general, the twisted geometry of the tube causes the fluid to interact with the tube walls in a more complex and unpredictable manner, leading to the generation of turbulent flow.

The turbulent flow within STHE also generates a higher pressure drop within the heat exchanger by means of the turbulent fluid flow type compared to laminar flow, considering twisted tubes and plain tubes within STHE. The higher pressure drop generated within the heat exchanger by turbulent flow is caused by the twist ratio of the twisted tube design. It was found that the pressure drop within the heat exchanger increases as the twist ratio of the tubes increases; thus, the twist ratio and the pressure drop are directly proportional within the heat exchanger design (Abdul Razzaq & Mushatet, 2021). Along with the increase of pressure drop generated by means of the turbulent flow, the swirling flow generated by the twisted tube geometry also enables one to enhance heat transfer between the tube wall and the shell side fluid by increasing the flow path and disrupting the thermal boundary layer.

The turbulent flow generated within twisted tubes thus causes a higher heat transfer rate within STHE when compared to conventional plain tubes due to the increased inclusions occurring between the tube wall and the fluid by means of the disrupted fluid flow.

2.1.5 Laminar Flow Type

Laminar flow is a type of fluid flow in which fluid moves smoothly in parallel layers with each other, where minimal disruption or mixing between layers occurs. This result of laminar flow is a predictable and stable flow pattern with no turbulence or chaotic movement. Laminar flow is characterised by a low Reynolds number, where the Reynolds number is a dimensionless quantity that describes the ratio of inertial forces to viscous forces within the fluid. Laminar flow usually occurs where the flow velocity is low, and the fluid viscosity is high.

Laminar flow is generated within plain tubes by means of a smooth and consistent flow of fluid with minimal disturbances or turbulence. Therefore, this flow type is typically achieved by controlling the velocity and pressure of the fluid as it moves through the tubes within the heat exchanger. By controlling the velocity and pressure of the fluid, the flow remains smooth and uniform within the tubes of the STHE.

Additionally, the diameter and length of the tubes can also play a vital role in promoting laminar flow. Larger diameter and shorter length tubes are more likely to facilitate a steady and controlled fluid flow compared to longer tubes with a smaller diameter. Using a straight and properly aligned tube without obstructions or rough surfaces can also help minimise disturbances and promote laminar flow within the heat exchanger tubes.

Heat exchangers are a very important factor to consider when considering the petrochemical industry, as they indulge crucial roles within this industry and thus affect the petrochemical industry in many ways. The primary reason for the use of heat exchangers in this industry is to generate heat exchange between two mediums. Heat exchangers are used to transfer heat from one medium to another without the mediums coming into direct contact with each other (Ipieca, 2022). This allows efficient temperature control in various petrochemical processes, such as distillation, condensation, and chemical reactions.

2.1.6 Tube Side Flow

The flow of the tube side is completely dependent on the type of tubes used for the heat exchanger design. The type of flow of the fluid within the tubes has a direct relation to the heat transfer rate of the heat exchanger when considering turbulent and laminar flow. Turbulent flow generates a higher heat transfer rate within the heat exchanger in comparison to laminar flow, although turbulent flow causes a higher pressure drop in comparison to laminar flow. The fluid flow type within the tubes can be completely determined and directed by means of inserts or selecting the correct tube design. It is all heat exchanger specific, depending on the required needs of the industry.

The characteristics and properties, considering the flow of the fluid within heat exchangers, is very important to analyse and consider when designing a heat exchanger for a specific operation.

2.1.7 Shell Side Flow

The fluid flowing through the shell of an STHE is the fluid flowing on the outside and over the tubes themselves. Variations in the type of heat exchanger, and its specific design can affect the flow on the shell side.

The shell side flow is usually intended to counter-flow or crossflow the tube side flow. The fluid enters the heat exchanger from one end in a crossflow arrangement and flows perpendicularly to the tube bundle. This type of flow allows for efficient heat transfer because of the turbulent fluid flow occurrence due to the perpendicular entry contact with the tubes.

The fluid flows in the opposite direction and parallel to the tube bundle in a counterflow configuration. This arrangement produces more effective heat transfer, which maximises the temperature differential between the fluids.

The fluid flow on the shell side can also be modified by means of baffles, which are plates or rods inserted inside the shell to guide the flow of the fluid. Baffles can alter the fluid flow on the shell

side, in order to ensure that the fluid contacts a greater area of the tube surface and improve heat transfer. Baffles provide a more regulated flow pattern on the shell side of heat exchangers.

Additionally, the shell-side fluid may have multiple shell-side passes over the tubes within the tube bundle, depending on the heat exchanger design and application due to baffle arrangements within the heat exchanger and the number of shell side passes (Figure 7). Multiple passes increase the heat transfer efficiency by allowing the fluid to make multiple passes over the tube bundle before exiting the heat exchanger, causing an increase in the overall heat transfer coefficient of the heat exchanger.

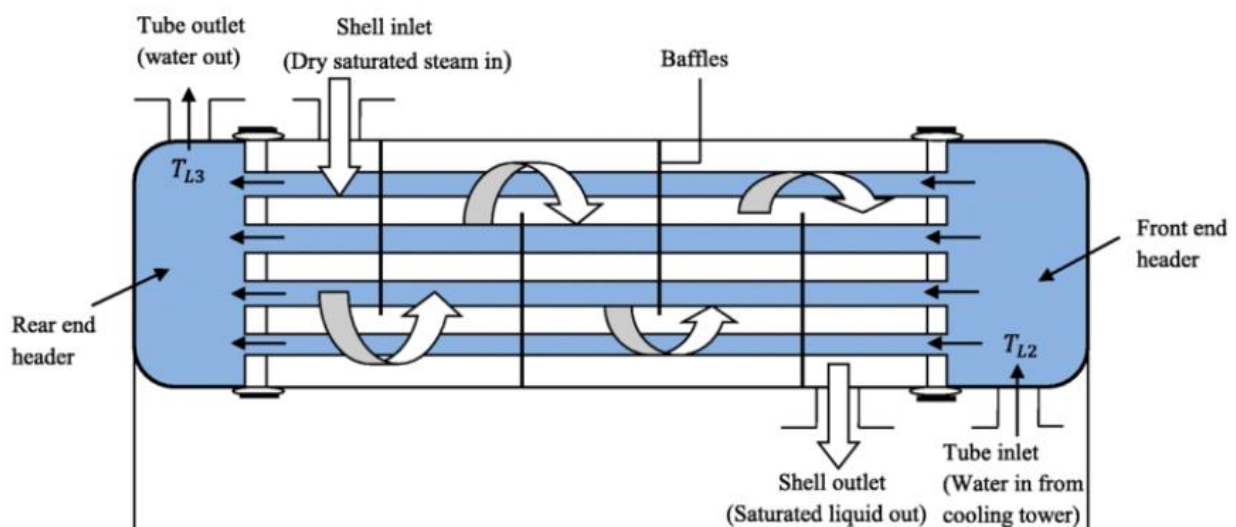


Figure 7: Schematic overview of the shell side flow in a shell-and-tube heat exchanger (Mishra, et al., 2020).

Overall, the fluid flow on the shell side of a heat exchanger is carefully designed to optimise heat transfer and achieve the desired performance based on the specific heat exchanger application.

2.2 Benefits and Disadvantages of Twisted Tube Shell-and-Tube heat Exchangers

This section elaborates on the expected benefits and disadvantages that the twisted tube design presents within STHE's. The section focuses on aspects that make the twisted tube unique within its design, as well as performance aspects to expect when using a TTSTHE.

Considering the TTSTHE design, there appears to be significant benefits towards the twisted tube design. Ljubicic (2004) described the advances in the range of design and operational reliability of TTSTHE's. It was mentioned within the study that when taking into account the size and convenience considerations, the twisted tube heat exchanger design can be cost effective in a wider range of applications of niches currently being used in the process industry (Ljubicic, 2004).

Ljubicic (2004) mentioned that most heat exchangers, including twisted tubes, have not yet been exploited commercially to their fullest potential. It was concluded that this occurrence is because historically, early heat transfer enhancement devices were limited mostly to gas flow applications (Ljubicic, 2004).

One of the benefits mentioned early within Ljubicic's study, is that twisted tube designed STHE's, regarding the tube bundle, generates a connection between the tubes with one another at their wider sides, six times over the length of one tube pitch (Ljubicic, 2004). This design regarding the twisted tubes, accommodates for vibration absorbance between the tubes, which causes the unit to be vibration free (Ljubicic, 2004).

Furthermore, Ljubicic mentioned that the twisted tube design generates the ability to provide considerable surface density, low pressure drop, and high heat transfer coefficients. These conditions are engineering conditions which is thoroughly considered during the design of a new heat exchanger. It is also mentioned that the twisted tube design enhances heat transfer on both the tube side and shell side of the heat exchanger (Ljubicic, 2004).

Ljubicic also mentioned that the use of twisted tubes within STHE's can provide a 30% equivalent reduction in heat exchanger size in order to fill the heat transfer required by the designed heat exchanger (Ljubicic, 2004).

Another attractive benefit of the twisted tube design is that the swirl-flow generated is not produced by any other component and/or device attached to the tubes. In this sense the tubes do not require any extra attention during assembly, maintenance, inspection or cleaning (Ljubicic, 2004).

It is also mentioned by Ljubicic that the tubes provide significant benefits regarding maintenance on STHE's. Ljubicic found that the tubes are firmly supported, and even though the tubes provide increased turbulence and a well-distributed flow without stagnant zones, the twisted tube design does not foul easily. The twisted tube design is also mentioned to be well-suited for chemical cleaning (Ljubicic, 2004).

When considering the economical factor of cost, Ljubicic also found that with the increased heat transfer, lower total heat exchanger weight, and the operating costs of the heat exchanger, the twisted tube design can be competitive and cost effective in the process industry (Ljubicic, 2004).

Ljubicic thus found that twisted tube heat exchangers provide substantial benefits to the process industry. It is also mentioned that twisted tubes provide high performance capabilities and have, in this way, been tested within the petrochemical industry (Ljubicic, 2004).

Ljubicic mentioned that TTSTHE's are generally more expensive than plain conventional STHE's, but Ljubicic also mentioned that the payback time on a twisted tube heat exchanger is not necessarily longer. Taking into account size and convenience considerations, these heat exchangers can be cost effective in a wider range of applications (Ljubicic, 2004).

It is thus clear that the twisted tube design does indeed provide recognizable results and performance improvements within the process industry. The design provides benefits regarding heat transfer capabilities, size, cleaning, maintenance, and in some sense cost.

Some studies have been done on the research topic of interest, and it can be seen within the following obtained results. In the next three sections, a few studies have been analysed and examined. These previously conducted studies provide a detailed concept of what previously have been researched and concluded considering plain- and twisted tube STHE's.

2.3 Improved Heat Recovery Using Twisted Tube Exchanger Technology (Barraza-Colon & Picon-Nunez, 2022).

This section presents the outcomes and results obtained within the study (Barraza-Colon & Picon-Nunez, 2022). It focuses on the inputs and results used and obtained during this study, and elaborates on the methods as well as the deliverable factors used within the study.

Barraza-Colon and Picon-Nunez found that the geometrical features of twisted tube heat exchangers enhance the thermo-hydraulic performance for the same fluid velocity compared to plain conventional heat exchangers. They also found that the benefit of the twisted tube design use becomes evident within the design, as the required heat duty is transferred directly with a smaller surface area when simultaneously generating an even lower pressure drop compared to plain conventional tube heat exchangers.

Within this study, a thermohydraulic model for the twisted tube geometry that relates pressure drop and heat transfer coefficient has been developed. Barraza-Colon and Picon-Nunez also used an alternative sizing approach, where the concept of full use of the pressure drop step is presented.

The study is presented in order to provide a comparison between plain conventional STHE's and thermohydraulic modelled twisted tube heat exchangers.

2.3.1 Study Calculated Outputs and Comparison

Within the study, the calculated outputs were compared to two different case studies. The studies used for the comparison are:

1. Yan, W., Gao, X., Xu, W., Ding, C., Luo, Z., Zhang, Z., 2017. Heat transfer performance of epoxy resin Flows in a horizontal twisted tube. Appl. Therm. Eng. 127, 28–34.
2. Barraza-Colón, J., 2021. Thermohydraulic model for the design and performance analysis of twisted-tube type heat exchangers (Original Spanish: Modelo termo-hidr'aulico para el dise~no y an'alisis del desempe~no de intercambiadores de calor del tipo twisted-tube). MSc Thesis. University of Guanajuato, Mexico.

2.3.1.1 Case Study 1:

Within the case study of Wanbo (2017), it was considered to heat epoxy resin using hot water. Figure 8 below shows that the variation of viscosity of the liquid resin with a change in temperature is significant. It can be seen that the viscosity decreases by 50% when the temperature increases with 10°C. In order to consider this variation, the design is done in infinitesimal segments.

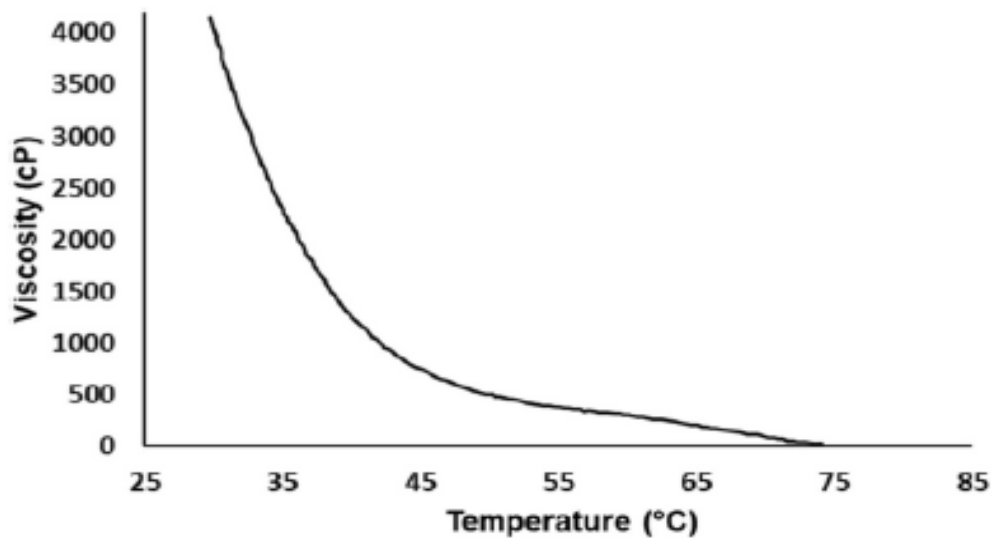


Figure 8: Viscosity vs temperature case study 1 (Barraza-Colon & Picon-Nunez, 2022).

The operating conditions and physical properties, as well as the calculated output comparison to the case study “Heat transfer performance of epoxy resin Flows in a horizontal twisted tube”, can be seen in the tables below.

Table 2 : Operating and physical properties, case study 1 (Barraza-Colon & Picon-Nunez, 2022).

Operating conditions and physical properties of case study 1		
	Hot fluid (shell side)	Cold fluid (tube side)
Heat duty (kW)	69.470	-
Mass flow rate (kg/s)	1.880	1.57
Inlet temperature (°C)	89.200	30.00
Outlet temperature (°C)	80.000	53.10
Allowable pressure drop (kPa)	42.000	42.00
Viscosity (cP)	0.354	-
Density (kg/m³)	971.800	1160.00
Thermal conductivity (W/m°C)	670	0.21
Heat capacity (J/kg°C)	4196.300	1920.00

Table 3 below shows that even with 18% less pressure drop consumption, the surface area of the twisted tube heat exchanger is 26% smaller than the conventional STHE. It is also clear from Table 3 below that the overall heat transfer coefficient of the twisted tube shell-and-tube heat exchanger increases by 33.8%.

Table 3: Comparison to case study 1 (Barraza-Colon & Picon-Nunez, 2022).

Design results for case study 1		
	Shell and tube	Twisted tube
Surface area (m²)	15.200	11.200
Overall heat transfer coefficient (W/m°C)	108.600	145.400
Tube side pressure drop (kPa)	42.000	31.000
Tube side velocity (m/s)	0.900	0.700
Tube length (m)	4.700	8.600
Tube hydraulic diameter (m)	0.012	0.018
Tube count (-)	38.000	23.000
Shell diameter (m)	0.203	0.120

2.3.1.2 Case Study 2:

Within the case study of Ruengpayungsak (2019), it was considered to preheat raw molasses from 20°C to 39.6°C, using hot treated molasses at 140°C. Figure 9 below shows the variation of the viscosity of the fluid with temperature. It can be seen that the viscosity of the fluid decreases with 50% when the temperature increases with 10°C.

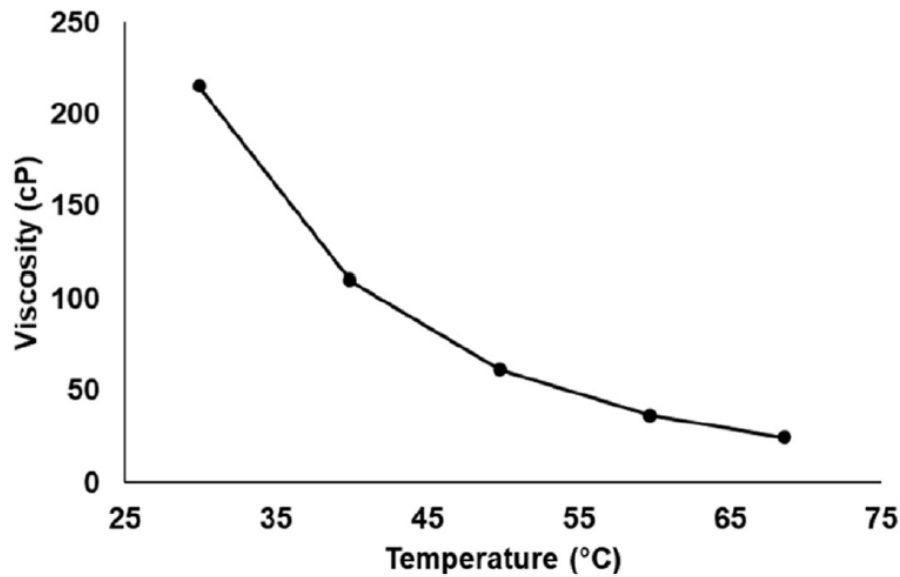


Figure 9: Temperature vs viscosity case study 2 (Barraza-Colon & Picon-Nunez, 2022).

The operating conditions and physical properties as well as the calculated output comparison to the case study “Thermohydraulic model for the design and performance analysis of twisted-tube heat exchangers”, can be seen in the tables below.

Table 4: Operating conditions and physical properties of case study 2 (Barraza-Colon & Picon-Nunez, 2022).

Operating conditions and physical properties of case study 2		
	Hot fluid (shell side)	Cold fluid (tube side)
Mass flow rate (kg/s)	0.445	0.445
Inlet temperature (°C)	140.000	20.000
Outlet temperature (°C)	120.300	39.600
Heat duty (kW)	18.690	-
Allowable pressure drop (kPa)	22.000	22.000
Density (kg/m ³)	1450.000	1450.000
Thermal conductivity (W/m°C)	0.300	0.300
Heat capacity (J/kg°C)	2140.000	2140.000

Table 5 below shows a reduction in heat transfer area of approximately 45% with respect to the use of a conventional plain tube heat exchanger. In the case of pressure drop, the twisted tube technology can meet the required heat duty with 15% less pressure drop occurring.

Table 5: Comparison of case study 2 (Barraza-Colon & Picon-Nunez, 2022).

Design results for case study 2		
	Shell and tube	Twisted tube
Surface area (m²)	8.000	4.400
Overall heat transfer coefficient (W/m²°C)	23.100	42.300
Tube side pressure drop (kPa)	22.700	19.300
Tube side velocity (m/s)	0.030	0.023
Tube length (m)	6.000	4.700
Tube hydraulic diameter (m)	0.027	0.027
Tube count (-)	16.000	11.000
Shell diameter (m)	0.203	0.120

Table 6 below shows the different tube design layouts used within the study, where different aspect ratios were used for the analysis of the study.

Table 6: Typical inner elliptical tube dimensions (Barraza-Colon & Picon-Nunez, 2022).

	a (mm)	b (mm)	R
#0	16.0	16.0	1.00
#1	19.8	9.4	2.10
#2	21.1	7.0	3.00
#3	21.8	5.8	3.75
#4	24.0	14.5	1.65
#5	28.0	9.0	3.10

2.3.2 Study Conclusion

From the results obtained within this study by Barraza-Colon and Picon-Nunez, it can be concluded that when comparing with plain conventional tube heat exchangers, the twisted tube geometry requires a smaller area and absorbs less pressure drop for the same heat duty to be generated. The twisted tube technology becomes more beneficial when in applications with high viscosity fluids where the surface area can be reduced with as much as 45% and pressure drop with 15%.

2.4 Experimental Research on Heat Transfer and Flow Resistance Properties in Spiral Twisted Tube Heat Exchanger (Dong, et al., 2020).

This section presets the outcomes and results obtained within the study (Dong, et al., 2020). This section focuses on the inputs and results used and obtained during this study. The section elaborates on the calculation methods and the deliverable factors used within the study.

Within this study, an experiment was conducted in order to gain the heat transfer and the flow resistance characteristics of a high-viscosity thermal oil and water within a twisted tube heat exchanger. The experiment conducted within this study, specified parameters in which the experiment was conducted. The parameters specified are:

- Mass fluxes of oil used, $1000 - 3000 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$.
- Mass fluxes of water used, $160 - 1200 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$
- Oil inlet temperature, 60°C
- Water inlet temperature, 15°C
- Shell side Reynolds number, $\text{Re} < 2050$
- Tube side Reynolds number, $12000 < \text{Re} < 45000$

The study outcomes calculated and/or measured was compared to that of PTSTHE's, as well as spiral-groove tube heat exchangers.

Dong (2020) found that the spiral twisted tube heat exchanger demonstrates a significant heat transfer enhancement effect in both laminar and turbulent flows when comparing to that of the plain tube heat exchanger and the spiral groove heat exchanger.

Dong's experimental data on the heat transfer and flow resistance of **spiral twisted tubes** led to a new correlation formula. When tested against conventional plain tubes, this formula proved highly accurate, with a deviation of less than $\pm 10\%$.

2.4.1 Study Calculated Outputs and Comparison.

2.4.1.1 Friction Factor Correlation Comparison

The study compared the calculated and measured friction factor correlation of the experimental study to four other completed studies. The studies used for the output comparison on the shell side can be seen in Table 7 below:

Table 7: Comparison studies, friction factor, shell side (Dong, et al., 2020).

The friction factor correlations and range		
Author	Working fluid	Correlation
Yang	Water Re: 600-55000	$f = 1.529Re^{-0.35} \left(\frac{A}{B}\right)^{1.686} \left(\frac{S}{D_h}\right)^{-0.366}$
Cheng	Water Re: 50-2000	$f = 5.448Re^{-0.627} \left(\frac{A}{B}\right)^{0.5}$
Khoshvaght-Aliabadi-1	Al ₂ O ₃ /water Re: 600-1800	$f = 2.1828Re^{-0.7551}(Pr)^{1.0296}(1 - \varphi)^{4.3461}$
Khoshvaght-Aliabadi-2	Air, Water, Engine Oil, & Ethylene Glycol Re: 600-1800	$f = 25.0105Re^{-0.7787} \left(\frac{P}{L}\right)^{-0.0653}$

The comparison to the literature studies regarding the friction factor correlation resulted in an average deviation of 35% on the shell side of the heat exchanger. The comparison can be seen in Figure 10 below.

Discussing the obtained and measured outputs reached within the graphs of Figure 10 below, presented the following outcomes.

- Within the first graph it can be seen that the experimental test bench study presented a measured friction factor deviation of up to 90% to that of the validation study.
- The second graph presented a maximum measured friction factor deviation of 50% to that of the validation study.
- Graph three also presented a maximum deviation of 50% to that of the validation study.
- Within the last graph, the maximum deviation measured for the shell side friction factor between the validation study and that of the experimental test bench presented to be 50%.

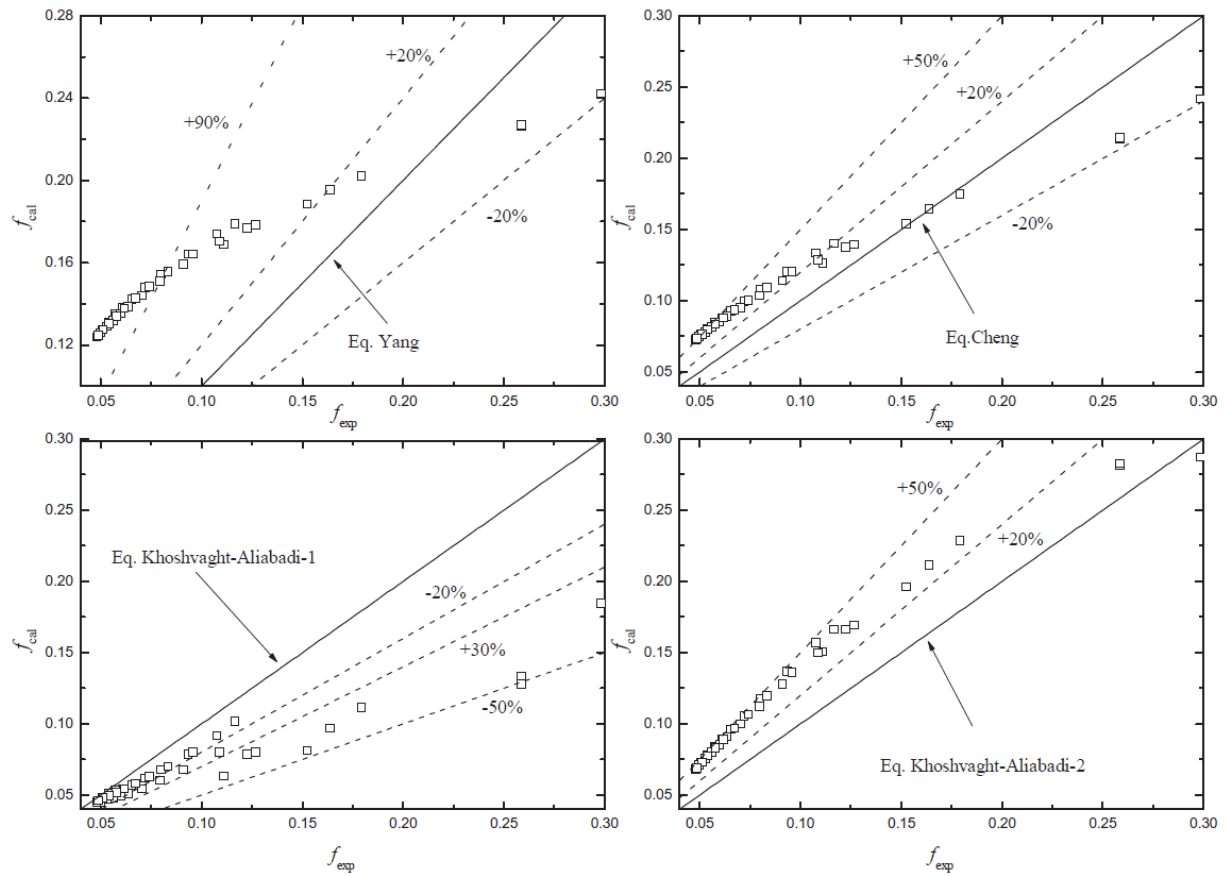


Figure 10: Comparison output to literature studies, friction factor, shell side (Dong, et al., 2020).

The studies used for the output comparison on the tube side can be seen in Table 8 below:

Table 8: Comparison studies, friction factor, tube side (Dong, et al., 2020).

The friction factor correlations and range.

Authors	Conditions	Medium	Correlations of f
Yang [24]	$600 < Re < 55,000$	water	$f = 1.529 Re^{-0.350} (A/B)^{1.686} (S/D_h)^{-0.366}$
Zhang [22]	$S:3500 < Re < 9000$	water	$f = 0.71496 Re^{0.07777} Pr^{-1.03974} (A/B)^{-0.76212} (S/D_h)^{-0.33393}$
Gao [23]	$5000 < Re < 20,000$	water	$f = 4.572 Re^{-0.521} (B/A)^{0.334} (S/D_h)^{-0.082}$
Tan [27]	$1000 < Re < 8000$	water	$f = 42.1244 Re^{-0.6576}$

On the tube side of the comparison to the literature studies, regarding the friction factor correlation, Xinyu Dong found the obtained results presented an average deviation of 30%. The comparison to the other mentioned literature studies can be seen in Figure 11 below.

The analysis of the obtained and measured outputs, as depicted in the graphs below, yielded the following outcomes.

- Graph one presented a maximum deviation for the tube side friction factor of 35% to that of the validation study.

- The second graph reached a maximum deviation of 10% for the tube side friction factor to that of the validation study.
- Within graph three, it can be seen that the experimental test bench reached a maximum deviation of 77% to that of the validation study for the tube side friction factor.
- The last graph presented a maximum tube side friction factor deviation of 40% to that of the validation study.

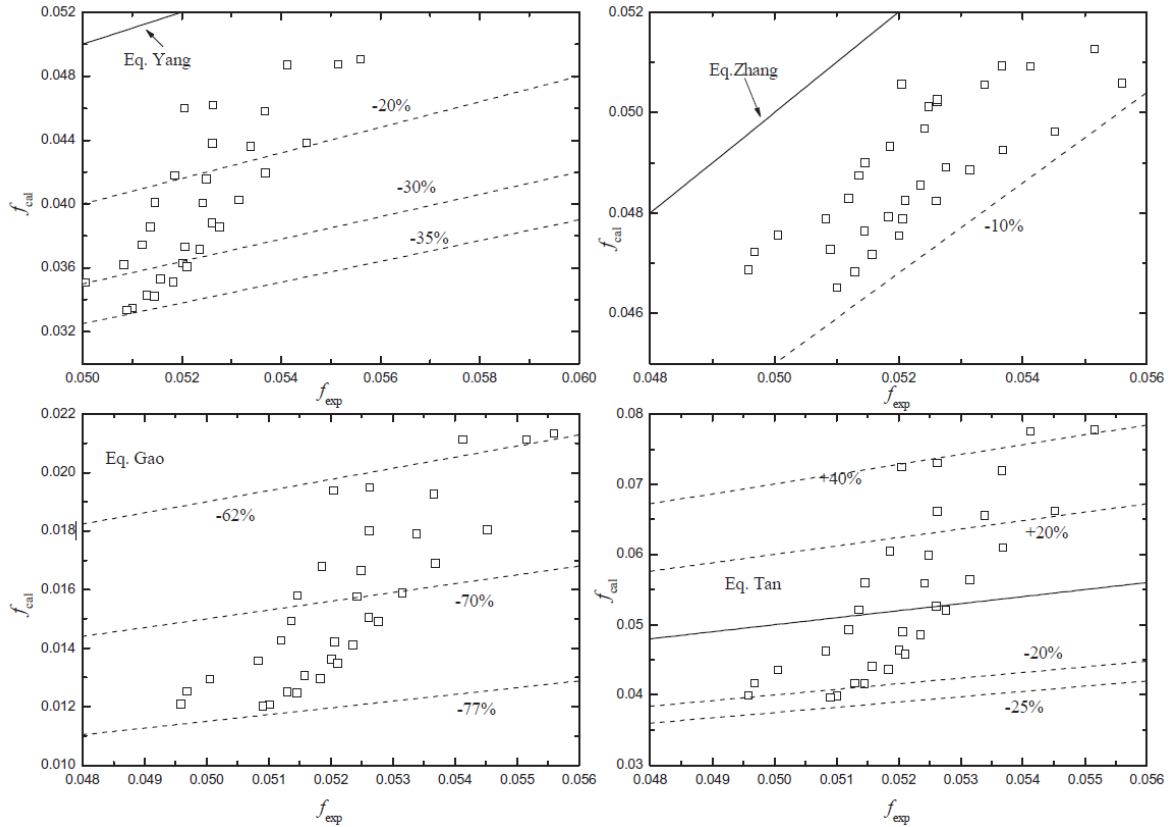


Figure 11: Comparison to literature studies, friction factor, tube side (Dong, et al., 2020).

2.4.1.2 Nusselt Number Correlation Comparison

The study also compared the calculated and measured Nusselt number of the experimental study to four other completed studies. The studies used for the Nusselt number correlation comparison on the shell side can be seen in Table 9 below:

Table 9: Comparison studies, Nusselt number, shell side (Dong, et al., 2020).

The heat transfer correlations and range		
Author	Working fluid	Correlation
Yang	Water Re: 600-55000	$Nu = 0.3496Re^{0.615}(Pr)^{0.33}\left(\frac{A}{B}\right)^{0.49}\left(\frac{S}{D_h}\right)^{-0.394}$

The heat transfer correlations and range		
Author	Working fluid	Correlation
Cheng	Water Re: 50-2000	$Nu = 0.205Re^{0.496}(Pr)^{0.432}\left(\frac{A}{B}\right)^{0.5}$
Khoshvaght-Aliabadi-1	Al ₂ O ₃ /water Re: 600-1800	$Nu = 0.0151Re^{0.7297}(Pr)^{0.6854}(1 - \varphi)^{0.0118}$
Khoshvaght-Aliabadi-2	Air, Water, Engine Oil, & Ethylene Glycol Re: 600-1800	$Nu = 0.0579Re^{0.5621}(Pr)^{0.6111}\left(\frac{P}{L}\right)^{0.0455}$

The comparison to the literature studies regarding the Nusselt number correlation on the shell side resulted in an average deviation of 35% on the shell side of the heat exchanger. The comparison can be seen in Figure 12 below (Dong, et al., 2020):

Discussing Figure 12 and the concluded results reached therein, presented the following outcomes.

- Graph one presented a maximum deviation of 35% for the shell side Nusselt number to that of the validation study mentioned within graph one.
- Within the second graph, it can be seen that the experimental test bench study presented a maximum deviation of 80% to that of the validation study mentioned within graph two.
- The third graph generated by means of the experimental test bench study presented a maximum deviation of 70% to that of the validation study mentioned within graph three.
- The last graph shows a maximum deviation value of 65% to the validation study mentioned within the last graph of Figure 12 below.

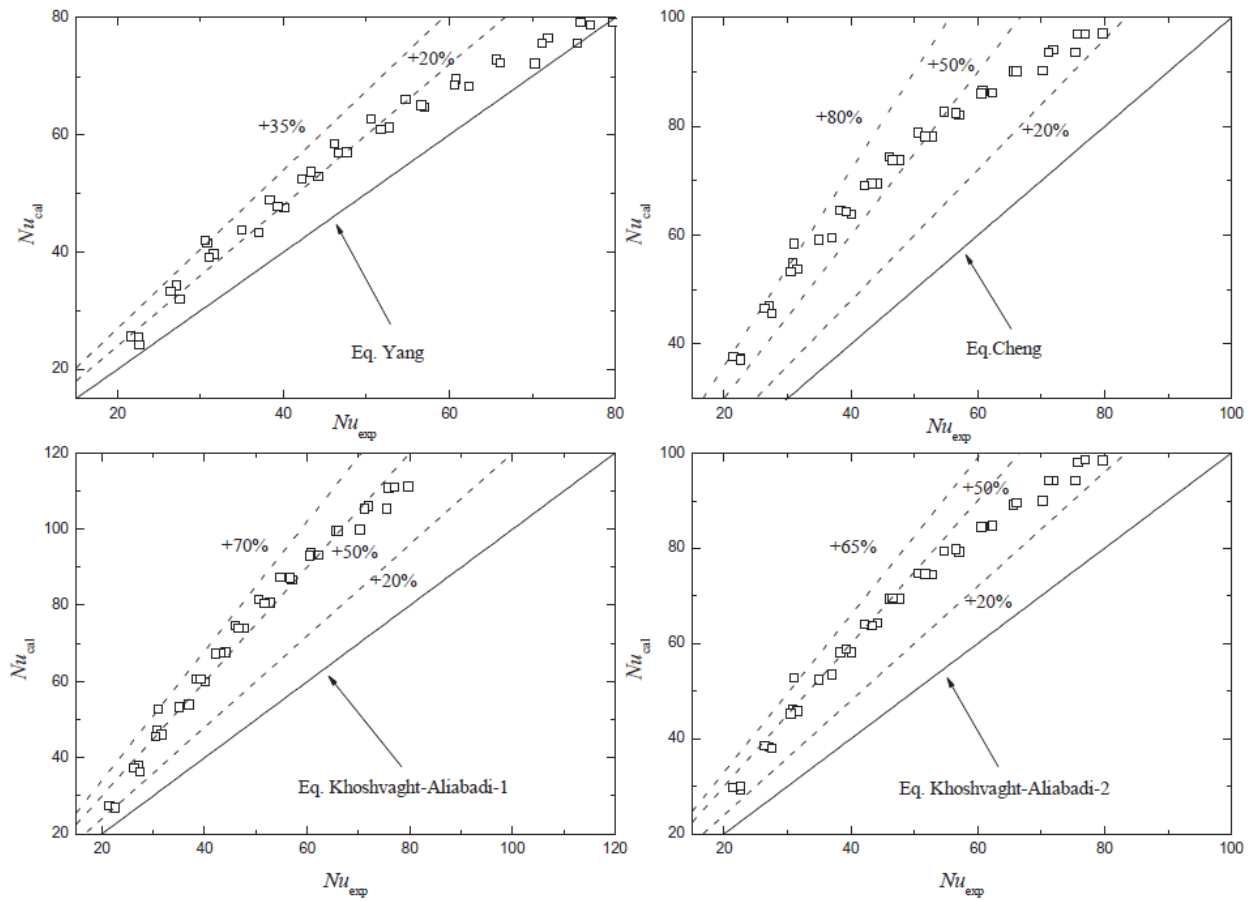


Figure 12: Comparison to literature studies, Nusselt number, shell side (Dong, et al., 2020).

The studies used for the output comparison on the tube side can be seen in Table 10 below:

Table 10: Comparison studies, Nusselt number, tubes side (Dong, et al., 2020).

The heat transfer correlations and range.

Authors	Conditions	Medium	Correlations of Nu
Yang [24]	$600 < Re < 55,000$	water	$Nu = 0.3496Re^{0.615}Pr^{1/3}(A/B)^{0.490}(S/D_h)^{-0.394}$
Gao [23]	$5000 < Re < 20,000$	water	$Nu = 0.034Re^{0.784}Pr^{0.333}(B/A)^{-5.90}(S/D_h)^{-0.165}$
Levlev [26]	$6000 < Re < 100,000$	air	$Nu = 0.019Re^{0.8}[1 + 0.547(S/A)^{0.83}]$
Tan [27]	$1000 < Re < 8000$ $Pr_M = 79$	water	$Nu = 0.1255Re^{0.67}Pr^{0.3}$

On the tube side of the comparison to the literature studies, regarding the Nusselt number correlation, the obtained results presented an average deviation of 26%. The comparison to the other mentioned literature studies can be seen in Figure 13 below.

Considering Figure 13, the following concluded results were reached.

- The first graph presented a maximum deviation of 10% regarding the tube side Nusselt number when comparing the measured outputs reached within the experimental test bench study to the calculated outputs reached within the validations study.

- Within graph two, it can be seen that the measured tube side Nusselt number generated a maximum deviation of 20% to that of the validation study output mentioned within graph two.
- The third graph presented a 40% tube side Nusselt number deviation to that of the validation study.
- The last graph reached a maximum deviational tube side Nusselt number outcome of 45% to that of the validation study mentioned within the last graph of Figure 13 below.

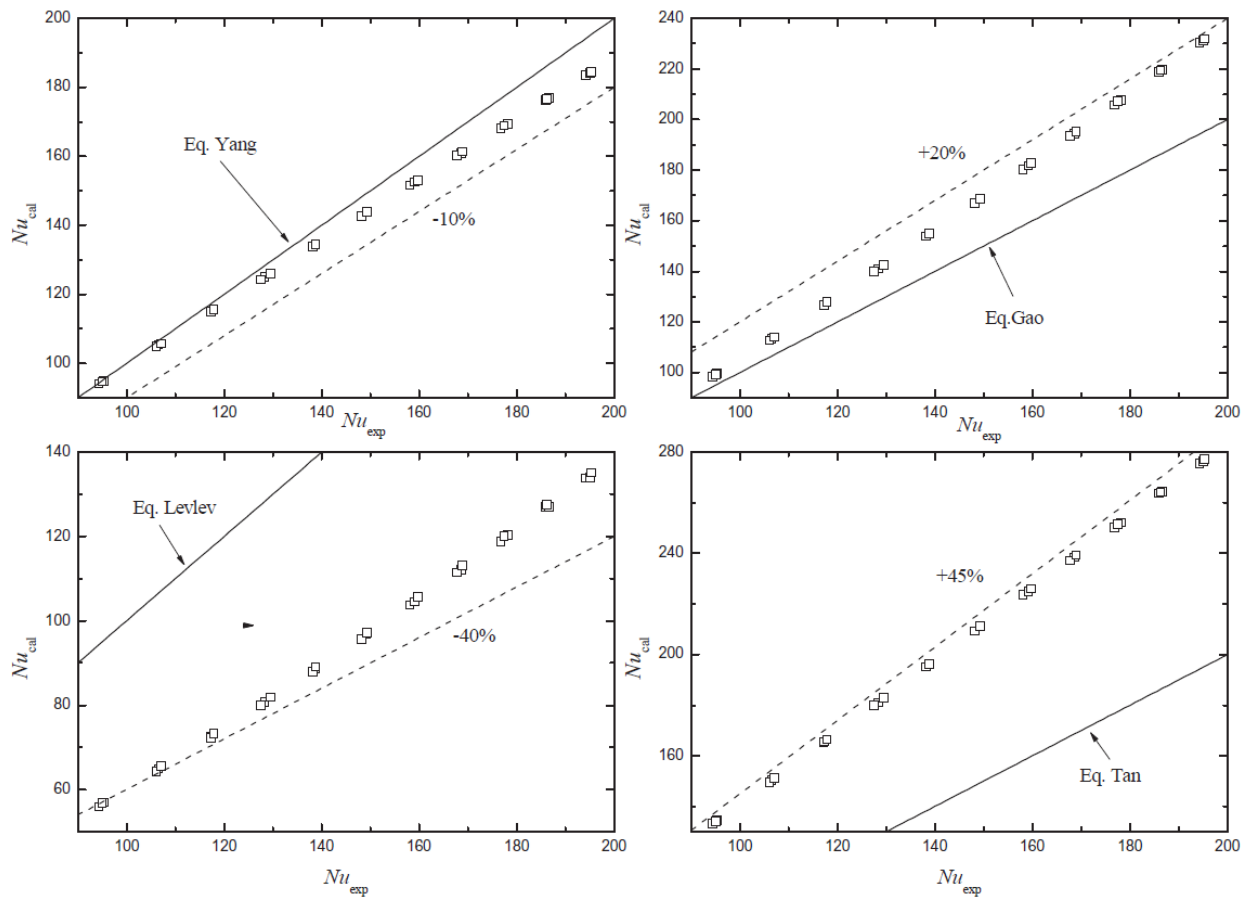


Figure 13: Comparison to literature studies, Nusselt number, tube side (Dong, et al., 2020).

2.4.2 Study Conclusion

Dong (2020) found that the enhanced heat transfer indices indicated that the spiral twisted tube heat exchanger in turbulent or laminar flow was better than that of the plain tube heat exchangers.

The experimental data were collected and measured in order to be compared with the values calculated via the literature correlations. The study found that the heat transfer and friction factor performance of spiral twisted tube heat exchanger could not be described by all the literature correlations.

Fittings of the friction factor and the Nusselt number on both sides of the spiral twisted tube in laminar and turbulent flow regions have been performed using the experiment data. The maximum fitting errors obtained of the friction factor for laminar and turbulent flows were $\pm 12\%$ and $\pm 1.5\%$, respectively, while the Nusselt number of the laminar and turbulent flows were $\pm 12\%$ and $\pm 10\%$, respectively.

These results obtained suggested that the fitting results were in good range for the experimental data, as a deviation of $\pm 20\%$ was considered acceptable in experimental projects. This experiment done provided a theoretical basis for the engineering modification and design of heat exchangers.

2.5 Convective Thermal-Hydraulic Performance in Twisted Tubes: From the Perspective of Symmetrical Cross-Sectional Shapes (Li, et al., 2023).

This section presets the outcomes and results obtained within the study (Li, et al., 2023). This section focuses on the inputs and results used and obtained during this study, and elaborates on the mathematical methods and deliverable factors used within the study.

Li explores the impact of intersecting surface shapes on thermal-fluid behaviour in twisted tubes under two conditions of equal area and equal perimeter of intersecting surfaces. Within this study, a 3D simulation method is used to examine the convection heat transfer characteristics of fluids with different viscosities in twisted tubes with typical symmetrical cross sections at low Reynold numbers, according to experimental verification.

Li found regular deformation of the helical twisted tube, due to induced secondary flow in the perpendicular direction of the main flow direction. The study found that this leads to bettering the convective heat transfer performance of the twisted tube design. The Reynolds number range used consisted of the Reynolds number being between 650 and 1550, using the operating fluid of water.

Li also found within the study that the twisted tube design, with equal cross-sectional perimeter, exhibits the topmost convection heat transfer performance, with a comprehensive performance evaluation factor of heat transfer of 1.85 – 1.87.

As for the equal cross-sectional perimeters with engine oil, Li found the twisted elliptical tubes to show the highest heat transfer, with a performance evaluation factor of 1.39 – 1.4 with the Reynolds number being below 1100.

Li also found that when the Reynolds number exceeds 1100, the twisted triangular tube displays the supreme heat transfer factor 1.33 – 1.37. It was found within the study that with the operating

fluid of water and engine oil, the twisted elliptical tube design presents the highest heat transfer factor.

2.5.1 Study Results and Validation

Li validated the obtained results from the 3D simulation with experimental data. The validation of the simulated output and the experimental obtained results can be seen in Figure 14 below.

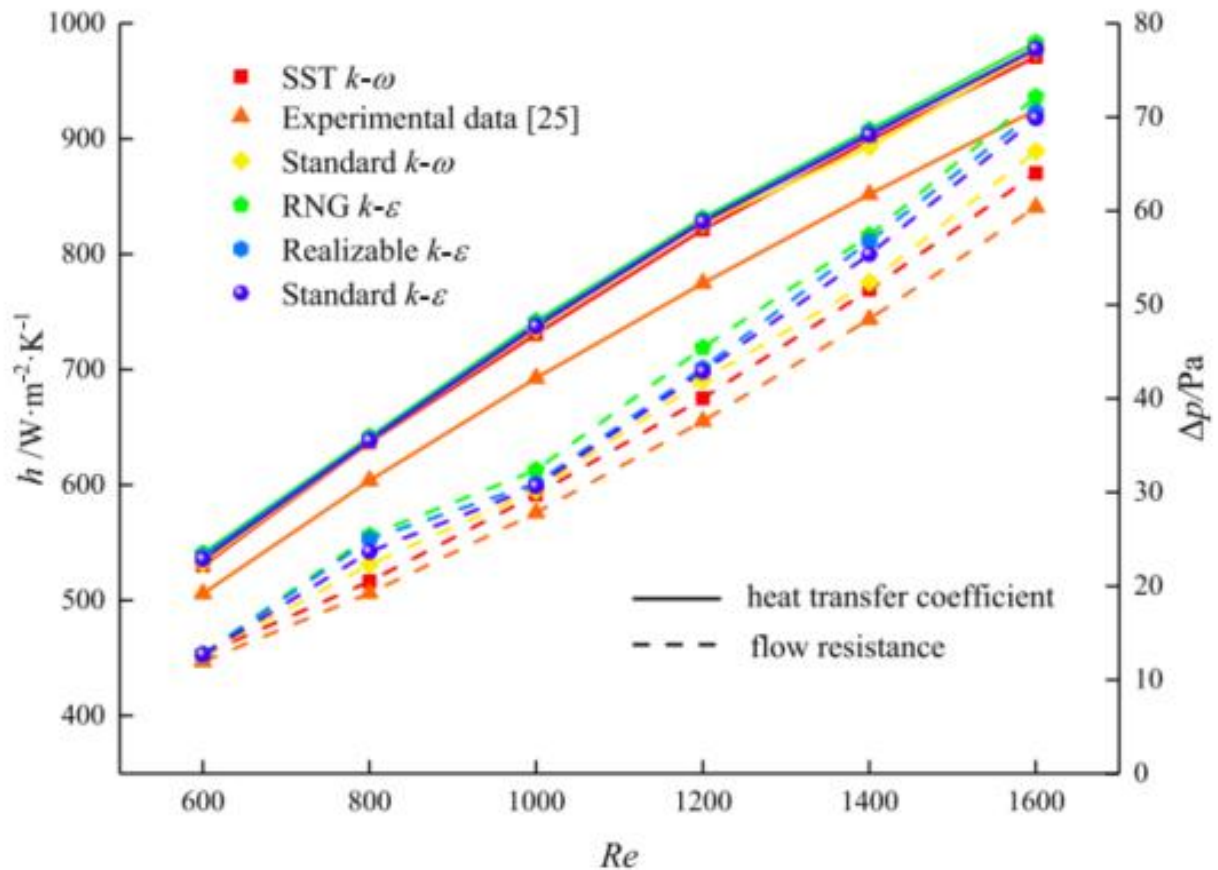


Figure 14: Validation of turbulence model through comparison with experimental data (Li, et al., 2023).

Within Figure 14 it can be seen that the study compared the different expressed turbulence models to experimental data reached within the validation study “Experimental study on convective heat transfer and flow resistance characteristics of water flow in twisted elliptical tubes” by Yang (2011).

Within Figure 14 above, it can be seen that as the tube side Reynolds number increases, the tube side heat transfer coefficient and pressure drop increases. It is also clear that the theoretically calculated output of the different turbulence models presented much higher heat transfer and pressure drop values than the values measured within the experimental study. The calculated turbulence models as well as the experimental measured results of the twisted tube design present the same gradient increases as the tube side Reynolds number increases.

Li also provided results of the flow characteristics and temperature distributions within the twisted tube design. The study mentioned that regardless of what fluid was specified, whether it be water or engine oil, the streamlines within the twisted tube design exhibit an obvious spiral flow. The generated streamline simulation can be seen in Figure 15 below.

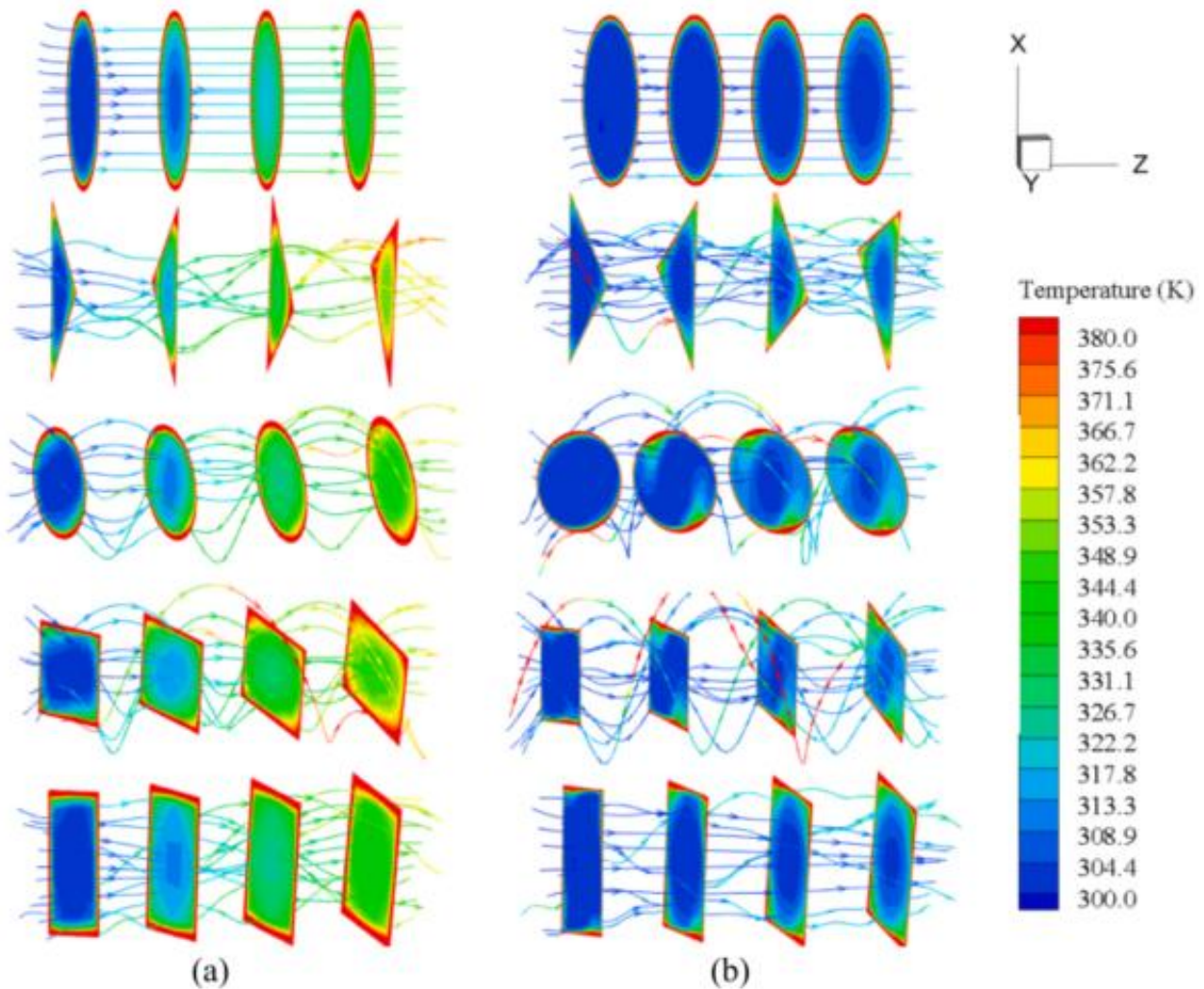


Figure 15: Streamline and temperature distribution on cross-sections in twisted tubes. a) equal cross-sectional perimeter, water as fluid. b) equal cross-sectional perimeter, engine oil as fluid (Li, et al., 2023).

Li explained the phenomenon as follows: the fluid only flows under the driving force of pressure, and there is no force perpendicular to the tube axis. As a result, no streamlines are shown. On the other hand, the fluid will be forced to produce a centrifugal force as it flows along the twisted tube due to the spiral deformation of the tubes. This will result in a spiral flow under the combined influence of centrifugal force and pressure, ignoring the effect of gravity. The fluid that is heated near the tube wall is thus mixed with the core fluid at a relatively low temperature at the centre of the twisted tube due to the scouring effect of the spiral flow. This lowers the mean temperature of the fluid that is adjacent to the tube wall, and somewhat magnifies the fluid temperature gradient.

Li concluded that the twisted tube helical deformation produces secondary flow, which eventually tend to be found in the sharp deformation domains, which includes the two ends of an ellipse's major axis, the four corners of a square or rectangle, and the three corners of a triangle. The secondary flows at the corners of the short side of the rectangle are merged and mixed due to the small distance between the corners at the ends of the short side, making the long side appear to have secondary flow only at its ends. The observed simulation within the study can be seen in Figure 16 below.

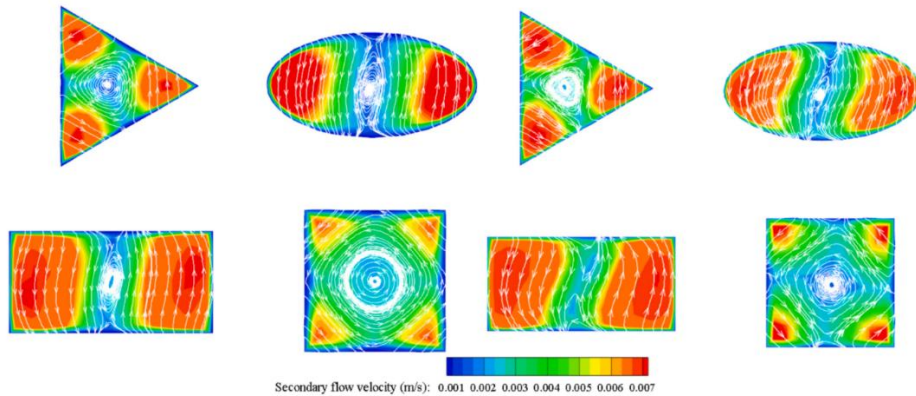


Figure 16: Secondary flow distribution on twisted tube cross-sections: (a) equal cross-sectional perimeter, water as fluid; (b) equal cross-sectional perimeter, engine oil as fluid (Li, et al., 2023).

The heat transfer coefficient results obtained within the study for all the different twisted tubes, regarding the Reynolds number, can be seen in Figure 17.

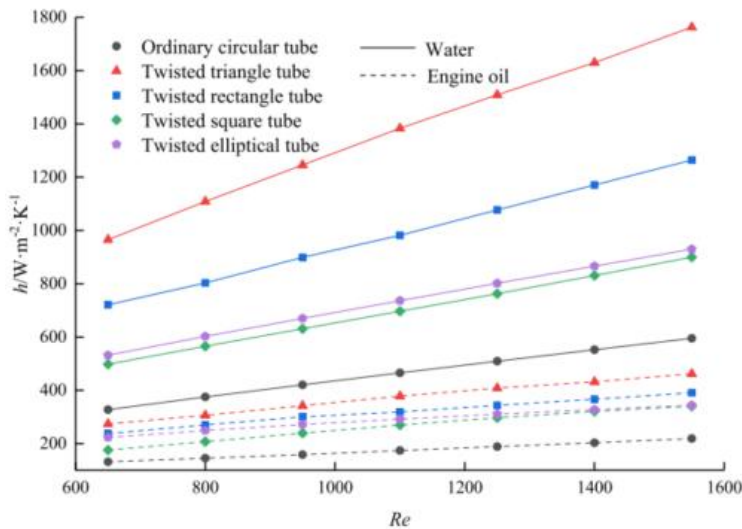


Figure 17: Variation of forced convective heat transfer coefficients with Re for the whole tubes: equal cross-sectional perimeter (Li, et al., 2023).

Considering Figure 17, it can be seen that the study evaluated the generated tube side heat transfer coefficient in correspondence with the tube side Reynolds number for different tube geometries with the same cross-sectional perimeter. The above Figure 17 expressed that the

twisted triangular tube generated the highest heat transfer coefficient as the tube side Reynolds number increases. It is also clear within Figure 17 that the difference between the generated outcomes of the different tube geometries for the operating fluid of water, are more independently structured than the outcomes generated for the operating fluid of oil.

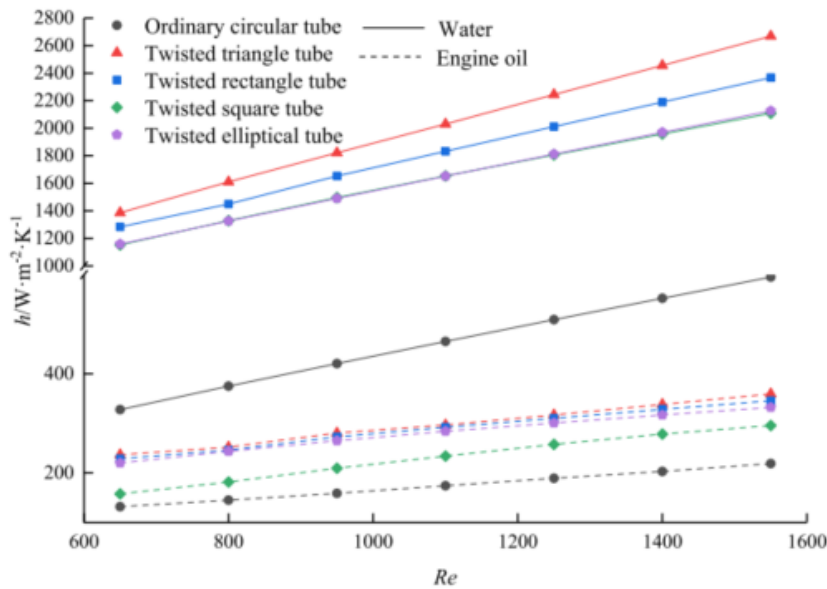


Figure 18: Variation of forced convective heat transfer coefficients with Re for the whole tubes: equal cross-sectional area (Li, et al., 2023).

Evaluating Figure 18 above, it can be seen that the study appraised the generated tube side heat transfer coefficient in correspondence with the tube side Reynolds number for different tube geometries with the same cross-sectional area. Figure 18 expressed that the twisted triangular tube again generated the highest heat transfer coefficient as the tube side Reynolds number increases. It is also clear within this figure that the difference between the generated outcomes of the different tube geometries for the operating fluid of water are more dependently structured, with the exception of the ordinary circular tube. The outcomes reached for the different tube geometries and the operating fluid of oil are closely related without any exception.

Evaluating the following graph, the study also compared the tube side pressure drop and the tube side Reynolds number for different tube geometries. The generated output can be seen in Figure 19 below.

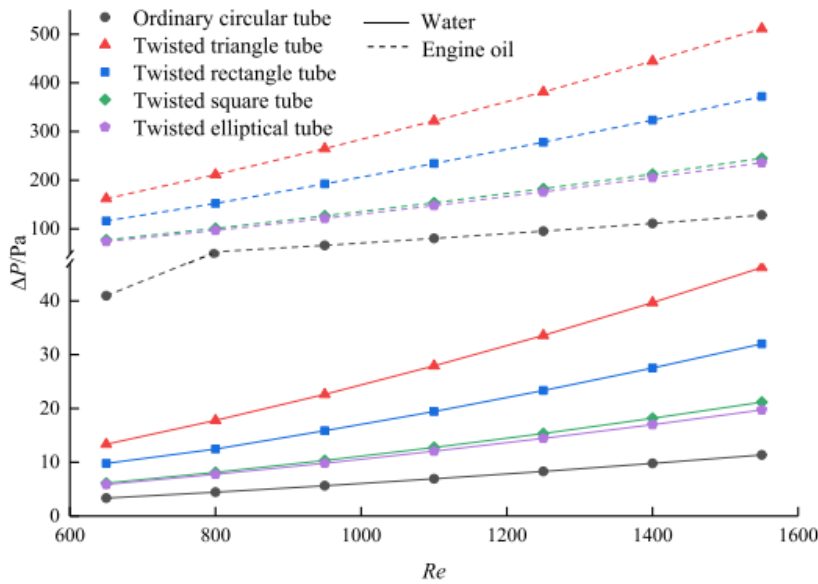


Figure 19: Variation of flow resistance with Re for the whole tubes: equal cross-sectional perimeter (Li, et al., 2023).

Analysing Figure 19, it can be seen that the study evaluated the generated tube side pressure drop in correspondence with the tube side Reynolds number for different tube geometries with the same cross-sectional perimeter. Figure 19 above expressed that the twisted triangular tube generated the highest pressure drop as the tube side Reynolds number increases with the operating fluid of oil. It is also clear within Figure 19 that the ordinary circular tube experienced a drastic increase in tube side pressure drop between the Reynolds number values of 650-800, whereafter the gradient increase remains constant for the operating fluid of oil.

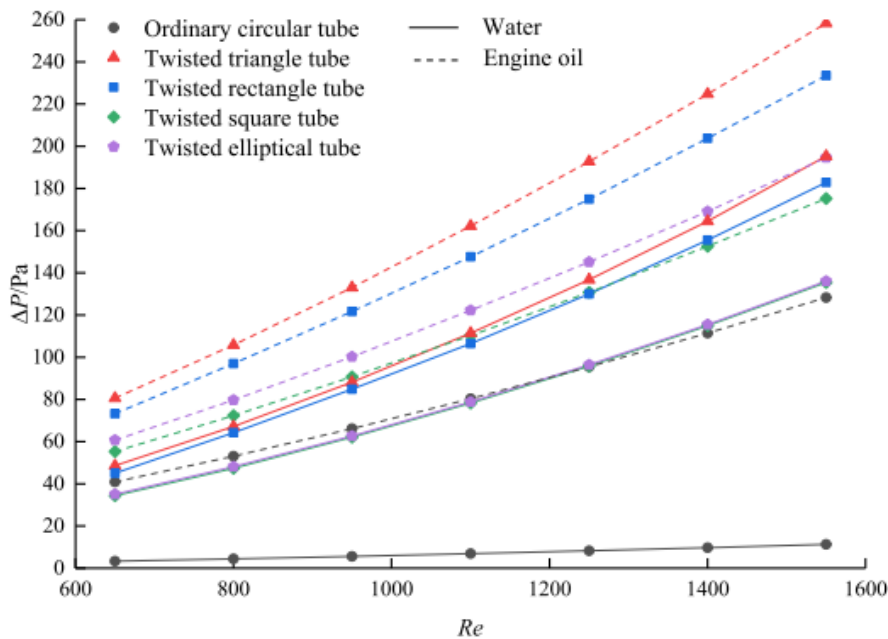


Figure 20: Variation of flow resistance with Re for the whole tubes: equal cross-sectional area (Li, et al., 2023).

Within Figure 20, it can be seen that the study evaluated the generated tube side pressure drop in correspondence with the tube side Reynolds number for different tube geometries with the same cross-sectional area. Figure 20 above expressed that the twisted triangular tube produced the highest pressure drop as the tube side Reynolds number increases. It is also clear within this figure, that the pressure drop for the ordinary circular tube with the operating fluid of water is the lowest when comparing to the other geometrical shapes and operating fluid of oil. It can also be seen within this figure that the difference between the generated outcomes of the different tube geometries for the operating fluid of water are structured in a pattern, with the exception of the ordinary circular tube.

The study also graphed the Performance Evaluation Factor (PEF) in accordance with the Reynolds number for the two fluids used: water and engine oil. The performance evaluation graphs can be seen in the figures below:

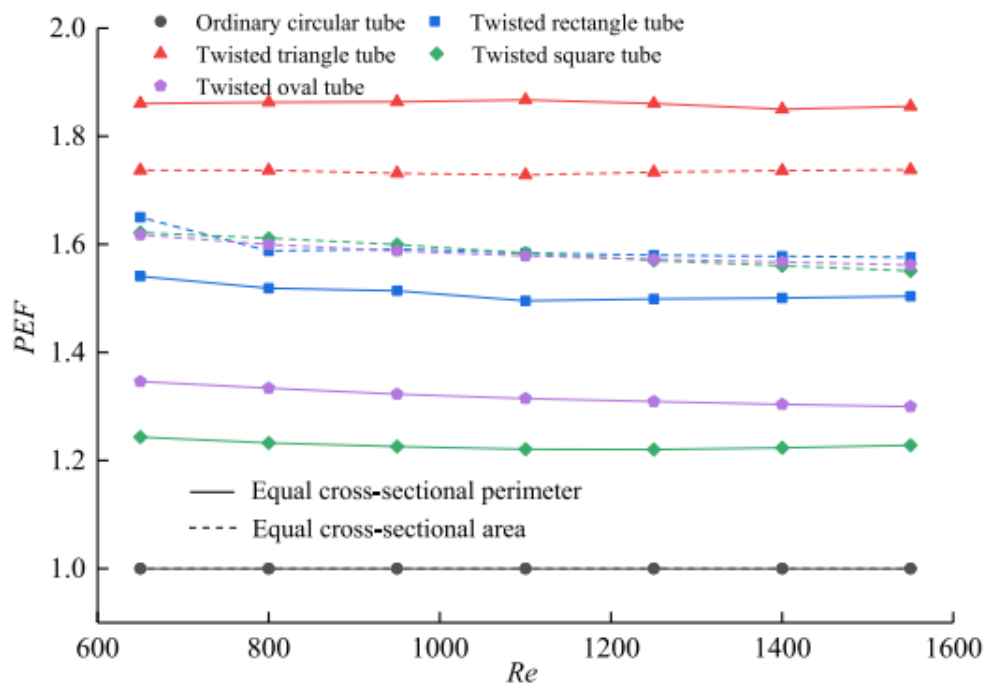


Figure 21: Variation of PEFs with Re for water inside the twisted tubes (Li, et al., 2023).

Evaluating Figure 21, it can be seen that all the tube geometries presented a vertically shaped output on the graphed axis considering the PEF and tube side Reynolds number. It can also be noted from Figure 21 that a difference in the PEF can be seen for tubes with the same geometrical form, but with the difference in perimeter and cross-sectional area. However, the exception is seen for the ordinary circular tube, whereas there is no difference within the output reached for a change in perimeter and cross-sectional area. Additionally, it is clear that for all geometrical shapes, the PEF is higher in tubes with the same cross-sectional area than for tubes with equal

perimeter; however the exception is for the twisted triangular tube, which generates the highest PEF, whereas the outcome presents to be different.

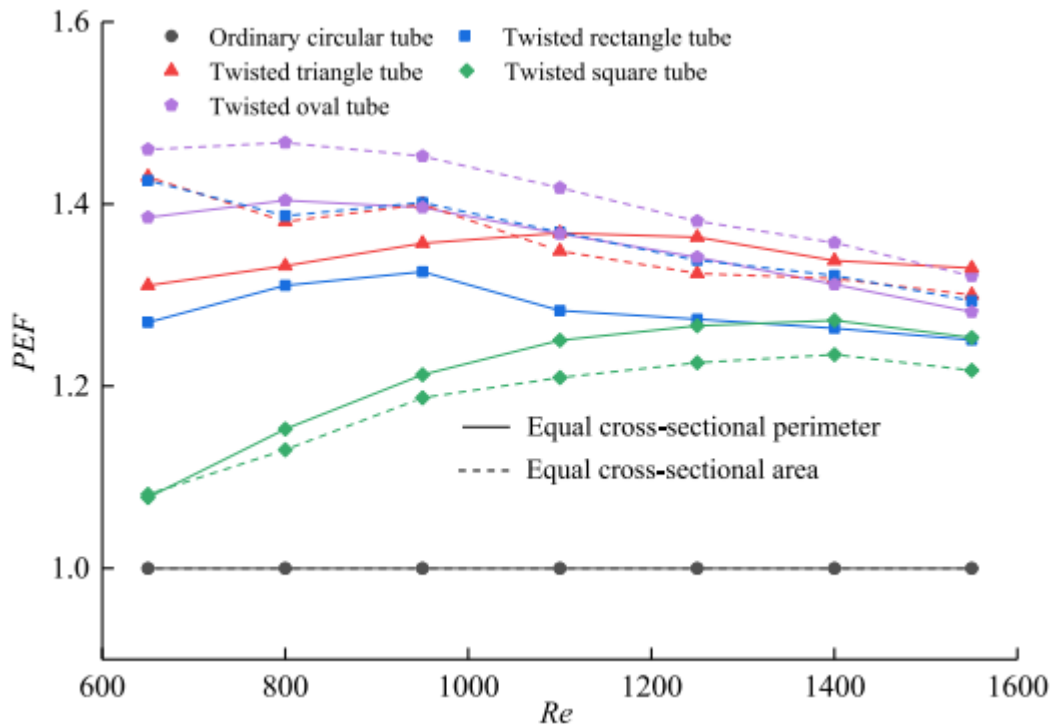


Figure 22: Variation of PEFs with Re for engine oil inside the twisted tubes (Li, et al., 2023).

Analysing Figure 22, it can be seen that the twisted oval tube with equal cross-sectional area presented the best PEF in correspondence to the tube side Reynolds number, considering the operating fluid of oil. It's also clear from Figure 22 that all the geometrical tube shapes trend to a coordinated commonly agreed PEF as the tube side Reynolds number increases. However, it can be concluded that the twisted square tube presented the lowest PEF for equal perimeter and cross-sectional area.

2.5.2 Study Conclusion

Li concluded that the secondary flow caused by the twisted tube spiral deformation grow, change, and ultimately converge in areas of sharp deformation, such as the major axis end region of the elliptical intersecting surface, the four corners of the square or rectangle cross section, and the three corners of the triangle cross section. The secondary flows facilitate the convective transfer of heat between the heat exchange medium and the twisted tube wall.

Among the twisted tube design with equal cross-sectional areas and cross-sectional perimeters, Li concluded that the twisted triangular tube has the best overall convection heat transfer performance within the range of Reynolds numbers from 650 to 1550 when the working fluid is water. This is followed by the twisted rectangular tube, twisted elliptical tube, twisted square tube,

and the ordinary circular tube. Furthermore, with a complete convection heat transfer performance assessment factor of 1.85 to 1.87, the twisted triangle tube manufactured by twisting the ordinary circular tube with an equal cross-sectional perimeter, exhibits the best overall convection heat transfer performance.

The twisted tube with elliptical cross section has the best comprehensive convection heat transfer performance when engine oil is used as the heat exchange fluid and the cross-sectional perimeters are identical. This is because when the Re is below 1100, the performance evaluation factor ranges from 1.39 to 1.40. The study therefore found that the twisted elliptical tube is followed in decreasing order of the comprehensive convection heat transfer performance generated by the triangular, rectangular, and square intersecting surfaces. The twisted elliptical tube, twisted rectangular tube, and twisted square tube have comprehensive performance evaluation factors that decrease from high to low. The twisted tube with a triangular intersecting surface has the best comprehensive convection heat transfer performance when the Reynolds number exceeds 1100, with a performance evaluation factor value of 1.33 to 1.37 (Li, et al., 2023).

With a comprehensive performance evaluation factor of 1.32 to 1.47, the twisted elliptical tube is the best type of tube for situations where the cross-sectional areas are equal. The comprehensive performance evaluation factors of the other twisted tubes are ranked from high to low as follows: twisted square tube, twisted rectangular tube, and twisted triangular tube (Li, et al., 2023).

2.6 Summary

2.6.1 Improved Heat Recovery Using Twisted Tube Exchanger Technology (Barraza-Colon & Picon-Nunez, 2022), Summary.

In this study, a thermohydraulic design model for twisted-tube heat exchangers is developed and validated, and its performance is compared to that of traditional shell-and-tube exchangers (STHE's). Through correlations for U , Nu , friction factors, surface area, and pressure drop on shell and tube sides, the model connects heat transfer, geometry, pressure drop, and operating conditions (Barraza-Colon & Picon-Nunez, 2022).

Barraza-Colon and Picon-Nunez also show that twisted tubes significantly enhance heat transfer and reduce required surface area and pressure drop at equal duty.

Key findings within the study include:

- For the same fluid velocity, twisted tubes improve thermo-hydraulic performance over plain tubes.

- Twisted tubes can achieve the same heat duty with less surface area and lower pressure drop, particularly beneficial for high viscosity fluids.
- Validation using, Danielsen (2009) showed 3.7% error, confirming that the model is high in accuracy.

Two industrial case studies demonstrated the following benefits:

- Case Study 1 (Epoxy Resin):
 - Surface area reduced 26%.
 - Overall heat transfer coefficient increased 33.8%.
 - Pressure drop reduced 18%.
- Case Study 2 (Molasses):
 - Surface area reduced 45%.
 - Overall heat transfer coefficient nearly doubled 83%.
 - Pressure drop reduced 15%.

Parametric plots show that higher Reynolds numbers increase heat transfer and reduce friction factor, while longer twist pitch decreases turbulence and thus reduces heat transfer. The model also illustrates how geometry (aspect ratio, tube pitch) affects heat duty and pressure drop, Barraza-Colon & Picon-Nunez, (2022).

Conclusion:

The twisted tube design outperformed the plain conventional tube STH's, especially for viscous fluids, reducing exchanger size and pressure drop while improving heat transfer.

2.6.2 Experimental Research on Heat Transfer and Flow Resistance Properties in Spiral Twisted Tube Heat Exchanger (Dong, et al., 2020), Summary.

Dong performs a controlled experimental comparison between a spiral twisted tube heat exchanger, a plain tube heat exchanger, and a spiral-groove tube heat exchanger, using high-viscosity oil and water under laminar (shell side) and turbulent (tube side) flow (Dong, et al., 2020).

Key observations Dong made during the study were the Reynolds number for the water flow of below 45,000 and 2050 for the oil flow. Calibrated flow loops and high-accuracy sensors with uncertainties under 3% were also used during the study, which ensured accurate results.

Key findings within the study:

- Spiral twisted tubes show strong heat transfer enhancement in both laminar and turbulent regimes compared to plain and spiral-groove tubes.
- The derived heat-transfer correlations showed a $\pm 10\%$ deviation error when compared to the predictions for the plain tube heat exchanger, which supports the reliability of the study.
- Test bench validations:
 - 0% deviation in mass flow.
 - 5% deviation in Nusselt number.
 - Less than 10% deviation in friction factor.
 - Less than 0.5% deviation in temperature.

Within the study, the friction factor and Nusselt number correlations were compared to multiple literature models:

- Shell side deviations: 35% average (up to 90% in some regions).
- Tube side deviations: 30% average (10–77% range).
- Nusselt number deviations:
 - Shell side: 35% average
 - Tube side: 26% average

Although deviations were larger in some regions, due to geometric differences and fluid properties, the results within this study consistently confirmed that spiral twisted tubes outperform conventional plain tube designs in heat transfer of STHE's, albeit with higher friction factors (Dong, et al., 2020).

Conclusion:

Spiral twisted tubes provide better heat transfer and acceptable pressure drop penalties in STHE's. It is also seen that experimental results support new correlations and validate their enhanced performance across a wide range of flow conditions.

2.6.3 Convective Thermal-Hydraulic Performance in Twisted Tubes: From the Perspective of Symmetrical Cross-Sectional Shapes (Li, et al., 2023), Summary.

Li (2023) investigates how intersecting surface shapes (elliptical, triangular, rectangular, square, circular) influence thermal fluid behaviour in twisted tubes. This study evaluates two different

scenarios, where the first is tubes with equal cross-sectional area and the second is tubes with equal cross-sectional perimeter (Li, et al., 2023).

The study employs 3D numerical simulation and validates results using experimental data. The fluids considered for this study include, water and engine oil, with low Reynolds numbers of 650 – 1550.

Key study assumptions:

- Fluid is steady, continuous, and incompressible.
- Inlet velocity is uniform.
- Physical properties may vary with temperature.
- Radiation heat transfer is neglected.

Li found that the simulations show strong spiral secondary flows caused by the geometrical twist within the twisted tube design. This occurrence caused mixing of the hot wall fluid with cooler core fluid within the tubes. It also enhanced the temperature gradients near the tube wall, which improved the convective heat transfer generated within the heat exchanger.

Li concluded that with water as the working fluid:

- For equal perimeter tubes:
The twisted triangular tube STHE has the highest heat transfer, with PEF between 1.85 – 1.87.
- For equal area tubes:
Heat transfer generated within the STHE hierarchy (from best to worst):
Triangular > Rectangular > Elliptical > Square > Circular

Li concluded that with engine oil as the working fluid:

- For equal perimeter tubes:
 - $Re < 1100$: Elliptical tube performs best with PEF between 1.39 – 1.40.
 - $Re > 1100$: Triangular tube performs best with PEF between 1.33 – 1.37.
- For equal area tubes:
 - The highest PEF generated was for the elliptical tube, with a PEF between 1.32 – 1.47.

- Heat transfer generated within the STHE hierarchy (from best to worst):
Elliptical > Square > Rectangular > Triangular

Li also found the following pressure drop occurrences during the analysis:

- The twisted geometries increase the pressure drop alongside the heat transfer.
- Triangular tubes consistently show the highest pressure drop.
- Circular tubes show the lowest pressure drop, especially with water.
- Oil shows smaller differences between geometries due to its higher viscosity.

Conclusion:

Spiral flow within twisted tubes induces strong secondary flow, greatly enhancing convective heat transfer. Li also concluded that the tube geometry significantly affects heat transfer, pressure drop, and PEF. However, within the study, the best performers during the analysis were:

- Water (equal perimeter): Twisted triangular tube
- Water (equal area): Twisted triangular tube
- Oil ($Re < 1100$): Twisted elliptical tube
- Oil ($Re > 1100$): Twisted triangular tube
- Oil (equal area): Twisted elliptical tube

Overall, the triangular, elliptical, and rectangular tubes offer consistent improvements over circular tubes.

CHAPTER 3 EVALUATION OF A PLAIN TUBE SHELL-AND-TUBE HEAT EXCHANGER

3.1 Introduction and Overview

This chapter is based on the factors that, considering the design of the PTSTHE, affect the overall efficiency and performance of the heat exchanger. Within this chapter, the factors will be analysed and validated for the PTSTHE design.

The PTSTHE design, Figure 23, is a commonly used design within the petrochemical industry of South Africa. Many availabilities present themselves when in search of an operating plain tube heat exchanger within the industry of interest. The design has shown great potential by giving important feedback on how well it performs and its ability to operate in the petrochemical industries. The design's important properties to consider and calculate have been identified and will be analysed in detail in order to provide an outcome of measurement regarding the capabilities and performance.

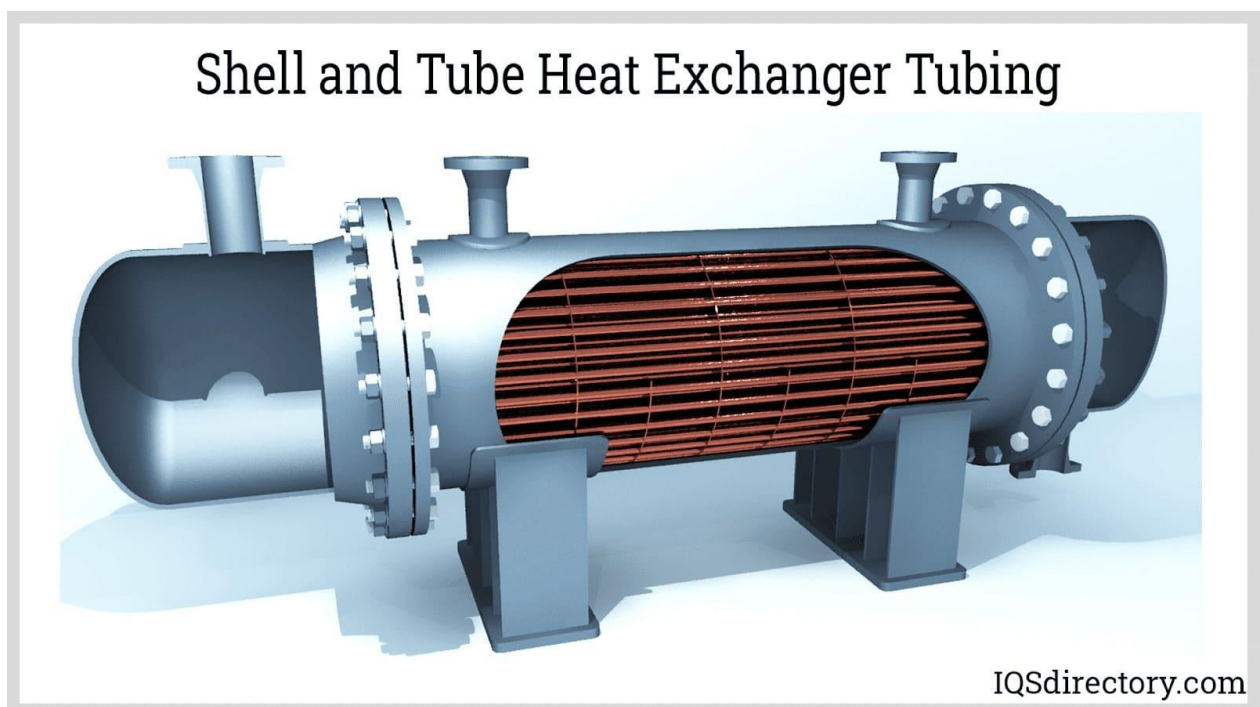


Figure 23: Plain tube shell-and-tube heat exchanger (Idiamart, 2014).

3.2 Plain Tube Shell-and-Tube Heat Exchanger Theory

The process that will be used to calculate the PTSTHE's performance and output has been identified and will be done using the following few steps and calculations.

The overall heat transfer coefficient U when considering the design of the PTSTHE is calculated by means of the following formula (Mills, 1999):

$$U = \frac{1}{\left(\frac{1}{h_i} + \frac{d_i \left(\ln \left(\frac{d_o}{d_i} \right) \right)}{k} + \frac{d_i}{h_o d_o} \right)} \quad 1$$

In the above-mentioned equation,

- d_i = Inside diameter of the tubes.
- d_o = Outside diameter of the tubes.
- k = Specific heat conductivity of the specified operating fluid.

3.2.1 Tube Side Heat Transfer Coefficient

The tube side heat transfer coefficients at the inlet and outlet of the PTSTHE can be determined by means of the following equations (Subramanian, 2024);

$$h = \frac{(C \cdot Re^n \cdot Pr^{1/3} \cdot k)}{d_o} \quad 2$$

Where:

- Re = Reynolds number.
- Pr = Prandtl number.
- d_o = Tube outside diameter (m).
- C = Constant
- k = Heat conductivity (W/mK).
- n = Constant

The constants n and C can be defined as seen in the tables below (Subramanian, 2024).

Table 11: Values for constant C (Subramanian, 2024).

S_n/D_o	1.25	1.5	2.0	3.0
S_p/D_o				
1.25	0.386	0.305	0.111	0.0703
1.50	0.407	0.278	0.112	0.0753
2.00	0.464	0.332	0.254	0.220
3.00	0.322	0.396	0.415	0.317

Table 12: Values for constant n (Subramanian, 2024).

S_n/D_o	1.25	1.5	2.0	3.0
S_p/D_o				
1.25	0.592	0.608	0.704	0.752
1.50	0.586	0.620	0.702	0.744
2.00	0.570	0.602	0.632	0.648
3.00	0.601	0.584	0.581	0.608

Where:

- S_n = Horizontal tube pitch (m)
- S_p = Vertical tube pitch (m)

The inlet and outlet Prandtl number can be calculated using the following equation (Karno & Ajib, 2006):

$$Pr = \frac{\mu \cdot C_p}{k}$$

3

Where:

- μ = The fluid viscosity (Pa-s).
- C_p = The specific heat capacity of the fluid (J/kg-K).
- k = The heat conductivity of the fluid (W/m-K).

However, to increase the calculation accuracy of the tube side heat transfer coefficient, the heat exchanger specific calculation was done using the average calculated fluid properties. This approach accounts for non-linearities, as it utilises the average fluid properties which is more representative for the fluid within the tube. These properties for the tube side heat transfer coefficient calculation are:

- Reynolds number
- Prandtl number
- Heat conductivity

The average for each of these properties were calculated using the formula below:

$$Average = \frac{Outlet - Inlet}{2}$$

4

3.2.2 Shell Side Heat Transfer Coefficient

The shell-side inlet and outlet heat transfer coefficient can be calculated using the following equation (Karno & Ajib, 2006):

$$h = \frac{k \cdot Nu}{\left(\frac{\pi}{2}\right) \cdot d_i}$$

5

Where:

- k = Heat conductivity (W/m-K).
- d_i = Tube inside diameter (m).
- Nu = The average Nusselt number.

The average Nusselt number can be calculated using the following formulas (Karno & Ajib, 2006);

$$Nu = f_A \cdot Nu_1$$

6

Where:

- f_A = The arrangement factor.
- Nu_1 = The Nusselt number.

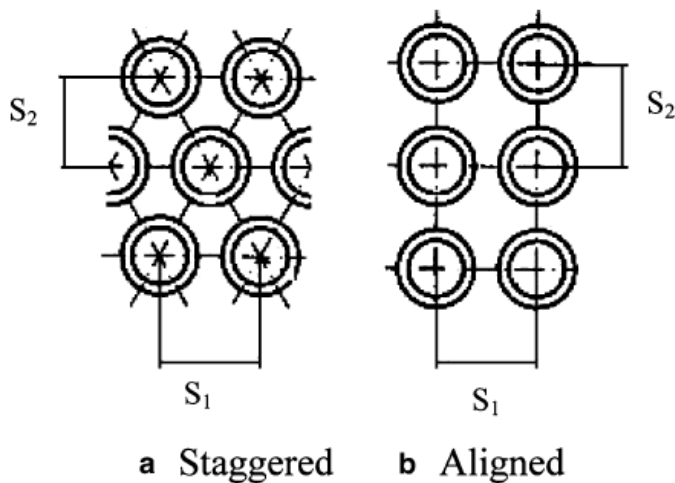


Figure 24: Different tube arrangements in shell-and-tube heat exchangers (Karno & Ajib, 2006).

The arrangement factor, as expressed in Figure 24 above, is dependent on the longitudinal and transverse tube spacing, and can be calculated by means of the following equation when the

tubes are aligned (Karno & Ajib, 2006). The formulas expressed below is not limited to the number of tubes.

$$f_{A,aligned} = 1 + \frac{0.7 \left(\frac{b}{a} - 0.3 \right)}{\psi^{1.5} \cdot \left(\frac{b}{a} + 0.7 \right)^2} \quad 7$$

Where the longitudinal and transverse tube spacing ratios can be defined as:

$$a = \frac{S_1}{d_o} \quad 8$$

$$b = \frac{S_2}{d_o} \quad 9$$

If the tubes are staggered, the arrangement factor can be calculated as (Karno & Ajib, 2006):

$$f_{A,stag} = 1 + \frac{2}{3b} \quad 10$$

The void fraction can be calculated by means of the following (Karno & Ajib, 2006):

$$\psi = 1 - \frac{\pi}{4a} \text{ for } b \geq 1 \quad 11$$

$$\psi = 1 - \frac{\pi}{4ab} \text{ for } b < 1 \quad 12$$

The Nusselt number expressed at the inlet and outlet can be calculated using the equations listed below (Gnielinski, 1993):

$$Nu_1 = 0.3 + \sqrt{Nu_{1,lam}^2 + Nu_{1,turb}^2} \quad 13$$

Where:

- $Nu_{1,turb}$ = The turbulent Nusselt number.
- $Nu_{1,lam}$ = The laminar Nusselt number.

The laminar and turbulent Nusselt numbers can be calculated using the following equations (Karno & Ajib, 2006), with the laminar flow in tubes being when $Re < 2100$, and turbulent flow when $Re > 2100$.

$$Nu_{1,lam} = 0.664 \cdot \sqrt{Re} \cdot \sqrt{3}Pr \quad 14$$

$$Nu_{1,turb,i} = \frac{0.037 \cdot (Re)^{0.8} \cdot Pr}{1 + 2.443 \cdot (Re)^{-0.1} \cdot (Pr^{2/3} - 1)} \quad 15$$

The Prandtl number, Pr , expressed in the above equations, can be calculated using the same formulas as detailed for the tube side in equation 3.

The Reynolds number, Re , for the inlet and outlet, can be calculated as (Karno & Ajib, 2006):

$$Re = \frac{\left(\frac{w \cdot \frac{\pi}{2} \cdot d_i}{v} \right)}{\psi} \quad 16$$

However, to also increase the calculation accuracy of the shell side heat transfer coefficient, the heat exchanger specific calculation was done using the average calculated fluid properties, as this approach better accounts for non-linearities. These properties for the shell side heat transfer coefficient calculation are:

- Nusselt number
- Heat conductivity

The average for each of these properties were calculated using equation 4.

3.2.3 Pressure Drop

Pressure drop is a factor within STHE design which can be affected by many other variable factors. The pressure drop refers to the loss of pressure throughout the cycle that a fluid takes within the STHE. It refers to the total pressure difference between the inlet and outlet of the STHE. The pressure drop of the PTSTHE can be calculated using the following formulas:

3.2.3.1 Tube Side Pressure Drop.

The total pressure drop generated on the tube side within the STHE can be calculated by means of the formula of Sinnot (1999):

$$\Delta P_{tube} = n_p \cdot \left(8j_f \cdot \left(\frac{L_t}{d_i} \right) \cdot \left(\frac{\mu}{\mu_w} \right)^{-m} + 2.5 \right) \cdot \left(\frac{\rho \cdot w_t^2}{2} \right) \quad 17$$

Where:

- n_p = The number of tube side passes.

- L_t = The tube length.
- d_i = The tube inside diameter.
- μ = The tube side fluid's viscosity at bulk temperature.
- μ_w = The tube side fluid's viscosity at wall temperature.
- m = Factor linked to laminar and turbulent flow.
- ρ = Tube side fluid density.
- w_t = Tube side fluid's flow velocity.
- j_f = Tube side friction factor obtained from Figure 25 below.

Within this study, it is assumed that the tubes used are smooth tubes, with an operating Reynolds number < 100000 , thus Blasius formula can be used to calculate the friction factor (Coetzee, 2015);

Laminar Flow

$$f = \frac{64}{Re}$$

18

Turbulent Flow

$$f = \frac{0.316}{Re^{0.25}}$$

19

Figure 25 below indicates the graphical selection of the tube side friction factor in accordance with the tube side Reynolds number (Sinnot, 1999):

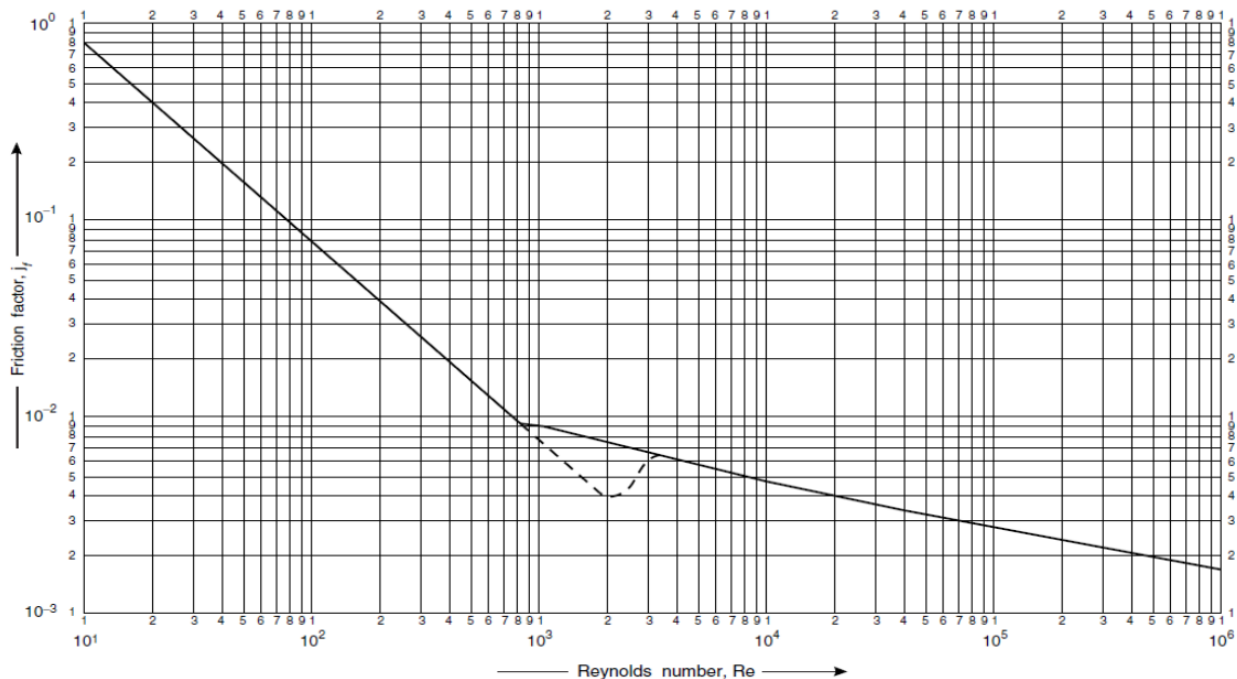


Figure 25: Tube side friction factors (Sinnot, 1999).

The factor 'm' mentioned within the formula above, which is linked to laminar and turbulent flow, can be expressed and defined as stated below (Sinnot, 1999):

Laminar flow $m = 0.25$ $Re \leq 2100$ 20

Turbulent flow $m = 0.14$ $Re > 2100$ 21

The total length of flow within the tubes, L_t , can be calculated by means of the following equation (Sinnot, 1999):

$$L_t = L \cdot (n_p) \quad 22$$

Where:

- L = The length of the tubes (m).
- n_p = The number of tube passes, which can be calculated as (Sinnot, 1999):

$$n_p = \frac{N_t}{N_{tp}} \quad 23$$

Where:

- N_t = The number of tubes.
- N_{tp} = The number of tubes per pass.

N_{tp} can be calculated by means of using the following formula (Sinnot, 1999):

$$N_{tp} = \frac{A_t}{\left(\frac{\pi \cdot (d_i)^2}{4}\right)} \quad 24$$

Where:

$$A_t = 2 \cdot \pi \cdot \frac{d_i}{2} \cdot L_{tube} \cdot n_t \quad 25$$

3.2.3.2 Shell Side Pressure Drop.

The total pressure drop generated on the shell side of the plain tube heat exchanger can be calculated and expressed using the following equation (Saffarian, et al., 2019):

$$\Delta P_{Shell} = f \cdot \left(\frac{D_s}{D_e}\right) \cdot (N_B + 1) \cdot \left(\frac{\rho V_s^2}{2}\right) \quad 26$$

Where:

- f = Friction factor.
- V_s = Flow velocity on the shell side (m/s²).
- D_s = Inside diameter of the shell (m).
- N_B = Number of baffles.
- ρ = Density of the shell-side fluid (kg/m³).
- D_e = Equivalent diameter (m).

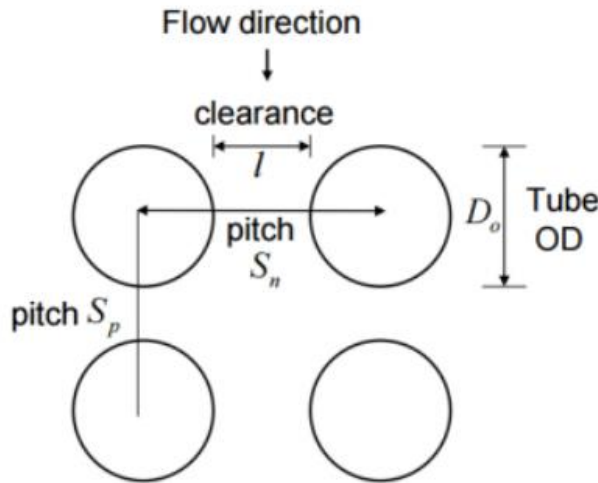


Figure 26: Tube clearance and pitch (Arveng Training, 2025).

The equivalent diameter, D_e , can be calculated by using the following equation, (Subramanian, 2024):

$$D_e = \frac{4 \left(C_p \cdot pitch^2 - \left(\frac{\pi \cdot d_o^2}{4} \right) \right)}{\pi \cdot d_o} \quad 27$$

Where:

- $pitch$ = Tube pitch (m), Figure 26.

And C_p can be defined as (Subramanian, 2024):

Square Tube Pitch	$C_p = 1$	28
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Triangular Tube Pitch	$C_p = 0.86$	29
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Considering the smooth tube assumption within this thesis, the friction factor can be calculated as defined within the equation below, where the factor is based on the Reynolds number. In this

case the Reynolds number can be calculated when considering the pressure drop within the STHE, as (Subramanian, 2024):

Turbulent Flow
$$f = \frac{0.316}{Re^{0.25}} \quad 30$$

Laminar Flow
$$f = \frac{64}{Re} \quad 31$$

Reynolds Number
$$Re = \frac{D_e \cdot G_s}{\mu} \quad 32$$

Figure 27 below indicates the graphical selection of the shell side friction factor in accordance with the shell side Reynolds number considering the shell diameter (Serth, 2007).

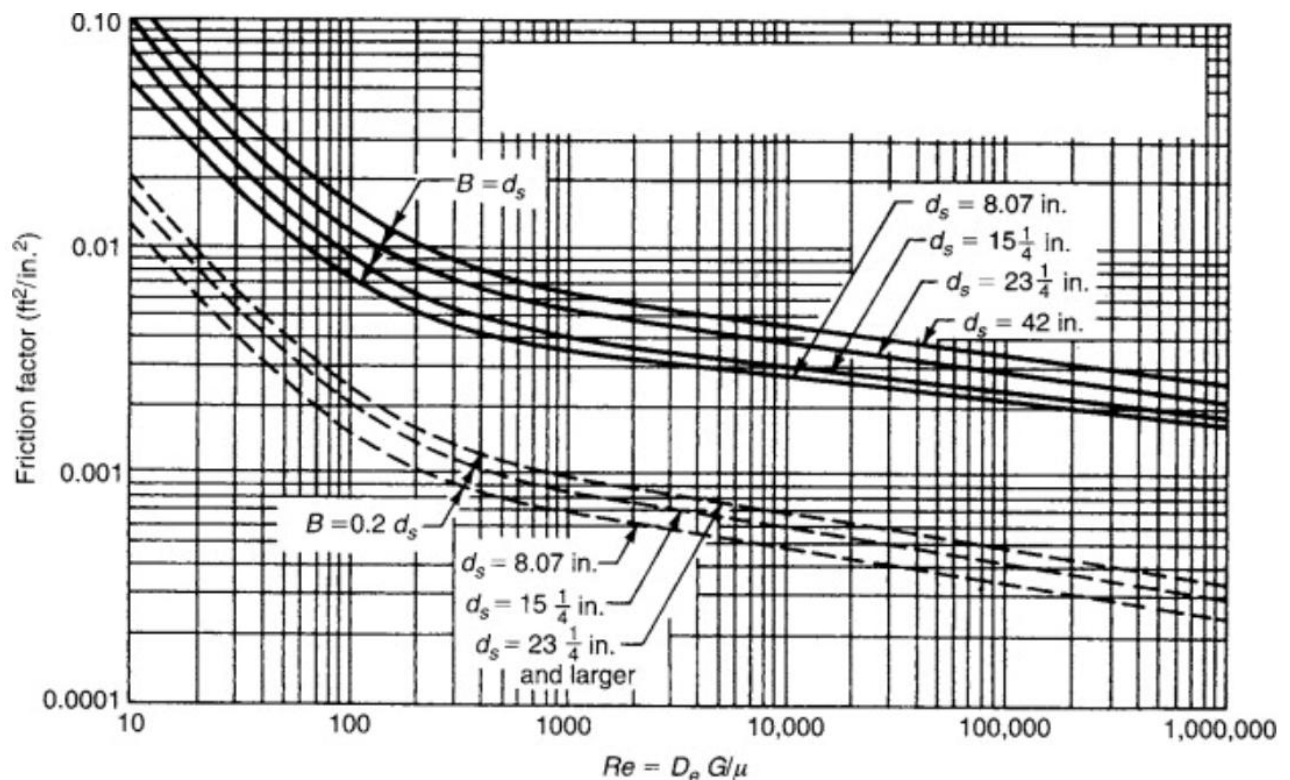


Figure 27: Shell side friction factor in accordance with shell side Reynolds number (Serth, 2007).

3.2.4 Effectiveness

Calculating the effectiveness of the STHE is a very important factor within the petrochemical industry. Heat exchangers play a crucial role in the energy-intensive processes of the petrochemical industry, such as distillation, cracking, and reforming. An efficient heat exchanger can thus help reduce energy consumption by effectively transferring heat between process streams, thereby improving overall plant efficiency.

When considering the STHE effectiveness analysis, if the fluid inlet and outlet temperatures are specified, known, or can be determined by means of basic energy balance equations, the NTU method can be used to calculate the effectiveness of the STHE. The calculation process can be seen within the following formulas, where the overall efficiency formulation has been analysed and formulated according to the NTU method below (Mills, 1999) (Fakheri, 2014):

$$\begin{array}{ll} \text{Maximum} & Q_{max} = C_{min}(T_{H,in} - T_{C,in}) \\ \text{Energy} & \end{array} \quad 33$$

$$\begin{array}{ll} \text{Tube Side} & Q_{tube} = C_c(T_{C,out} - T_{C,in}) \\ \text{Energy} & \end{array} \quad 34$$

$$\begin{array}{ll} \text{Shell Side} & Q_{shell} = C_H(T_{H,in} - T_{H,out}) \\ \text{Energy} & \end{array} \quad 35$$

$$\begin{array}{ll} \text{Overall} & \\ \text{Efficiency} & eff = \frac{\tanh \cdot \left(\frac{N_{tu}}{2} \cdot (1 + m_1 \cdot R_c^{n_1})^{\frac{1}{n_1}} \right)}{\frac{N_{tu}}{2} \cdot (1 + m_1 \cdot R_c^{n_1})^{\frac{1}{n_1}}} \end{array} \quad 36$$

$$\begin{array}{ll} \text{One Shell} & \\ \text{Side Pass} & \varepsilon = \varepsilon_1 = 2 \left[1 + R_c + (1 + R_c^2)^{0.5} \left(\frac{1 + \exp \left[-N_{tu} \cdot (1 + R_c^2)^{0.5} \right]}{1 - \exp \left[-N_{tu} \cdot (1 + R_c^2)^{0.5} \right]} \right) \right]^{-1} \end{array} \quad 37$$

$$\begin{array}{ll} \text{Multiple} & \\ \text{Shell Side} & \varepsilon = \left[\left(\frac{1 - \varepsilon_1 \cdot R_c}{1 - \varepsilon_1} \right)^{n_2} - 1 \right] \left[\left(\frac{1 - \varepsilon_1 \cdot R_c}{1 - \varepsilon_1} \right)^{n_2} - R_c \right]^{-1} \\ \text{Pass} & \end{array} \quad 38$$

Where:

- Q_{max} = The maximum energy.
- $T_{H,in}$ = Temperature of hot side fluid inlet.
- $T_{H,out}$ = Temperature of hot side fluid outlet.
- $T_{C,in}$ = Temperature of cold side fluid inlet.
- $T_{C,out}$ = Temperature of cold side fluid outlet.
- N_{tu} = The number of transfer units.
- R_c = The capacity ratio.
- m_1 = Fa coefficient obtained within Table 13 below.
- n_1 = Fa coefficient obtained within Table 13 below.
- n_2 = The number of shell side passes

Table 13: Fin analogy number parameters for different heat exchangers (Fakheri, 2014).

HX Type	m	n
Cross flow	-1.00	1.00
Parallel	1.00	1.00

The capacity ratio in the above-mentioned equation, R_c , can be expressed as (Mills, 1999):

$$R_c = \frac{C_{min}}{C_{max}} \quad 39$$

Where:

- C_{min} = The minimum heat capacity rate (W/K).
- C_{max} = The maximum heat capacity rate (W/K).

The minimum and maximum heat capacity rates can be calculated using the following equations (Mills, 1999):

$$C_{min} = \min(C_H, C_C) \quad 40$$

$$C_{max} = \max(C_H, C_C) \quad 41$$

Where:

- C_H = The specific heat capacity of the fluid on the hot side (J/kg-K).
- C_C = The specific heat capacity of the fluid on the cold side (J/kg-K).

The number of transfer units, NTU, in the above-mentioned equation is defined as (Mills, 1999):

$$N_{tu} = \frac{U \cdot A}{C_{min}} \quad 42$$

Where:

- U = The overall heat transfer coefficient.
- A = The total tube surface area.
- $\dot{m}$ = The tube side mass flow rate.
- C_p = The specific heat capacity of the tube side fluid.

3.2.5 Energy Balance

In order to ensure that the expressed heat exchanger design has been accurately formulated, the energy balance was calculated by using the below formulas. This energy balance is crucial to ensure accuracy within the calculation based on the fundamental principle of energy conservation. This principle states that energy cannot be created or destroyed, which leads to a balance between the energy in and the energy out of the heat exchanger (Mills, 1999):

$$\sum H_i + \sum Q_i + \sum W_i = \sum H_o + \sum Q_o + \sum W_o \quad 43$$

Where:

- H_i = Inlet Enthalpy
- Q_i = Inlet heat energy
- W_i = Inlet work energy
- H_o = Outlet Enthalpy
- Q_o = Outlet heat energy
- W_o = Outlet work energy

The above-mentioned equation can be simplified for a single STH as seen below (Mills, 1999):

$$(m_H h_H + m_C h_C)_{x=0} = (m_H h_H + m_C h_C)_{x=L} \quad 44$$

$$m_H (h_{H,o} - h_{H,L}) = m_C (h_{C,L} - h_{C,o}) \quad 45$$

$$Q_{in} = m_H (h_{H,o} - h_{H,L}) \quad 46$$

$$Q_{out} = m_C (h_{C,L} - h_{C,o}) \quad 47$$

Where:

- m_H = The shell side (hot) fluid mass flow rate
- m_C = The tube side (cold) fluid mass flow rate
- h_H = The shell side (hot) fluid enthalpy.
- h_C = The tube side (cold) fluid enthalpy.

By means of using the above equations, the energy balance of the STH can be calculated and determined.

3.3 Plain Tube Shell-and-Tube Heat Exchanger Calculation Process

The process that will be used in order to calculate the PTSTHE's output and performance, can be seen in the below expressed diagrams.

3.3.1 Tube side heat transfer coefficient calculation.

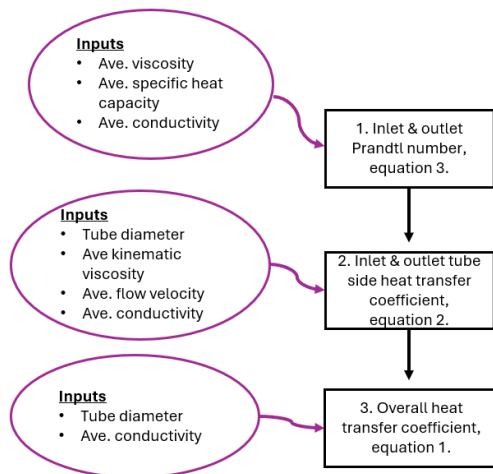


Figure 28: Plain tube, tube side heat transfer coefficient calculation.

3.3.2 Shell side heat transfer coefficient calculation.

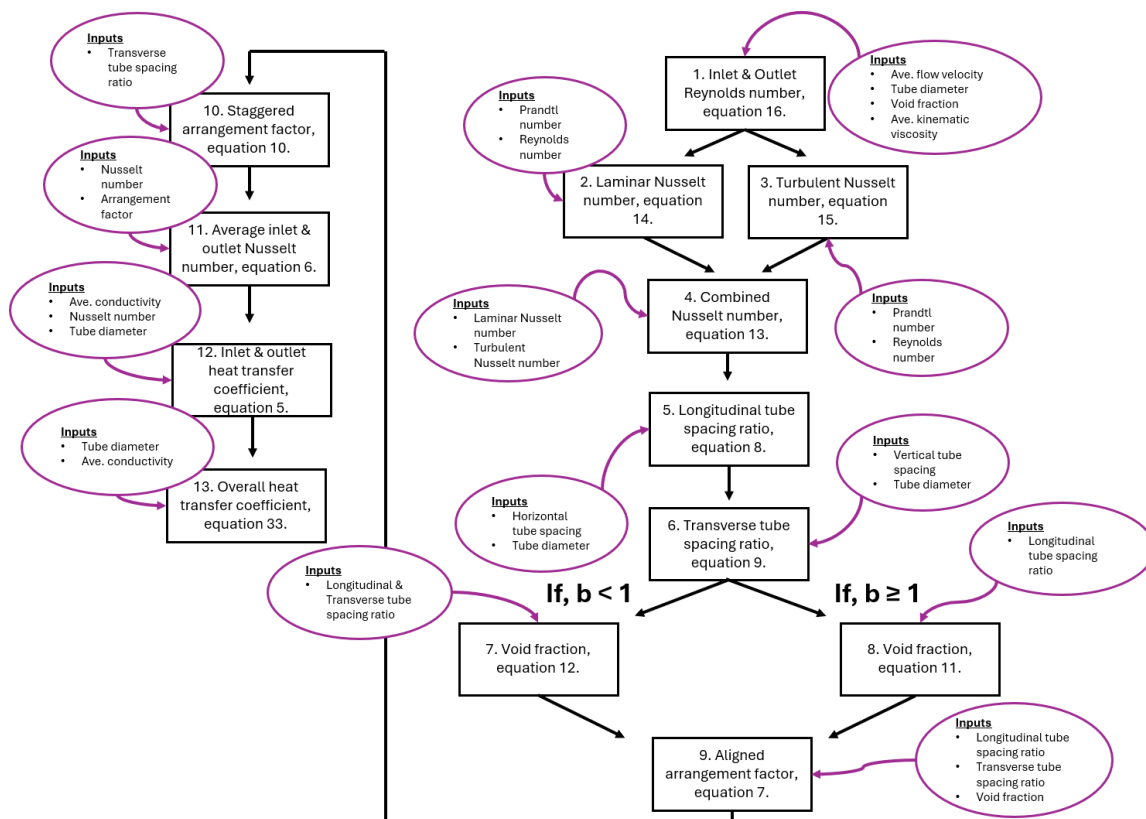


Figure 29: Plain tube, shell side heat transfer coefficient calculation.

3.3.3 Tube side pressure drop calculation.

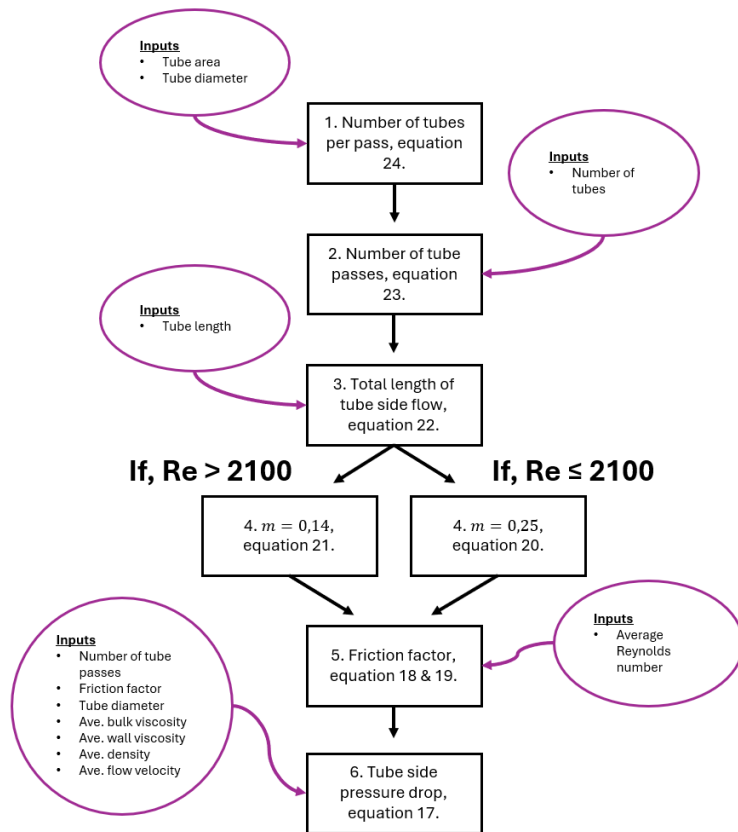


Figure 30: Plain tube, tube side pressure drop calculation.

3.3.4 Shell side pressure drop calculation.

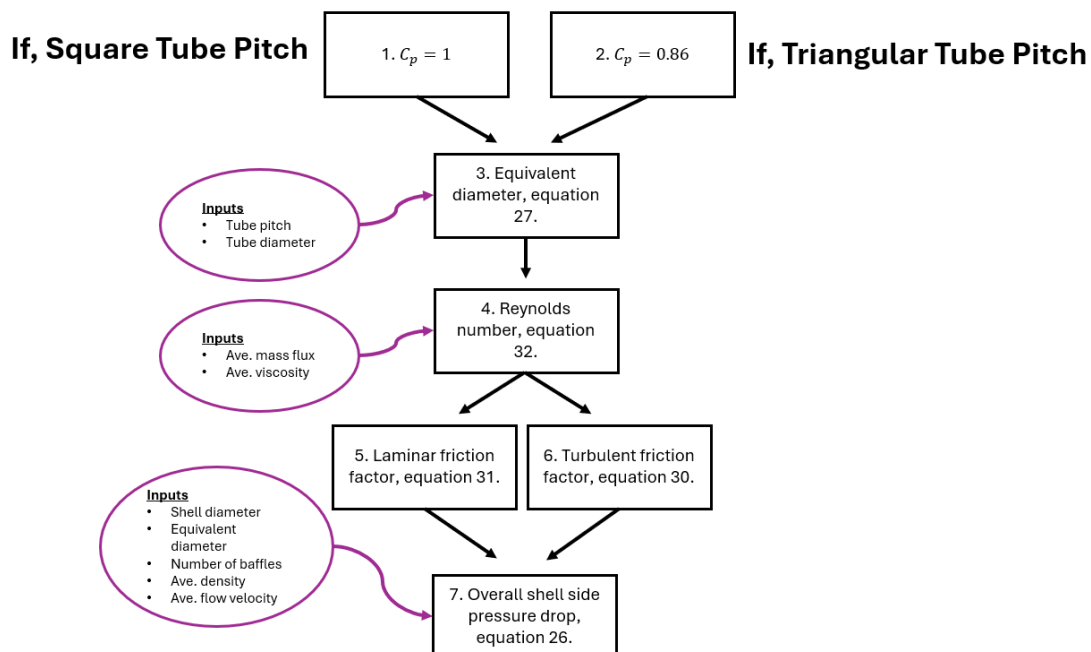


Figure 31: Plain tube, shell side pressure drop calculation.

3.3.5 Effectiveness calculation.

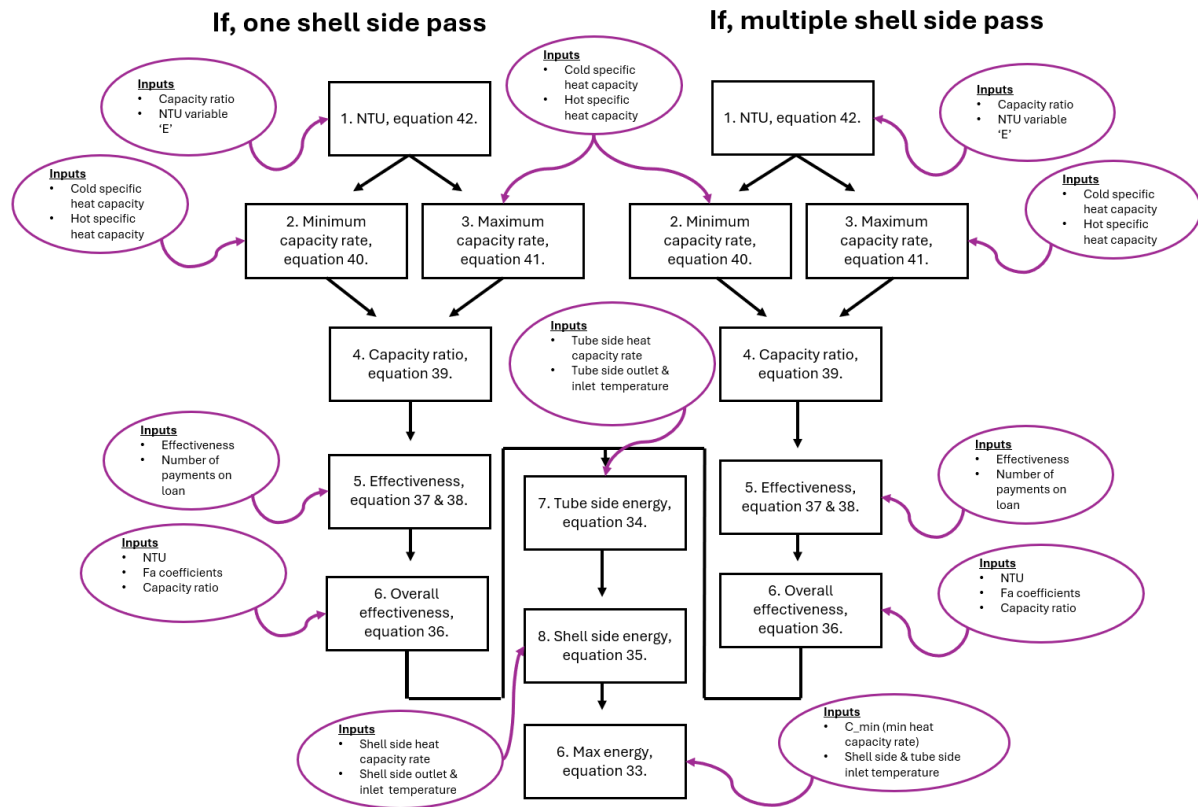


Figure 32: Plain tube, effectiveness calculation.

3.3.6 Energy Balance

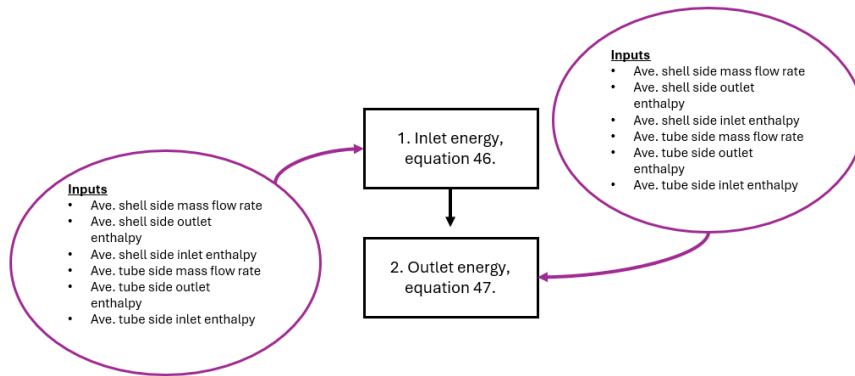


Figure 33: Plain tube, energy balance calculation.

3.4 Calculated Output and Validation of Calculated Results

In order to validate the calculated results of the thesis for the plain tube design, the following outputs will be compared to previous completed studies in the similar field of interest:

- Shell side pressure drop vs the mass flow rate within the shell.

- Tube side heat transfer coefficient vs the tube side fluid Reynolds number.
- Shell side pressure drop vs the shell side Reynolds number.
- Tube side pressure drop vs the tube side Reynolds number.
- Shell side mass flow rate vs the heat exchanger effectiveness.

3.4.1 Shell Side Pressure Drop vs the Shell Side Mass Flow Rate

The design parameters and conditions used within the validation study of Saffarian, et al. (2019) can be seen in the table below. The design data, stipulated below, was used within the EES calculation for the validation of the calculation.

Table 14: Design data and parameters of the validation study (Saffarian, et al., 2019).

Validation Study			
Variable	Value		Unit
Shell Diameter	200		mm
Tube Diameter	15		mm
Tube Length	500		mm
Number of Tubes	37		-
Shell Inlet/Outlet Diameter	100		mm
Number of Baffles	4		-
Variable	Water	Air	
Dynamic Viscosity	0.0003545	0.00001754	Pa-s
Specific Heat Ratio	1.0000000	1.40000000	-
Specific Heat Capacity	4194.0000000	1006.00000000	J/kg-K
Density	971.8000000	1.26900000	kg/m ³
Thermal Conductivity	0.6562000	0.02401000	W/m-K

The obtained results calculated within the study “Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes” along with the overlaying simulated output, can be seen in Figure 34 below (Saffarian, et al., 2019).

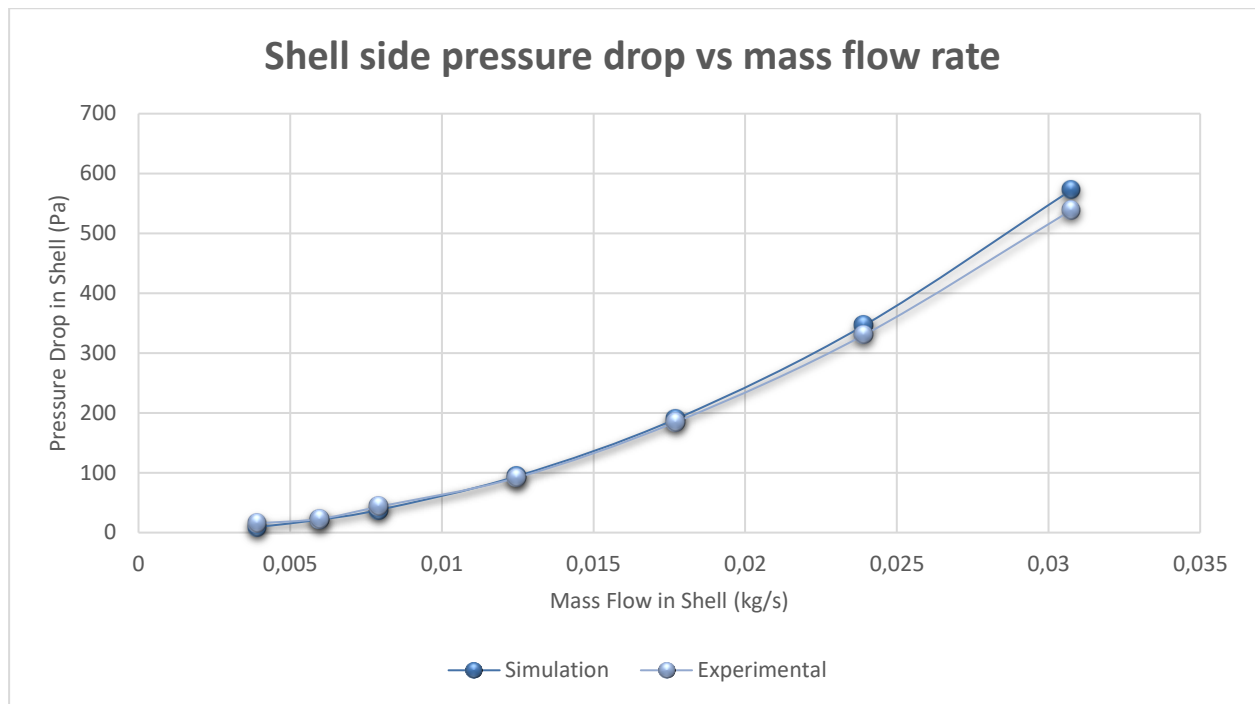


Figure 34: Plain tube, tube side mass flow rate vs tube side pressure drop validation output.

- $Re \geq 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made:

Table 15: Plain tube, shell side pressure drop vs shell side mass flow rate, graphical annalistic output.

Variable	Value	Unit
Average percentage error	0.59	%
Maximum percentage error	1.41	%
Standard deviation	0.43	-
r^2 value	0.99	-

3.4.2 Tube Side Heat Transfer Coefficient vs Tube Side Fluid Reynolds Number

The design parameters and conditions used within the validation study of Saffarian et al. (2019) can be seen within Table 14. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

The heat transfer coefficient and corresponding Reynolds number calculated can be validated using the calculated trend of the obtained results within the validation study “Numerical study of

shell and tube heat exchanger with different cross-section tubes and combined tubes”. Said study compared the different heat transfer coefficients calculated to their corresponding Reynolds number for tubes of different cross-sections, and focused on circular tubes (CT) and elliptical tubes (ET) with the attack angles of 0° and 90° .

The obtained results calculated within the study “Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes” along with the overlaying simulated output, can be seen in Figure 35 below (Saffarian, et al., 2019).

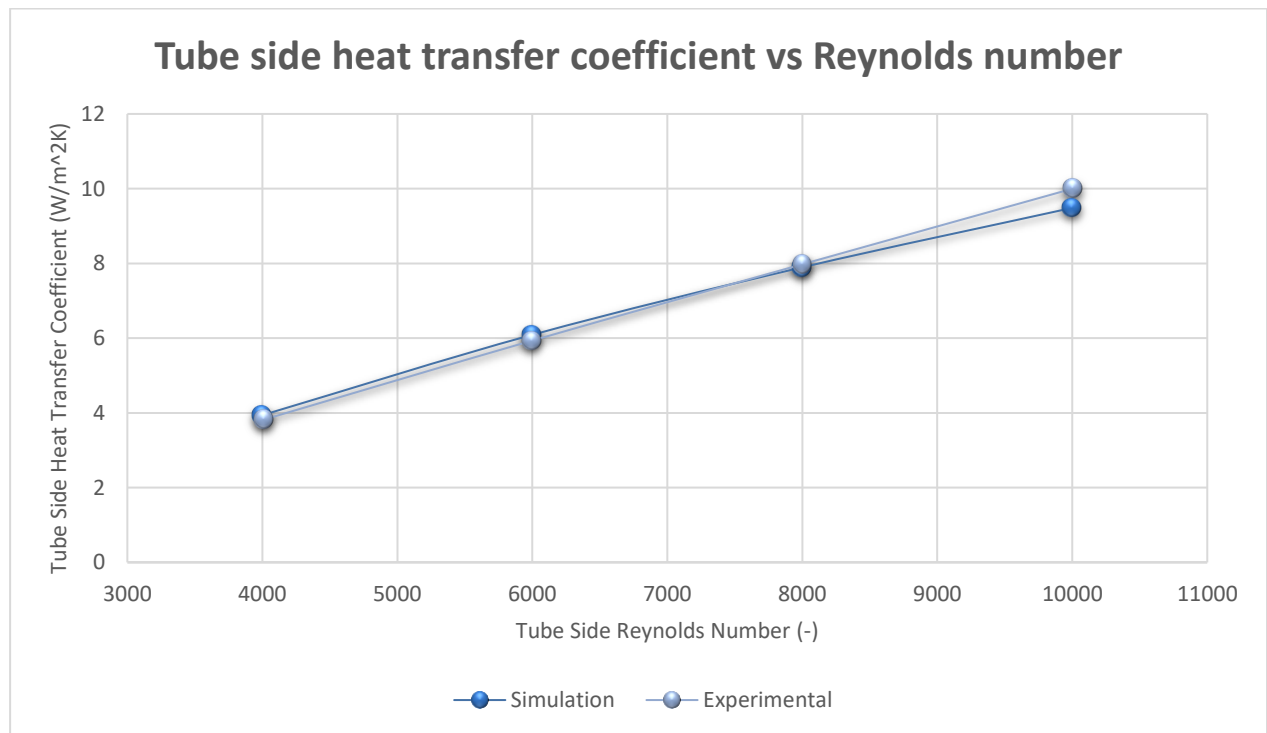


Figure 35: Plain tube, tube side heat transfer coefficient vs tube side fluid Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $Re \geq 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 16: Plain tube, tube side heat transfer coefficient vs tube side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	3.00	%
Maximum percentage error	5.48	%
Standard deviation	1.83	-
<i>r² value</i>	0.93	-

3.4.3 Shell Side Pressure Drop vs the Shell Side Reynolds Number

The design parameters and conditions used within the validation study of Saffarian et al. (2019) can be seen in Table 14. The design data, stipulated within the table, was used in the EES calculation for the validation of the calculation.

The shell side pressure drop in correspondence with the shell side fluid Reynolds number can be validated using the study “Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes”. The obtained result within the mentioned study can then be used to validate the calculated output by comparing the obtained graph trend between the two outputs. The validation output expresses that little to no deviation occurred during the validation calculation.

The obtained results calculated within the study “Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes”, along with the overlaying simulated output, can be seen in Figure 36 below (Saffarian, et al., 2019).

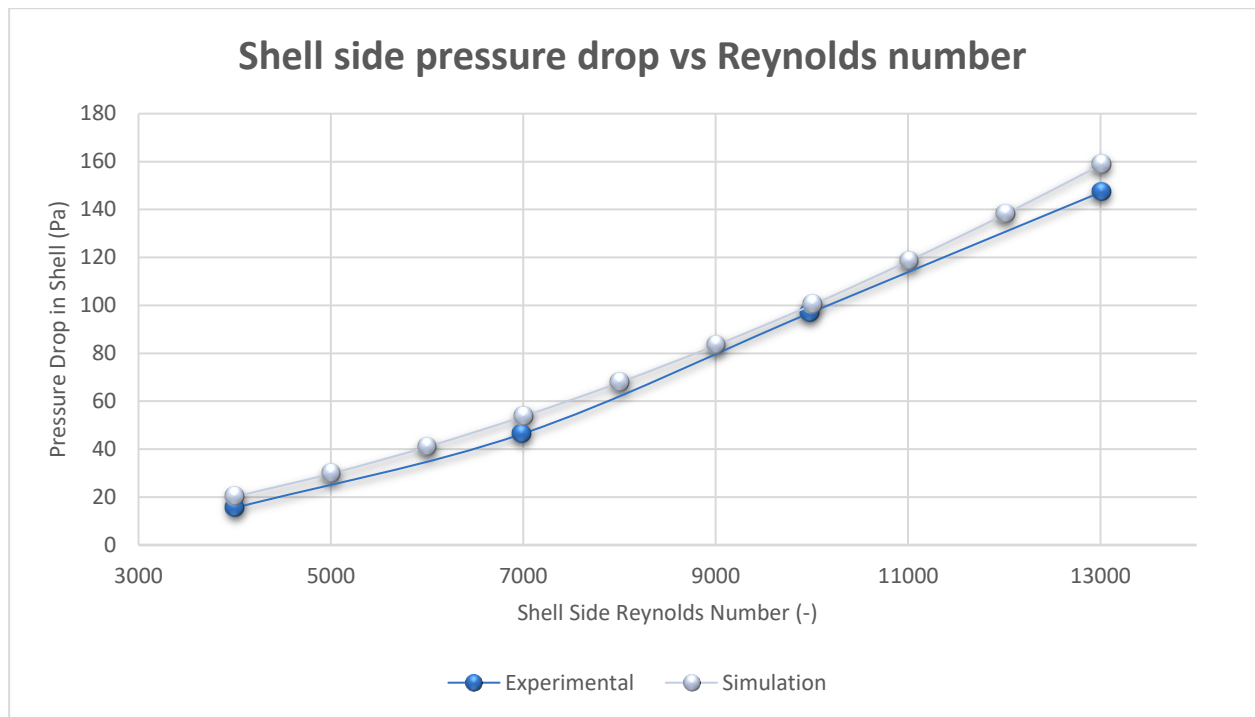


Figure 36: Plain tube, shell side pressure drop vs shell side Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $Re \geq 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 17: Plain tube, shell side pressure drop vs shell side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	4.85	%
Maximum percentage error	7.22	%
Standard deviation	1.75	-
r^2 value	0.97	-

3.4.4 Tube Side Pressure Drop vs the Tube Side Reynolds Number

The design parameters and conditions used within the validation study of Mohammad Reza Saffarian (2019), can be seen within Table 14. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

The tube side pressure drop with the corresponding Reynolds numbers can be validated using the obtained results with the study, “Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes”. The graphical trend calculated can be used to validate the accuracy of the EES output. The validation output expresses that little to no deviation occurred during the validation calculation.

The obtained results calculated within the said study, along with the overlaying simulated output, can be seen in Figure 37 below.

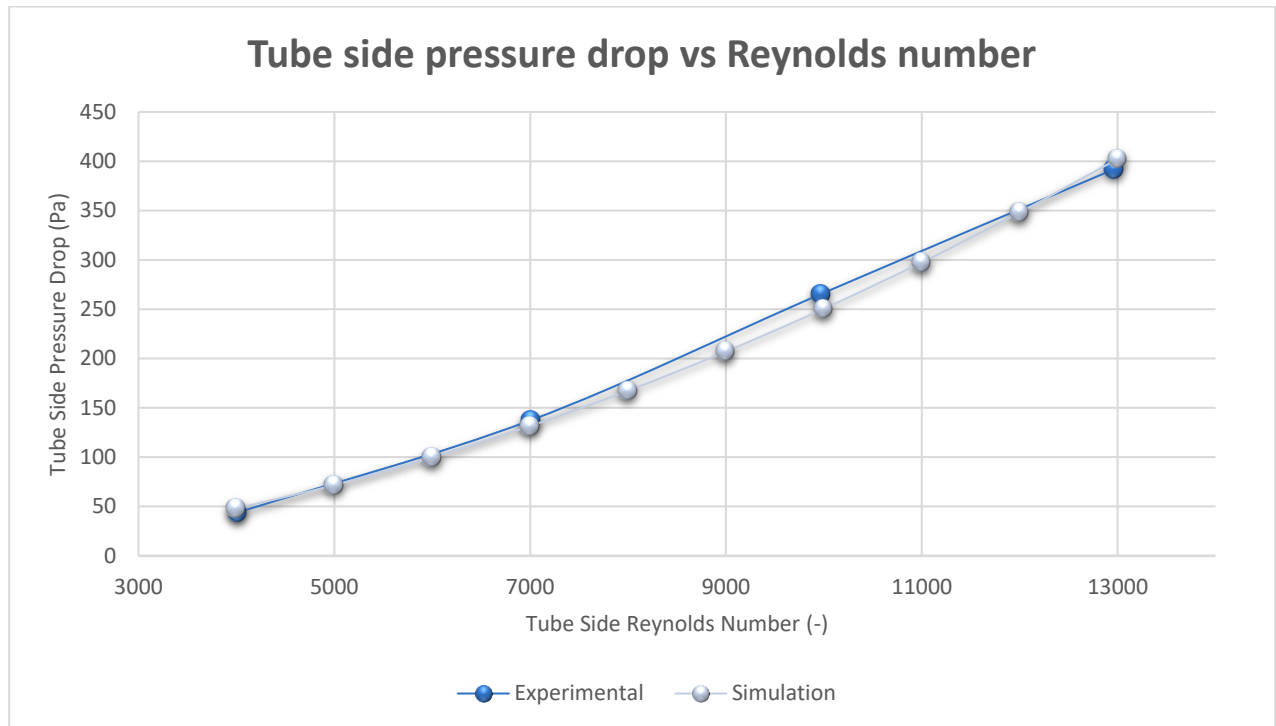


Figure 37: Plain tube, tube side pressure drop vs tube side Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $Re \geq 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 18: Plain tube, tube side pressure drop vs tube side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	5.39	%
Maximum percentage error	7.19	%

Standard deviation	1.96	-
r^2 value	0.99	-

3.4.5 Shell Side Mass Flow Rate vs the Heat Exchanger Effectiveness

The design parameters and conditions used within the validation study of Hussein et al. (2021) can be seen in the table below. The design data, stipulated below, was used in the EES calculation for the validation of the calculation.

Table 19: Design data and parameters of the validation study (Hussein, et al., 2021).

Validation Study		
Variable	Value	Unit
Shell side fluid	Water	-
Tube side fluid	Water	-
Internal tube diameter	0.0254	m
Outside tube diameter	0.0257	m
Internal shell diameter	0.1544	m
Outside shell diameter	0.1564	m
Tube side mass flow rate	1.5	kg/s
Shell side mass flow rate	1.5	kg/s
Tube length	2.0	m
Number of baffles	22.0	-
Number of tubes	12.0	-
Inlet Tube temperature	313.0	K
Outlet tube temperature	304.4	K
Inlet shell temperature	313.0	K
Outlet shell temperature	311.5	K
Inlet tube pressure	100.3	kPa
Outlet tube pressure	103.7	kPa
Inlet shell pressure	100.3	kPa
Outlet shell pressure	115.5	kPa
Shell side fluid density	994.75	kg/m ³
Tube side fluid density	996.1	kg/m ³
Shell side fluid specific heat capacity	4178.375	J/kg-K
Tube side fluid specific heat capacity	4179.125	J/kg-K

Validation Study		
Variable	Value	Unit
Shell side fluid viscosity	698.0	Pa-s
Tube side fluid viscosity	786.0	Pa-s
Shell side fluid inlet conductivity	0.624	W/m-K
Shell side fluid outlet conductivity	0.632	W/m-K
Tube side fluid inlet conductivity	0.616	W/m-K
Tube side fluid outlet conductivity	0.614	W/m-K
a	1.25	-
b	1.5	-
n	1.0	-
Inlet tube Prandtl number	1000.0	-
Outlet tube Prandtl number	2000.0	-
Inlet tube side Reynolds number	3000.0	-
Outlet tube side Reynolds number	10000.0	-
Inlet tube side fluid velocity	0.887	m/s
Outlet tube side fluid velocity	0.443	m/s

The mass flow rate and effectiveness calculated output can be validated using the study “Design A Prototype Module for Al-Mussaib Thermal Power Plant” by Hussein (2021). The study’s focus is directly toward designing a prototype shell and tube heat exchanger for Al-Mussaib thermal power plant (Hussein, et al., 2021).

The obtained results calculated within the study “Design A Prototype Module for Al-Mussaib Thermal Power Plant”, along with the overlaying simulated output, can be seen in Figure 38 below (Hussein, et al., 2021).

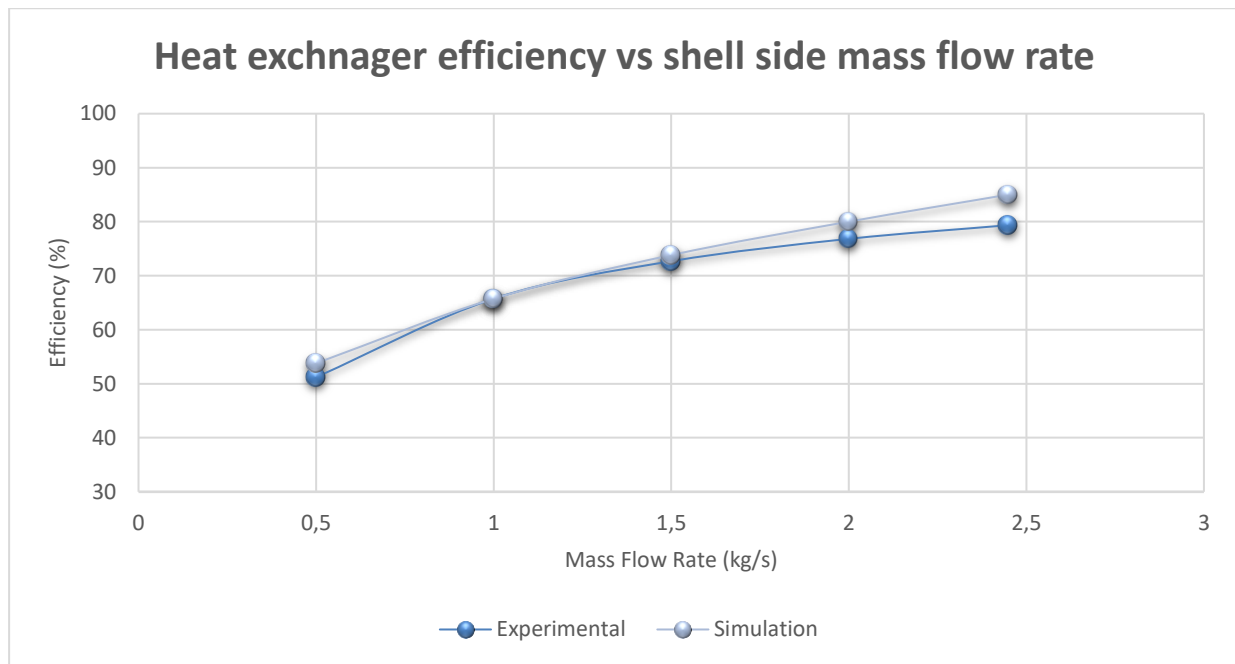


Figure 38: Plain tube heat exchanger efficiency vs shell side mas flow rate validation output.

The graph above is simulated only for the following non-dimensional variables:

- $Re \geq 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 20: Plain tube heat exchanger efficiency vs shell side mass flow rate, graphical annalistic output.

Variable	Value	Unit
Average percentage error	3.39	%
Maximum percentage error	6.65	%
Standard deviation	2.63	-
r^2 value	0.97	-

3.4.6 Energy Balance

Considering the studies used for the validation of the EES simulation, the following energy balances were obtained for each validation study used.

Table 21: Plain tube, validation study energy balance output.

Study	Energy Balance [kW]	Difference [%]
“Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes” (Saffarian, et al., 2019).	Inlet Energy = 721 kW Outlet Energy = 719 kW <u>Balance = 2 kW</u>	0.28 %
“Design A Prototype Module for Al-Mussaib Thermal Power Plant” (Husseini, et al., 2021).	Inlet Energy = 203.4 kW Outlet Energy = 203 kW <u>Balance = 0.4 kW</u>	0.19 %

Considering the obtained outputs for the energy balance calculated, it can be seen that little to no difference is obtained for the energy produced by the inlet of the heat exchanger model to that measured at the outlet of the heat exchanger. The small difference within the obtained energy balance could be explained in the sense that some heat energy is absorbed by the environment during the operation of the heat exchanger.

3.5 Summary

The validation of the plain-tube STHE involved comparing the EES calculated results with data from two previously conducted and established studies, namely: Mohammad Reza Saffarian (2019) and Husseini (2021).

Using identical geometrical configurations and fluid properties, five key performance relationships were examined for the plain tube STHE during the validations of the EES calculation model. These relations include:

- Shell-side pressure drop versus shell-side mass flow rate.
- Tube-side heat-transfer coefficient versus Reynolds number.
- Shell-side pressure drop versus shell-side Reynolds number.
- Tube-side pressure drop versus tube-side Reynolds number.
- Shell-side mass flow rate versus heat-exchanger effectiveness.

In each case, the simulated outputs followed the reference trends of the validation studies closely, demonstrating a strong alignment between the EES calculation model predictions and the published results.

Considering the validations found within Mohammad Reza Saffarian (2019), the shell side and tube-side pressure drop correlations showed excellent agreement, with average errors below 6% and r^2 values of 0.97 - 0.99. Similarly, the tube side heat transfer coefficient comparison produced an average error of just 2.21%, confirming that the EES model accurately captures turbulent flow heat transfer behaviour for Reynolds numbers above 4000. Only minor deviations were observed during the validations across all datasets, remaining well within acceptable engineering tolerances.

Analysing the validation performed using Hussein (2021) focused on heat exchanger effectiveness as a function of shell side mass flow rate. Again, the EES simulation closely matched the published results, yielding an average error of 3.05% and an r^2 value of 0.98. This supported the reliability of the effectiveness calculations under the examined operating conditions.

Finally, the energy balance checks done for both studies showed extremely small differences, between inlet and outlet energy values, less than 0.3%. This confirms that the model is thermodynamically consistent and that any minor discrepancies can be attributed to small heat losses to the environment. Overall, the validation results demonstrate that the plain tube STH model is accurate, consistent, and robust across all evaluated performance metrics.

CHAPTER 4 EVALUATION OF A TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER

4.1 Introduction and Overview

This chapter is based on the factors that, considering the design of the TTSTHE, affect the overall efficiency and performance of the heat exchanger. Within this chapter, the factors will be analysed and calculated for the TTSTHE design within the petrochemical industry of South Africa.

Considering this chapter, the design conditions of an operational PTSTHE will be implemented into the TTSTHE design, Figure 39, in order to evaluate the operating and performance capabilities of the twisted tube heat exchanger under the design conditions of a PTSTHE operating within the petrochemical industry of South Africa. This chapter will consist of the calculation process followed in order to calculate and evaluate the TTSTHE under the mentioned conditions. It will also include the validation of the calculated result for the twisted tube heat exchanger by means of a comparison to previously conducted studies within other countries. The properties to consider as identified within the previous chapter for the PTSTHE, will also be analysed and calculated within this chapter for the twisted tube heat exchanger, in order to provide a comparison between the plain tube and twisted tube heat exchanger designs under the same conditions, by evaluating the same properties for each exchanger as identified.

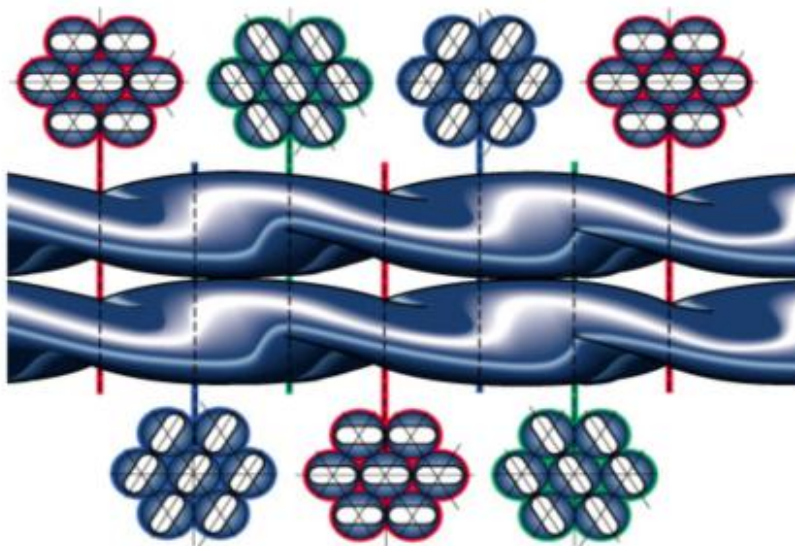


Figure 39: Twisted tube pitch section view (Danielsen, 2009).

4.2 Twisted Tube Shell-and-Tube Heat Exchanger Theory

The process that will be used to calculate the TTSTHE's performance and output will be done using the following few steps and calculations.

The overall heat transfer coefficient of the TTSTHE can be calculated as follows (Barraza-Colon & Picon-Nunez, 2022):

$$U = \left(\frac{1}{h_i} + \frac{1}{h_o} + R_w + R_f \right)^{-1}$$

48

Where:

- R_w = The thermal resistance due to conduction through the wall (m^2K/W)
- R_f = The thermal resistance due to fouling (m^2K/W)

4.2.1 Tube Side Heat Transfer Coefficient

Calculating the tube side heat transfer coefficient of the TTSTHE, consists of the following few steps and formulas.

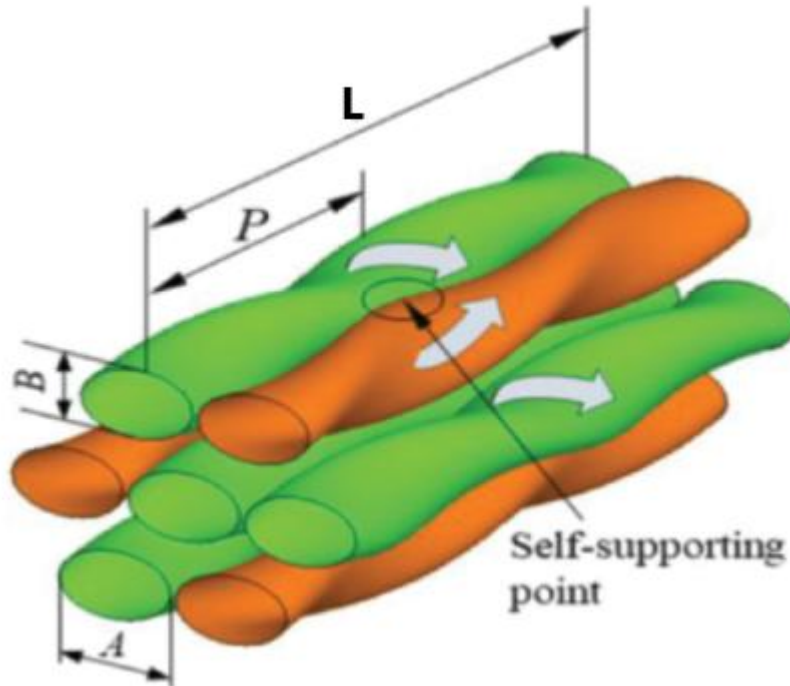


Figure 40: Twisted tube dimensional constraints visual specification (Razzaq & Mushatet, 2023).

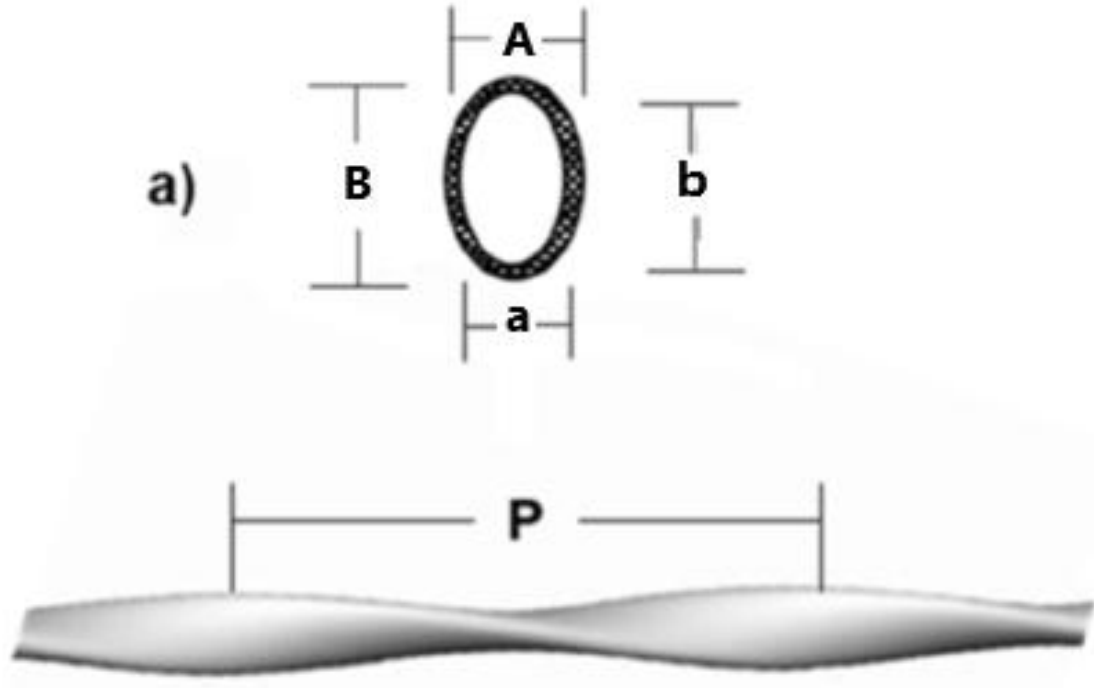


Figure 41: Twisted tube geometry and pitch (Barraza-Colon & Picon-Nunez, 2022).

The tube side heat transfer coefficient at the inlet and outlet of the TTSTHE can be calculated using the following (Barraza-Colon & Picon-Nunez, 2022):

$$h = 0.034 \cdot \left(\frac{Re}{w}\right)^{0.8} \cdot Pr^{0.33} \cdot \left[\left(\frac{a}{b}\right)^{-0.59}\right] \cdot \left[\frac{P}{d_{h,t}}\right]^{-0.165} \cdot \frac{k}{d_{h,t}}$$

49

Where:

- $d_{h,t}$ = Hydraulic diameter of the tubes (m).
- w = Velocity of tube side fluid (m/s).
- a = Vertical tube inside diameter (m).
- b = Horizontal tube inside diameter (m).
- P = Twisted tube twist pitch length (m), Figure 41.
- k = Thermal conductivity of the tube side fluid (W/m-K).

The hydraulic diameter, d_h , of the twisted tubes was calculated as for an ellipse, using the below formulas (Barraza-Colon & Picon-Nunez, 2022):

$$d_{h,t} = \frac{4 \cdot A}{Perimeter}$$

50

$$A = \pi \cdot L \cdot d_h$$

51

$$Perimeter = 2 \cdot \pi \cdot a + 4 \cdot (b - a)$$

52

Where:

- L = Tube length (m), Figure 40.

The vertical and horizontal tube inside diameters, as well as the twisted tube twist pitch length, can be expressed as seen in Figure 41 above.

The inlet and outlet Prandtl numbers can be expressed by means of the following equations (Barraza-Colon & Picon-Nunez, 2022):

$$Pr = \frac{\mu \cdot C_p}{k}$$

53

Where:

- μ = The fluid viscosity (Pa-s).
- C_p = The specific heat capacity of the fluid (J/kg-K).
- k = The heat conductivity of the fluid (W/m-K).

The tube side inlet and outlet Reynolds numbers can be calculated using the following formula (Barraza-Colon & Picon-Nunez, 2022):

$$Re = \frac{f \cdot \rho \cdot w^2 \cdot L}{\mu \cdot d_{h,t}}$$

54

Where:

- w = Fluid flow velocity at the shell side (m/s).
- ρ = The density of the shell side fluid (kg/m³).

The friction factor, f , can be calculated using the below equation (Barraza-Colon & Picon-Nunez, 2022):

$$f = 9.461 \cdot Re^{-0.521} \cdot \left(\frac{P}{d_{h,t}} \right)^{-0.082} \cdot \left(\frac{a}{b} \right)^{-0.33}$$

55

However, to increase the calculation accuracy of the tube side heat transfer coefficient, the heat exchanger specific calculation was done using the average calculated fluid properties, as this approach better accounts for non-linearities. These properties for the tube side heat transfer coefficient calculation are:

- Reynolds number
- Fluid velocity
- Prandtl number
- Thermal conductivity

The average for each of these properties were calculated using equation 4.

4.2.2 Shell Side Heat Transfer Coefficient

The shell side heat transfer coefficient of the TTSTHE can be calculated using the following few formulas, where the inlet and outlet heat transfer coefficients of the twisted tube heat exchanger can be expressed as (Barraza-Colon & Picon-Nunez, 2022):

$$h = k_{s3} \cdot (G_s)^{0.8} \quad 56$$

Where:

- k_{s3} = Defined parameter.

The mass flux G_s , can be calculated using the below formula (Barraza-Colon & Picon-Nunez, 2022):

$$G_s = \frac{\dot{m}}{A_c} \quad 57$$

Where:

- A_c = The free flow area (m²).

The parameter, k_{s3} , can be defined by means of the following formula (Barraza-Colon & Picon-Nunez, 2022):

$$k_{s3} = \frac{0.2379 \cdot \left(\frac{D_e}{\mu}\right) \cdot k \cdot Pr^{0.33} \cdot Fr_m^{-0.4347} \cdot [1 + 3.6 \cdot (Fr_m)^{-0.357}]}{D_e} \quad 58$$

Where:

- k = The heat conductivity of the fluid (W/m-K).
- Pr = Shell side Prandtl number, above equation 53.
- Fr_m = Modified Froude number.
- μ = Viscosity of tube side fluid (Pa-s).

The Prandtl number as specified within the above equation can be calculated using the same formula as specified within equation 3, with the selected shell side conditions.

The free flow area, A_c , at the inlet of the shell can be calculated using the following equation (Barraza-Colon & Picon-Nunez, 2022):

$$A_c = \frac{\pi \cdot d_{h,s}^2 \cdot N_t}{4 \cdot N_p} \quad 59$$

Where:

- N_t = Number of tubes.
- N_p = Number of tube passes.

The modified Froude number can be calculated as (Munoz-Carpena, et al., 1993):

$$Fr_m = \frac{w}{(g \cdot d)^{0.5}} \quad 60$$

Where:

- w = The flow velocity (m/s).
- g = The gravitational acceleration (m/s²).
- d = The depth of the flow (m).

The flow velocity can be calculated as (Munoz-Carpena, et al., 1993):

$$w = \left(\frac{g \cdot A_{c,t}}{B_{s,t}} \right)^{0.5} \quad 61$$

Where:

- $A_{c,t}$ = Cross sectional area of the tubes (m²).
- $B_{s,t}$ = Surface width of the tubes (m).

However, to also increase the calculation accuracy of the shell side heat transfer coefficient, the heat exchanger specific calculation was done using the average calculated fluid properties, as this approach better accounts for non-linearities. These properties for the shell side heat transfer coefficient calculation are:

- Fluid viscosity
- Prandtl number

- Heat conductivity
- Froude number
- Mass flux

The average for each of these properties were calculated using equation 4.

4.2.3 Tube Side Pressure Drop

The tube side pressure drop can be calculated and expressed by means of the following formula (Tan, et al., 2012):

$$\Delta P_t = \frac{f_t \cdot L \cdot \rho_t \cdot w_t^2}{2 \cdot d_{h,t}} \quad 62$$

Where:

- f_t = The tube side friction factor.
- ρ_t = The tube side fluid density (kg/m³).
- w_t = The tube side fluid flow velocity (m/s).
- $d_{h,t}$ = The hydraulic diameter of the tubes (m).

The friction factor, Reynolds number $5000 < Re < 100000$, tube side fluid flow velocity, and the hydraulic diameter of the tubes can be calculated using the following equations (Tan, et al., 2012):

$$f_t = 0.143 \cdot Re_t^{-0.169} \quad 63$$

$$Re_t = \frac{\rho_t \cdot w_t \cdot d_{h,t}}{\mu_t} \quad 64$$

$$w_t = \frac{V_t}{A_t} \quad 65$$

$$d_{h,t} = \frac{4 \cdot A_{t,sec}}{\mathcal{P}} \quad 66$$

Where:

- Re_t = Tube side Reynolds number.
- μ_t = Tube side fluid viscosity (Pa-s).
- V_t = Tube side volumetric flow rate (m³/s).
- A_t = Tube side flow area (m²).
- $A_{t,sec}$ = Tube sectional area (m²).

If the twisted tubes within the TTSTHE are manufactured from circular round tube, the perimeter of the twisted tubes can be expressed as (Tan, et al., 2012):

$$P_t = 2 \cdot d_{i,tube} + 2 \cdot L \quad 67$$

Where:

- $d_{i,tube}$ = The inside diameter of the initial tube (m).

4.2.4 Shell Side Pressure Drop

The TTSTHE's shell side pressure drop can be calculated by means of the following equation (Tan, et al., 2012):

$$\Delta P_s = \frac{f_s \cdot L_s \cdot \rho_s \cdot w_s^2}{2 \cdot d_{h,s}} \quad 68$$

Where:

- f_s = The shell side friction factor.
- L_s = The shell length (m).
- ρ_s = The shell side fluid density (kg/m³).
- w_s = The shell side fluid flow velocity (m/s).
- $d_{h,s}$ = The hydraulic diameter of the shell (m).

The friction factor, Reynolds number $5000 < Re < 100000$, tube side fluid flow velocity, and the hydraulic diameter of the tubes can be calculated using the following equations (Tan, et al., 2012):

$$f_s = 0.143 \cdot Re_s^{-0.169} \quad 69$$

$$Re_s = \frac{\rho_s \cdot w_s \cdot d_{h,s}}{\mu_s} \quad 70$$

$$w_s = \frac{V_s}{A_s} \quad 71$$

$$d_{h,s} = \frac{4 \cdot A_{s,sec}}{2 \cdot d_{i,shell} + 2 \cdot L_s} \quad 72$$

Where:

- Re_s = Shell side Reynolds number.
- μ_s = Shell side fluid viscosity (Pa-s).

- V_s = Shell side volumetric flow rate (m^3/s).
- A_s = Shell side flow area (m^2).
- $A_{s,sec}$ = Shell sectional area (m^2).
- $d_{i,shell}$ = Inside shell diameter (m).
- L_s = Length of the shell (m).

4.2.5 Effectiveness

The effectiveness of the TTSTHE can be calculated and expressed in terms of the NTU method by means of the same formulas used in section 3.2.4.

4.2.6 Energy Balance

In order to ensure that the expressed heat exchanger design has been accurately formulated, the energy balance was calculated by using the same formulas as listed in 3.2.5.

4.3 Twisted Tube Shell-and-Tube Heat Exchanger Calculation Process

The process that will be used in order to calculate the TTSTHE's output and performance, can be seen in the below expressed diagrams.

4.3.1 Tube side heat transfer coefficient calculation

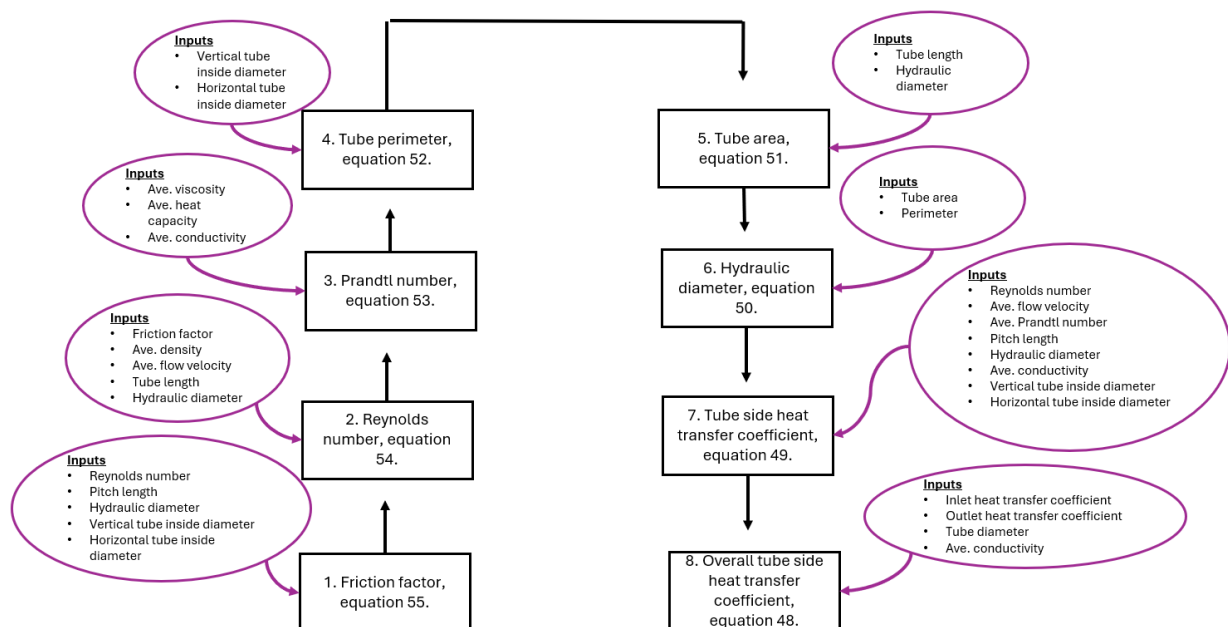


Figure 42: Twisted tube, tube side heat transfer coefficient calculation.

4.3.2 Shell side heat transfer coefficient calculation

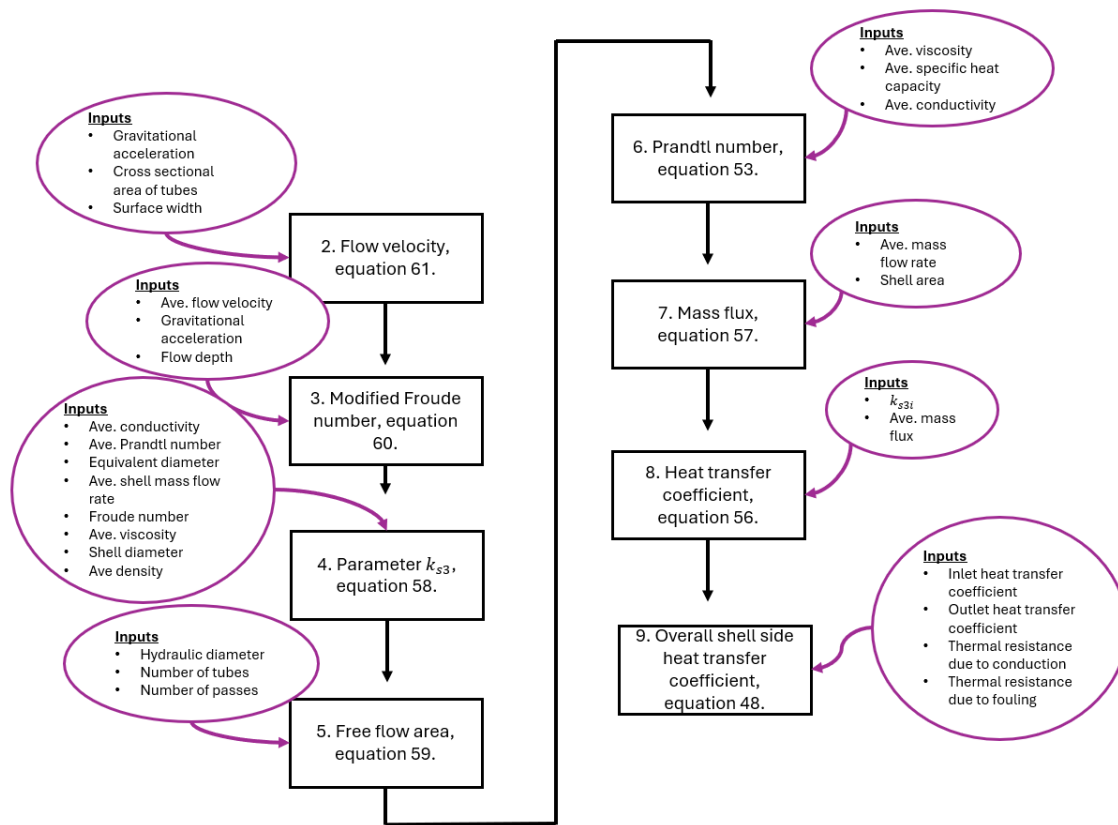


Figure 43: Twisted tube, shell side heat transfer coefficient calculation.

4.3.3 Tube side pressure drop calculation

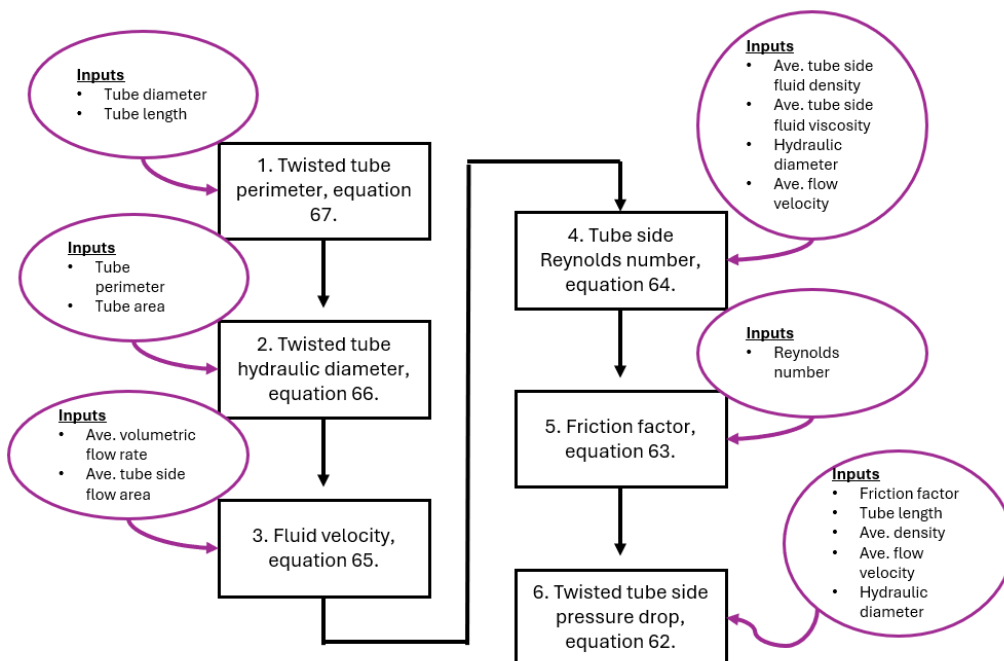


Figure 44: Twisted tube, tube side pressure drop calculation

4.3.4 Shell side pressure drop calculation

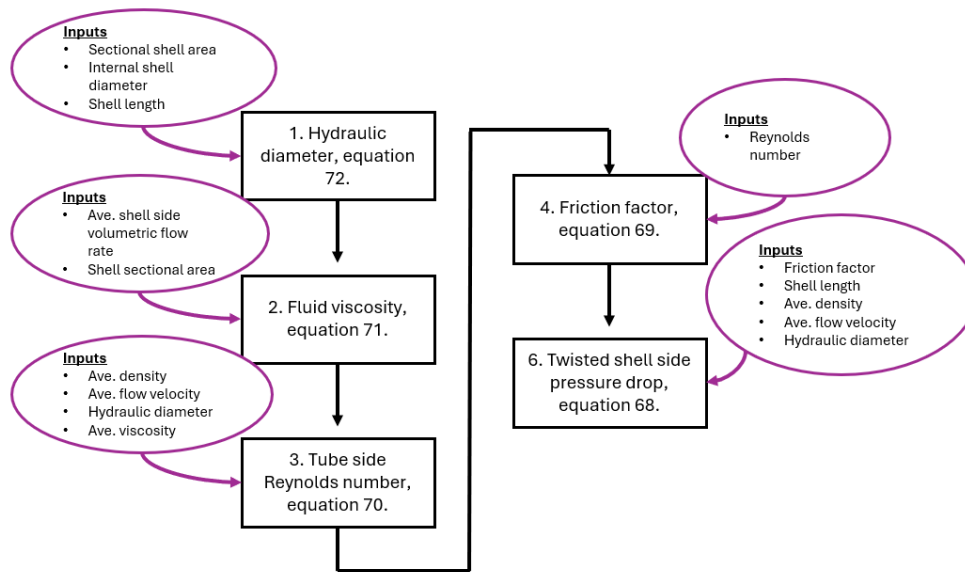


Figure 45: Twisted tube, shell side pressure drop calculation.

4.3.5 Effectiveness calculation

The process that will be used in order to calculate the TTSTHE's effectiveness can be seen as indicated in Figure 32.

4.3.6 Energy Balance

The process that will be used in order to calculate the TTSTHE's energy balance can be seen as indicated in Figure 33.

4.4 Calculated Output and Validation of Calculated Results

In order to validate the calculated results of the thesis for the twisted tube design, the following outputs will be compared to previous completed studies in the similar field of interest.

- Tube side heat transfer coefficient vs the tube side fluid Reynolds number.
- Shell side pressure drop vs the shell side Reynolds number.
- Tube side pressure drop vs the tube side Reynolds number.
- Shell side mass flow rate vs the heat exchanger effectiveness.
- Tube side heat transfer coefficient vs the twist pitch.

4.4.1 Tube Side Heat Transfer Coefficient vs Tube Side Fluid Reynolds Number

The design parameters and conditions used within the validation study of Barraza-Colon and Picon-Nunez (2022) can be seen in the table below. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

Table 22: Design data and parameters for the validation study (Barraza-Colon & Picon-Nunez, 2022).

Validation Study			
Variable	Value		Unit
Tube Length	9920.00		mm
Shell diameter	1390.00		mm
Hydraulic tube diameter	8.76		mm
Number of Tubes	1601.00		-
Tube side pressure drop	23.44		kPa
Variable	Sea Water	A gas	
Mass flow rate	1.570	1.880	kg/s
Inlet temperature	30.000	89.200	°C
Viscosity	-	0.354	cP
Density	1160.000	971.800	kg/m ³
Conductivity	0.210	670.000	W/m-°C

The tube side heat transfer coefficient with the corresponding tube side Reynolds number, can be validated using the obtained results with the study “Improved heat recovery using twisted tube exchanger technology”. The graphical trend calculated can be used to validate the accuracy of the EES simulated output.

The obtained results calculated within the study “Improved heat recovery using twisted tube exchanger technology”, along with the overlaying simulated output, can be seen in Figure 46 below (Barraza-Colon & Picon-Nunez, 2022).

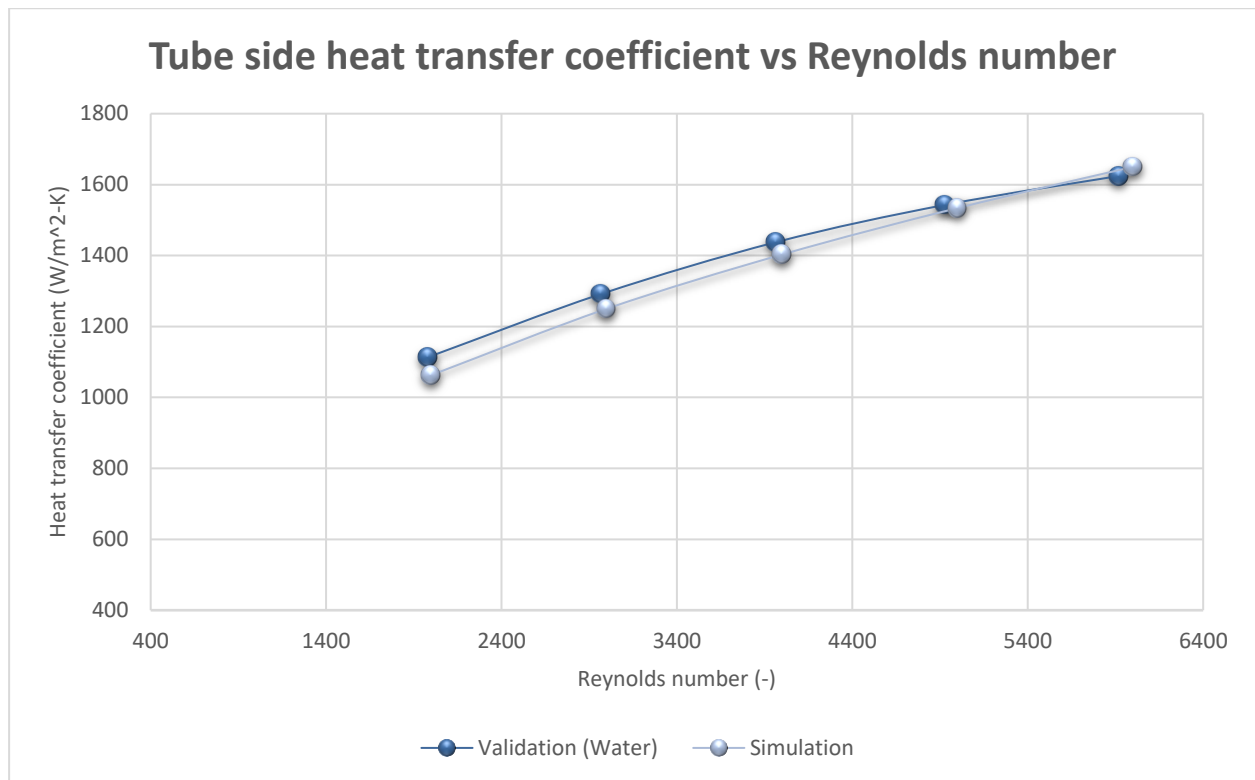


Figure 46: Twisted tube, tube side heat transfer coefficient vs tube side Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $1500 < Re < 5500$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 23: Twisted tube, tube side heat transfer coefficient vs tube side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	2.47	%
Maximum percentage error	4.53	%
Standard deviation	1.51	-
<i>r</i> ² value	0.99	-

4.4.2 Shell Side Pressure Drop vs the Shell Side Reynolds Number

The design parameters and conditions used within the validation study of Dong (2020) can be seen in the table below. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

Table 24: Design data and parameters of the validation study (Dong, et al., 2020).

Validation Study			
Variable	Value		Unit
Tube Diameter	19		mm
Tube Length	1500		mm
Tube Thickness	1.5		mm
Number of Tubes	7		-
a	235		mm
b	135		mm
Variable	Water	Thermal Oil	
Velocity	3.0	1.40	m/s
Inlet temperature	17.0	60.00	°C
Reynolds number	12000-45000	300-2050	-

The shell side pressure drop in correspondence with the shell side fluid Reynolds number can be validated using the following study: “Experimental research on heat transfer and flow resistance properties in spiral twisted tube heat exchanger”. The obtained result within the mentioned study can be used to validate the calculated output by comparing the obtained graphical trend between the two outputs, as well as the accuracy of the data points obtained between the two studies.

The obtained results calculated within the study along with the overlaying simulated output, can be seen in Figure 47 below (Dong, et al., 2020).

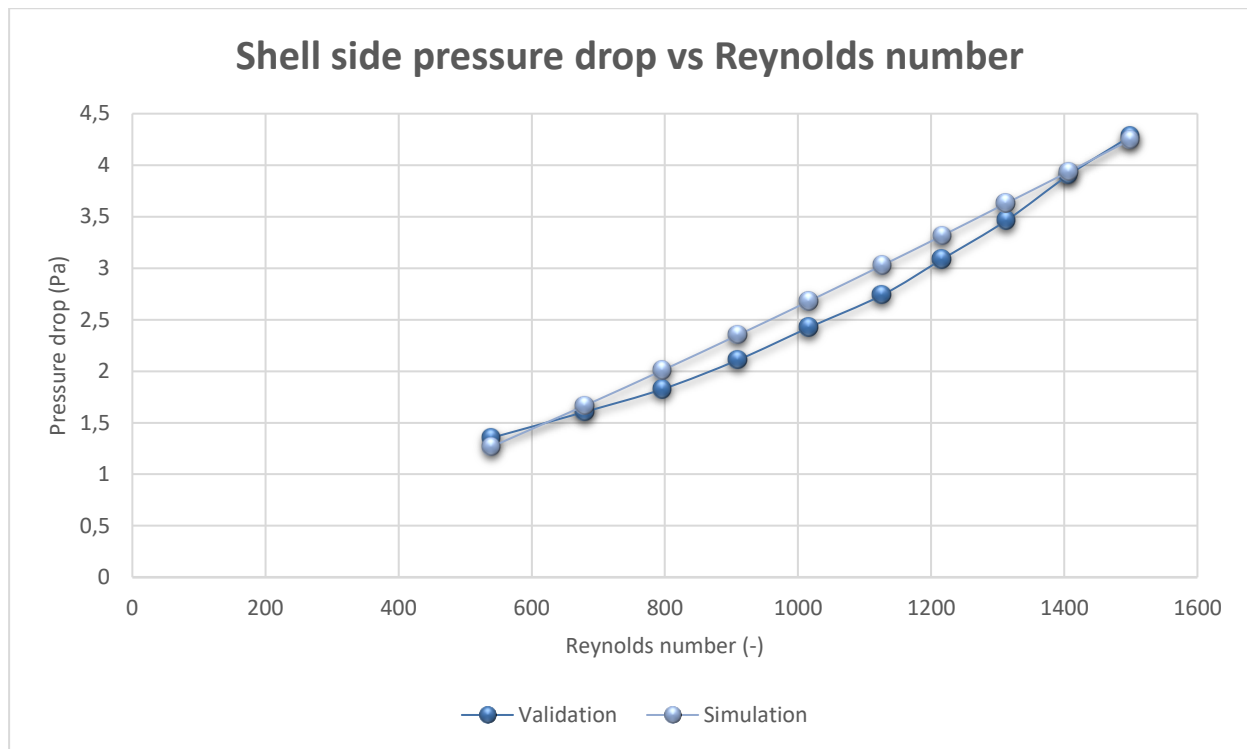


Figure 47: Twisted tube, shell side pressure drop vs shell side Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $500 < Re < 1500$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 25: Twisted tube, shell side pressure drop vs shell side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	3.98	%
Maximum percentage error	6.72	%
Standard deviation	1.8	-
r^2 value	0.98	-

4.4.3 Tube Side Pressure Drop vs the Tube Side Reynolds Number

The design parameters and conditions used within the validation study of Dong (2020) can be seen in Table 24. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

The tube side pressure drop with the corresponding Reynolds numbers, can be validated using the obtained results with the study: “Experimental research on heat transfer and flow resistance properties in spiral twisted tube heat exchanger”. The graphical trend calculated can then be used to validate the accuracy of the EES output.

The obtained results calculated within the study “Experimental research on heat transfer and flow resistance properties in spiral twisted tube heat exchanger”, along with the overlaying simulated output, can be seen in Figure 48 below (Dong, et al., 2020).

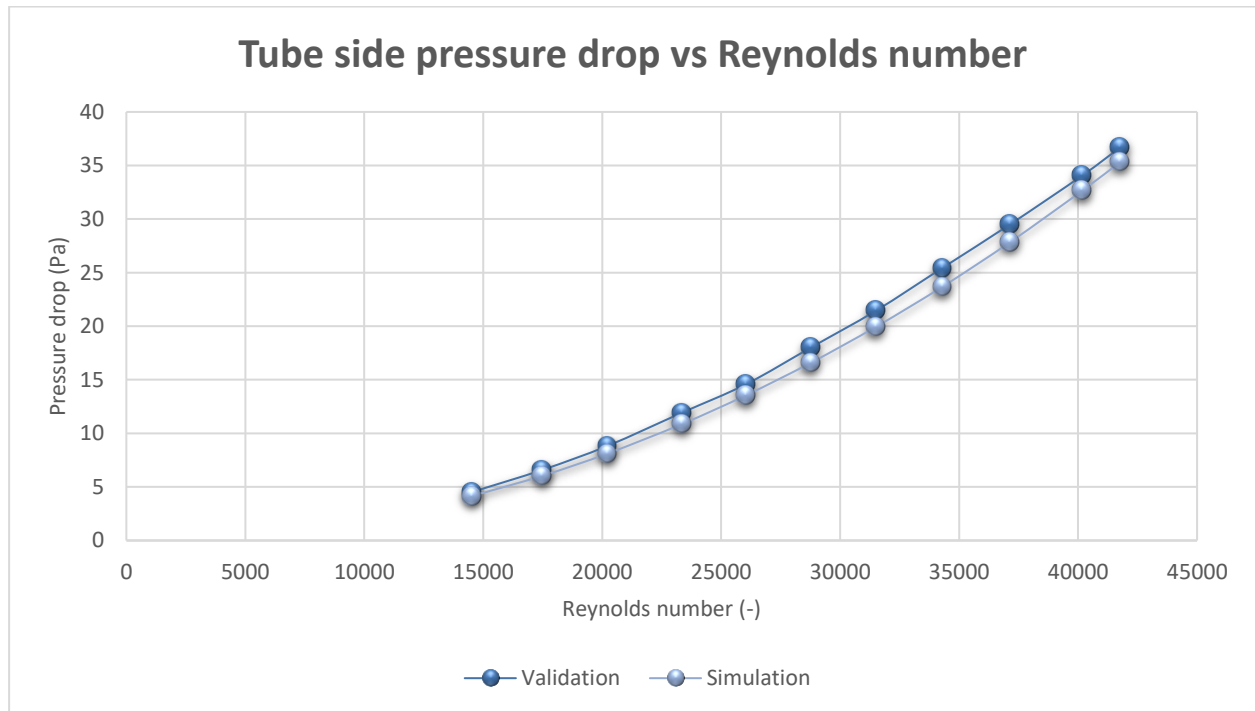


Figure 48: Twisted tube, tube side pressure drop vs tube side Reynolds number validation output.

The graph above is simulated only for the following non dimensional variables:

- $14000 < Re < 42000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 26: Twisted tube, tube side pressure drop vs tube side Reynolds number, graphical annalistic output.

Variable	Value	Unit
Average percentage error	3.96	%
Maximum percentage error	4.43	%
Standard deviation	0.34	-
r^2 value	0.99	-

4.4.4 Shell Side Mass Flow Rate vs the Heat Exchanger Effectiveness

The design parameters and conditions used within the validation study of Lokesh (2017) can be seen in the table below. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

Table 27: Design data and parameters for the validation study (Lokesh, et al., 2017).

Validation Study			
Variable	Water	Water	Unit
Tube length	450		mm
Density	998.200	998.200	kg/m ³
Specific heat capacity	4182.000	4182.000	J/kg-K
Conductivity	0.600	0.600	W/m-K
Viscosity	0.001	0.001	Pa-s
Mass flow rate	0.600	0.200	g/s
Inlet temperature	330.000	380.000	°C

The mass flow rate and effectiveness calculated output, can be validated using the study: “Heat transfer enhancement of double pipe exchanger using twisted tape inserts”. The study compared normal tubes to twisted tape inserted tubes considering different flow angles, such as parallel flow and counter flow.

The obtained results calculated within the study “Heat transfer enhancement of double pipe exchanger using twisted tape inserts”, along with the overlaying simulated output, can be seen in Figure 49 below (Lokesh, et al., 2017).

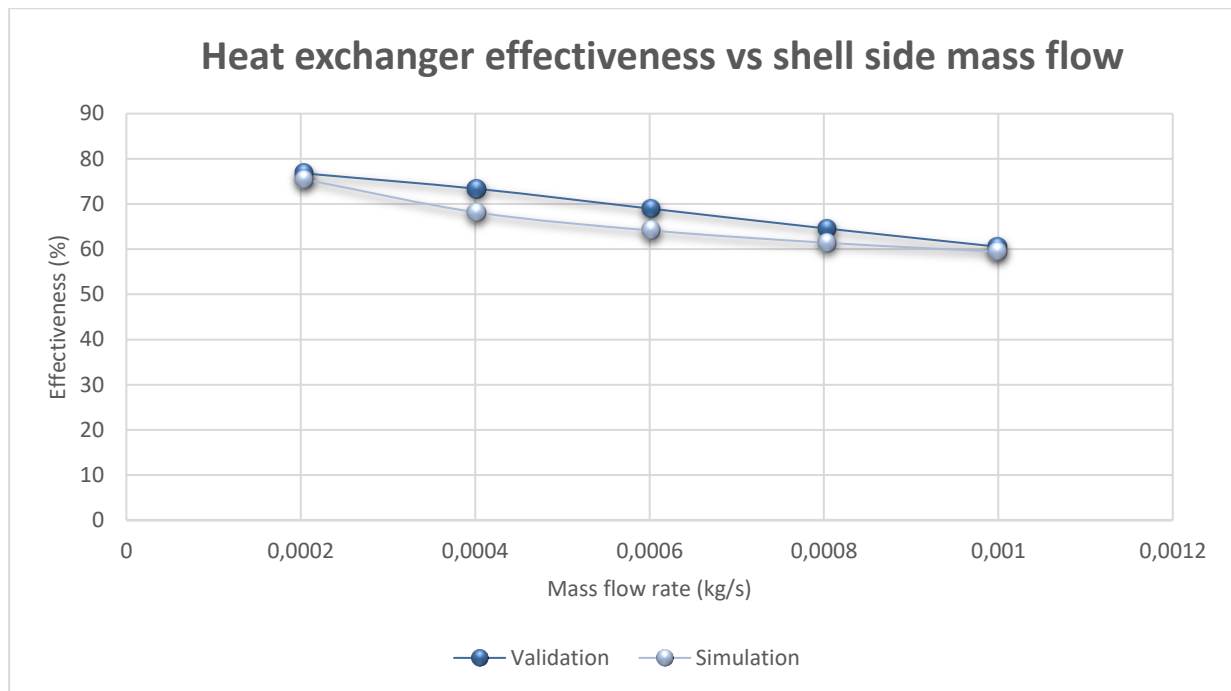


Figure 49: Twisted tube, shell side mass flow rate vs heat exchanger effectiveness validation output.

The graph above is simulated only for the following non dimensional variables:

- $Re > 4000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 28: Twisted tube, shell side mass flow rate vs heat exchanger effectiveness, graphical annalistic output.

Variable	Value	Unit
Average percentage error	4.51	%
Maximum percentage error	7.13	%
Standard deviation	2.65	-
r^2 value	0.91	-

4.4.5 Tube Side Heat Transfer Coefficient vs Tube Pitch

The design parameters and conditions used within the validation study of Barraza-Colon and Picon-Nunez (2022) can be seen in the table below. The design data, stipulated within the table, was used within the EES calculation for the validation of the calculation.

Table 29: Design data and parameters for the validation study (Barraza-Colon & Picon-Nunez, 2022).

Validation Study			
Variable	Value		Unit
Tube Length	9920.00		mm
Shell diameter	1390.00		mm
Hydraulic tube diameter	0.01		mm
Number of Tubes	1601.00		-
Tube side pressure drop	23.44		kPa
Variable	Sea Water	A gas	
Mass flow rate	1.57	1.88	kg/s
Inlet temperature	30.00	89.20	°C
Viscosity	-	0.35	cP
Density	1160.00	971.80	kg/m ³
Conductivity	0.21	670.00	W/m-°C

The tube side heat transfer coefficient with the corresponding tube pitch, can be validated using the obtained results with the study: “Improved heat recovery using twisted tube exchanger technology”. The graphical trend calculated can be used to validate the accuracy of the EES simulated output.

The obtained results compared with Barraza-Colon and Picon-Nunez (2022) can be seen in Figure 50 below.

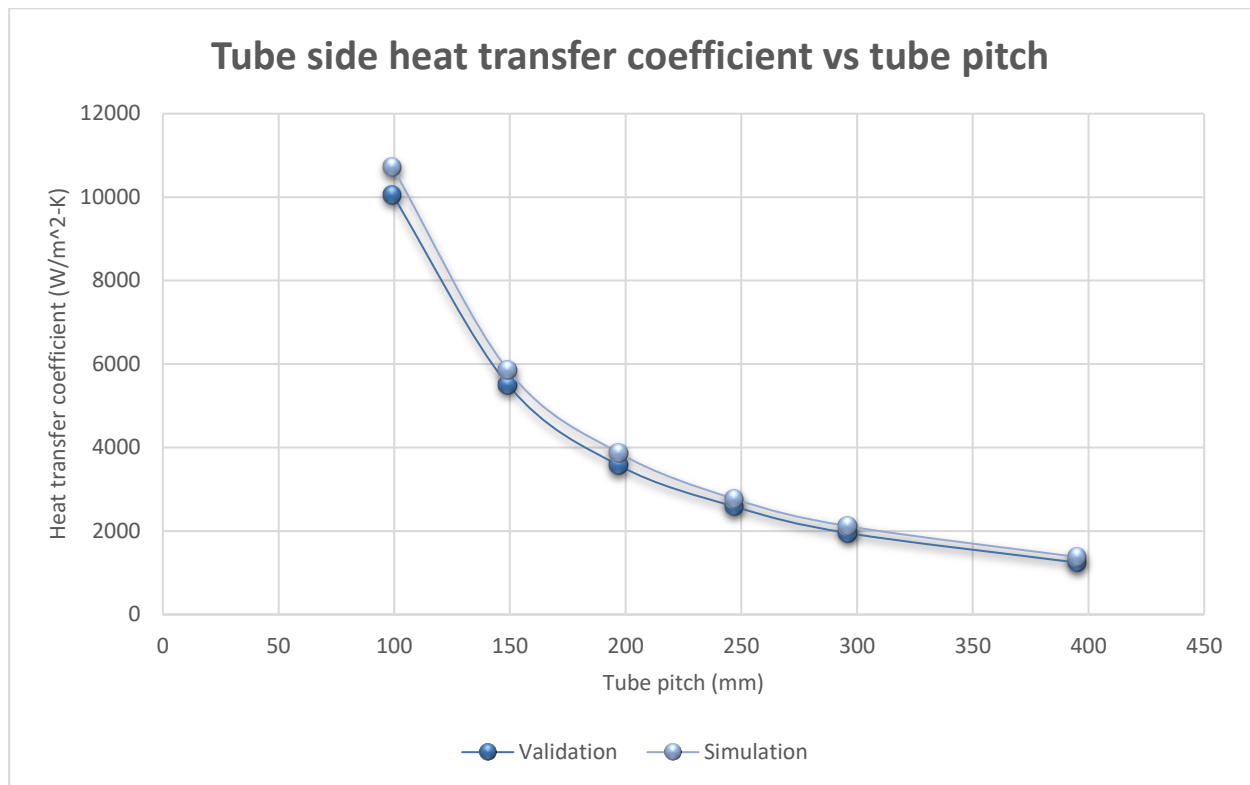


Figure 50: Twisted tube, tube side heat transfer coefficient vs tube pitch validation output.

The graph above is simulated only for the following non dimensional variables:

- $2000 < Re < 6000$

By means of analysing the above graphical validation output, the following calculated conclusions can be made.

Table 30: Twisted tube, tube side heat transfer coefficient vs tube pitch, graphical annalistic output.

Variable	Value	Unit
Average percentage error	6.85	%
Maximum percentage error	7.74	%
Standard deviation	0.84	-
r^2 value	0.99	-

4.4.6 Energy Balance

Considering the studies used for the validation of the EES simulation, the following energy balances were obtained for each validation study used.

Table 31: Twisted tube, validation study energy balance output.

Study	Energy Balance [kW]	Difference [%]
“Convective thermal-hydraulic performance in twisted tubes: From the perspective of symmetrical cross-sectional shapes” Li (2023) .	Inlet Energy = 151.6 kW Outlet Energy = 150 kW <u>Balance = 1.6 kW</u>	1.05 %
“Experimental research on heat transfer and flow resistance properties in spiral twisted tube heat exchanger” Dong (2020) .	Inlet Energy = 20.6 kW Outlet Energy = 20.4 kW <u>Balance = 0.2 kW</u>	0.97 %
“Heat transfer enhancement of double pipe exchanger using twisted tape inserts” Lokesh (2017) .	Inlet Energy = 18.4 W Outlet Energy = 18.3 kW <u>Balance = 0.1 kW</u>	0.54 %
“Improved heat recovery using twisted tube exchanger technology” Barraza-Colon & Picon-Nunez (2022) .	Inlet Energy = 72.8 kW Outlet Energy = 72.7 kW <u>Balance = 0.1 kW</u>	0.14 %

The calculated energy balance indicates minimal to no difference between the inlet of the heat exchanger model to that measured at the outlet of the heat exchanger. The small difference within the obtained energy balance could be explained in the sense that some heat energy is absorbed by the environment during the operation of the heat exchanger.

4.5 Summary

The twisted-tube heat exchanger model was validated by comparing numerical and experimental data from multiple well-established investigations with EES computed outputs. Heat transfer coefficients, pressure drop, Reynolds numbers, mass flow rates, and heat exchanger efficiency were important performance indicators. To ensure the model appropriately captures both fluid flow and heat transfer phenomena in twisted-tube geometries, each validation scenario concentrated on a distinct aspect of thermal-hydraulic behaviour.

Using data from Barraza-Colon and Picon-Nunez (2022), the comparison of the tube side heat transfer coefficients and Reynolds numbers showed excellent correspondence, with an R^2 value of 0.99 and average errors of 3%. Similar to this, the pressure drop on the shell side and tube side were verified against Dong (2020), demonstrating once more a high correlation and small error margins in the turbulent, transitional, and laminar regimes. These results confirm that the model reliably captures flow resistance characteristics influenced by twisting effects and secondary flows within the tubes.

Shell side mass flow rate and effectiveness were compared to the dataset of Lokesh (2023) in order to further assess the heat exchanger performance. The overall trend and accuracy remained acceptable, with an R^2 of 0.91, despite the effectiveness predictions showing slightly higher variability as expected due to geometric differences between twisted tubes and twisted tape inserts. Additionally, data from Barraza-Colon and Picon-Nunez (2022) was used to validate the impact of the twist pitch on the tube side heat transfer coefficient, demonstrating that the model accurately depicts pitch dependent enhancement mechanisms.

Energy balance assessments for all validation cases revealed very small differences between inlet and outlet energy, ranging from 0.65% to 2.13%. These small discrepancies indicate strong thermodynamic consistency within the EES model, and suggest that any minor imbalances may be due to unavoidable environmental heat losses or rounding effects. Overall, the validation results of the above chapter demonstrate that the EES simulation accurately reproduces published thermal-hydraulic behaviour of twisted tube heat exchangers.

CHAPTER 5 PETROCHEMICAL OPERATED SHELL-AND-TUBE HEAT EXCHANGER SIMULATION OUTPUT & COMPARISON

5.1 Introduction

This chapter is based on comparing the calculated factors that, considering the design of the plain tube and twisted tube shell and tube heat exchanger, affects the overall efficiency and performance of the STHE. Within this chapter, the factors will be analysed and compared for the PTSTHE and the TTSTHE design within the petrochemical industry of South Africa.

By means of a detailed comparison, both heat exchanger designs can be analysed and compared using the same geometrical outline and operating conditions. Both heat exchanger designs also provide a financial implication to the petrochemical plant, which will be analysed based on the steam usage and water supply during the operation of the two different designed heat exchangers.

The detailed comparison of both the PTSTHE and the TTSTHE can be seen in the following few sections.

5.2 Petrochemical Operated Plain Tube Shell-and-Tube Heat Exchanger Design Conditions

The design conditions used for the heat exchanger calculations is obtained from an operational PTSTHE within Mpumalanga, South Africa. The design conditions specified for the operational heat exchanger can be seen in the table below. These design conditions were manipulated and multiplied with a specifically chosen factor in order to still be able to make use of accurate data for the calculation, but to protect the petrochemical company's right to privacy.

Table 32: Design conditions of plain tube shell-and-tube heat exchanger.

Variable	Shell Side	Tube Side	Unit
Product	Steam	Water	-
Design Pressure	500	500	kPa
Design Temperature	200	150	°C
Inlet Temperature	123	80	°C
Outlet Temperature	105	90	°C
Inlet Pressure	306	150	kPa
Outlet Pressure	305	135	kPa
Inlet Fluid Velocity	15	55	m/s

Variable	Shell Side	Tube Side	Unit
Outlet Fluid Velocity	13	20	m/s
Bulk Tube Pressure		138	kPa
Bulk Tube Temperature		88	°C
Baffle Spacing		300	mm
Wall Pressure		170	kPa
Wall Temperature		102.5	°C
Horizontal Tube Spacing (S_1)		24	mm
Vertical Tube Spacing (S_2)		18	mm
Tube Outside Diameter		19.05	mm
Tube Thickness		1.65	mm
Tube Length		4260	mm
Shell Length		4260	mm
Clearance		4.95	mm
Pitch		24	mm
Pump Efficiency		58	%
Corrosion Allowance	0	0	mm
Capacity	0.58	0.69	m ³
Number of Passes	1	1	-
Mass (Full/Empty)		4054/2723	kg

5.3 Twisted Tube Parameter Investigation

In order to select the best suited parameters for the twisted tube design, the industry available parameters will be evaluated and analysed by means comparing the performance of the different parameters to each other, whereafter the best performing parameters will be selected and used for the evaluation of the TTSTHE.

The data used for the parameter investigation, considering the aspect ratio and twist pitch length of the TTSTHE, was obtained from the study “Improved heat recovery using twisted tube exchanger technology” (Barraza-Colon & Picon-Nunez, 2022). The used data can be seen in the table below.

Table 33: Standard aspect ratios for twisted tube design (Barraza-Colon & Picon-Nunez, 2022).

Item	a (mm)	b (mm)	Aspect Ratios	Unit
1	7.0	35.0	0,20	-
2	9.0	33.0	0,27	-
3	11.0	31.0	0,36	-
4	13.0	29.0	0,45	-
6	15.0	27.0	0,56	-
7	17.0	25.0	0,68	-
8	19.0	23.0	0,83	-
9	21.0	21.0	1,00	-
10	23.0	19.0	1,21	-
11	25.0	17.0	1,47	-
13	27.0	15.0	1,80	-
15	29.0	13.0	2,23	-
16	31.0	11.0	2,82	-
17	33.0	9.0	3,67	-
18	35.0	7.0	5,00	-

Table 34: Standard twist pitch length for twisted tube design (Barraza-Colon & Picon-Nunez, 2022).

Item	Twist Pitch Length	Unit
1	160	mm
2	200	mm
3	250	mm
4	300	mm
5	400	mm

5.3.1 Aspect ratio

In order to select the best suited aspect ratio for the twisted tube design, the different ratios were inserted into the EES calculation, and the results of each on the heat transfer coefficient and tube side pressure drop were analysed. According to specialists working within the twisted tube heat exchanger design industry, the aspect ratios can be personalized for each application according to the deliverable requirements of the heat exchanger. Thus, the aspect ratios that were used for the analysis, were specified considering the tube outside diameter of 19.05 mm used for the plain

tube heat exchanger analysis. This specification was made to ensure that the same tubes are used for both applications. The used aspect ratios can be seen in Table 33.

By means of using the available aspect ratios specified within the table above, the following output was reached when analysing the ratios within the EES calculation.

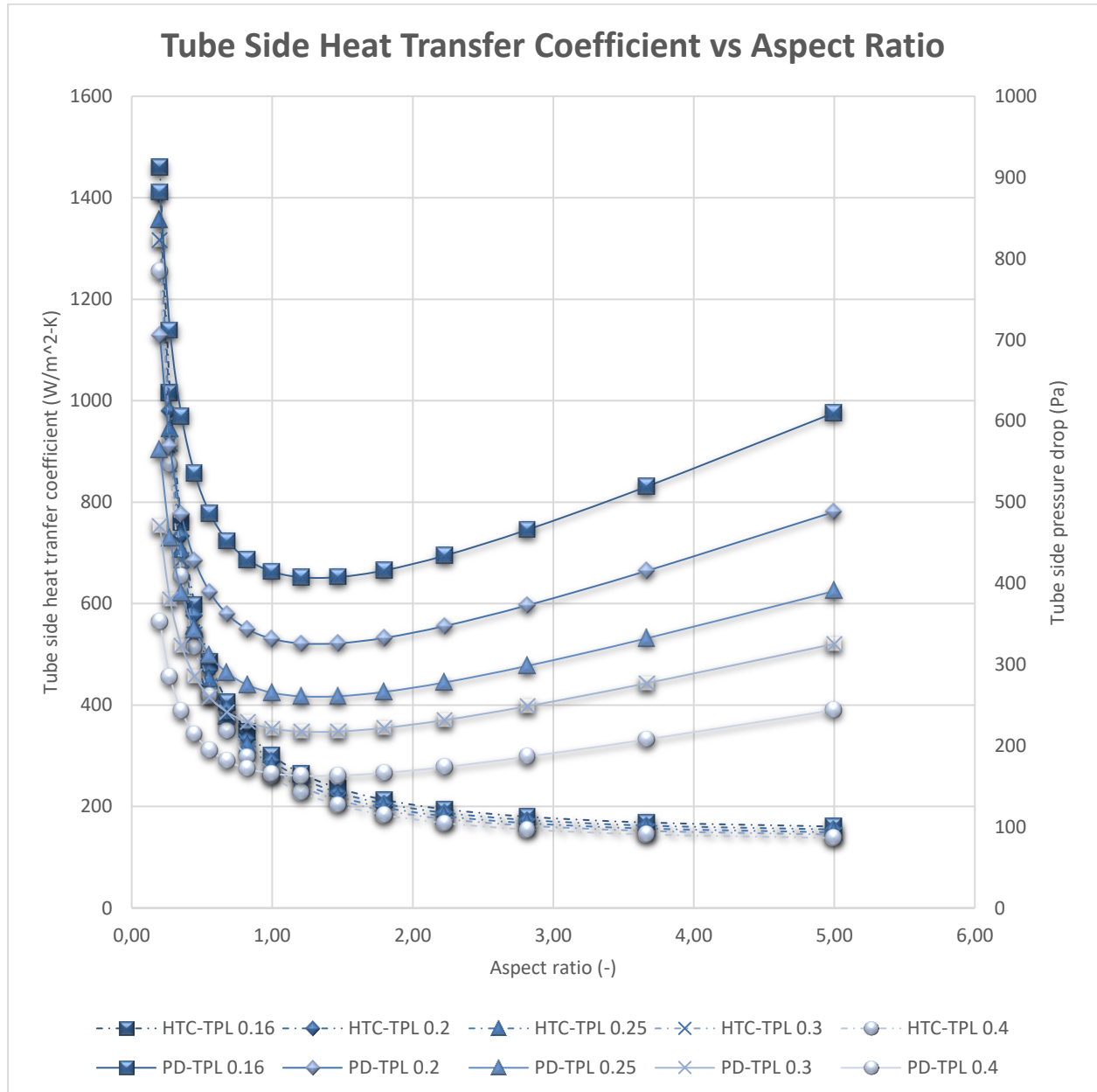


Figure 51: Heat transfer coefficient and pressure drop analysis for a change in aspect ratio

Considering the output within the figure above, it can be seen that the tube side pressure drop decreases as the aspect ratios increases from 0.2 to 1.21, and furthermore increases as the aspect ratios increases from 1.21 to 5. It is also clear that the tube side pressure drop decreases as the aspect ratio increases.

Analysing the above figure, it is clear that the optimal pressure drop for the TTSTHE would be found in the aspect ratio ranges of 0.35 to 2.11. Thus, the output was redefined according to the optimal ranges for the pressure drop experienced by the TTSTHE. The redefined output can be seen in Figure 52 below.

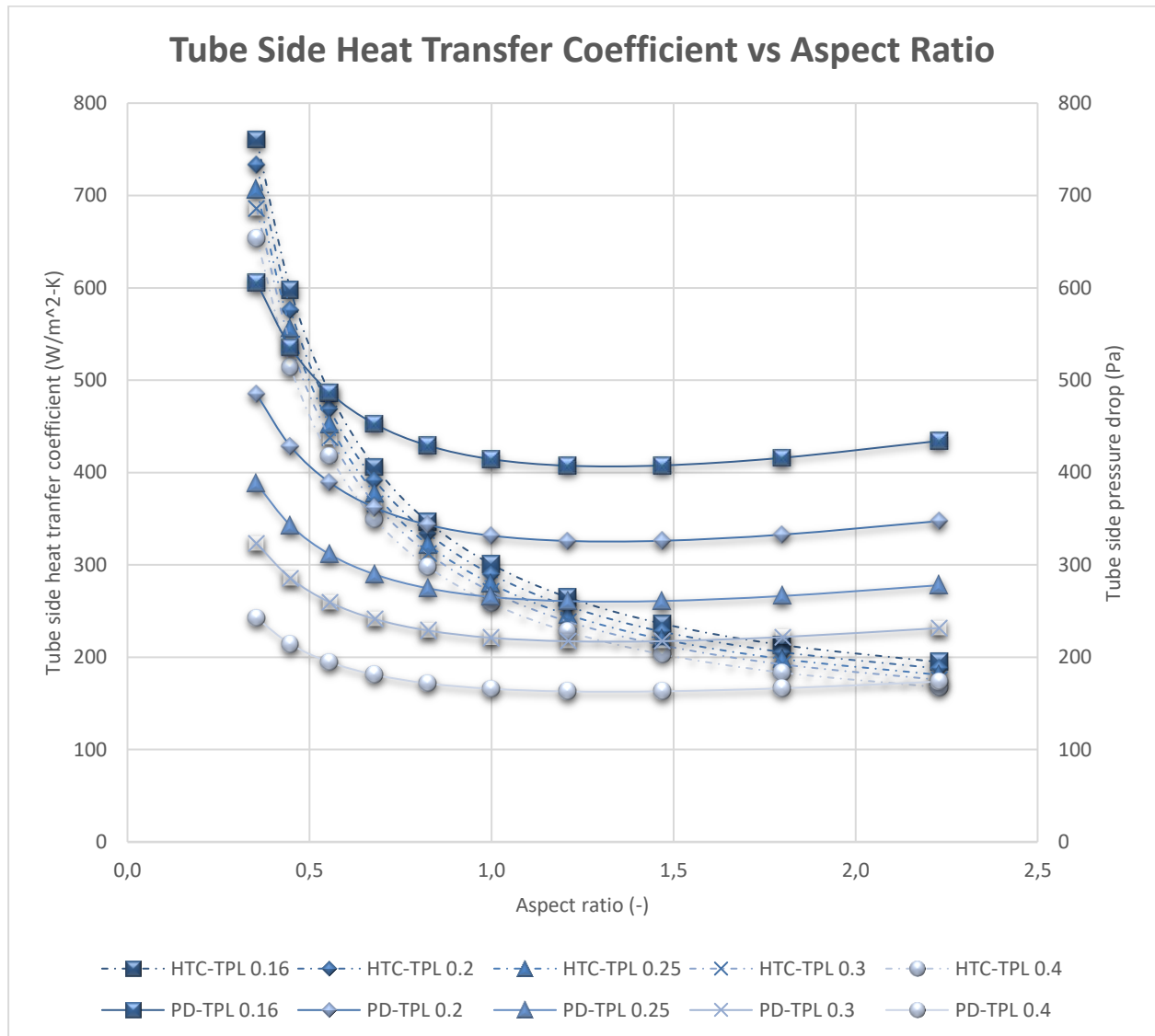


Figure 52: Tube side heat transfer coefficient and aspect ratio analysis.

Analysing the above figure, it is clear that as the tube aspect ratio increases, the tube side heat transfer coefficient decreases; however, the tube side pressure drop expresses a parabolic reaction to the aspect ratio increase. It is clear that at the specified aspect ratio of 1.21, the pressure drop of each twist pitch length analysed, is at its lowest.

In order to better understand the effect that a change in the specified aspect ratio has on the TTSTHE, an additional evaluation was done. It compares the tube side pressure drop of the

TTSTHE with the tube surface area. This evaluation expresses during which aspect ratio the best tube surface area is found and what the effect on the tube side pressure drop is.

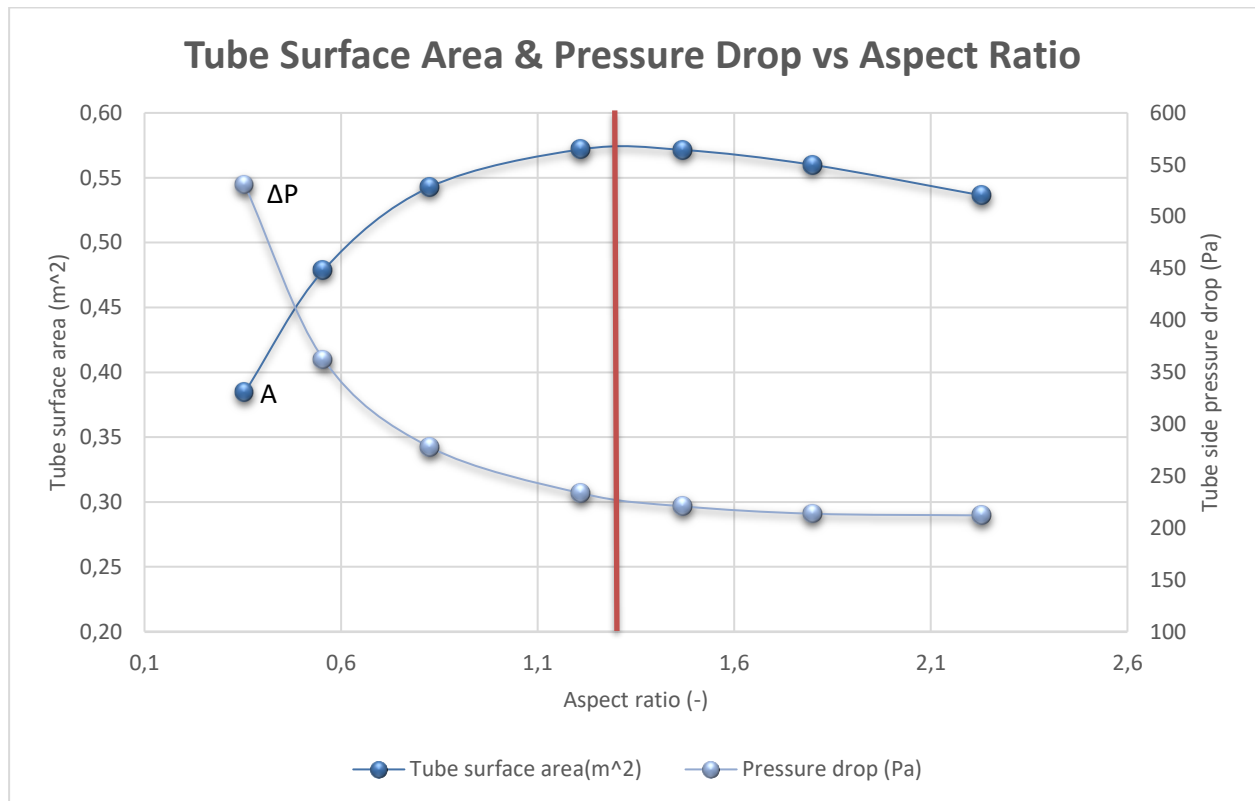


Figure 53: Parameter plot to show the design space and opportunity for optimisation of the aspect ratio with a constant heat duty (Q).

Evaluating the effect of the change in aspect ratios on the tube side heat transfer coefficient and pressure drop, it can be seen that the specified aspect ratio of 1.21 presented the best tube surface area and a respectfully low tube side pressure drop within the above figure.

However, in order to better express how the calculated pressure drop within the above figure affects the TTSTHE, the below table was added to indicate the % pressure drop used when considering the operational allowable pressure drop of the TTSTHE.

Table 35: Tube side pressure drop % analysis.

Twist Length (m)	Pitch	Design Pressure (kPa)	Pressure Drop (kPa)	% Design Pressure (%)
	0.16	15	0.644	4.29
	0.20	15	0.516	3.44
	0.25	15	0.412	2.75
	0.30	15	0.343	2.29
	0.40	15	0.258	1.72

Analysing the above outputs generated by means of the EES calculation, it can be seen that an aspect ratio of 1.21 will be the best selection for the twisted tube design using the operating conditions of a petrochemical heat exchanger. This aspect ratio was selected because it provides the best tube surface area and the best-balanced pressure drop for the TTSTHE. The aspect ratio of 1.21 also provides a relatively high tube side heat transfer coefficient, which ensures a balanced TTSTHE.

5.3.2 Twisted tube, twist pitch length

In order to select the best suited twist pitch length for the twisted tube design, the different lengths were inserted into the EES calculation and the results of each on the tube side heat transfer coefficient and pressure drop were analysed. The twist pitch lengths that were used for the analysis can be seen in Table 34 above.

In order to avoid an overload of results but still provide an accurate conclusion, it was chosen to only analyse five data sets, using the coloured data shown in Table 33 above.

By means of using the available twist pitch lengths and aspect ratios specified within the tables above, the following outputs were reached when analysing the EES calculation, as can be deduced from the concluded figure below.

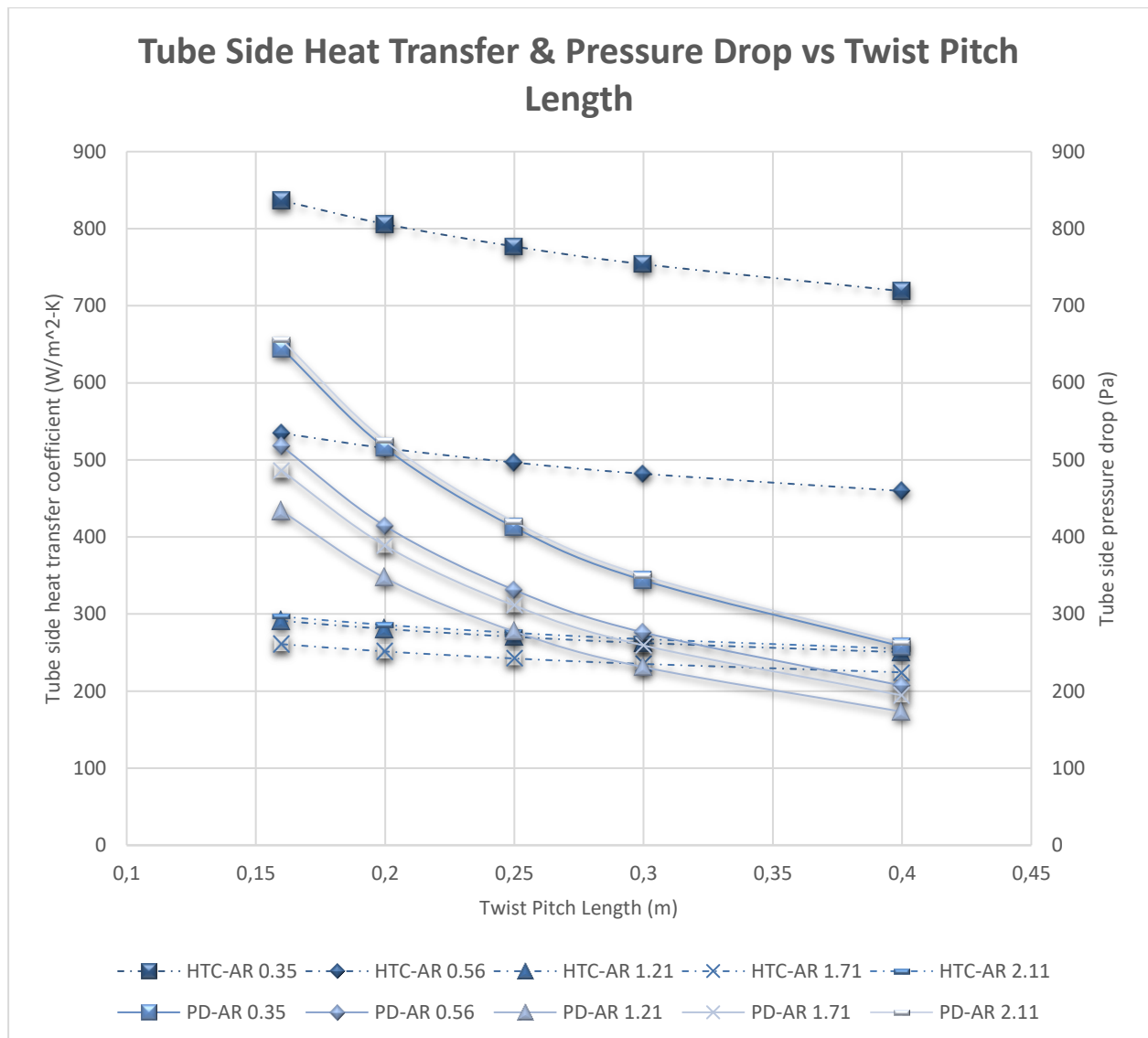


Figure 54: Compared analysis comparison considering twist pitch lengths for different aspect ratios.

As seen in Figure 54 above, the tube side heat transfer coefficient decreases as the twist pitch length increases. It is also clear from the above graph that an increase in the tube's aspect ratio provides a decrease in the generated tube side heat transfer coefficient. The above figure also indicates that as the tube pitch length increases, the tube side pressure drop decreases.

Considering the above graph and the aspect of evaluation – the tube twist pitch length – it is important to select a twist pitch length that will be best suited for the operation of the heat exchanger system. Analysing the outputs above, one can argue that a small twist pitch length of 160 mm provides the best heat transfer coefficient and highest pressure drop to occur with the

TTSTHE. Whereas a larger twist pitch length of 400 mm provides the lowest tube side heat transfer coefficient and pressure drop to occur within the TTSTHE.

In order to better define Figure 53 above, and present the output reached for the twist pitch length over the generated tube surface area and pressure drop, the following figure was generated.

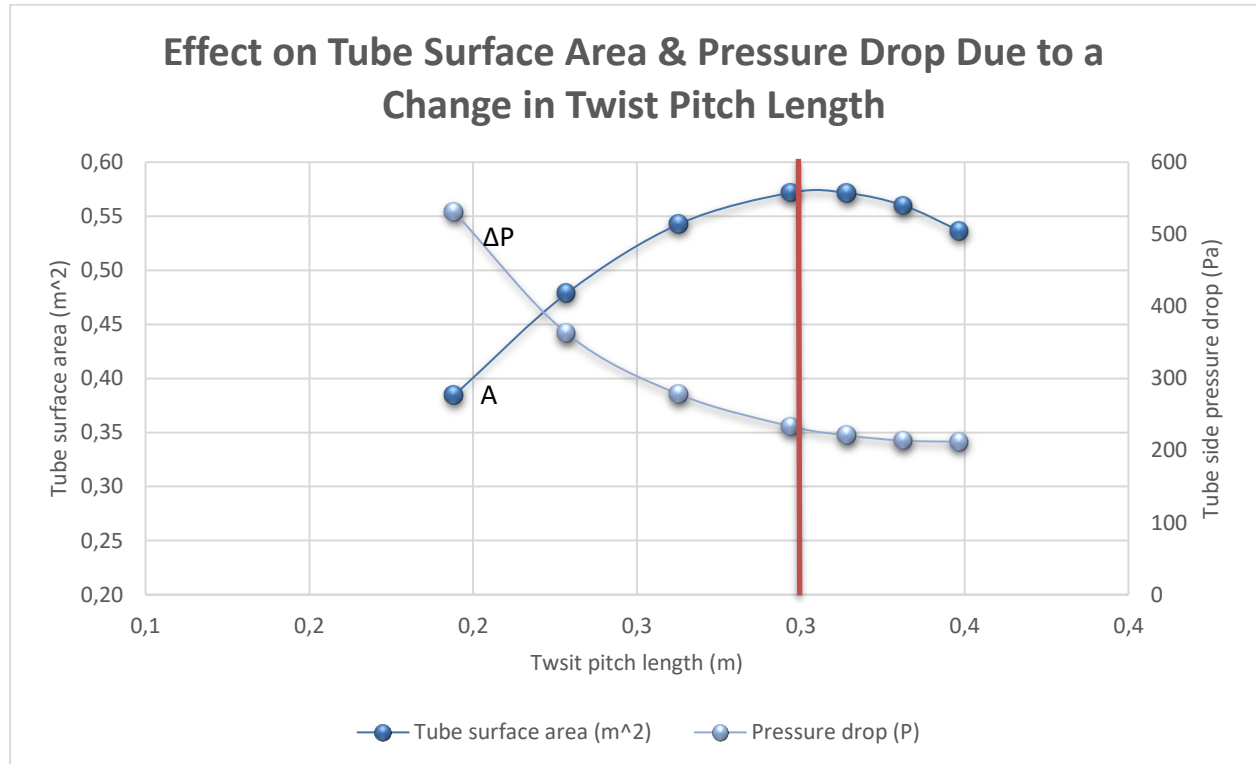


Figure 55: Parameter plot to show the design space and opportunity for optimisation of the twist pitch length with a constant heat duty (Q) due to the change observed regarding the pressure drop and tube surface area.

Within Figure 55 above, it can be seen that during the parameter plot, the tube side pressure drop decreases as the twist pitch length of the tube increases from 160 mm to 350 mm. It is also clear that the tube surface area increases as the twist pitch length increases from 160 mm to 300 mm, but furthermore decreases as the twist pitch length increases from 300 mm to 350 mm. It can thus be seen that a twist pitch length of 300 mm provides the best balance for the selection between the generated pressure drop and tube surface area. Thus, a twist pitch length of 300 mm was selected for the TTSTHE.

5.4 Petrochemical Operated Twisted Tube Shell-and-Tube Heat Exchanger Design Conditions

The design conditions that will be used for the calculation of the TTSTHE design will be the same as specified for the PTSTHE. The design conditions for the twisted tube heat exchanger can thus be seen in Table 32. The use of the data within this table allows for an accurate comparison

between the plain tube and twisted tube heat exchanger designs with the same specified conditions.

The outlet temperature for the twisted tube heat exchanger was calculated based on the conclusions in Samruaisin (2019), where it was realised that the twisted tube design presents a 59.4% increase in presented heat transfer results in comparison to plain smooth tubes.

The TTSTHE design also presents fixed geometry parameters that should be known and accounted for during the analysis. The parameters can be seen in the table below.

Table 36: Twisted tube conditions and geometry constraints.

Variable	Shell Side	Tube Side	Unit
Outlet Temperature		90.0	°C
Tube Length (L)	-	4260.0	mm
Twist Pitch Length (P)	-	300.0	mm
Vertical Tube Outside Diameter (B)	-	22.3	mm
Vertical Tube Inside Diameter (b)	-	19	mm
Horizontal Tube Outside Diameter (A)	-	26.3	mm
Horizontal Tube Inside Diameter (a)	-	23	mm
Aspect Ratio	-	1.21	-

5.4.1 Specific Comparison

In order to compare the specific outcomes for each heat exchanger design under the specified petrochemically operated conditions, the following table was constructed. The table below presents a detailed comparison between the PTSTHE and TTSTHE designs for the operating conditions of the petrochemical plant obtained within Mpumalanga, South Africa.

Table 37: Petrochemical plant operation comparison between PT & TT, STHE.

Variable	PTSTHE	TTSTHE	Units
Tube side fluid	Water	Water	-
Shell side fluid	Steam	Steam	-
Tube side heat transfer coefficient	342.50	539.40	W/m ² K
Tube side mass flow rate	572.70	440.90	kg/s
Shell side mass flow rate	320.70	284.50	kg/s
Tube side pressure drop	115.3	332.6	Pa

Variable	PTSTHE	TTSTHE	Units
Shell side pressure drop	814.2	1087.1	Pa
% Tube side pressure drop of allowable pressure drop	0.76	2.22	%
% Shell side pressure drop of allowable pressure drop	5.43	7.25	%
Tube side heat duty	24050.0	24050.0	kW
Shell side heat duty	24050.0	24050.0	kW

Considering the above table, the outputs for the specific heat exchanger design with the specified operating conditions as obtained from the petrochemical industry within Mpumalanga, South Africa, were analysed and compared for both the PTSTHE and the TTSTHE designs.

Analysed, it can be seen that the TTSTHE presented a much higher heat transfer coefficient comparing to the output reached for that of the PTSTHE. It is however also clear that the twisted tube design generated a higher-pressure drop than that of the plain tube design for the specific design conditions of the operational STHE located within Mpumalanga, South Africa. However, the pressure drop difference between the PTSTHE and TTSTHE is low, only a 1.82% difference between the two STHE designs. As a result of the constant heat duty produced, it can also be seen that the TTSTHE used much lower shell and tube side mass flow rates to achieve the desired heat duty for the designed heat exchanger than the PTSTHE.

5.5 General Comparison

This section will consist of the comparison between the obtained EES calculations for the plain tube and TTSTHE's. The results will be compared and explained in terms of the heat exchanger performance with the specified South African petrochemical industry's tube conditions and geometry constraints.

The aspects and outputs that will be compared within this chapter entails:

- Tube side heat transfer coefficient vs the tube side fluid Reynolds number.
- Shell side pressure drop vs the shell side Reynolds number.
- Tube side pressure drop vs the tube side Reynolds number.
- Shell side mass flow rate vs the heat exchanger effectiveness.

5.5.1 Tube side heat transfer coefficient vs the tube side fluid Reynolds number.

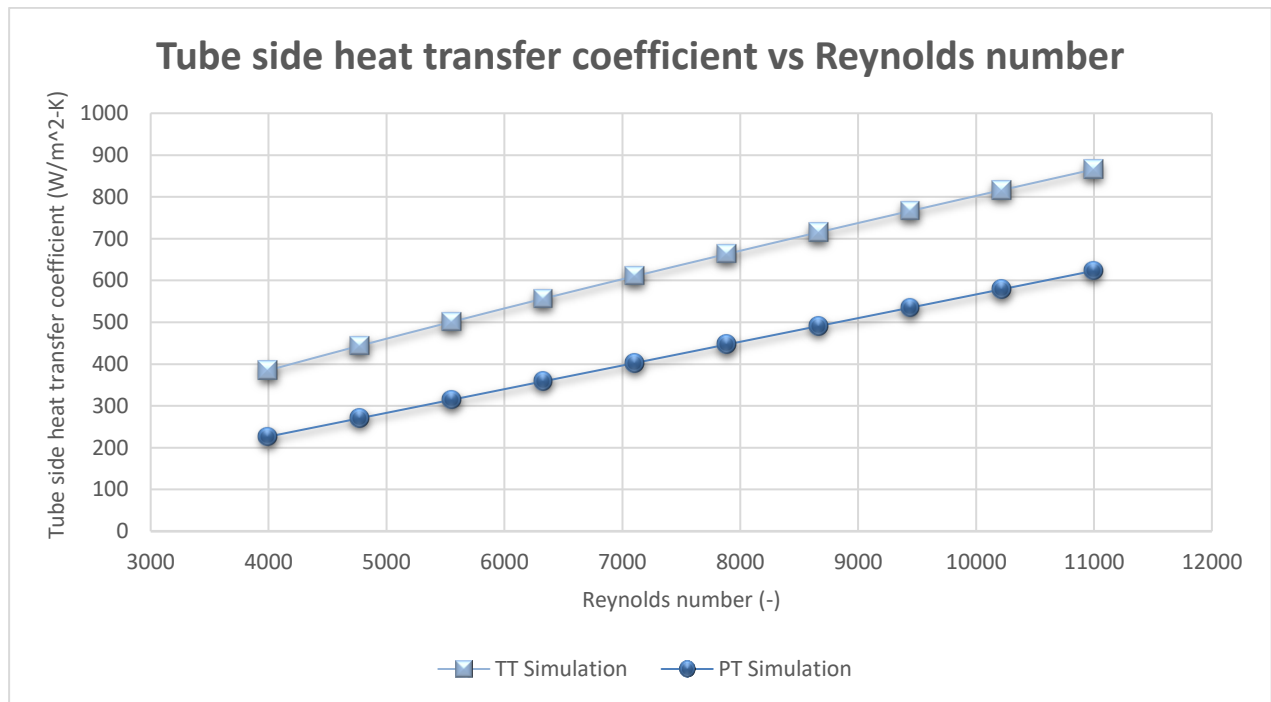


Figure 56: Plain tube vs twisted tube heat transfer coefficient over Reynolds number comparison.

When analysing the obtained output from the plain tube heat exchanger in comparison to that of the twisted tube heat exchanger, it can be seen that both heat exchangers present an increase in the physical calculated heat transfer coefficient as the fluid Reynolds number increases.

Comparing the physical heat transfer generated between the plain tube and the twisted tube, the following conclusion was reached.

Table 38: Plain tube vs twisted tube heat transfer coefficient over fluid Reynolds number output comparison.

	Reynolds Number	Heat Transfer Coefficient (W/m ² -K)
PTSTHE	4000	620
TTSTHE	4000	795
PTSTHE	11000	900
TTSTHE	11000	1750
Change of TTSTHE	4000-11000	955
Change of PTSTHE	4000-11000	280

From the calculated output, it is clear that the twisted tube heat exchanger design presented better physical heat transfer during operation than the plain tube heat exchanger. An increase in the produced Reynolds number generates a larger change in the physical heat transfer coefficient

generated, when comparing the output reached by means of the twisted tube heat exchanger design to that of the plain tube heat exchanger design.

It can be seen from the graphic comparison between the plain tube and TTSTHE, that the twisted tube heat exchanger exerted $175 \text{ W/m}^2\text{-K}$ more than the plain tube heat exchanger at a Reynolds number of 4000. Also, the twisted tube presented $850 \text{ W/m}^2\text{-K}$ more than the plain tube heat exchanger at a Reynolds number of 11000.

Additionally, the change in heat transfer coefficient produced by the PTSTHE for the tube side Reynolds number range of 4000-11000 can be calculated as $280 \text{ W/m}^2\text{-K}$. The change in heat transfer coefficient produced by the TTSTHE for the tube side Reynolds number range of 4000-11000 can be calculated as $955 \text{ W/m}^2\text{-K}$. Thus, the twisted tube exerted $675 \text{ W/m}^2\text{-K}$ more than the PTSTHE for the change in Reynolds numbers 4000-11000.

Overall, it is a clear conclusion that when considering the tube side fluid Reynolds number and the heat transfer coefficient generated in accordance therewith, considering both heat exchanger designs, the twisted tube heat exchanger design presented a much better heat transfer coefficient than the PTSTHE.

5.5.2 Shell side pressure drop vs the shell side Reynolds number.

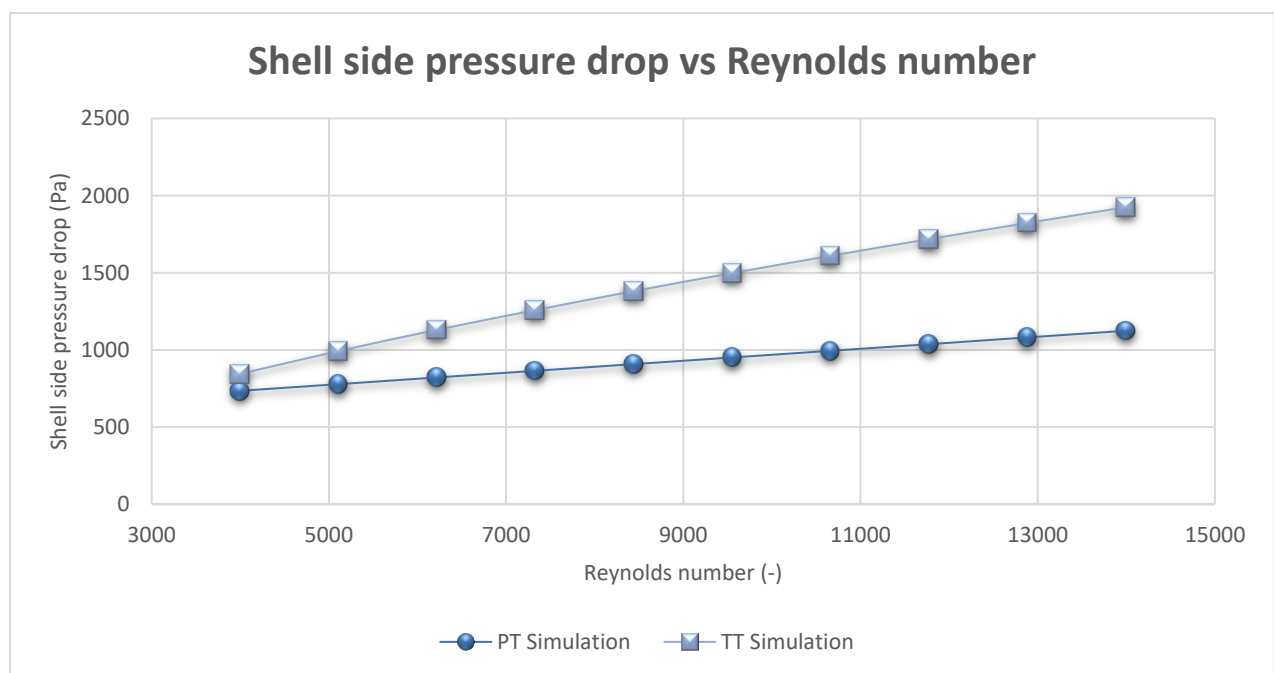


Figure 57: Plain tube vs twisted tube; shell side pressure drop over Reynolds number comparison.

When considering the obtained output from the plain tube heat exchanger in comparison to that of the twisted tube heat exchanger, both heat exchangers present an increase in the measured shell side pressure drop as the fluid Reynolds number increases.

Comparing the pressure drop measured between the plain tube and the twisted tube, the following conclusion was reached.

Table 39: Plain tube vs twisted tube; shell side pressure drop over fluid Reynolds number output comparison.

	Reynolds Number	Shell Side Pressure Drop (Pa)
PTSTHE	4000	740
TTSTHE	4000	810
PTSTHE	14000	1150
TTSTHE	14000	1900
Change of PTSTHE	4000-14000	410
Change of TTSTHE	4000-14000	1090

Comparing the obtained results between the PTSTHE and that of the TTSTHE, it can be seen that the twisted tube heat exchanger experiences more pressure drop on the shell side to occur than that of the plain tube heat exchanger.

Considering the change in pressure drop between the two heat exchanger designs, it is clear that the plain tube heat exchanger experienced a 410 Pa increase in shell side pressure drop from a Reynolds number of 4000-14000. Analysing the change in shell side pressure drop for the twisted tube heat exchanger design, the twisted tube heat exchanger experienced a 1090 Pa increase in pressure drop from a Reynolds number of 4000-14000.

Evidently, the pressure drop experienced by the two heat exchanger designs differ with 70 Pa at a Reynolds number of 4000, and with 750 Pa at a Reynolds number of 14000. The reasoning behind the increased twisted tube shell side pressure drop over the normal shell side pressure drop of the plain tube heat exchanger, could be explained in the sense of increased fluid friction generated. This increased friction is caused by the induced turbulence created by the swirl-like flow from the twisted geometry of the tubes. As the Reynolds number increases, the turbulence generated on the shell side of the heat exchanger also linearly increases, causing an increase in friction and turbulence, overall increasing the generated pressure drop.

Therefore, it can be concluded that the TTSTHE design experiences a higher change in shell side pressure drop than that of the PTSTHE, as the Reynolds number of the shell side fluid increases.

5.5.3 Tube side pressure drop vs the tube side Reynolds number.

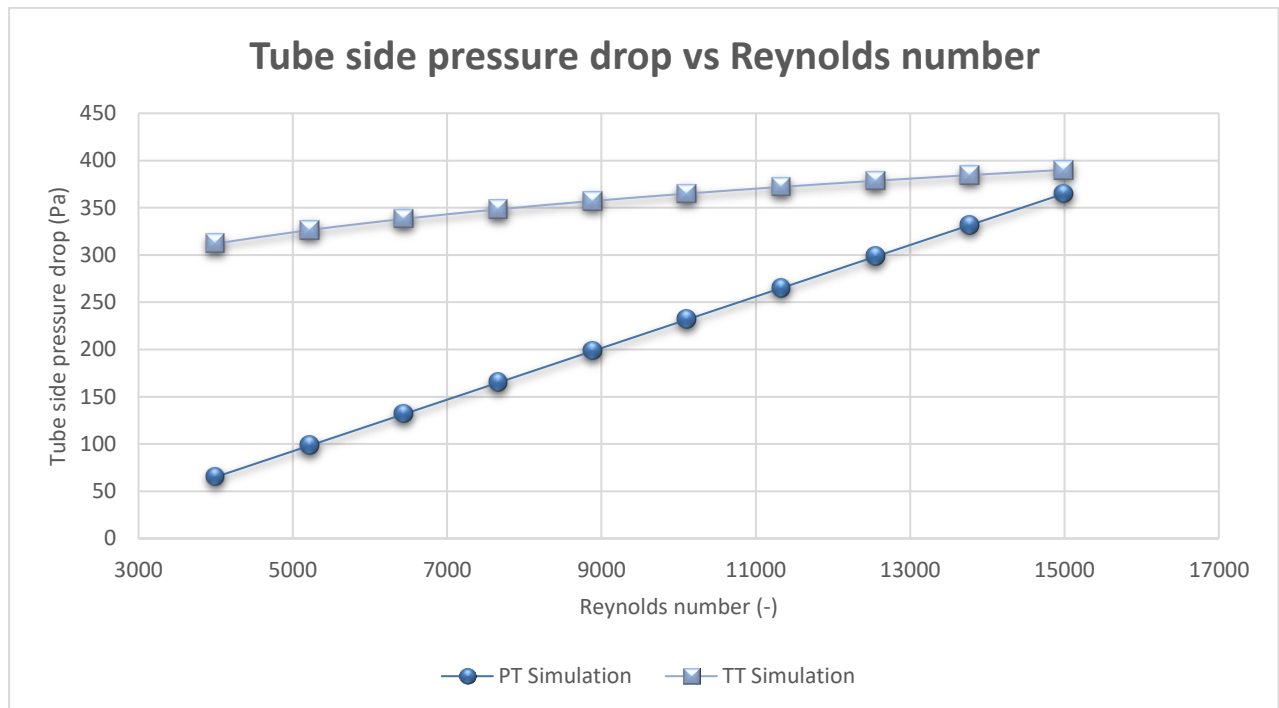


Figure 58: Plain tube vs twisted tube; tube side pressure drop over Reynolds number comparison.

When analysing the obtained output from the plain tube heat exchanger in comparison to that of the twisted tube heat exchanger, it can be seen that both heat exchangers present an increase in the measured tube side pressure drop as the fluid Reynolds number increases.

Comparing the tube side pressure drop measured between the plain tube and the twisted tube, the following conclusion was reached.

Table 40: Plain tube vs twisted tube; tube side pressure drop over fluid Reynolds number output comparison.

	Reynolds Number	Tube Side Pressure Drop (Pa)
PTSTHE	4000	60
TTSTHE	4000	320
PTSTHE	15000	360
TTSTHE	15000	400
Change of PTSTHE	4000-15000	300
Change of TTSTHE	4000-15000	80

Comparing the obtained results between the PTSTHE and that of the TTSTHE, it can be seen that the twisted tube heat exchanger experiences more pressure drop on the tube side to occur than that of the plain tube heat exchanger. Thus, analysing the graphed EES outputs for both

heat exchangers, it can be concluded that the twisted tube experiences more tube side pressure drop than the plain tube heat exchanger with the same design conditions specified.

Considering the change in pressure drop between the two heat exchanger designs, the twisted tube heat exchanger experienced an 80 Pa increase in tube side pressure drop from a Reynolds number of 4000 to 15000. Analysing the change in tube side pressure drop for the plain tube heat exchanger design, it is clear that the plain tube heat exchanger experienced a 300 Pa increase in pressure drop from a Reynolds number of 4000 to 15000.

The pressure drop experienced by the two heat exchanger designs differ with 260 Pa at a Reynolds number of 4000, and with 40 Pa at a Reynolds number of 15000.

Overall, it can be concluded that the TTSTHE design experiences a higher tube side pressure drop than that of the PTSTHE, as the Reynolds number of the tube side fluid increases.

5.5.4 Shell side mass flow rate vs the heat exchanger effectiveness.

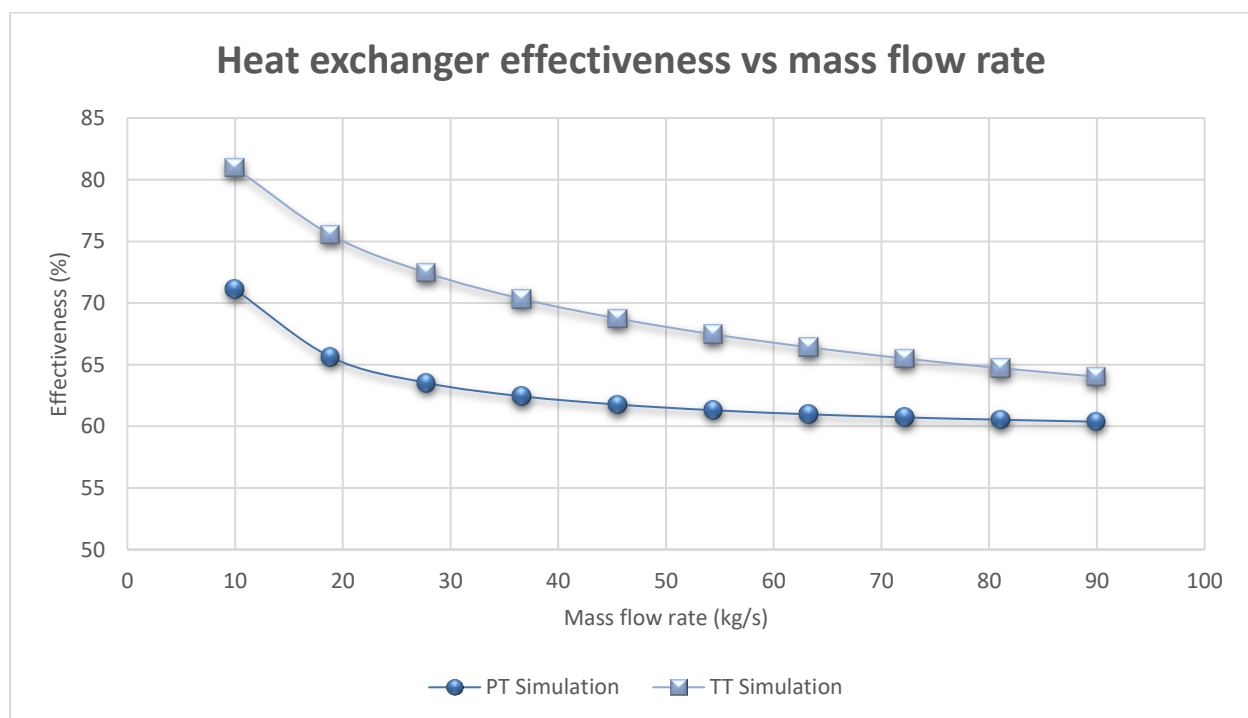


Figure 59: Plain tube vs twisted tube, heat exchanger effectiveness over shell side mass flow rate comparison.

Analysing the obtained output from the plain tube and twisted tube heat exchanger, it can be seen that both heat exchangers' effectiveness decrease as the shell side mass flow rate increases. It is however clear that the effectiveness of the twisted tube heat exchanger is more than that of the plain tube heat exchanger under the same petrochemically operated design conditions, even if both heat exchanger designs experience an exponential decrease in effectiveness as the shell side mass flow rate increases.

Explaining the above figure, both heat exchangers experience a decrease in effectiveness as the shell side mass flow rate increases. As indicated in Figure 57, the pressure drop increases as the Reynolds number increases due to an increase in fluid turbulence. Thus, as the mass flow rate of the shell side fluid increases, the Reynolds number increases, causing an increase in shell side pressure drop. This occurrence results in more energy consumption for all process equipment involved, causing a decrease in process efficiency.

Comparing the heat exchanger effectiveness measured between the plain tube and the TTSTHE's, the following conclusion was reached.

Table 41: Plain tube vs twisted tube, shell side mass flow rate over the heat exchanger effectiveness output comparison.

	Mass Flow Rate (kg/s)	Heat Exchanger Effectiveness (%)
PTSTHE	10	72
TTSTHE	10	82
PTSTHE	90	61
TTSTHE	90	64
Change of PTSTHE	10-90	11
Change of TTSTHE	10-90	18

From the calculated output, it can be seen that the effectiveness of both heat exchangers decreases as the mass flow rate increases. However, at a mass flow rate of 10 kg/s, both heat exchangers present to be the best effective for their design.

Analysing the graph shows that the twisted tube heat exchanger design presents better effectiveness than the plain tube heat exchanger. This appearance can be explained in the sense that the relationship between the amount of pressure drop experienced and the heat transfer coefficient generated by each heat exchanger, influence their effectiveness.

Overall, it is a clear conclusion that when considering the shell side mass flow rate and the heat exchanger effectiveness measured, the effectiveness decreases as the mass flow rate increases for both STHE designs.

5.6 Output Comparison

5.6.1 Tube Length Comparison

In order to fully calculate and compare the difference in tube length needed for both the plain- and twisted tube geometrical setups, the following formula was used (Mills, 1999).

$$L_{tube} = \frac{Q_{tube}}{U_{tube} \cdot P_{tube} \cdot \Delta T_{tube}}$$

Where:

- Q_{tube} = The tube side rate of heat transfer (W)
- U_{tube} = The overall tube side heat transfer rate (W/m²-K)
- P_{tube} = The tube perimeter (m)
- ΔT_{tube} = The LMTD tube side temperature difference (K)

The outputs reached by means of the above equation can be seen in the table below, as well as within the attached annexures.

Table 42: Plain tube heat transfer rate versus tube length.

Heat Transfer Rate	(kW)	Tube Length	(m)
Q_{tube}	24050	L_{tube} (Given)	4.26
Q_{tube}	24050	L_{tube_check} (Calculated)	4.14
		$L_{difference}$	0.12

Table 43: Twisted tube heat transfer rate versus tube length.

Heat Transfer Rate	(kW)	Tube Length	(m)
Q_{tube}	24050	L_{tube} (Given)	4.26
Q_{tube}	24050	L_{tube_check} (Calculated)	3.98
		$L_{difference}$	0.28

By means of analysing the above tables, the following conclusion can be made.

Table 44: Twisted tube and plain tube length evaluation on heat duty generated

Heat Transfer Rate	(kW)	Tube Length	(m)
Q_{tube}	24050	L_{tube_check} (Plain Calculated)	4.14
Q_{tube}	24050	L_{tube_check} (Twisted Calculated)	3.98
		$L_{difference}$	0.16

Analysing the above table, it can be stated that a much shorter twisted tube heat exchanger is needed in order to generate the same heat duty than the plain tube heat exchanger. In this sense we can state that the TTSTHE is more economically efficient and practical for smaller space limitations.

5.6.2 Enthalpy, Heat Transfer, and Mass Flow Comparison

Comparing the generated Enthalpy (H) over the rate of heat transfer (Q), in comparison with the mass flow rate (m), using the same length of tubes for both the plain and the twisted tube design, the following formula was used (Mills, 1999).

$$Q = m \cdot \Delta H$$

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Where:

- Q = The rate of heat transfer (kW)
- m = The mass flow rate (kg/s)
- ΔH = The generated enthalpy (kJ/kg)

By means of using the above formula, the following conclusion can be made.

Table 45: H, Q & m comparison between twisted and plain tube designs

	Enthalpy (kJ/kg)	Heat Transfer (kW)	Mass Flow (kg/s)
Shell Side (Plain Tube)			
Inlet	516.60	24050	320.70
Outlet	441.60	24050	320.70
Shell Side (Twisted Tube)			
Inlet	516.60	24050	284.50
Outlet	432.10	24050	284.50

Analysing the above table, it can be stated that a much slower mass flow rate is needed for the twisted tube heat exchanger in order to generate the same heat duty than the plain tube heat exchanger. In this sense we can state that the TTSTHE is more economically efficient and practical.

5.6.3 Temperature and Mass Flow Comparison

In order to examine the effect of the tube side mass flow rate on the tube side outlet temperature, an additional calculation was done within EES. Within this analysis, a 6°C deviation was considered from the fixed design parameters for the produced outlet temperature of the tube side fluid. The output calculated can be seen in the figure below.

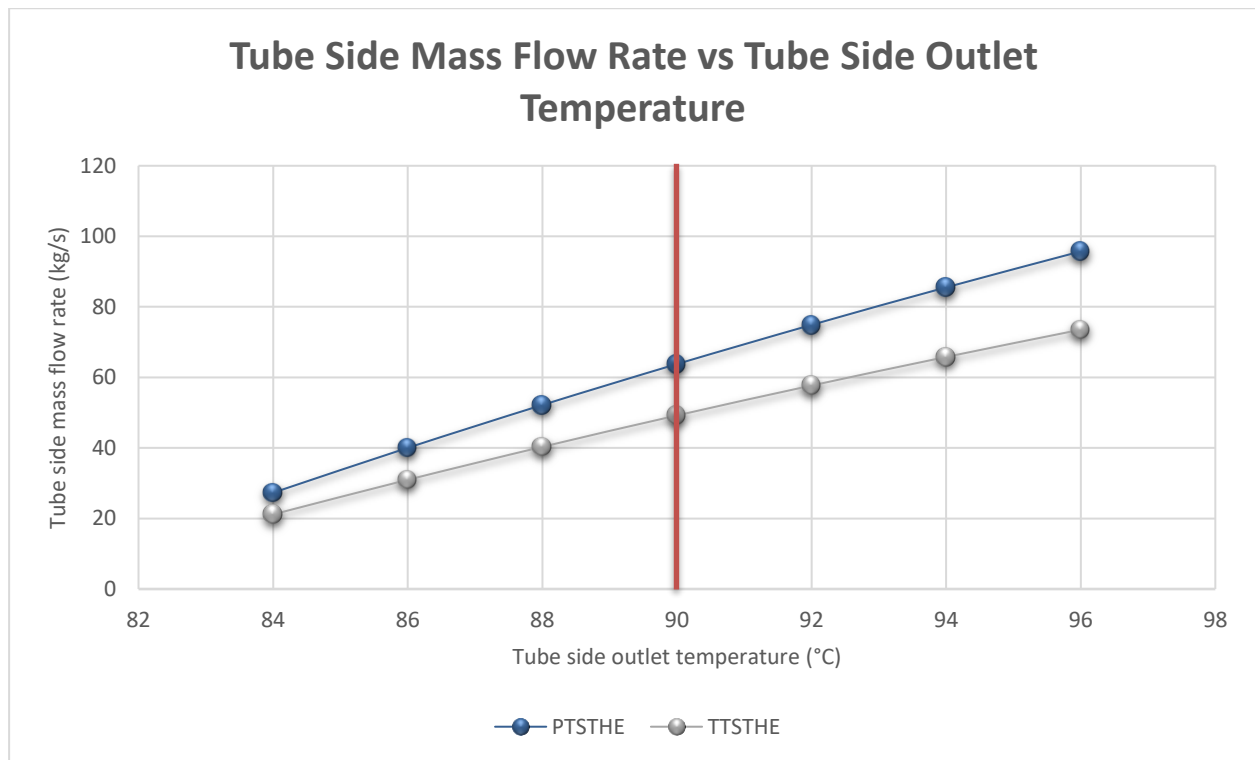


Figure 60: Tube side mass flow rate vs. tube side outlet temperature

Analysing the above figure, it is clear that both heat exchangers present an increase in tube side outlet temperature as the tube side mass flow rate increases. However, the twisted tube heat exchanger generates a higher tube side outlet temperature than the plain tube heat exchanger when having the same tube side mass flow rate.

This occurrence can be explained when considering the geometrical outline of the twisted tube heat exchanger. The twisted tube geometry causes the tube side fluid to flow in a swirl-like manner, which causes better mixing of the tube side fluid. The swirl-like flow within the tubes causes the outside fluid on the tube wall of the heat exchanger to flow faster than the inner fluid near the centre of the tube. This occurrence causes the second fluid boundary layer to break, leading to an increase in energy release, which causes a higher tube side outlet temperature.

5.6.4 Pressure Drop Comparison

In order to evaluate the effect of the pressure drop calculated for the TTSTHE and PTSTHE, the following table was generated to indicate the allowable pressure over the measured pressure drop for both heat exchanger designs considering the petrochemical operated heat exchanger design conditions.

Tube/Shell Side	PTSTHE/TTSTHE	Allowable Pressure Drop (kPa)	Calculated Pressure Drop (kPa)	% Pressure of Allowable Pressure (%)
Tube Side	PTSTHE	15	0.104	0.69
	TTSTHE		0.343	2.29
Shell Side	PTSTHE	15	0.673	4.49
	TTSTHE		1.356	9.04

Analysing the above table, the pressure drop experienced by the PTSTHE and TTSTHE based on the petrochemical operated heat exchanger's design conditions, are low. However, the pressure drop calculated for the TTSTHE is higher than the pressure drop calculated for the PTSTHE. The maximum pressure drop experienced is 1.356 kPa, which is measured and calculated on the shell side of the TTSTHE, which only conducts 9.04% of the allowable pressure for the designed heat exchanger.

Thus, it can be stated that the pressure drop experienced by both heat exchangers are very low and will have little to no effect on the operating conditions of the designed heat exchanger.

5.7 Financial Comparison

In order to fully understand whether the TTSTHE is viable for installation considering the financial implications, a considerable cost comparison was done between the plain and twisted tube heat exchanger designs using South African prices and quotations. However, since no information was obtainable for the manufacturing cost of the TTSTHE, the manufacturing cost was calculated based on the findings in the study "Improved heat recovery using twisted tube exchanger technology" (Barraza-Colon & Picon-Nunez, 2022). The study concluded that the manufacturing of twisted tube heat exchangers is roughly 40% more expensive than that of the plain tube heat exchanger. The manufacturing cost for the plain tube heat exchanger, specified below, was directly obtained from a requested quotation. The quotation was provided by one of the most constantly used manufactures for the petrochemical plant.

The costing information that was used for the analysis can be seen in the table below. The information shown in the table below was directly obtained from an operational petrochemical plant within Mpumalanga, South Africa. This costing information were manipulated and multiplied with the same specifically chosen factor as mentioned within the design conditions (section 5.1), in order to still be able to make used of accurate data for the calculation, but to protect the petrochemical company's right to privacy. The production costs provided by the petrochemical

plant as used within the tables below, were calculated by engineers of the petrochemical plant, with the values including the maintenance of the equipment.

Within the costing analysis of both heat exchangers, the outlet temperatures of the tube side and the shell side of the heat exchanger were considered constant, as the heat duty needed for the petrochemical plant remained constant. Thus, a lower mass flow rate was calculated for the shell side of the twisted tube heat exchanger due to its ability to generate more heat transfer than the plain tube heat exchanger. A retrofitting cost was also considered for the transformation from a PTSTHE to a TTSTHE, considering a retrofitting will be done within two weeks. The financial effects can be seen in the table below.

Table 46: Costing variables for viability determination

Variables	Unit	PTSTHE	TTSTHE
Manufacturing Cost	R	3 712 961.00	5 198 145.40
Retrofitting Cost for Two Weeks of Shutdown	R	129 949 779.50	129 949 779.50
Steam Production Cost	R/ton	303.90	303.90
Steam Mass Flow Rate	kg/s	320.70	284.50
	ton/h	1272.64	1272.64
Operating Time	h/year	8030.00	8030.00
Yearly Steam Usage	ton/year	10 219 299.20	9 065 709.40
Yearly Operational Cost	R/year	3 105 645 026.88	2 755 069 086.66
Total Cost	R/year	3 235 594 806.00	2 885 018 866.00

Although the calculated steam usage in the above table is high, it should be noted that the petrochemical plant operation is an enclosed process. Thus, the steam that is used by the specified heat exchanger is also used for other equipment and processes throughout the plant. It should thus not be seen as only the specified heat exchangers steam usage, but the steam usage for a whole process.

By means of the information specified above and considering that the heat exchangers operate 24 hours every day, with only a month of shutdown during a year, the following can be calculated.

- Steam production
- Operational cost
- Yearly steam usage value
- Return on investment time

In order to clearly evaluate the two heat exchangers considering the above factors, the below table was used to specify the comparison.

Table 47: Costing variables based on ROI.

Variables	Unit	PTSTHE & TTSTHE
Water Production Cost	R/ton	376.77
Water Mass Flow Rate	kg/s	572.70
	ton/h	2272.66
Operating Time	h/year	8030.00
Yearly Water Production	ton/year	18 249 459.80
Yearly Income Value	R/year	6 875 848 969.00

By means of the information analysed within the above two tables, the following can be calculated and illustrated.

Table 48: ROI of twisted tube heat exchanger.

Variables	Unit	PTSTHE	TTSTHE
Yearly Benefit Value	R/year	3 640 254 163.00	3 990 830 103.00
Yearly Benefit Difference	R/year	0	350 575 939.00
ROI	hours	8.9	11.4

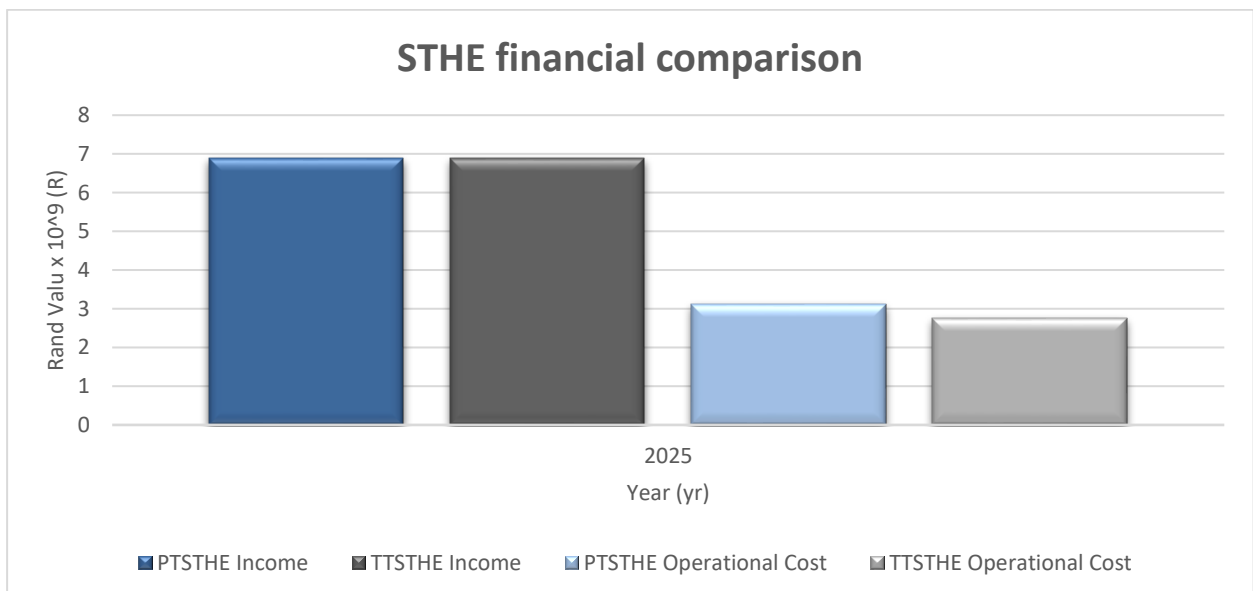


Figure 61: STHE financial comparison.

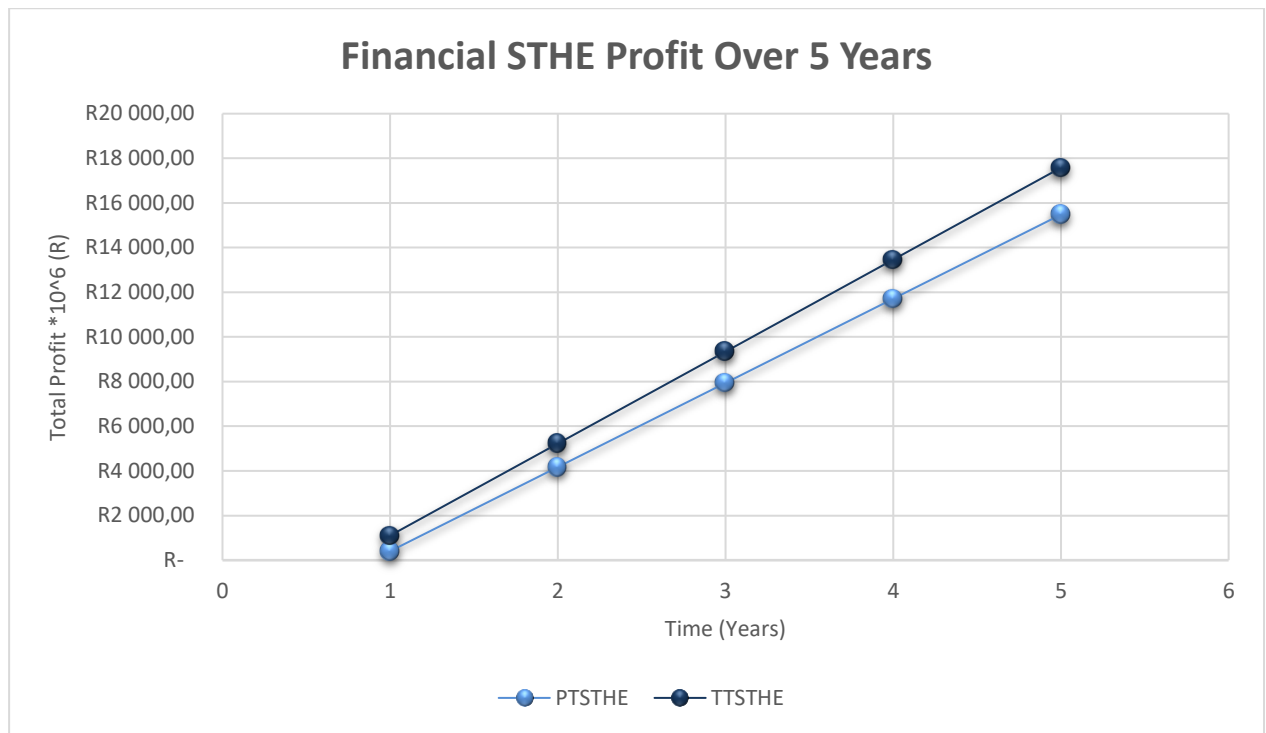


Figure 62: Financial STHE profit results over 5 years.

As seen and specified within the above table and figure, it is thus a valued decision to rather install the TTSTHE. The twisted tube heat exchanger provides a considerable higher amount of benefit in comparison to the plain tube heat exchanger over an annual period.

CHAPTER 6 CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

During the completion of this study, it was insightful to learn all the unseen positives and negatives from both heat exchanger designs. More so the depth in which one can analyse a STHE. Since the petrochemical industry is a constantly operating environment, it was absorbingly interesting to evolve my understanding of heat transfer and the operating performances and capabilities regarding STHE. Completing this study ensured a better understanding of STHE's, and generated a wider knowledge of consideration regarding the STHE design.

The inputs and geometrical constraints used within the heat exchanger designs were obtained from a petrochemical STHE, operating within Mpumalanga, South Africa. The designs for both heat exchangers remained constant during the study. Only the tube designs between the heat exchanger designs changed, by making use of plain tubes and twisted tubes during the analysis. Analysing the study and the obtained outputs, the following conclusions can be made.

When considering the tube side heat transfer coefficient, and how an increase in the tube side fluid Reynolds number affects the calculated heat transfer coefficient, it can be seen that as the tube side fluid Reynolds number increases, the TTSTHE presents to be more beneficial than the PTSTHE for the Reynolds number range of 4000 – 11000. The TTSTHE presented an average of 58% increase in the produced heat transfer coefficient when comparing to the output reached for the PTSTHE. It can thus be concluded for the selected heat exchanger designs, that the plain tubes provided less heat transfer to occur than the twisted tubes within the STHE's, as the tube side fluid Reynolds number increases.

Analysing the obtained output reached within EES, considering the shell side pressure drop and the effect that the shell side fluid Reynolds number has thereon, it can be seen that the plain tube shell and tube heat exchanger expresses much lower pressure drop to occur during the change in shell side fluid Reynolds number than the TTSTHE. The PTSTHE presented an average pressure drop of 45% lower when comparing to the pressure drop determined for the TTSTHE. It can thus be concluded that for the specified heat exchanger designs, the plain smooth tubes provided less shell side pressure drop than the twisted tubes as the shell side fluid Reynolds number increases from 4000 – 14000.

Comparing the obtained output reached within EES for the tube side pressure drop and the effect that the tube side fluid Reynolds number has thereon, it can be seen that the TTSTHE

experienced diminutive more pressure drop to occur than the PTSTHE, as the tube side fluid Reynolds number increases from 4000 – 15000. However, it is also clear that as the Reynolds number increases from 4000 – 15000, the pressure drop difference between the TTSTHE and PTSTHE decreases. It can thus be concluded that the plain tubes prove to be a better selection within the specified heat exchanger design for a Reynolds number range of 4000 – 15000, when considering the pressure drop calculated on the tube side of the heat exchanger as the tube side fluid Reynolds number increases.

When analysing the effectiveness of the two tube type STHE's, it is important to remember that the calculated heat exchanger efficiency is a percentage relation of the heat exchanger's performance for the selected design. Considering the simulated comparison output for both heat exchanger designs, it can be seen that both heat exchangers experienced an exponential decrease regarding their effectiveness as the shell side mass flow rate increases. Analysing the output reached, it can be concluded that on average the TTSTHE presented to be 10% more effective than the PTSTHE. Taking into account this study's obtained output, it can also be concluded that, for the specified heat exchanger geometrical design and constraints, both heat exchangers presented to be at their best efficiency with the mass flow rate of 10 kg/s. However, both heat exchangers presented a stagnating result towards their produced efficiency as the shell side mass flow rate increases. The effectiveness difference between the PTSTHE and the TTSTHE can be explained when considering the nature of the fluid flow occurring within the STHE. As the mass flow rate increases, the mixing and turbulence of the shell side fluid increase. Simultaneously, increasing the tube side flow causes the centrifugal forces within the tubes to increase, causing better mixing between the hotter fluid on the tube wall and the colder fluid in the centre of the tube. The increased flow rate causes the thermal boundary layer along the tube wall to break more effectively, generating a better release of energy and absorption, overall leading to a more effective STHE. Thus, it can be concluded that as the shell side mass flow rate increases, the twisted tube heat exchanger design presents the best efficiency when comparing to the output calculated for the PTSTHE.

Analysing the cost of both heat exchanger designs, it can be seen that the TTSTHE is roughly R1.48 million more expensive than the PTSTHE. This results in a 40% increase in the heat exchanger purchase when comparing to the PTSTHE's manufacturing cost. This occurrence can be explained when considering the specialities needed to manufacture both tube bundle designs. However, it can also be seen that the twisted tube design is roughly R350 million more profitable when comparing to the PTSTHE over a yearly operating period. This results in a 10% increase in the provided profit when comparing to the profit provided by the PTSTHE. This change in value is due to the increased heat transfer generated and the reduced maintenance on the TTSTHE

due to its self-reinforcing design. However, a higher pressure drop on the tube and shell side of the TTSTHE can require an upgrade on other equipment involved, such as pumps etc., within the process when considering a retrofitting of an installed PTSTHE. These costs should also be taken into account. Since this is completely dependent on the operating process and equipment of the plant, this calculation will differ for all plants. This variation has not been taken into account for the calculation presented within this thesis, but should be measured when considering the change from PTSTHE to TTSTHE.

It can overall be concluded that both heat exchanger designs provide calculated positives and negatives. The twisted tube design generated the best effectiveness for the specified petrochemical conditions and design. The plain tube design generated the lowest tube side and shell side pressure drop due to the change in fluid turbulence. The twisted tube design provided the best heat transfer coefficient as the tube side fluid's Reynolds number increases. The twisted tube bundle design provided a significant financial benefit over the plain tube bundle design for a yearly period. Overall, due to the efficiencies calculated, which is an accurate measure of each heat exchanger's performance for the selected design, the TTSTHE deemed to outperform the PTSTHE.

6.2 Recommendations

This thesis has been investigating and comparing twisted tubes and plain tubes within STHE's. This study focused on the following few important aspects.

- Comparing the output calculated for a plain tube and a TTSTHE, using operating and design data from a petrochemical plant within Mpumalanga, South Africa.
- Using one specific selected operating STHE design from a petrochemical plant within Mpumalanga, South Africa, for the analysis.
- Presenting reliable information calculated from the output reached by analysing the two different STHE designs.
- Comparing the two different heat exchanger efficiencies with the specified operating conditions and geometry retrieved from a petrochemical plant within Mpumalanga, South Africa.

Considering the aspects used within this thesis, other possible areas of development and improvement were identified as the thesis was analysed and compiled. In this case it would have been interesting to analyse the effect that the tube twist pitch length will have on the heat transfer coefficient of the twisted tubes, when it is altered. Ultimately, a change in twist pitch length could cause higher turbulence and an increase in second boundary flow velocity, which could force the

fluid to break the second boundary layer within itself, relieving more energy and causing an impact on the heat transfer coefficient generated.

Another factor that would have been interesting to analyse in depth would have been to investigate the effect that different fluid types would have on the efficiency and heat transfer coefficient of the twisted tubes. Due to the difference in viscosity and density of the different fluids, it is recommended that one analyses how higher viscose fluids would impact the twisted tube's performance within STHE's.

It would also have been interesting to analyse the twisted tube length and the effect thereof on the cost of the heat exchanger. By means of the twisted tube configuration generating a higher heat transfer coefficient, a smaller heat exchanger could be used for the same heat duty to be produced, thus reducing the size and cost of the heat exchanger.

Another considering factor would be to investigate the effect that different materials for the twisted tube design would have on the heat transfer performance of the TTSTHE. Different materials present different material properties, which will either benefit or handicap the heat transfer output measured in terms of the generated heat transfer coefficient.

This thesis is only focused on the petrochemical industry in Mpumalanga, South Africa. It would have also been desirable to investigate the performance that this technology would have on other industries over the world. For example, analysing the performance of the twisted tube heat exchanger within the automotive industry, could also provide significant benefits.

BIBLIOGRAPHY

Abdul Razzaq, A. K. & Mushatet, K. S., 2021. A numerical study for a double twisted tube heat exchanger. *International journal of heat and technology*, p. 1583.

Abdullah, Z., Othman, M. H. & Taip, F. S., 2019. *ResearchGate*. [Online]
Available at: https://www.researchgate.net/figure/Schematic-diagram-of-shell-and-tube-heat-exchanger-1_fig1_337503247
[Accessed 14 08 2024].

Ahmed, F., 2024. *Heat Exchangers in Oil and Gas: A Critical Component for Energy Efficiency*. [Online]
Available at: <https://egyptoil-gas.com/features/heat-exchangers-in-oil-and-gas-a-critical-component-for-energy-efficiency/>
[Accessed 29 01 2025].

Alfa Laval, 2002. *African chemical giant Sasol*. [Online]
Available at: <https://www.alfalaval.com/media/stories/industries/african-chemical-giant-sasol/>
[Accessed 2024/02/04].

Alsulaiei, Z. M., Hasan, H. M. & Fagr, M. H., 2021. Flow and heat transfer in obstructed twisted tubes. *Case Studies in Thermal Engineering*, pp. 1-5.

Amit, K. & Ajay, T., 2019. Experimental analysis and contrast of LMTD and effectiveness for corrugated tube (60 degree pitch) in parallel flow arrangement with shell and pipe exchanger. *Think India*, p. 1536.

Arveng Training, 2025. *Arveng Training & Engineering*. [Online]
Available at: <https://arvengtraining.com/en/basics-of-shell-tube-heat-exchangers/>
[Accessed 19 August 2025].

Barraza-Colon, J. & Picon-Nunez, M., 2022. Improved heat recovery using twisted tube exchanger technology. *Cleaner Engineering and Technology*, pp. 2666-7908.

Coetzee, N., 2015. *Heat transfer coefficients of smooth tubes in the turbulent flow regime*. Pretoria: University of Pretoria.

Danielsen, S. O., 2009. Investigation of a twisted-tube type shell-and-tube heat exchanger. *Norwegian University of Science and Technology*, p. 3.

Dong, X. et al., 2020. Experimental research on heat transfer and flow resistance properties in spiral twisted tube heat exchanger. *Applied Thermal Engineering*, pp. 1-15.

Doran, P. M., 2013. Fluid Flow. *Bioprocess Engineering Principles*, p. 202.

Eiamsa-ard, S., 2023. Heat transfer mechanism in ribbed twisted oval tubes.. *International Journal of Thermal Science*, p. Volume 193.

Fakheri, A., 2014. Efficiency analysis of heat exchangers and heat exchanger networks. *International journal of heat and mass transfer*, pp. 100-101.

Geckointech, 2024. *Geckointech*. [Online]

Available at: <https://geckointech.com/product/twisted-tube/#:~:text=Applicable%20materials%20for%20fabrication%20of,copper%2C%20Monel%20alloy%2C%20etc.>

[Accessed 26 09 2024].

Gnielinski, V., 1993. *Heat transfer in cross-flow around individual tubes, wires and profiled cylinders*. Dusseldorf: VDI-heat atlas.

Horizon, 2025. *South Africa Shell And Tube Heat Exchanger Market Size & Outlook*. [Online]

Available at: <https://www.grandviewresearch.com/horizon/outlook/shell-and-tube-heat-exchanger-market/south-africa>

Husseini, R. A., Alwan, A. A. & Abed, I. M., 2021. Design a prototype module for Al-Mussaib thermal power plant. *Advances in Mechanics, Volume 9, Issue 3*, pp. 408-424.

HVAC, Chemicals and Petrochemicals, Power Generation, 2021. *Heat Exchangers Market*, Canada: Fact.Mr.

Idiamart, 2014. *Shell and tube heat exchanger*. [Online]

Available at: <https://www.indiamart.com/proddetail/rlht-109-shell-and-tube-heat-exchanger-2852577207473.html?srsId=AfmBOoqCubL43ayMUqE3bV3-pL57zGtMGLaZjzuWGatuA0E5selwqw2o>

[Accessed 12 July 2024].

Ipieca, 2022. *Heat Exchangers (2022)*. [Online]

Available at: <https://www.ipieca.org/resources/energy-efficiency-database/heat-exchangers-2022#:~:text=Heat%20exchangers%20are%20used%20to,may%20be%20in%20direct%20contact.>

[Accessed 25 09 2024].

Karno, A. & Ajib, S., 2006. Effect of tube pitch on heat transfer in shell-and-tube heat exchanger - new simulation software. *Heat Mass Transfer*, pp. 263-270.

Khaled, R. A. & Mushatet, K. S., 2023. CFD analysis for a twisted elliptical double tube heat exchangers Integrated with a twisted tape.. *International journal of heat and technology*, p. 1305.

Liu, S., Tu, A., Li, Y. & Zhu, D., 2023. Optimization Design and Comparative Analysis of Enhanced Heat Transfer and Anti-Vibration Performance of Twisted-Elliptic-Tube Heat Exchanger: A Case Study. *Energies*, pp. 5-6.

Li, X. et al., 2023. Convective thermal-hydraulic performance in twisted tubes: From the perspective of symmetrical cross-sectional shapes. *Heat and mass transfer*, pp. 1-13.

Ljubicic, B., 2004. *Twisted tube heat exchangers: technology and application*, Houston: Technical briefings and case studies on: Heat exchangers.

Lokesh, K. et al., 2017. Heat transfer enhancement of double pipe heat exchanger using twisted tape inserts. *International journal of mechanical engineering and technology*, p. 424.

Market, Heat Exchanger, 2024. *Heat Exchanger Market*. [Online]
Available at: https://www.marketsandmarkets.com/Market-Reports/heat-exchanger-market-750.html?gad_source=1&qclid=CjwKCAjw9cCyBhBzEiwAJTUWNeu9dD4Lqi5701LwP-eRPOTvzsnOHVdGx8ijltbvEH7CDLHVkD25nRoCZzkQAvD_BwE

Mills, A., 1999. *Heat Transfer, Second Edition*. Clifornia at Los Angeles: Prentice Hall.

Mishra, B., Srivastava, A. & Yadav, L., 2020. Performance Analysis of Cooling Tower Using Desiccant. *Heat and Mass Transfer*, p. 56.

Morgan, R. D., 1980. Twisted Tube Heat Exchanger Technology. *Convventional Shell And Tube Design*, p. 79.

Munoz-Carpena, R., Parsons, J. & Gilliam, J., 1993. Numerical Approach to the Overland Flow Process in Vegetative Filter Strips. *Biological and Agricultural Engineering*, p. 763.

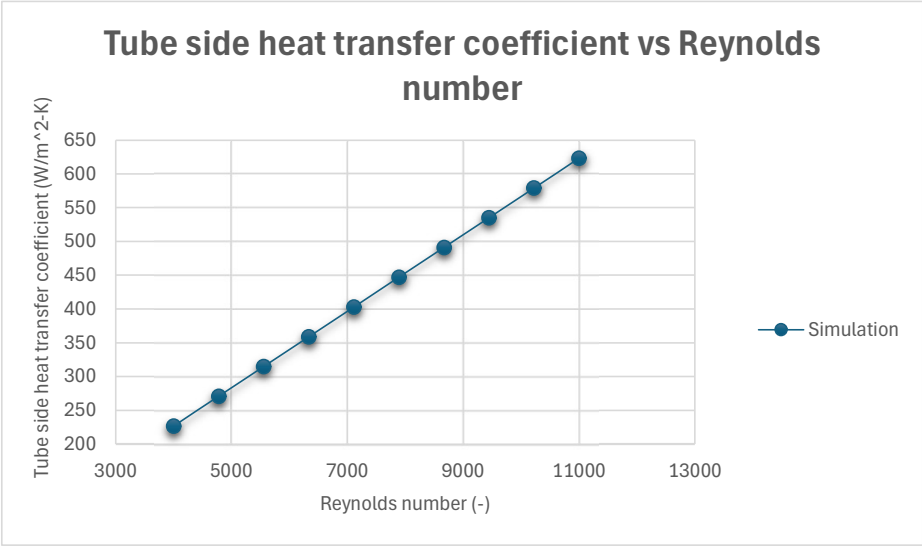
Razzaq, A. A. k. & Mushatet, K. S., 2023. A review for a twisted tube heat exchanger. *Journal of Nanofluids*. Vol 12., pp. 299 - 317.

- Ruengpayungsak, K., Saysroy, A., Wongcharee, K. & Eiamsa-ard, S., 2019. Thermohydraulic performance evaluation of heat exchangers equipped with centrally perforated twisted tape: Laminar and turbulent flows. *Thermal science and technology*, pp. 1-18.
- Saffarian, M. R., Fazelpour, F. & Sham, M., 2019. Numerical study of shell and tube heat exchanger with different cross-section tubes and combined tubes. *Energy and Environmental Engineering*, pp. 39-44.
- Samah, Z. N., Mohammed, Q. K. & Ammar, A. A., 2018. Performance analysis of shell and tube heat exchanger: Parametric study. *Case studies in thermal engineering*, pp. 563-568.
- Samruaisin, P. et al., 2019. Intensification of convective heat transfer and heat exchanger performance by the combined influence of a twisted tube and twisted tape. *Case studies in thermal engineering*, p. 55.
- Serth, R., 2007. *Process Heat Transfer*. Kingville, Texas: Elsevier Ltd..
- Services, Fossil Consulting, 2024. *Fossil Consulting Services*. [Online]
Available at: <https://www.fossilconsulting.com/blog/power-plant-fundamentals/laminar-flow/>
- Sinnot, K. R., 1999. *Heat transfer design guidelines*, Butterworth-Heinemann: Elsevier, Chemical Engineering Design (4th Edition).
- Subramanian, R. S., 2024. *Department of Chemical and Biomolecular Engineering*. [Online]
Available at:
<https://web2.clarkson.edu/projects/subramanian/ch330/notes/Shell%20Side%20Pressure%20Drop%20in%20a%20Shell%20and%20Tube%20Heat%20Exchanger.pdf>
- Tan, X.-h., Zhu, D.-s., Zhou, G.-y. & Zeng, L.-d., 2012. Heat transfer and pressure drop performance of twisted oval tube. *Applied Thermal Energy*, pp. 374-383.
- Transfer, Koch Heat, 2024. *www.kochheattransfer.com*. [Online]
Available at: <https://www.kochheattransfer.com/technical-resources>
- Wanbo, Y. et al., 2017. Heat transfer performance of epoxy resin flows in horizontal twisted tube. *Applied thermal engineering*, pp. 28-34.
- Yang, S., Zhang, L. & Hong, X., 2011. Experimental study on convective heat transfer and flow resistance characteristics of water flow in twisted elliptical tubes. *Applied Thermal Engineering*, pp. 2981-2991.

**APPENDIX A: PLAIN TUBE SHELL-AND-TUBE HEAT EXCHANGER
PETROCHEMICAL OUTPUT ANALYSIS**

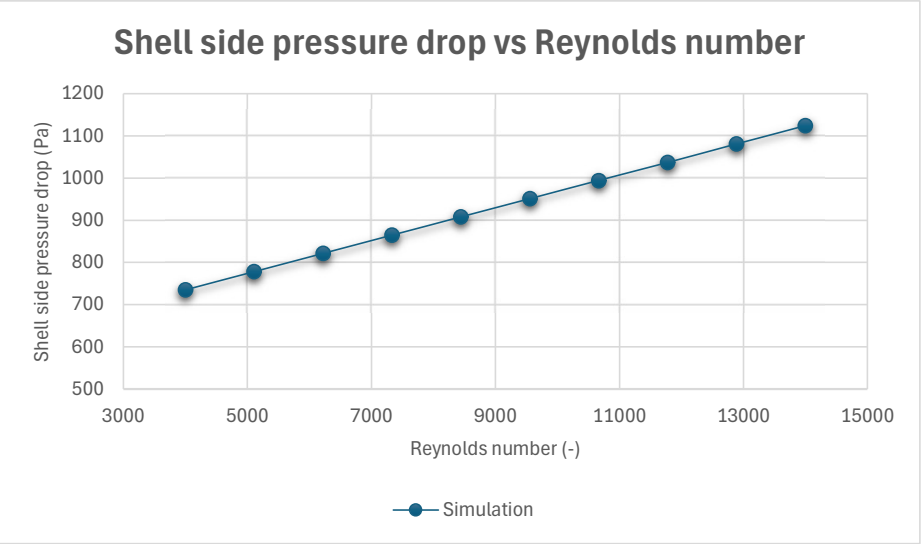
Tube side heat transfer coefficient vs Reynolds number

x	y
-	W/m^2-K
4000	226,6
4778	270,7
5556	314,8
6333	358,9
7111	402,9
7889	447
8667	491,1
9444	535,1
10222	579,2
11000	623,3



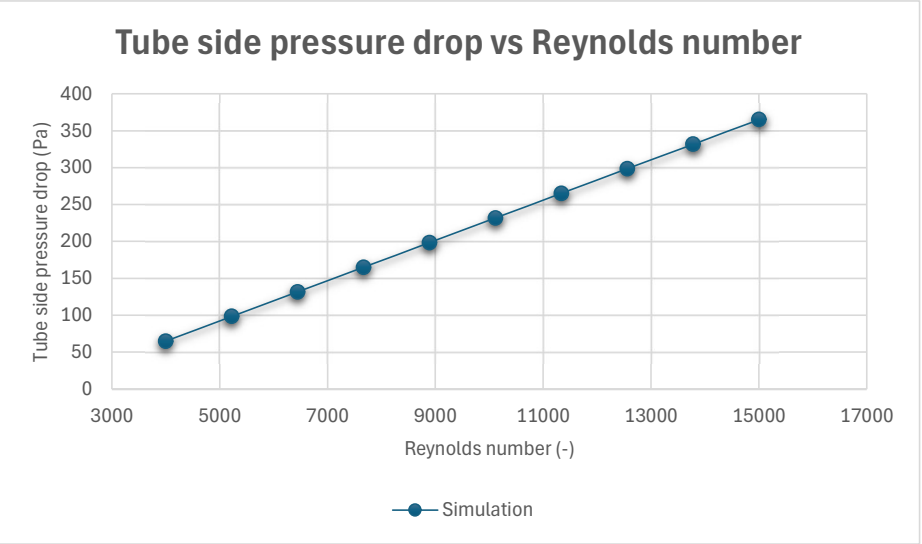
Shell side pressure drop vs Reynolds number

x	y
-	Pa
4000	734,8
5111	778
6222	821,3
7333	864,5
8444	907,7
9556	950,9
10667	994,2
11778	1037
12889	1081
14000	1124



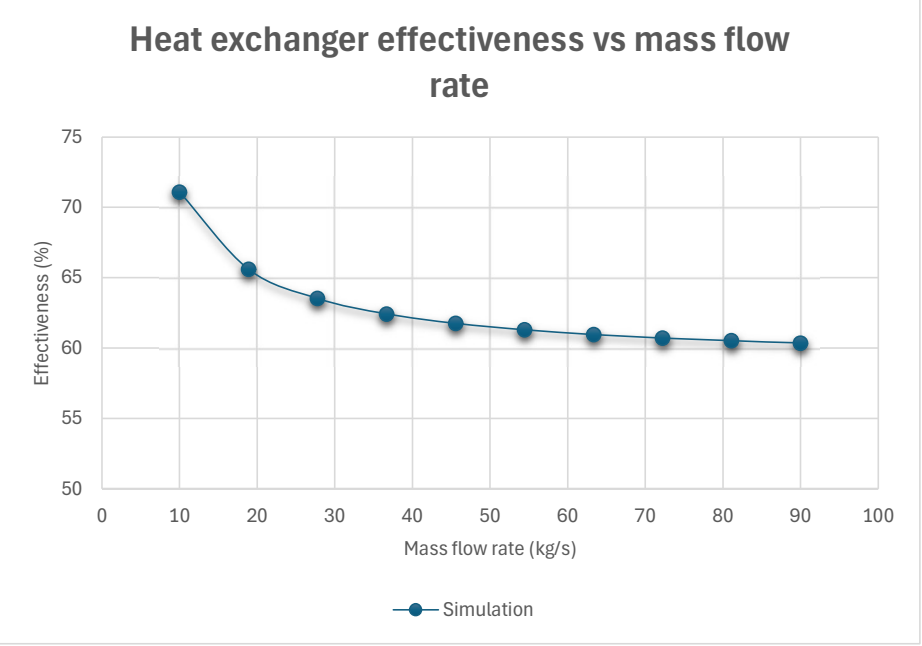
Tube side pressure drop vs Reynolds number

x	y
-	Pa
4000	65,12
5222	98,45
6444	131,8
7667	165,1
8889	198,5
10111	231,8
11333	265,1
12556	298,5
13778	331,8
15000	365,1



Heat exchanger eff vs shell side mass flow rate

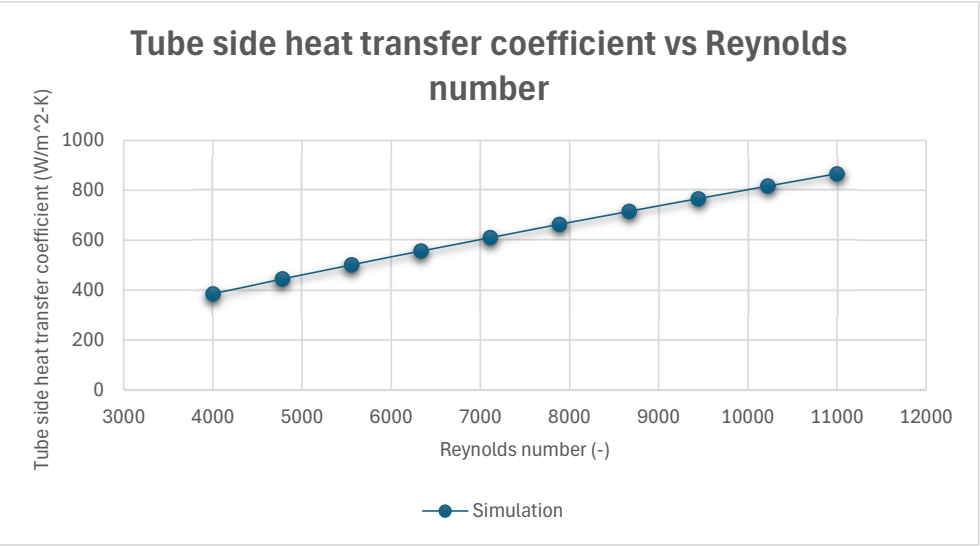
x	y
kg/s	%
10	71,09
18,89	65,61
27,78	63,52
36,67	62,43
45,56	61,76
54,44	61,3
63,33	60,97
72,22	60,72
81,11	60,53
90	60,37



APPENDIX B: TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER PETROCHEMICAL OUTPUT ANALYSIS

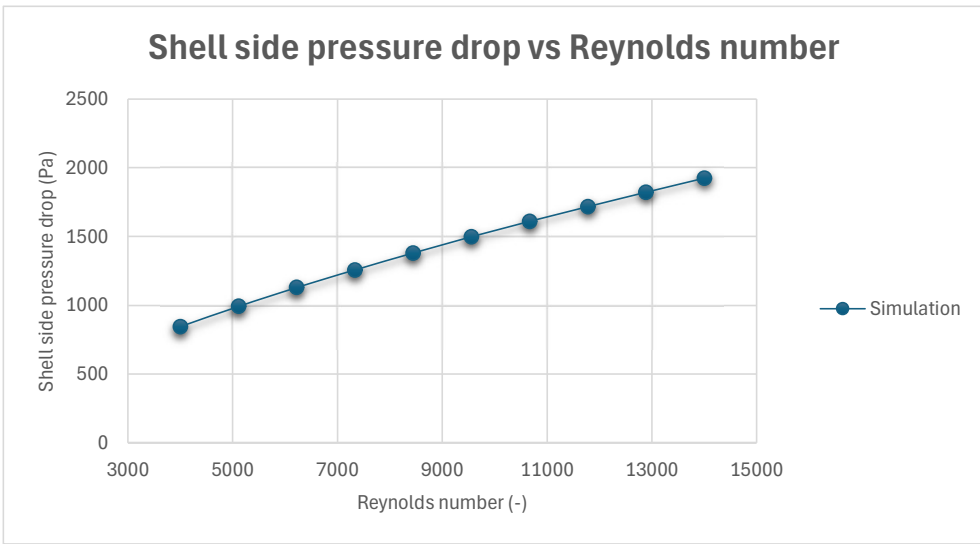
Tube side heat transfer coefficient vs Reynolds number

x	y
-	W/m^2-K
4000	385,4
4778	444,3
5556	501,3
6333	556,7
7111	610,7
7889	663,6
8667	715,5
9444	766,4
10222	816,5
11000	865,8



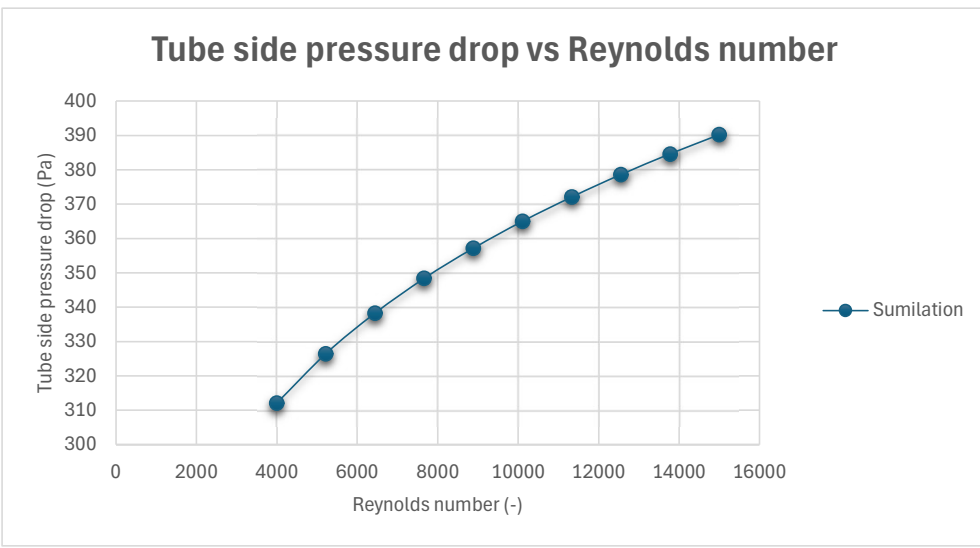
Shell side pressure drop vs Reynolds number

x	y
-	Pa
4000	844,7
5111	992,5
6222	1130
7333	1258
8444	1381
9556	1498
10667	1610
11778	1718
12889	1823
14000	1925



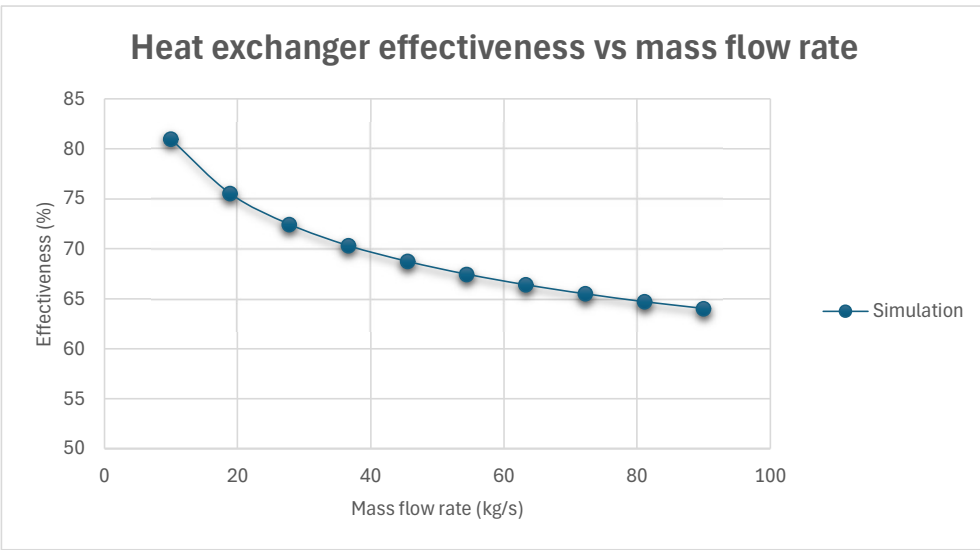
Tube side pressure vs Reynolds number

x	y
-	Pa
4000	312,1
5222	326,5
6444	338,3
7667	348,4
8889	357,2
10111	365
11333	372,1
12556	378,6
13778	384,6
15000	390,2



Heat exchanger effectiveness vs mass flow rate

x	y
kg/s	%
10	80,96
18,89	75,52
27,78	72,44
36,67	70,32
45,56	68,73
54,44	67,45
63,33	66,4
72,22	65,49
81,11	64,71
90	64,02



APPENDIX C: PLAIN TUBE SHELL-AND-TUBE HEAT EXCHANGER VALIDATION

EES Validation Output

Re_sheit (-)	P_sheit (Pa)	R*2
4006,5	20,21	408,444
5008	29,87	1688,39
6009	41,09	6975,59
7010	53,81	25249,2
8011	67,97	34321,6
9012	83,52	20,21
10014	100,4	41,09
11015	118,7	83,52
12016	138,2	158,9
13017	158,9	

LPH-kg/s = 1LPH = 0,00028 kg/s				Average Percentage Error	Average percentage error	Average Percentage Error
m (kg)	eff (%)	eff (%)	R ²			
15	70,51	0,7051	4971,66	0,590322581	2,904290429	5,129473858
23,33	53,59	0,5359	2871,89	1,408554185	0,658923961	3,401609055
30	43,81	0,4381	1913,32	0,246412442	1,057497467	3,455179283
40	33,84	0,3384	1145,15	0,268664564	5,481762598	7,218376337
50	23,87	0,2387	597,484	0,461027714		
56,67	20,72	0,2427	589,033	0,589457148		
60	22,96	0,2296	12414,5	0,590377405	3,002518614	4,852474633
75	18,46	0,1846	70,51	1,408554185	5,481762598	7,218376337
81,67	16,98	0,1698	53,59	Standard Deviation	Standard Deviation	Standard Deviation
90	15,43	0,1543	43,81	4,393961358	#DIV/0!	3,289653134

3	0,426410027	#DIV/0!	1,747406569
5	R^2 Value	R^2 Value	R^2 Value
2	0,999891156	0,932463328	0,96957219

Average Percentage Error	Average Percentage Error
7,191923397	0,333286059
6,129086525	1,983565396
5,607054036	5,384160757
2,626880898	3,452939772
	4,100304981

7,191923397	5,384160757
Standard Deviation	Standard Deviation

2,931664715	0,166170094
26,7781338	0,614475793

41,21018321	1,221880518
7.283199846	1.540078569

2,070408655
1.609375034

1 955770101	1 951385484
-------------	-------------

R^2 Value	R^2 Value
0.0048732025	0.078134024

0,994873935 0,976134224
efficiency:

Experimental Results		CFD Results		R ²	Re, h _o (W/m ² *K)	
m (kg/s)	P _{shell} (Pa)	m (kg/s)	P _{shell} (Pa)		Re _{tube} (-)	h _o tube (W/m ² *K)
				14.627565	4014.2	8.2846
0.00392	15.528	0.00392	15.528	35.196929	5997.2	5.9327
0.00599	22.981	0.00599	22.981	63.67242	8002.8	8.7985
0.00794	43.478	0.00794	43.478	101.12604	10008	10.006
0.01248	91.925	0.01248	103.11	213.61895	Re _{tube} (-)	h _o tube (W/m ² *K)
0.01773	185.09	0.01773	200		4002.8	5.0731
0.02392	330.43	0.02386	354.66		5985.8	6.7725
					8014.2	10.088
0.03077	539.13	0.03071	572.67		9845.8	12.175
R²					Re _{tube} (-)	h _o tube (W/m ² *K)
	241.11874				4014.2	5.2573
	528.126361				6019.8	7.8772
	1890.33668				8002.8	10.374
	8450.20543				9997.2	12.605
	34258.3081					
	109183.985					
	290661.157					
Sum	445213.237	493263			4002.8	5.6257
					6008.5	8.1637

Re_shell (-)	P_shell (Pa)
3971,7	6,6337
7021,8	22,079
10020	46,733
13017	82,673

Re_shell (-)	P_shell (Pa)
4006,5	8,4158
6986,9	28,317
10002	60,693
13000	106,73

Re_shell (-)	P_shell (Pa)
3989,1	9,3069
7004,4	30,99
10020	64,851
13017	117,43

Re_shell (-)	P_shell (Pa)	R^2
4006,5	15,545	241,647
6989,9	46,406	2153,52
9984,7	96,931	9395,62
13017	147,43	21735,6

33526,4

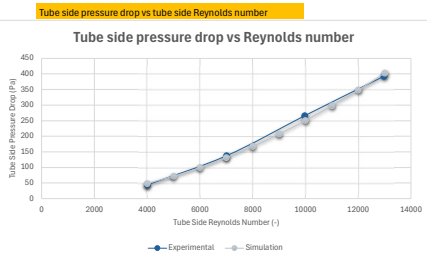
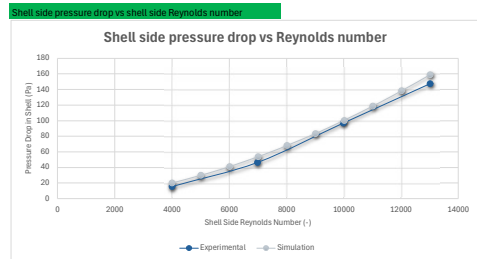
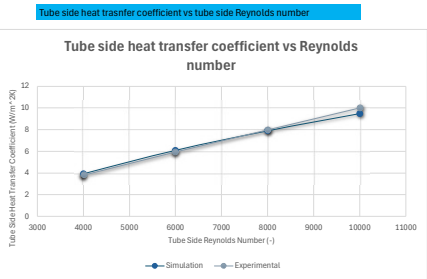
Re_tube (-)	P_tube (Pa)
3995,6	25,161
6991,2	82,461
10040	167,31
12965	287,42

Re_tube (-)	P_tube (Pa)
4013,2	31,773
6973,6	95,685
9986,8	197,06
12965	332,6

Re_tube (-)	P_tube (Pa)
4013,2	43,894
7008,8	137,56
9969,2	265,38
12968,2	392,1

Re_tube (-)	P_tube (Pa)	245018
3995,6	58,219	
6973,6	178,33	
9986,8	338,11	
12982	567,31	

11/11/2019



APPENDIX D: TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER VALIDATION

Tube side heat transfer coefficient vs Reynolds number

Validation			EES		
x	y		x	y	
Water	1981.1	1113.5	2000	1063	
	2969.5	1291.9		3000	1250
	3965.1	1437.8		4000	1403
	4931.7	1543.2		5000	1534
	5920.1	1624.3		6000	1650
Oil	651.7	223.11			
	801.36	252.83			
	949.66	272.64			
	1099.3	295.75			
	1249	315.57			
	1398.6	332.08			
	1548.3	348.58			

Shell Side Pressure Drop vs the Shell Side Reynolds Number

Validation			EES		
x	y		x	y	
540	1.3543		540	1.269	
	680	1.6063		680	1.667
	796.67	1.8268		796.7	2.01
	910	2.1102		910	2.352
	1016.7	2.4252		1017	2.682
	1126.7	2.7402		1127	3.029
	1216.7	3.0881		1217	3.317
	1313.3	3.4646		1313	3.631
	1406.7	3.9055		1407	3.938
	1500	4.2835		1500	4.249

Tube Side Heat Transfer Coefficient vs Tube Pitch

Validation			EES		
x	y		x	y	
99.304	10051		99.304	10710	
	149.3	5493.7		149.3	5845
	197.13	3569.6		197.13	3868
	247.13	2582.3		247.13	2765
	296.26	1949.4		296.26	2113
395.39	1240.5		395.39	1376	

Tube Side Pressure Drop vs the Tube Side Reynolds Number

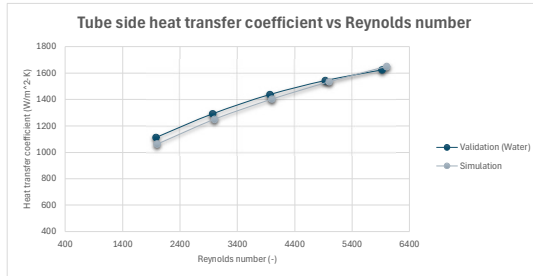
Validation			EES		
x	y		x	y	
14525	4.5208		14525	4.148	
	17494	6.5974		17494	6.049
	20226	8.8339		20226	8.119
	23373	11.949		23373	10.89
	26045	14.585		26045	13.56
	28777	18.019		28777	16.6
	31508	21.454		31508	19.95
	34299	25.447		34299	23.69
	37150	29.521		37150	27.86
	40178	34.073		40178	32.66
	41781	36.709		41781	35.35

Tube Side Mass Flow Rate vs the Heat Exchanger Effectiveness

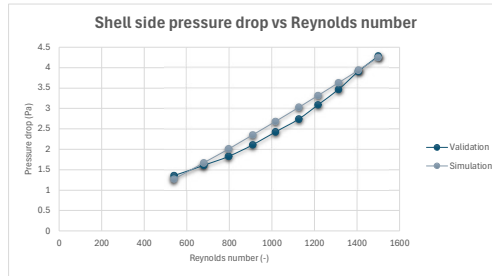
Validation			EES		
x	y		x	y	
0.000204	76.771		0.000204	75.43	
	0.000402	73.363		0.000402	68.13
	0.000602	68.953		0.000602	64.13
	0.000804	64.543		0.000804	61.4
	0.001	60.535		0.001	59.43
0.000202	69.354		0.000202	62.739	
	0.000402	62.739		0.000402	59.131
	0.000602	59.131		0.000602	55.122
	0.000804	55.122		0.000804	50.111
	0.001	50.111			

	6.721828	4.123164			
	3.641272	4.156183			
4.535249	4.985075	4.046344	1.746753	6.153128	
3.243285	4.449804	4.431333	7.133023	6.010265	
2.420364	2.170022	3.513884	6.99462	7.714581	
0.596164	1.951139	3.93751	4.869622	6.607595	
1.58222	0.811956	3.505174	1.82539	7.742546	
2.475457	3.986523	3.959085	4.513882	6.845623	
Max % err	4.535249	6.721828	4.431333	7.133023	7.742546
STD dev	1.513684	1.803069	0.341835	2.646956	0.835702
R^2 value	0.996283	0.982585	0.992477	0.914776	0.999956

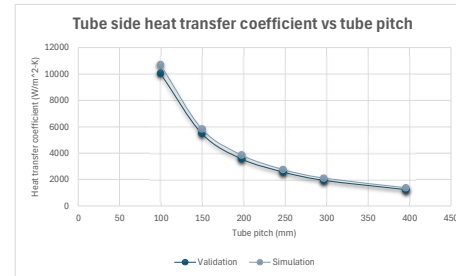
Tube side heat transfer coefficient vs Reynolds number



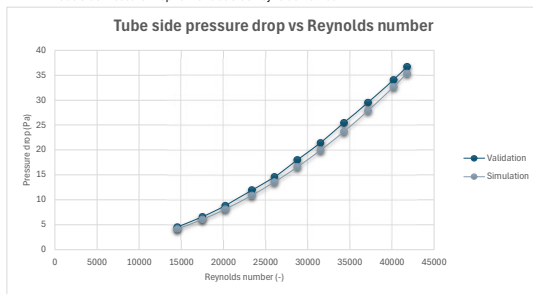
Shell Side Pressure Drop vs the Shell Side Reynolds Number



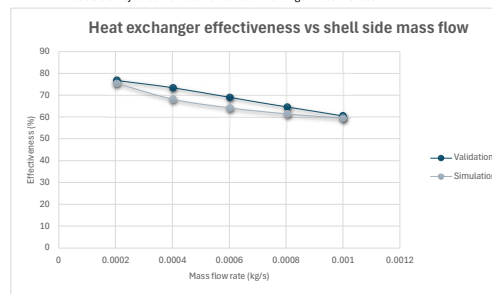
Tube Side Heat Transfer Coefficient vs Tube Pitch



Tube Side Pressure Drop vs the Tube Side Reynolds Number

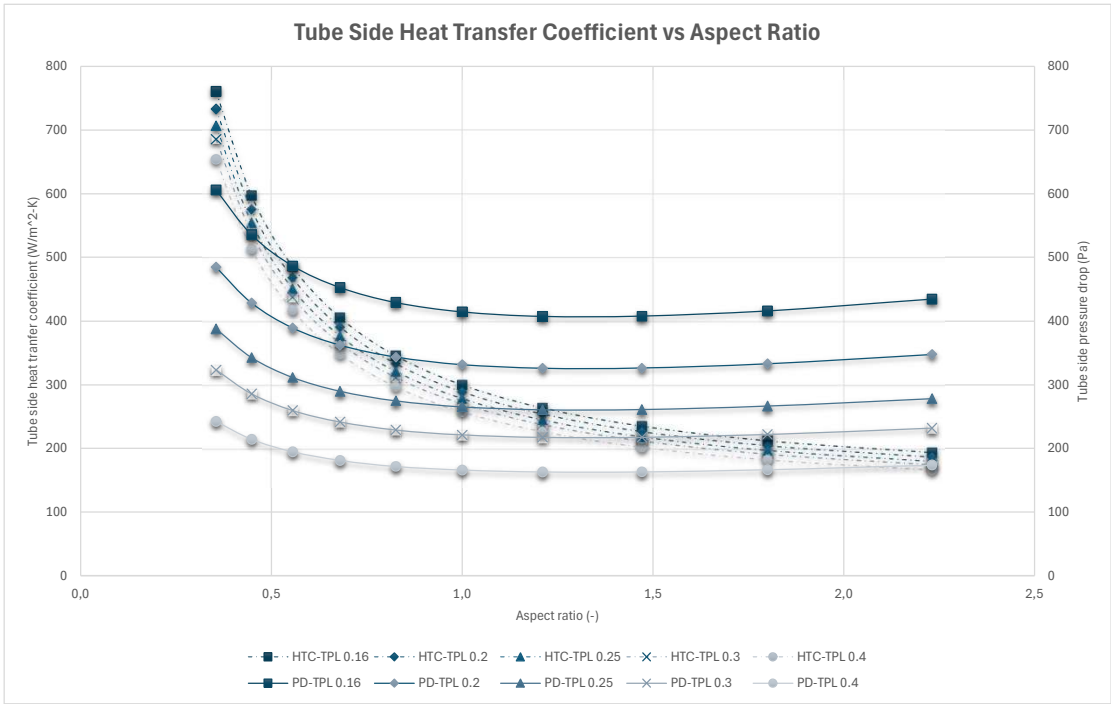
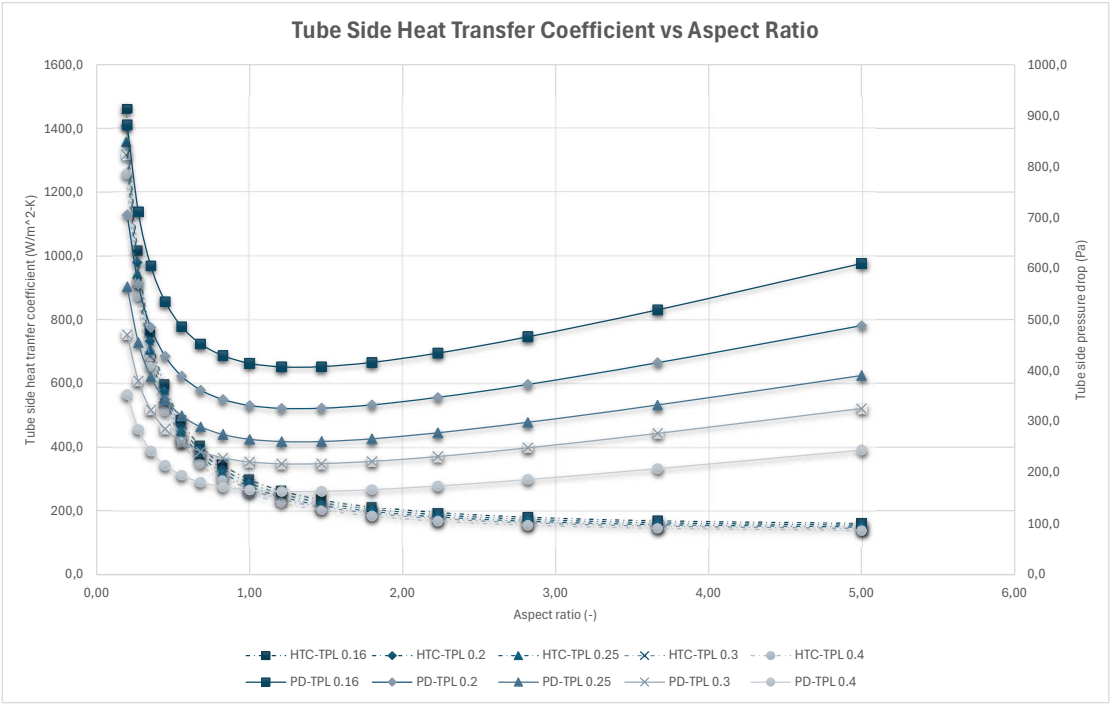


Tube Side Reynolds Number vs the Heat Exchanger Effectiveness



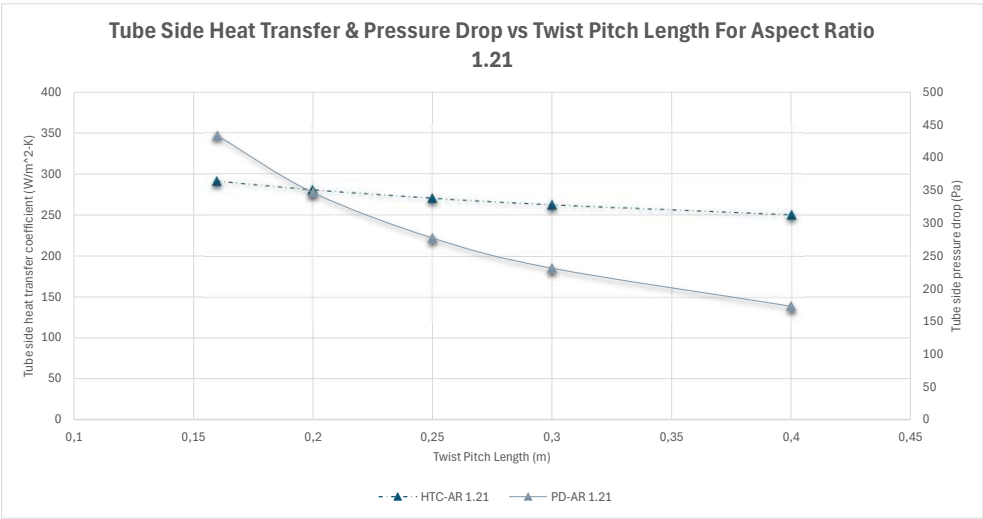
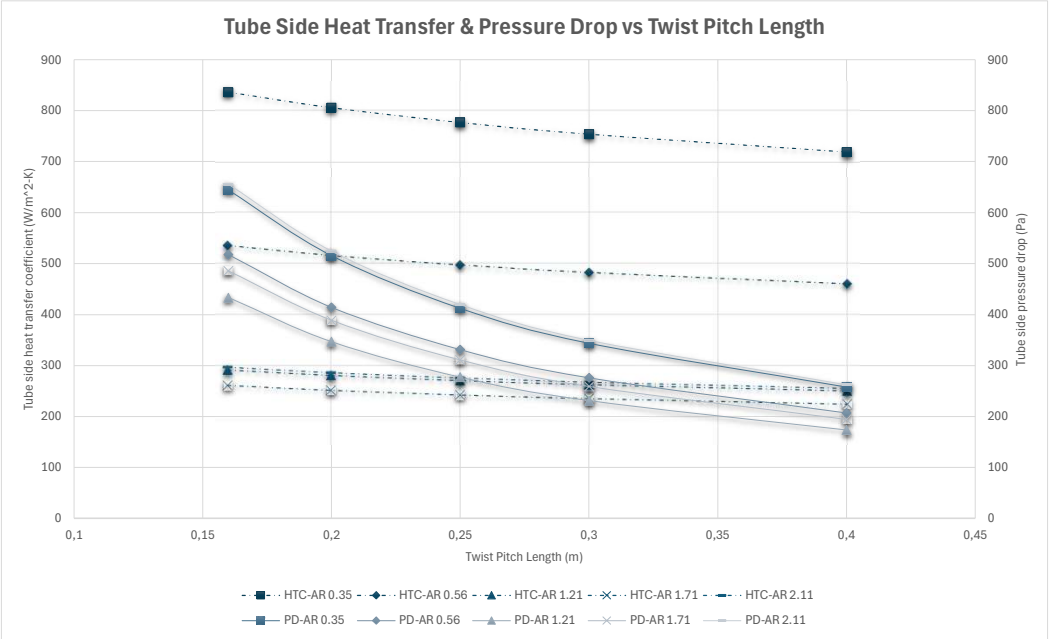
APPENDIX E: ASPECT RATION EVALUATION

a	b	Aspect ratio	TPL: 0.16		TPL: 0.2		TPL: 0.25		TPL: 0.3		TPL: 0.4	
			h i tube	P tube	h i tube	P tube	h i tube	P tube	h i tube	P tube	h i tube	P tube
			W/m^2K	Pa	W/m^2K	Pa	W/m^2K	Pa	W/m^2K	Pa	W/m^2K	Pa
1	7	35	0,20	1460,0	881,9	1407,0	705,5	1356,0	564,4	1316,0	470,3	1255,0
2	9	33	0,27	1016,0	711,5	979,3	569,2	943,9	455,3	915,9	379,5	873,5
3	11	31	0,35	760,5	605,7	733,0	484,6	706,5	387,7	685,6	323,1	653,8
4	13	29	0,45	597,5	535,3	575,9	428,2	555,1	342,6	538,7	285,5	513,7
5	15	27	0,56	486,1	486,5	468,6	389,2	451,6	311,4	438,2	259,5	417,9
6	17	25	0,68	406,1	452,4	391,4	361,9	377,3	289,6	366,1	241,3	349,1
7	19	23	0,83	346,4	429,1	333,9	343,3	321,8	274,6	312,3	228,9	297,8
8	21	21	1,00	300,6	414,4	289,7	331,6	279,3	265,2	271,0	221,0	258,4
9	23	19	1,21	264,7	407,4	255,2	325,9	245,9	260,7	238,6	217,3	227,6
10	25	17	1,47	236,2	407,7	227,6	326,1	219,4	260,9	212,9	217,4	203,0
11	27	15	1,80	213,2	416,0	205,5	332,8	198,1	266,3	192,2	221,9	183,3
12	29	13	2,23	194,7	434,3	187,7	347,5	180,9	278,0	175,5	231,6	167,4
13	31	11	2,82	180,0	466,2	173,5	373,0	167,2	298,4	162,3	248,7	154,7
14	33	9	3,67	168,6	519,3	162,5	415,4	156,6	332,3	152,0	277,0	145,0
15	35	7	5,00	160,6	610,1	154,8	488,1	149,2	390,5	144,8	325,4	138,1

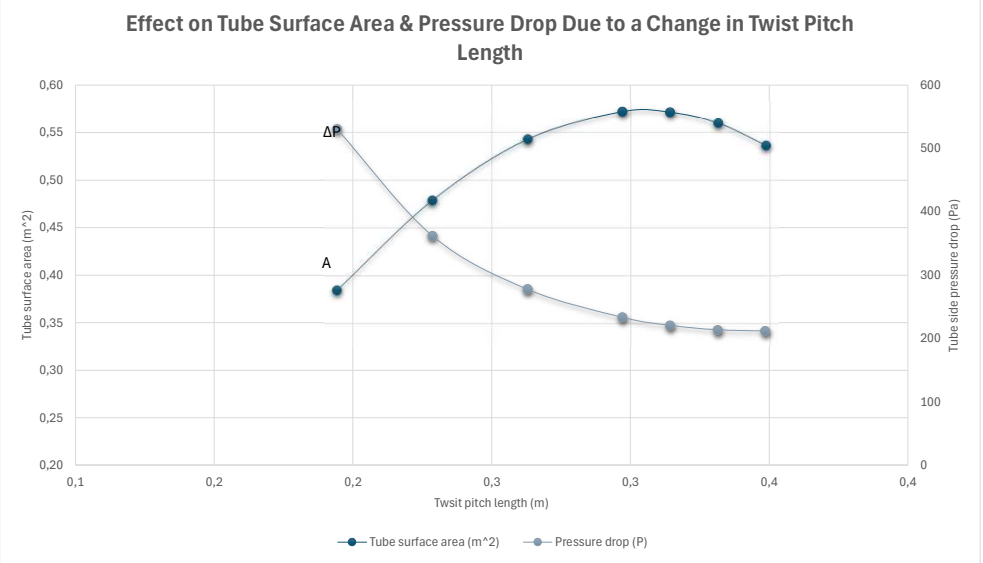
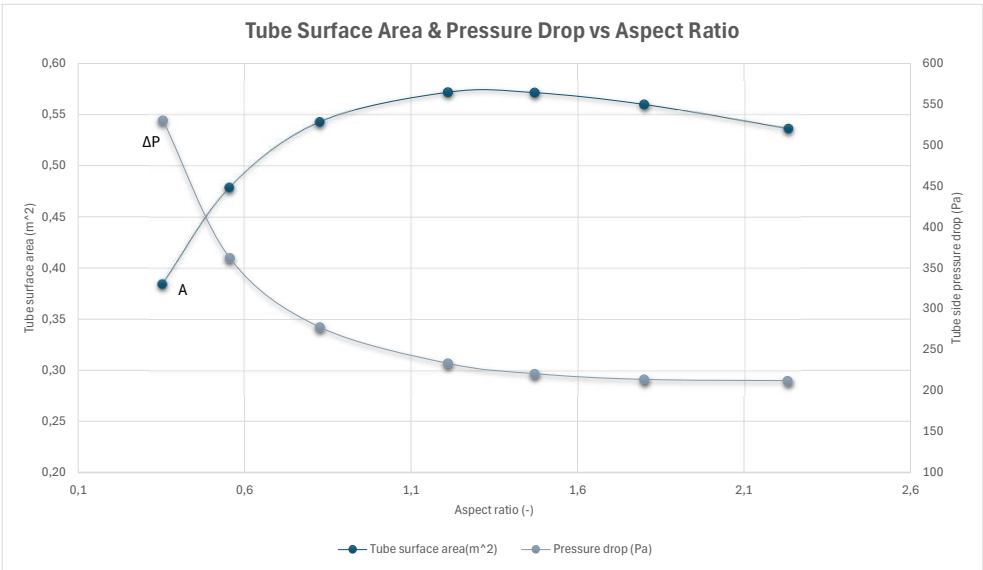


APPENDIX F: TWIST PITCH LENGTH EVALUATION

Name	a mm	b mm	Ratio =	Tube pitch length m	h i tube W/m^2K	Delta P Pa
Aspect ratio : 0,35	11	31	0,35	0,16	836,3	644,4
Aspect ratio : 0,35	11	31	0,35	0,2	806	515,5
Aspect ratio : 0,35	11	31	0,35	0,25	776,9	412,4
Aspect ratio : 0,35	11	31	0,35	0,3	753,9	343,7
Aspect ratio : 0,35	11	31	0,35	0,4	718,9	257,8
Aspect ratio : 0,56	15	27	0,56	0,16	534,6	517,6
Aspect ratio : 0,56	15	27	0,56	0,2	515,2	414,1
Aspect ratio : 0,56	15	27	0,56	0,25	496,6	331,2
Aspect ratio : 0,56	15	27	0,56	0,3	481,9	276
Aspect ratio : 0,56	15	27	0,56	0,4	459,6	207
Aspect ratio : 1,21	23	19	1,21	0,16	291,1	433,4
Aspect ratio : 1,21	23	19	1,21	0,2	280,6	346,7
Aspect ratio : 1,21	23	19	1,21	0,25	270,4	277,3
Aspect ratio : 1,21	23	19	1,21	0,3	262,4	231,1
Aspect ratio : 1,21	23	19	1,21	0,4	250,3	173,3
Aspect ratio : 1,71	24	14	1,71	0,16	260,9	485,9
Aspect ratio : 1,71	24	14	1,71	0,2	251,4	388,7
Aspect ratio : 1,71	24	14	1,71	0,25	242,3	311
Aspect ratio : 1,71	24	14	1,71	0,3	235,2	259,2
Aspect ratio : 1,71	24	14	1,71	0,4	224,3	194,4
Aspect ratio : 2,11	19,8	9,4	2,11	0,16	296,7	655,6
Aspect ratio : 2,11	19,8	9,4	2,11	0,2	285,9	524,5
Aspect ratio : 2,11	19,8	9,4	2,11	0,25	275,6	419,6
Aspect ratio : 2,11	19,8	9,4	2,11	0,3	267,4	349,7
Aspect ratio : 2,11	19,8	9,4	2,11	0,4	255	262,2



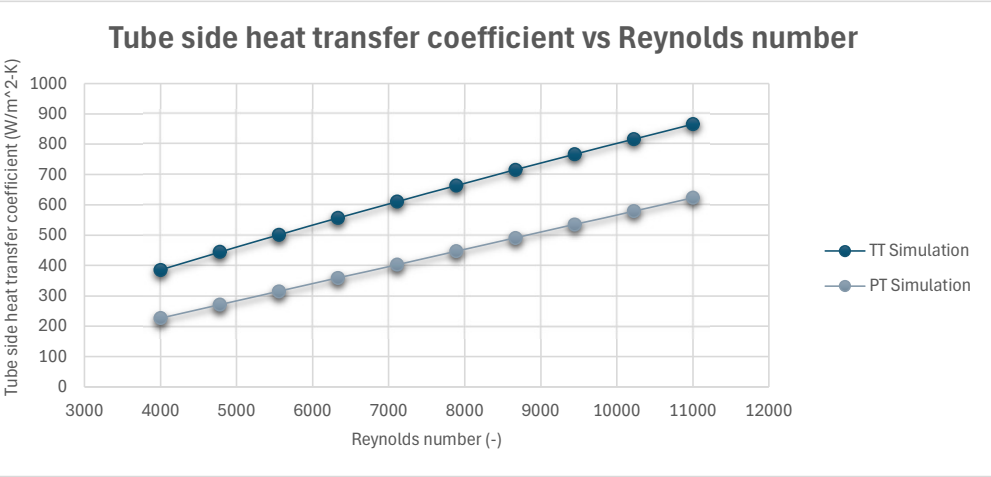
a mm	b mm	Ratio =	Tube pitch length m	A tube m^2	h i tube W/m^2-K	Delta P Pa
0,011	0,031	0,4	0,2	0,38	809,90	531
0,015	0,027	0,6	0,2	0,48	504,00	362
0,019	0,023	0,8	0,3	0,54	350,90	278
0,023	0,019	1,2	0,3	0,57	262,80	233
0,025	0,017	1,5	0,3	0,57	232,30	221
0,027	0,015	1,8	0,3	0,56	207,90	214
0,029	0,013	2,2	0,3	0,54	188,30	212
0,031	0,011	2,8	0,4	0,50	172,70	217
0,033	0,009	3,7	0,4	0,45	160,50	231
0,035	0,007	5,0	0,4	0,38	151,90	260



APPENDIX G: PLAIN TUBE VS. TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER PETROCHEMICAL OUTPUT COMPARISON

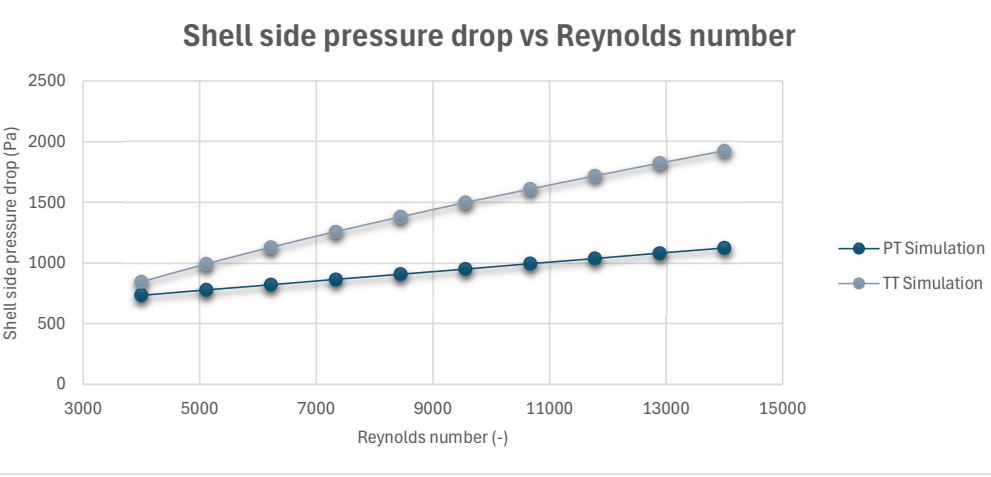
Tube side heat transfer coefficient vs Reynolds number

	PT	TT
x	y	y
	4000	226,6
	4778	270,7
	5556	314,8
	6333	358,9
	7111	402,9
	7889	447
	8667	491,1
	9444	535,1
	10222	579,2
	11000	623,3
		865,8



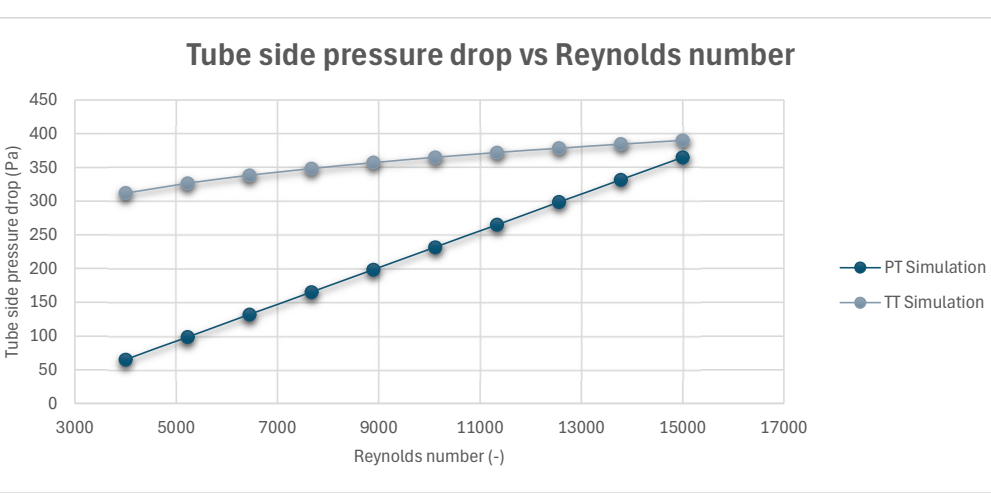
Shell side pressure drop vs Reynolds number

	PT	TT
x	y	y
	4000	734,8
	5111	778
	6222	821,3
	7333	864,5
	8444	907,7
	9556	950,9
	10667	994,2
	11778	1037
	12889	1081
	14000	1124
		1925



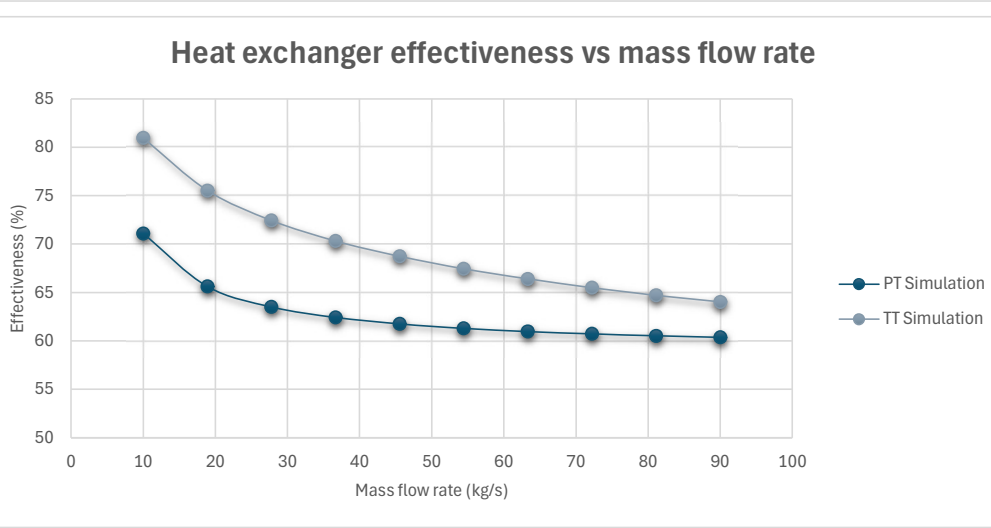
Tube side pressure vs Reynolds number

	PT	TT
x	y	y
	4000	65,12
	5222	98,45
	6444	131,8
	7667	165,1
	8889	198,5
	10111	231,8
	11333	265,1
	12556	298,5
	13778	331,8
	15000	365,1
		390,2



Heat exchanger effectiveness vs mass flow rate

	PT	TT
x	y	y
	10	71,09
	18,89	65,61
	27,78	63,52
	36,67	62,43
	45,56	61,76
	54,44	61,3
	63,33	60,97
	72,22	60,72
	81,11	60,53
	90	60,37
		64,02



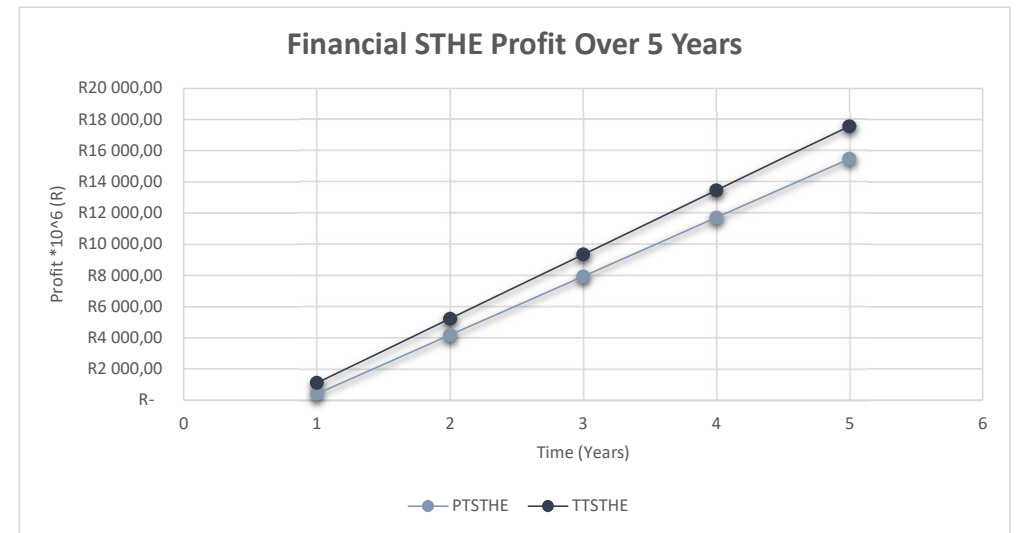
APPENDIX H: PLAIN TUBE VS. TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER FINANCIAL COMPARISON

PTSTHE

Year	Cost	Profit	Result
1	R 3 235,59	R 3 640,25	R 404,66
2	R 3 105,65	R 6 875,85	R 4 174,86
3	R 3 109,36	R 6 875,85	R 7 941,35
4	R 3 105,65	R 6 875,85	R 11 711,56
5	R 3 109,36	R 6 875,85	R 15 478,05
Total	R 15 665,60	R 31 143,65	R 39 710,48

TTSTHE

Year	Cost	Profit	Result
1	R 2 885,02	R 3 990,83	R 1 105,81
2	R 2 755,07	R 6 875,85	R 5 226,59
3	R 2 760,27	R 6 875,85	R 9 342,17
4	R 2 755,07	R 6 875,85	R 13 462,95
5	R 2 760,27	R 6 875,85	R 17 578,53
Total	R 13 915,69	R 31 494,23	R 46 716,06



APPENDIX I: PLAIN TUBE SHELL-AND-TUBE HEAT EXCHANGER EES CALCULATIONS

Main

$$\text{RUN}_{\text{Eff\$}} = \text{'True'}$$

$$\text{RUN}_{\text{HEAT\$}} = \text{'True'}$$

$$\text{RUN}_{\text{PRE\$}} = \text{'True'}$$

$$\text{RUN}_{\text{EB\$}} = \text{'True'}$$

$$\text{fluid}_{\text{tube;side\$}} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Steam'}$$

$$P_{i;\text{tube}} = 150 \text{ [kPa]}$$

$$P_{o;\text{tube}} = 135 \text{ [kPa]}$$

$$T_{o;\text{tube}} = (90 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;\text{shell}} = 306 \text{ [kPa]}$$

$$P_{o;\text{shell}} = 305 \text{ [kPa]}$$

$$T_{i;\text{tube}} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{i;\text{shell}} = (123 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{\text{bulk}} = 138 \text{ [kPa]}$$

$$T_{\text{bulk}} = (88 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{\text{wall}} = 170 \text{ [kPa]}$$

$$T_{\text{wall}} = (102,5 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{\text{bulk;shell}} = 305,5 \text{ [kPa]}$$

$$T_{\text{bulk;shell}} = (118 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{\text{wall;shell}} = 30 \text{ [kPa]}$$

$$T_{\text{wall;shell}} = (100 + 273,15) \cdot 1 \text{ [K]}$$

$$d_{o;\text{tube}} = 0,01 \text{ [m]}$$

$$d_{\text{inlet;tube}} = 0,1168 \text{ [m]}$$

$$\text{tube}_{\text{thk}} = 0,00165 \text{ [m]}$$

$$d_{i;\text{tube}} = d_{o;\text{tube}} - 2 \cdot \text{tube}_{\text{thk}}$$

$$d_{o;\text{shell}} = 0,6 \text{ [m]}$$

$$\text{shell}_{\text{thk}} = 0,016 \text{ [m]}$$

$$d_{i;\text{shell}} = d_{o;\text{shell}} - 2 \cdot \text{shell}_{\text{thk}}$$

$$d_{\text{inlet;shell}} = 0,1701 \text{ [m]}$$

$$w_{i;\text{tube}} = 55 \text{ [m/s]}$$

$$w_{o,tube} = 10 \text{ [m/s]}$$

$$w_{ave,tube} = \left| \frac{w_{i,tube} + w_{o,tube}}{2} \right|$$

$$w_{i,shell} = 15 \text{ [m/s]}$$

$$w_{o,shell} = 53 \text{ [m/s]}$$

$$w_{ave,shell} = \left| \frac{w_{o,shell} + w_{i,shell}}{2} \right|$$

$$S_1 = 0,024 \text{ [m]}$$

$$S_2 = 0,018 \text{ [m]}$$

$$a = \frac{S_1}{d_{o,tube}}$$

$$b = \frac{S_2}{d_{o,tube}}$$

$$L_{tube} = 4,26 \text{ [m]}$$

$$n_t = 442 \text{ [-]}$$

$$L_b = 0,3 \text{ [m]}$$

$$z = 130000 \text{ [-]}$$

$$C = 0,386 \text{ [-]}$$

$$\text{clearance} = 0,00495 \text{ [m]}$$

$$\text{pitch} = 0,024 \text{ [m]}$$

$$c_p = 0,86 \text{ [-]}$$

$$n_b = 4 \text{ [-]}$$

$$C_{if} = 2, \times 10^6 \text{ [R]}$$

$$c_i = 10000 \text{ [R]}$$

$$i_{\%} = 11 \text{ [%]}$$

$$\text{opp}_{\text{time}} = 20 \text{ [yr]}$$

$$C_s = 1,5 \times 10^6 \text{ [R]}$$

$$n = 1 \text{ [-]}$$

$$n_p = 58 \text{ [%]}$$

$$c_{\text{pump}} = 0,00288 \text{ [R/Wh]}$$

$$I = 0,38 \cdot m_{\text{shell}} \cdot 1 \text{ [s/kg]}$$

$$tt = 7300 \text{ [h/yr]}$$

$$j = 1,88 \text{ [R/W]}$$

$$q = 780 \text{ [W/h]}$$

$$e = 2,718 \text{ [-]}$$

$$m = 0,14 \text{ [-]}$$

$$k_{i,tube} = k(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$k_{o,tube} = k(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$k_{i,shell} = k(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$k_{o,shell} = k(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$k_{ave,tube} = \left| \frac{k_{o,tube} + k_{i,tube}}{2} \right|$$

$$k_{ave,shell} = \left| \frac{k_{o,shell} + k_{i,shell}}{2} \right|$$

$$\mu_{i,tube} = \text{Visc}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$\mu_{o,tube} = \text{Visc}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\mu_{i,shell} = \text{Visc}(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$\mu_{o,shell} = \text{Visc}(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\mu_{ave,tube} = \left| \frac{\mu_{o,tube} + \mu_{i,tube}}{2} \right|$$

$$\mu_{ave,shell} = \left| \frac{\mu_{o,shell} + \mu_{i,shell}}{2} \right|$$

$$\rho_{i,tube} = \rho(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$\rho_{o,tube} = \rho(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\rho_{i,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\rho_{ave,tube} = \left| \frac{\rho_{o,tube} + \rho_{i,tube}}{2} \right|$$

$$\rho_{ave,shell} = \left| \frac{\rho_{o,shell} + \rho_{i,shell}}{2} \right|$$

$$v_{i,tube} = \text{KinematicViscosity}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$v_{o,tube} = \text{KinematicViscosity}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$C_{p_{o; tube; kJ/kgK}} = C_p(\text{fluid}_{tube; side\$} ; T = T_{o; tube} ; P = P_{o; tube})$$

$$C_{p_{i:shell:kJ/kgK}} = C_p(\text{fluid}_{\text{shell:side}}; T = T_{i:shell}; P = P_{i:shell})$$

$$C_{p_{o;shell;kJ/kgK}} = C_p(\text{fluid}_{shell;side\$} ; T = T_{o;shell} ; P = P_{o;shell})$$

$$C_{p_{ave;shell}} = \frac{C_{p_{o;shell;kJ/kgK}} + C_{p_{i;shell;kJ/kgK}}}{2}$$

$$C_{p_{ave,tube}} = \frac{C_{p_{o,tube;kJ/kgK}} + C_{p_{i,tube;kJ/kgK}}}{2}$$

$$\mu_{\text{bulk}} = \text{Visc}(\text{fluid}_{\text{shell:side\$}} ; T = T_{\text{bulk}} ; P = P_{\text{bulk}})$$

$$\mu_{wall} = \mathbf{Visc}(\text{fluid}_{shell:side\$} ; T = T_{wall} ; P = P_{wall})$$

$$\mu_{\text{bulk:shell}} = \mathbf{Visc}(\text{fluid}_{\text{shell:side\$}}; T = T_{\text{bulk:shell}}; P = P_{\text{bulk:shell}})$$

$$\mu_{\text{wall;shell}} = \mathbf{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{\text{wall;shell}} ; P = P_{\text{wall;shell}})$$

$$Cp_{i;\text{tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{i;\text{tube};\text{kJ/kgK}}$$

$$C_{p_{o; tube}} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot C_{p_{o; tube; kJ/kgK}}$$

$$Cp_{i,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{i,shell;kJ/kgK}$$

$$C_{p_{o;shell}} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot C_{p_{o;shell};kJ/kgK}$$

$$i = \left| 0,01 \cdot \frac{\text{dim}}{\%} \right| \cdot i_{\%}^2$$

Tube side heat transfer coefficient

and-tube heat exchanger calcula

Heat Transfer Coefficient Calculation

$$Pr_{i;tube} = \frac{\mu_{i;tube} \cdot Cp_{i;tube}}{k_{i;tube}}$$

$$\text{Pr}_{\text{o;tube}} = \frac{\mu_{\text{o;tube}} \cdot \text{Cp}_{\text{o;tube}}}{k_{\text{o;tube}}}$$

$$f_a = 1 + \frac{2}{3 \cdot b}$$

$$y = 1 - \frac{\pi}{4 \cdot a \cdot b}$$

This section is not calculated for validation 2

$$Re_o = \frac{w_{o,tube} \cdot \frac{\pi}{2} \cdot d_{o,tube}}{v_{o,tube} \cdot y}$$

End of section

$$Re_i = \frac{w_{i,tube} \cdot \frac{\pi}{2} \cdot d_{i,tube}}{v_{i,tube} \cdot y}$$

$$v_{i,lam} = 0,664 \cdot \sqrt{Re_i} \cdot \sqrt{3} \cdot Pr_{i,tube}$$

$$v_{o,lam} = 0,664 \cdot \sqrt{Re_o} \cdot \sqrt{3} \cdot Pr_{o,tube}$$

$$v_{i,turb} = \frac{0,037 \cdot Re_i^{0,8} \cdot Pr_{i,tube}}{1 + 2,443 \cdot \frac{1}{Re_i^{0,1}} \cdot (Pr_{i,tube} - 1)}$$

$$v_{o,turb} = \frac{0,037 \cdot Re_o^{0,8} \cdot Pr_{o,tube}}{1 + 2,443 \cdot \frac{1}{Re_o^{0,1}} \cdot (Pr_{o,tube} - 1)}$$

$$v_{1,i} = 0,3 + \sqrt{v_{i,lam}^2 + v_{i,turb}^2}$$

$$v_{1,o} = 0,3 + \sqrt{v_{o,lam}^2 + v_{o,turb}^2}$$

$$v_{i,shell} = f_a \cdot v_{1,i}$$

$$v_{o,shell} = f_a \cdot v_{1,o}$$

$$h_{i,tube} = \frac{C \cdot Re_i^n \cdot \frac{Pr_{i,tube}^1}{3} \cdot k_{i,tube}}{d_{o,tube}}$$

$$h_{o,tube} = \frac{C \cdot Re_o^n \cdot \frac{Pr_{o,tube}^1}{3} \cdot k_{o,tube}}{d_{o,tube}}$$

$$h_{ave,tube} = \frac{h_{i,tube} + h_{o,tube}}{2}$$

$$h_{i,shell} = \frac{k_{i,shell} \cdot v_{i,shell}}{\frac{\pi}{2} \cdot d_{i,shell}}$$

$$h_{o,shell} = \frac{k_{o,shell} \cdot v_{o,shell}}{\frac{\pi}{2} \cdot d_{o,shell}}$$

$$h_{ave,shell} = \frac{h_{i,shell} + h_{o,shell}}{2}$$

$$U_{tube} = \frac{1}{\frac{1}{h_{i,tube}} + \frac{d_{i,tube} \cdot \ln\left[\frac{d_{o,tube}}{d_{i,tube}}\right]}{k_{o,tube}} + \frac{d_{i,tube}}{h_{o,tube} \cdot d_{o,tube}}}$$

$$U_{shell} = \frac{1}{\frac{1}{h_{i,shell}} + \frac{d_{i,shell} \cdot \ln\left[\frac{d_{o,shell}}{d_{i,shell}}\right]}{k_{o,shell}} + \frac{d_{i,shell}}{h_{o,shell} \cdot d_{o,shell}}}$$

Pressure Drop Calculation

$$P_{tube,i} = 1,5 \cdot \rho_{i,tube} \cdot \frac{w_{i,tube}^2}{2}$$

$$P_{tube,o} = 1,5 \cdot \rho_{o,tube} \cdot \frac{w_{o,tube}^2}{2}$$

$$s_m = d_{i,shell} \cdot L_b \cdot \frac{\text{clearance}}{\text{pitch}}$$

$$m_{tube} = \rho_{i,tube} \cdot w_{i,tube} \cdot \pi \cdot \left[\frac{d_{inlet,tube}}{2}\right]^2$$

$$G_s = \frac{m_{tube}}{s_m}$$

$$d_e = \frac{4 \cdot \left[c_p \cdot \text{pitch}^2 - \frac{\pi \cdot d_{o,tube}^2}{4}\right]}{\pi \cdot d_{o,tube}}$$

This section is not calculated for validation 4

$$Re_{tube} = \frac{w_{o,tube} \cdot L_{tube}}{v_{o,tube}}$$

End of section

$$f_{tube} = \frac{0,316}{|Re_{tube}|^{0,25}}$$

This section is not calculated for validation 3

$$Re_{shell} = \frac{d_e \cdot G_s}{\mu_{o,shell}}$$

End of section

$$f_{shell} = \frac{0,316}{|Re_{shell}|^{0,25}}$$

$$x = \frac{\mu_{bulk}}{\mu_{wall}}$$

$$P_{tubes} = \frac{2 \cdot f_{tube} \cdot \rho_{o,tube} \cdot w_{o,tube}^2 \cdot L_{tube}}{d_{i,tube}} \cdot x$$

This section is not calculated for validation 1

$$m_{shell} = \rho_{i,shell} \cdot w_{i,shell} \cdot \pi \cdot \left[\frac{d_{inlet,shell}}{2} \right]^2$$

End of section

$$P_{tube} = n_t \cdot \left[8 \cdot f_{tube} \cdot \frac{L_{tube}}{d_{i,tube}} \cdot \left[\frac{\mu_{i,tube}}{\mu_{wall}} \right]^{-m} + 2,5 \right] \cdot \frac{\rho_{o,tube} \cdot w_{i,tube}^2}{2}$$

$$P_{shell} = f_{shell} \cdot \frac{d_{o,shell}}{d_e} \cdot (n_b + 1) \cdot \rho_{o,shell} \cdot w_{o,shell}^2$$

Effectiveness Calculation

$$C_h = m_{shell} \cdot Cp_{i,shell}$$

$$C_c = m_{tube} \cdot Cp_{i,tube}$$

$$C_{min} = \text{Min}(C_h; C_c)$$

$$C_{max} = \text{Max}(C_h; C_c)$$

$$R_c = \frac{C_{min}}{C_{max}}$$

$$A_{tube} = 2 \cdot \pi \cdot \frac{d_{i,tube}}{2} \cdot L_{tube} \cdot n_t$$

$$N_{tu} = \frac{U_{tube} \cdot A_{tube} \cdot I}{C_{min}}$$

$$e_1 = 2 \cdot \left[\frac{1}{1 + R_c + (1 + R_c^2)^{0,5}} \cdot \left[\frac{1 + e^{(-N_{tu} \cdot (1 + R_c^2)^{0,5})}}{1 - e^{(-N_{tu} \cdot (1 + R_c^2)^{0,5})}} \right] \right] \cdot 100 \text{ [%]}$$

$$m_1 = 1$$

$$n_1 = 1$$

$$Q_{\max} = C_{\min} \cdot (T_{i;\text{shell}} - T_{i;\text{tube}})$$

$$Q_{\text{tube}} = C_c \cdot (T_{o;\text{tube}} - T_{i;\text{tube}})$$

$$Q_{\text{shell}} = C_h \cdot (T_{i;\text{shell}} - T_{o;\text{shell}})$$

$$\text{eff} = \frac{\left[\frac{\tanh\left[\frac{N_{tu}}{2} \cdot (1 + m_1 \cdot R_c^{n_1})^{\left[\frac{1}{n_1}\right]}\right]}{\frac{N_{tu}}{2} \cdot (1 + m_1 \cdot R_c^{n_1})^{\left[\frac{1}{n_1}\right]}} \right]}{\left[\frac{N_{tu}}{2} \cdot (1 + m_1 \cdot R_c^{n_1})^{\left[\frac{1}{n_1}\right]} \right]} \cdot 100 \text{ [%]}$$

$$e_2 = \frac{\left[\left[\frac{1 - \frac{e_1}{1 \text{ [%]}} \cdot R_c}{1 - \frac{e_1}{1 \text{ [%]}}} \right]^n - 1 \right] \cdot \left[\frac{1}{\left[\frac{1 - \frac{e_1}{1 \text{ [%]}} \cdot R_c}{1 - \frac{e_1}{1 \text{ [%]}}} \right]^n - R_c} \right]}{100} \cdot 100 \text{ [%]}$$

$$\delta T_{\text{tube}} = \frac{T_{o;\text{shell}} - T_{o;\text{tube}} - (T_{i;\text{shell}} - T_{i;\text{tube}})}{\ln\left[\frac{T_{o;\text{shell}} - T_{o;\text{tube}}}{T_{i;\text{shell}} - T_{i;\text{tube}}}\right]}$$

$$L_{\text{tube;check}} = \frac{Q_{\text{tube}}}{U_{\text{tube}} \cdot d_{i;\text{tube}} \cdot \delta T_{\text{tube}}}$$

Energy Balance

$$\text{enthalpy}_{i;\text{shell}} = h(\text{fluid}_{\text{shell;side\$}}; T = T_{i;\text{shell}}; P = P_{i;\text{shell}})$$

$$\text{enthalpy}_{o;\text{shell}} = h(\text{fluid}_{\text{shell;side\$}}; T = T_{o;\text{shell}}; P = P_{o;\text{shell}})$$

$$\text{enthalpy}_{i;\text{tube}} = h(\text{fluid}_{\text{tube;side\$}}; T = T_{i;\text{tube}}; P = P_{i;\text{tube}})$$

$$\text{enthalpy}_{o;\text{tube}} = h(\text{fluid}_{\text{tube;side\$}}; T = T_{o;\text{tube}}; P = P_{o;\text{tube}})$$

$$Q_{\text{in}} = Q_{\text{out}}$$

$$Q_{\text{in}} = m_{\text{shell}} \cdot (\text{enthalpy}_{i;\text{shell}} - \text{enthalpy}_{o;\text{shell}})$$

$$Q_{\text{out}} = m_{\text{tube}} \cdot (\text{enthalpy}_{o;\text{tube}} - \text{enthalpy}_{i;\text{tube}})$$

$$m_{i;\text{tube}} = \frac{Q_{\text{out}}}{\text{enthalpy}_{i;\text{tube}}}$$

$$m_{o;\text{tube}} = \frac{Q_{\text{out}}}{\text{enthalpy}_{o;\text{tube}}}$$

$$m_{i;\text{shell}} = \frac{Q_{\text{in}}}{\text{enthalpy}_{i;\text{shell}}}$$

$$m_{o;shell} = \frac{Q_{in}}{\text{enthalpy}_{o;shell}}$$

Plain Tube Inputs**Variables**

$$RUN_{Eff\$} = 'True'$$

$$RUN_{HEAT\$} = 'True'$$

$$RUN_{PRE\$} = 'True'$$

$$RUN_{EB\$} = 'True'$$

$$\text{fluid}_{tube;side\$} = 'Water'$$

$$\text{fluid}_{shell;side\$} = 'Steam'$$

$$P_{i;tube} = 150 \text{ [kPa]}$$

$$P_{o;tube} = 135 \text{ [kPa]}$$

$$T_{o;tube} = (90 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;shell} = 306 \text{ [kPa]}$$

$$P_{o;shell} = 305 \text{ [kPa]}$$

$$T_{i;tube} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{i;shell} = (123 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{bulk} = 138 \text{ [kPa]}$$

$$T_{bulk} = (88 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{wall} = 170 \text{ [kPa]}$$

$$T_{wall} = (102,5 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{bulk;shell} = 305,5 \text{ [kPa]}$$

$$T_{bulk;shell} = (118 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{wall;shell} = 30 \text{ [kPa]}$$

$$T_{wall;shell} = (100 + 273,15) \cdot 1 \text{ [K]}$$

$$d_{o;tube} = 0,01 \text{ [m]}$$

$$d_{inlet;tube} = 0,1168 \text{ [m]} \quad 4 \text{ \#150 Nozzle}$$

$$\text{tube}_{thk} = 0,00165 \text{ [m]}$$

$$d_{i;tube} = d_{o;tube} - 2 \cdot \text{tube}_{thk}$$

$$d_{o;shell} = 0,6 \text{ [m]}$$

$$\text{shell}_{\text{thk}} = 0,016 \text{ [m]}$$

$$d_{i,\text{shell}} = d_{o,\text{shell}} - 2 \cdot \text{shell}_{\text{thk}}$$

$$d_{\text{inlet,shell}} = 0,1701 \text{ [m]}$$

6 #150 Nozzle

$$w_{i,\text{tube}} = 55 \text{ [m/s]}$$

$$w_{o,\text{tube}} = 10 \text{ [m/s]}$$

$$w_{\text{ave,tube}} = \left| \frac{w_{i,\text{tube}} + w_{o,\text{tube}}}{2} \right|$$

$$w_{i,\text{shell}} = 15 \text{ [m/s]}$$

$$w_{o,\text{shell}} = 53 \text{ [m/s]}$$

$$w_{\text{ave,shell}} = \left| \frac{w_{o,\text{shell}} + w_{i,\text{shell}}}{2} \right|$$

$$S_1 = 0,024 \text{ [m]} \quad \text{Horizontal tube spacing}$$

$$S_2 = 0,018 \text{ [m]} \quad \text{Vertical tube spacing}$$

$$a = \frac{S_1}{d_{o,\text{tube}}} \quad \text{Longitudinal tube spacing ratio}$$

$$b = \frac{S_2}{d_{o,\text{tube}}} \quad \text{Transverse tube spacing ratio}$$

$$L_{\text{tube}} = 4,26 \text{ [m]}$$

$$n_t = 442 \text{ [-]} \quad \text{Number of tubes}$$

$$L_b = 0,3 \text{ [m]}$$

$$z = 130000 \text{ [-]}$$

$$C = 0,386 \text{ [-]} \quad \text{Baffle spacing}$$

$$\text{clearance} = 0,00495 \text{ [m]}$$

$$\text{pitch} = 0,024 \text{ [m]}$$

$$c_p = 0,86 \text{ [-]} \quad \text{Triangular tube pitch constant}$$

$$n_b = 4 \text{ [-]} \quad \text{Number of baffles}$$

$$C_{if} = 2, \times 10^6 \text{ [R]}$$

$$c_i = 10000 \text{ [R]}$$

$$i_{\%} = 11 \text{ [%]}$$

$$\text{opp}_{\text{time}} = 20 \text{ [yr]}$$

$$C_s = 1,5 \times 10^6 \text{ [R]}$$

$$n = 1 \text{ [-]}$$

$$n_p = 58 \text{ [%]}$$

$$C_{\text{pump}} = 0,00288 \text{ [R/Wh]}$$

$$l = 0,38 \cdot m_{\text{shell}} \cdot 1 \text{ [s/kg]}$$

$$tt = 7300 \text{ [h/yr]}$$

$$j = 1,88 \text{ [R/W]}$$

$$q = 780 \text{ [W/h]}$$

$$e = 2,718 \text{ [-]}$$

$$m = 0,14 \text{ [-]}$$

$$k_{i;\text{tube}} = k(\text{fluid}_{\text{tube;side\$}} ; T = T_{i;\text{tube}} ; P = P_{i;\text{tube}})$$

$$k_{o;\text{tube}} = k(\text{fluid}_{\text{tube;side\$}} ; T = T_{o;\text{tube}} ; P = P_{o;\text{tube}})$$

$$k_{i;\text{shell}} = k(\text{fluid}_{\text{shell;side\$}} ; T = T_{i;\text{shell}} ; P = P_{i;\text{shell}})$$

$$k_{o;\text{shell}} = k(\text{fluid}_{\text{shell;side\$}} ; T = T_{o;\text{shell}} ; P = P_{o;\text{shell}})$$

$$k_{\text{ave;tube}} = \left| \frac{k_{o;\text{tube}} + k_{i;\text{tube}}}{2} \right|$$

$$k_{\text{ave;shell}} = \left| \frac{k_{o;\text{shell}} + k_{i;\text{shell}}}{2} \right|$$

$$\mu_{i;\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{i;\text{tube}} ; P = P_{i;\text{tube}})$$

$$\mu_{o;\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{o;\text{tube}} ; P = P_{o;\text{tube}})$$

$$\mu_{i;\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{i;\text{shell}} ; P = P_{i;\text{shell}})$$

$$\mu_{o;\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{o;\text{shell}} ; P = P_{o;\text{shell}})$$

$$\mu_{\text{ave;tube}} = \left| \frac{\mu_{o;\text{tube}} + \mu_{i;\text{tube}}}{2} \right|$$

$$\mu_{\text{ave;shell}} = \left| \frac{\mu_{o;\text{shell}} + \mu_{i;\text{shell}}}{2} \right|$$

$$\rho_{i;\text{tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{i;\text{tube}} ; P = P_{i;\text{tube}})$$

$$\rho_{o;\text{tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{o;\text{tube}} ; P = P_{o;\text{tube}})$$

$$\rho_{i;\text{shell}} = \rho(\text{fluid}_{\text{shell;side\$}} ; T = T_{i;\text{shell}} ; P = P_{i;\text{shell}})$$

$$\rho_{o;\text{shell}} = \rho(\text{fluid}_{\text{shell;side\$}} ; T = T_{o;\text{shell}} ; P = P_{o;\text{shell}})$$

$$\rho_{\text{ave;tube}} = \left| \frac{\rho_{o;\text{tube}} + \rho_{i;\text{tube}}}{2} \right|$$

$$\rho_{ave;shell} = \left| \frac{\rho_{o;shell} + \rho_{i;shell}}{2} \right|$$

$$v_{i;tube} = \text{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{i;tube} ; P = P_{i;tube})$$

$$v_{o;tube} = \text{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$v_{ave;tube} = \left| \frac{v_{o;tube} + v_{i;tube}}{2} \right|$$

$$Cp_{i;tube;kJ/kgK} = Cp(\text{fluid}_{tube;side\$} ; T = T_{i;tube} ; P = P_{i;tube})$$

$$Cp_{o;tube;kJ/kgK} = Cp(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$Cp_{i;shell;kJ/kgK} = Cp(\text{fluid}_{shell;side\$} ; T = T_{i;shell} ; P = P_{i;shell})$$

$$Cp_{o;shell;kJ/kgK} = Cp(\text{fluid}_{shell;side\$} ; T = T_{o;shell} ; P = P_{o;shell})$$

$$Cp_{ave;shell} = \left| \frac{Cp_{o;shell;kJ/kgK} + Cp_{i;shell;kJ/kgK}}{2} \right|$$

$$Cp_{ave;tube} = \left| \frac{Cp_{o;tube;kJ/kgK} + Cp_{i;tube;kJ/kgK}}{2} \right|$$

$$\mu_{bulk} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{bulk} ; P = P_{bulk})$$

$$\mu_{wall} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{wall} ; P = P_{wall})$$

$$\mu_{bulk;shell} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{bulk;shell} ; P = P_{bulk;shell})$$

$$\mu_{wall;shell} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{wall;shell} ; P = P_{wall;shell})$$

convert units

$$Cp_{i;tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{i;tube;kJ/kgK}$$

$$Cp_{o;tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o;tube;kJ/kgK}$$

$$Cp_{i;shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{i;shell;kJ/kgK}$$

$$Cp_{o;shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o;shell;kJ/kgK}$$

$$i = \left| 0,01 \cdot \frac{\text{dim}}{\%} \right| \cdot i_{\%}^2$$

PT.V1

Plain Tube STHE Validation 1

Shell Side Mass Flow Rate vs. Shell Side Pressure Drop

RUN_{Eff}\$ = 'False'

RUN_{HEAT}\$ = 'False'

RUN_{PRE}\$ = 'True'

RUN_{EB}\$ = 'False'

Variables

fluid_{tube;side}\$ = 'Water'

fluid_{shell;side}\$ = 'Air'

d_{i;shell} = 0,1 [m]

d_{inlet;shell} = 0,1 [m]

d_{o;shell} = 0,2 [m]

d_{i;tube} = 0,015 [m]

d_{inlet;tube} = 0,015 [m]

d_{o;tube} = 0,017 [m]

s₂ = 0,02 [m]

b = $\frac{s_2}{d_{o;tube}}$

A_{shell} = $\pi \cdot \left[\frac{d_{o;shell}}{2} \right]^2$

L_{tube} = 0,5 [m]

n_t = 37 [-] Number of tubes

n_b = 4 [-] Number of baffles

d_{h;tube} = 0,015 [m]

L_b = 0,125 [m]

pitch = 0,02 [m]

clearance = pitch - d_{i;tube}

c_p = 0,86 [-]

z = 1 [-]

m = 0,14 [-] Turbulent flow constant

μ_{wall} = 0,0003545 [Pa-s]

μ_{bulk} = 0,00001754 [Pa-s]

$$\mu_{\text{wall;shell}} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{\text{bulk;shell}} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{\text{i;tube}} = \mathbf{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{i;tube}} ; P = P_{\text{i;tube}})$$

$$\mu_{\text{o;tube}} = \mathbf{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{o;tube}} ; P = P_{\text{o;tube}})$$

$$\mu_{\text{i;shell}} = \mathbf{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{\text{i;shell}})$$

$$\mu_{\text{o;shell}} = \mathbf{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{\text{o;shell}})$$

$$\mu_{\text{ave;tube}} = \left| \frac{\mu_{\text{o;tube}} + \mu_{\text{i;tube}}}{2} \right|$$

$$\mu_{\text{ave;shell}} = \left| \frac{\mu_{\text{o;shell}} + \mu_{\text{i;shell}}}{2} \right|$$

$$\text{SP}_{\text{heat;ratio;tube}} = 1 \text{ [-]}$$

$$\text{SP}_{\text{heat;ratio;shell}} = 1,4 \text{ [-]}$$

$$\nu_{\text{i;tube}} = \mathbf{KinematicViscosity}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{i;tube}} ; P = P_{\text{i;tube}})$$

$$\nu_{\text{o;tube}} = \mathbf{KinematicViscosity}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{o;tube}} ; P = P_{\text{o;tube}})$$

$$\nu_{\text{ave;tube}} = \left| \frac{\nu_{\text{o;tube}} + \nu_{\text{i;tube}}}{2} \right|$$

$$\text{Cp}_{\text{i;tube}} = 4,194 \text{ [kJ/kg-K]}$$

$$\text{Cp}_{\text{i;shell}} = 1,006 \text{ [kJ/kg-K]}$$

$$\text{Cp}_{\text{i;tube;calc}} = \mathbf{Cp}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{i;tube}} ; P = P_{\text{i;tube}})$$

$$\text{Cp}_{\text{o;tube;kJkgK}} = \mathbf{Cp}(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{o;tube}} ; P = P_{\text{o;tube}})$$

$$\text{Cp}_{\text{o;shell;kJkgK}} = \mathbf{Cp}(\text{fluid}_{\text{shell;side\$}} ; T = T_{\text{o;shell}})$$

$$\text{Cp}_{\text{ave;tube}} = \frac{\text{Cp}_{\text{i;tube}} + \text{Cp}_{\text{o;tube;kJkgK}}}{2}$$

$$\text{Cp}_{\text{ave;shell}} = \frac{\text{Cp}_{\text{i;shell}} + \text{Cp}_{\text{o;shell;kJkgK}}}{2}$$

$$\rho_{\text{i;tube}} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{\text{i;shell}} = 1,269 \text{ [kg/m}^3\text{]}$$

$$\rho_{\text{o;tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{\text{o;tube}} ; P = P_{\text{o;tube}})$$

$$\rho_{\text{o;shell}} = \rho(\text{fluid}_{\text{shell;side\$}} ; T = T_{\text{o;shell}} ; P = P_{\text{o;shell}})$$

$$\rho_{\text{ave;tube}} = \left| \frac{\rho_{\text{o;tube}} + \rho_{\text{i;tube}}}{2} \right|$$

$$\rho_{ave;shell} = \left| \frac{\rho_{o;shell} + \rho_{i;shell}}{2} \right|$$

$$k_{i;tube} = 0,6562 \text{ [W/m-K]}$$

$$k_{i;shell} = 0,02401 \text{ [W/m-K]}$$

$$k_{o;tube} = k(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$k_{o;shell} = k(\text{fluid}_{shell;side\$} ; T = T_{o;shell})$$

$$k_{ave;tube} = \left| \frac{k_{o;tube} + k_{i;tube}}{2} \right|$$

$$k_{ave;shell} = \left| \frac{k_{o;shell} + k_{i;shell}}{2} \right|$$

$$Re_i = 4000 \text{ [-]}$$

$$Re_o = 10000 \text{ [-]}$$

$$P_{i;tube} = 101,325 \text{ [kPa]}$$

$$P_{o;shell} = 85 \text{ [kPa]}$$

$$P_{i;shell} = 150 \text{ [kPa]}$$

$$P_{o;tube} = 10 \text{ [kPa]}$$

$$T_{i;tube} = 324,1 \text{ [K]}$$

$$T_{o;tube} = 348 \text{ [K]}$$

$$T_{i;shell} = 340 \text{ [K]}$$

$$T_{o;shell} = 281 \text{ [K]}$$

$$w_{i;shell} = m_{i;shell} \cdot \frac{w_{i;tube}}{m_{i;tube}} \quad \text{Relation calculated}$$

$$w_{o;shell} = \frac{m_{o;shell}}{\rho_{o;shell} \cdot A_{shell}} \quad \text{Relation calculated}$$

$$w_{i;tube} = 0,2 \text{ [m/s]}$$

$$w_{o;tube} = 0,1 \text{ [m/s]}$$

$$m_{i;tube} = 0,76049 \text{ [kg/s]}$$

$$m_{o;tube} = 0,38025 \text{ [kg/s]}$$

$$m_{i;shell} = 0,0099697 \text{ [kg/s]}$$

$$m_{o;shell} = 0,0039857 \text{ [kg/s]}$$

Convert Units

$$Cp_{o,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,shell;kJ/kgK}$$

$$Cp_{o,tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,tube;kJ/kgK}$$

Uncompiled equations within \$IF conditional statements

//Re_tube = 8000 [-]

//Re_shell = 8000 [-]

PT V2

Plain Tube STHE Validation 2

Tube Side Heat Transfer Coefficient vs. Reynolds number

RUN_Eff\$ = 'False'

RUN_HEAT\$ = 'True'

RUN_PRE\$ = 'False'

RUN_EB\$ = 'False'

Variables

fluid_tube;side\$ = 'Water'

fluid_shell;side\$ = 'Air'

Re_shell = 8000 [-]

Re_tube = 8000 [-]

P_i,tube = 101,325 [kPa]

P_o,shell = 85 [kPa]

P_i,shell = 150 [kPa]

P_o,tube = 10 [kPa]

T_i,tube = 324,1 [K]

T_o,tube = 348 [K]

T_i,shell = 340 [K]

T_o,shell = 281 [K]

d_i,shell = 0,1 [m]

d_inlet,shell = 0,1 [m]

d_o,shell = 0,2 [m]

d_i,tube = 0,015 [m]

d_inlet,tube = 0,015 [m]

$$d_{o,tube} = 0,017 \text{ [m]}$$

$$s_2 = 0,02 \text{ [m]}$$

$$s_1 = 0,02 \text{ [m]}$$

$$a = \frac{s_1}{d_{o,tube}}$$

$$b = \frac{s_2}{d_{o,tube}}$$

$$A_{shell} = \pi \cdot \left[\frac{d_{o,shell}}{2} \right]^2$$

$$L_{tube} = 0,5 \text{ [m]}$$

$$n_t = 37 \text{ [-] Number of tubes}$$

$$n_b = 4 \text{ [-] Number of baffles}$$

$$d_{h,tube} = 0,015 \text{ [m]}$$

$$L_b = 0,125 \text{ [m]}$$

$$\text{pitch} = 0,02 \text{ [m]}$$

$$\text{clearance} = \text{pitch} - d_{i,tube}$$

$$c_p = 0,86 \text{ [-]}$$

$$m = 0,14 \text{ [-]}$$

$$C = 0,386 \text{ [-]}$$

$$n = 0,492 \text{ [-] Turbulent flow constant}$$

$$\mu_{wall} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{bulk} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{wall,shell} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{bulk,shell} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{i,tube} = \mathbf{Visc}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$\mu_{o,tube} = \mathbf{Visc}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\mu_{i,shell} = \mathbf{Visc}(\text{fluid}_{shell,side\$} ; T = T_{i,shell})$$

$$\mu_{o,shell} = \mathbf{Visc}(\text{fluid}_{shell,side\$} ; T = T_{o,shell})$$

$$\mu_{ave,tube} = \left| \frac{\mu_{o,tube} + \mu_{i,tube}}{2} \right|$$

$$\mu_{ave;shell} = \left| \frac{\mu_{o;shell} + \mu_{i;shell}}{2} \right|$$

$$sp_{heat;ratio;tube} = 1 \text{ [-]}$$

$$sp_{heat;ratio;shell} = 1,4 \text{ [-]}$$

$$v_{o;tube} = \text{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$v_{i;tube} = \text{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{i;tube} ; P = P_{i;tube})$$

$$v_{ave;tube} = \left| \frac{v_{o;tube} + v_{i;tube}}{2} \right|$$

$$Cp_{i;tube} = 4194 \text{ [J/kg-K]}$$

$$Cp_{i;shell} = 1,006 \text{ [kJ/kg-K]}$$

$$Cp_{o;tube;kJkgK} = Cp(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$Cp_{o;shell;kJkgK} = Cp(\text{fluid}_{shell;side\$} ; T = T_{o;shell})$$

$$Cp_{ave;shell} = \left| \frac{Cp_{o;shell;kJkgK} + Cp_{i;shell}}{2} \right|$$

$$\rho_{i;tube} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{i;shell} = 1,269 \text{ [kg/m}^3\text{]}$$

$$\rho_{o;tube} = \rho(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$\rho_{o;shell} = \rho(\text{fluid}_{shell;side\$} ; T = T_{o;shell} ; P = P_{o;shell})$$

$$\rho_{ave;tube} = \left| \frac{\rho_{o;tube} + \rho_{i;tube}}{2} \right|$$

$$\rho_{ave;shell} = \left| \frac{\rho_{o;shell} + \rho_{i;shell}}{2} \right|$$

$$k_{i;tube} = 0,6562 \text{ [W/m-K]}$$

$$k_{i;shell} = 0,02401 \text{ [W/m-K]}$$

$$k_{o;tube} = k(\text{fluid}_{tube;side\$} ; T = T_{o;tube} ; P = P_{o;tube})$$

$$k_{o;shell} = k(\text{fluid}_{shell;side\$} ; T = T_{o;shell})$$

$$k_{ave;tube} = \left| \frac{k_{o;tube} + k_{i;tube}}{2} \right|$$

$$k_{ave;shell} = \left| \frac{k_{o;shell} + k_{i;shell}}{2} \right|$$

$$m_{i;tube} = 0,76049 \text{ [kg/s]}$$

$$m_{o;tube} = 0,38025 \text{ [kg/s]}$$

$$m_{i,shell} = 0,0099697 \text{ [kg/s]}$$

$$m_{o,shell} = 0,0039857 \text{ [kg/s]}$$

$$m_{ave,tube} = \frac{m_{i,tube} + m_{o,tube}}{2}$$

$$m_{ave,shell} = \frac{m_{i,shell} + m_{o,shell}}{2}$$

$$w_{i,shell} = m_{i,shell} \cdot \frac{w_{i,tube}}{m_{i,tube}} \quad \text{Relation calculated}$$

$$w_{o,shell} = \frac{m_{o,shell}}{\rho_{o,shell} \cdot A_{shell}} \quad \text{Relation calculated}$$

$$w_{i,tube} = 0,2 \text{ [m/s]}$$

$$w_{o,tube} = 0,1 \text{ [m/s]}$$

Convert Units

$$Cp_{o,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,shell;kJ/kgK}$$

$$Cp_{o,tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,tube;kJ/kgK}$$

PT V3

Plain Tube STHE Validation 3

Shell Side Pressure Drop vs. Shell Side Reynolds number

$$RUN_{Eff\$} = \text{'False'}$$

$$RUN_{HEAT\$} = \text{'False'}$$

$$RUN_{PRE\$} = \text{'True'}$$

$$RUN_{EB\$} = \text{'False'}$$

Variables

$$fluid_{tube;side\$} = \text{'Water'}$$

$$fluid_{shell;side\$} = \text{'Air'}$$

$$d_{i,shell} = 0,1 \text{ [m]}$$

$$d_{inlet,shell} = 0,1 \text{ [m]}$$

$$d_{o,shell} = 0,2 \text{ [m]}$$

$$d_{i,tube} = 0,015 \text{ [m]}$$

$$d_{\text{inlet,tube}} = 0,015 \text{ [m]}$$

$$d_{\text{o,tube}} = 0,017 \text{ [m]}$$

$$s_2 = 0,02 \text{ [m]}$$

$$b = \frac{s_2}{d_{\text{o,tube}}}$$

$$A_{\text{shell}} = \pi \cdot \left[\frac{d_{\text{o,shell}}}{2} \right]^2$$

$$L_{\text{tube}} = 0,5 \text{ [m]}$$

$$n_t = 37 \text{ [-] Number of tubes}$$

$$n_b = 4 \text{ [-] Number of baffles}$$

$$d_{\text{h,tube}} = 0,015 \text{ [m]}$$

$$L_b = 0,125 \text{ [m]}$$

$$\text{pitch} = 0,02 \text{ [m]}$$

$$\text{clearance} = \text{pitch} - d_{\text{i,tube}}$$

$$c_p = 0,86 \text{ [-]}$$

$$z = 1 \text{ [-]}$$

$$m = 0,14 \text{ [-] Turbulent flow constant}$$

$$\mu_{\text{wall}} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{\text{bulk}} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{\text{wall,shell}} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{\text{bulk,shell}} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{\text{i,tube}} = \text{Visc}(\text{fluid}_{\text{tube,side}}; T = T_{\text{i,tube}}; P = P_{\text{i,tube}})$$

$$\mu_{\text{o,tube}} = \text{Visc}(\text{fluid}_{\text{tube,side}}; T = T_{\text{o,tube}}; P = P_{\text{o,tube}})$$

$$\mu_{\text{o,shell}} = \text{Visc}(\text{fluid}_{\text{shell,side}}; T = T_{\text{o,shell}})$$

$$\mu_{\text{i,shell}} = \text{Visc}(\text{fluid}_{\text{shell,side}}; T = T_{\text{i,shell}})$$

$$\mu_{\text{ave,tube}} = \left| \frac{\mu_{\text{o,tube}} + \mu_{\text{i,tube}}}{2} \right|$$

$$\mu_{\text{ave,shell}} = \left| \frac{\mu_{\text{o,shell}} + \mu_{\text{i,shell}}}{2} \right|$$

$$sp_{\text{heat, ratio, tube}} = 1 \text{ [-]}$$

$$sp_{\text{heat, ratio, shell}} = 1,4 \text{ [-]}$$

$$v_{o,tube} = \text{KinematicViscosity}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$v_{i,tube} = \text{KinematicViscosity}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$v_{ave,tube} = \left| \frac{v_{o,tube} + v_{i,tube}}{2} \right|$$

$$cp_{i,tube} = 4,194 \text{ [kJ/kg-K]}$$

$$cp_{i,shell} = 1,006 \text{ [kJ/kg-K]}$$

$$cp_{i,tube,calc} = Cp(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$Cp_{o,tube,kJ/kgK} = Cp(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$Cp_{o,shell,kJ/kgK} = Cp(\text{fluid}_{shell,side\$} ; T = T_{o,shell})$$

$$\rho_{i,tube} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{i,shell} = 1,269 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,tube} = \rho(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\rho_{ave,tube} = \left| \frac{\rho_{o,tube} + \rho_{i,tube}}{2} \right|$$

$$\rho_{ave,shell} = \left| \frac{\rho_{o,shell} + \rho_{i,shell}}{2} \right|$$

$$k_{i,tube} = 0,6562 \text{ [W/m-K]}$$

$$k_{i,shell} = 0,02401 \text{ [W/m-K]}$$

$$k_{o,tube} = k(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$k_{o,shell} = k(\text{fluid}_{shell,side\$} ; T = T_{o,shell})$$

$$k_{ave,tube} = \left| \frac{k_{o,tube} + k_{i,tube}}{2} \right|$$

$$k_{ave,shell} = \left| \frac{k_{o,shell} + k_{i,shell}}{2} \right|$$

$$Re_i = 4000 \text{ [-]}$$

$$Re_o = 10000 \text{ [-]}$$

$$P_{i,tube} = 101,325 \text{ [kPa]}$$

$$P_{o,shell} = 85 \text{ [kPa]}$$

$$P_{i,shell} = 150 \text{ [kPa]}$$

$$P_{o,tube} = 10 \text{ [kPa]}$$

$$T_{i,tube} = 324,1 \text{ [K]}$$

$$T_{o,tube} = 348 \text{ [K]}$$

$$T_{i,shell} = 340 \text{ [K]}$$

$$T_{o,shell} = 281 \text{ [K]}$$

$$w_{i,shell} = \frac{Re_{shell} \cdot \mu_{i,shell}}{\rho_{i,shell} \cdot d_{o,shell} \cdot 2} \quad \text{Relation calculated}$$

$$w_{o,shell} = \frac{Re_{shell} \cdot \mu_{o,shell}}{\rho_{o,shell} \cdot d_{o,shell} \cdot 2} \quad \text{Relation calculated}$$

$$w_{i,tube} = 0,2 \text{ [m/s]}$$

$$w_{o,tube} = 0,1 \text{ [m/s]}$$

$$m_{i,tube} = 0,76049 \text{ [kg/s]}$$

$$m_{o,tube} = 0,38025 \text{ [kg/s]}$$

$$m_{i,shell} = 0,0099697 \text{ [kg/s]}$$

$$m_{ave,tube} = \frac{m_{i,tube} + m_{o,tube}}{2}$$

$$m_{ave,shell} = \frac{m_{i,shell} + m_{o,shell}}{2}$$

$$m_{o,shell} = 0,0039857 \text{ [kg/s]}$$

Convert Units

$$Cp_{o,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,shell;kJ/kgK}$$

$$Cp_{o,tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot Cp_{o,tube;kJ/kgK}$$

PT_V4

Plain Tube STHE Validation 4

Tube Side Pressure Drop vs. Tube Side Reynolds number

$$RUN_{Eff\$} = \text{'False'}$$

$$RUN_{HEAT\$} = \text{'False'}$$

$$RUN_{PRE\$} = \text{'True'}$$

$$RUN_{EB\$} = \text{'False'}$$

Variables

$$fluid_{tube;side\$} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Air'}$$

$$d_{i;\text{shell}} = 0,1 \text{ [m]}$$

$$d_{\text{inlet;shell}} = 0,1 \text{ [m]}$$

$$d_{o;\text{shell}} = 0,2 \text{ [m]}$$

$$d_{i;\text{tube}} = 0,015 \text{ [m]}$$

$$d_{\text{inlet;tube}} = 0,015 \text{ [m]}$$

$$d_{o;\text{tube}} = 0,017 \text{ [m]}$$

$$s_2 = 0,02 \text{ [m]}$$

$$b = \frac{s_2}{d_{o;\text{tube}}}$$

$$A_{\text{shell}} = \pi \cdot \left[\frac{d_{o;\text{shell}}}{2} \right]^2$$

$$L_{\text{tube}} = 0,5 \text{ [m]}$$

$$n_t = 37 \text{ [-]} \quad \text{Number of tubes}$$

$$n_b = 4 \text{ [-]} \quad \text{Number of baffles}$$

$$d_{h;\text{tube}} = 0,015 \text{ [m]}$$

$$L_b = 0,125 \text{ [m]}$$

$$\text{pitch} = 0,02 \text{ [m]}$$

$$\text{clearance} = \text{pitch} - d_{i;\text{tube}}$$

$$c_p = 0,86 \text{ [-]}$$

$$z = 1 \text{ [-]}$$

$$m = 0,14 \text{ [-]} \quad \text{Turbulent flow constant}$$

$$\mu_{\text{wall}} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{\text{bulk}} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{\text{wall;shell}} = 0,0003545 \text{ [Pa-s]}$$

$$\mu_{\text{bulk;shell}} = 0,00001754 \text{ [Pa-s]}$$

$$\mu_{i;\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}}; T = T_{i;\text{tube}}; P = P_{i;\text{tube}})$$

$$\mu_{o;\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}}; T = T_{o;\text{tube}}; P = P_{o;\text{tube}})$$

$$\mu_{o;\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}}; T = T_{o;\text{shell}})$$

$$\mu_{i;\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}}; T = T_{i;\text{shell}})$$

$$\mu_{ave,tube} = \left| \frac{\mu_{o,tube} + \mu_{i,tube}}{2} \right|$$

$$\mu_{ave,shell} = \left| \frac{\mu_{o,shell} + \mu_{i,shell}}{2} \right|$$

$$Sp_{heat,ratio,tube} = 1 \text{ [-]}$$

$$Sp_{heat,ratio,shell} = 1,4 \text{ [-]}$$

$$v_{o,tube} = \text{KinematicViscosity}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$cp_{i,tube} = 4,194 \text{ [kJ/kg-K]}$$

$$cp_{i,shell} = 1,006 \text{ [kJ/kg-K]}$$

$$cp_{i,tube,calc} = Cp(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$Cp_{o,tube,kJ/kgK} = Cp(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$Cp_{o,shell,kJ/kgK} = Cp(\text{fluid}_{shell,side\$} ; T = T_{o,shell})$$

$$\rho_{i,tube} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{i,shell} = 1,269 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,tube} = \rho(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\rho_{ave,tube} = \left| \frac{\rho_{o,tube} + \rho_{i,tube}}{2} \right|$$

$$\rho_{ave,shell} = \left| \frac{\rho_{o,shell} + \rho_{i,shell}}{2} \right|$$

$$k_{i,tube} = 0,6562 \text{ [W/m-K]}$$

$$k_{i,shell} = 0,02401 \text{ [W/m-K]}$$

$$k_{o,tube} = k(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$k_{o,shell} = k(\text{fluid}_{shell,side\$} ; T = T_{o,shell})$$

$$k_{ave,tube} = \left| \frac{k_{o,tube} + k_{i,tube}}{2} \right|$$

$$k_{ave,shell} = \left| \frac{k_{o,shell} + k_{i,shell}}{2} \right|$$

$$Re_i = 4000 \text{ [-]}$$

$$Re_o = 10000 \text{ [-]}$$

$$P_{i,tube} = 101,325 \text{ [kPa]}$$

$$P_{o,shell} = 85 \text{ [kPa]}$$

$$P_{i,\text{shell}} = 150 \text{ [kPa]}$$

$$P_{o,\text{tube}} = 10 \text{ [kPa]}$$

$$T_{i,\text{tube}} = 324,1 \text{ [K]}$$

$$T_{o,\text{tube}} = 348 \text{ [K]}$$

$$T_{i,\text{shell}} = 340 \text{ [K]}$$

$$T_{o,\text{shell}} = 281 \text{ [K]}$$

$$w_{i,\text{shell}} = \frac{Re_{\text{shell}} \cdot \mu_{i,\text{shell}}}{\rho_{i,\text{shell}} \cdot d_{o,\text{shell}} \cdot 2} \quad \text{Relation calculated}$$

$$w_{o,\text{shell}} = \frac{Re_{\text{shell}} \cdot \mu_{o,\text{shell}}}{\rho_{o,\text{shell}} \cdot d_{o,\text{shell}} \cdot 2} \quad \text{Relation calculated}$$

$$w_{i,\text{tube}} = \frac{Re_{\text{tube}} \cdot \mu_{i,\text{tube}}}{\rho_{i,\text{tube}} \cdot d_{o,\text{tube}}} \cdot 2,2$$

$$w_{o,\text{tube}} = \frac{Re_{\text{tube}} \cdot \mu_{o,\text{tube}}}{\rho_{o,\text{tube}} \cdot d_{o,\text{tube}}} \cdot 2,2$$

$$w_{\text{ave,tube}} = \frac{w_{i,\text{tube}} + w_{o,\text{tube}}}{2}$$

$$w_{\text{ave,shell}} = \frac{w_{i,\text{shell}} + w_{o,\text{shell}}}{2}$$

$$m_{i,\text{tube}} = 0,76049 \text{ [kg/s]}$$

$$m_{o,\text{tube}} = 0,38025 \text{ [kg/s]}$$

$$m_{i,\text{shell}} = 0,0099697 \text{ [kg/s]}$$

$$m_{\text{ave,tube}} = \frac{m_{i,\text{tube}} + m_{o,\text{tube}}}{2}$$

$$m_{\text{ave,shell}} = \frac{m_{i,\text{shell}} + m_{o,\text{shell}}}{2}$$

$$m_{o,\text{shell}} = 0,0039857 \text{ [kg/s]}$$

Convert Units

$$Cp_{o,\text{shell}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{o,\text{shell};\text{kJ/kgK}}$$

$$Cp_{o,\text{tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{o,\text{tube};\text{kJ/kgK}}$$

PT V5

Plain Tube STHE Validation 5

Heat Exchanger Efficiency vs. Shell Side Mass Flow Rate

Tube Side Fluid = Water

Shell Side Fluid = Water

 $RUN_{Eff\$} = 'True'$ $RUN_{HEAT\$} = 'True'$ $RUN_{PRE\$} = 'False'$ $RUN_{EB\$} = 'False'$

Variables

 $fluid_{tube;side\$} = 'Water'$ $fluid_{shell;side\$} = 'Water'$ $d_{i,tube} = 0,0254 \text{ [m]}$ $d_{o,tube} = 0,0257 \text{ [m]}$ $d_{i,shell} = 0,1544 \text{ [m]}$ $d_{o,shell} = 0,1564 \text{ [m]}$ $m_{tube} = 1,5 \text{ [kg/s]}$

This section is not calculated for validation 5

 $m_{shell} = 1,5 \text{ [kg/s]}$

End of section

 $L_{tube} = 2 \text{ [m]}$ $I = 0,38 \cdot m_{shell} \cdot 1 \text{ [s/kg]}$ $n_b = 22 \text{ [-]}$ $n_t = 12 \text{ [-]}$ $a = 1,25 \text{ [-]}$ $b = 1,5 \text{ [-]}$ $e = 2,718 \text{ [-]}$ $C = 0,386 \text{ [-]}$ $n = 0,492 \text{ [-]}$ $T_{o,shell} = 311,52 \text{ [K]}$

$$T_{i,shell} = 313 \text{ [K]}$$

$$T_{i,tube} = 303,55 \text{ [K]}$$

$$T_{o,tube} = 304,44 \text{ [K]}$$

$$P_{i,shell} = 100,27659 \text{ [kPa]}$$

$$P_{o,shell} = 115,50156 \text{ [kPa]}$$

$$P_{i,tube} = 100,27659 \text{ [kPa]}$$

$$P_{o,tube} = 103,65992 \text{ [kPa]}$$

$$\rho_{i,shell} = 994,75 \text{ [kg/m}^3\text{]}$$

$$\rho_{i,tube} = 996,1 \text{ [kg/m}^3\text{]}$$

$$Cp_{i,shell} = 4178,375 \text{ [J/kg-K]}$$

$$Cp_{i,tube} = 4179,125 \text{ [J/kg-K]}$$

$$Cp_{o,tube;kJ/kgK} = \mathbf{Cp}(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\mu_{i,shell} = 698 \text{ [Pa-s]}$$

$$\mu_{i,tube} = 786 \text{ [Pa-s]}$$

$$\mu_{o,tube} = \mathbf{Visc}(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\mu_{ave,tube} = \left| \frac{\mu_{o,tube} + \mu_{i,tube}}{2} \right|$$

$$k_{i,shell} = 0,624 \text{ [W/m-K]}$$

$$k_{o,shell} = 0,632 \text{ [W/m-K]}$$

$$k_{i,tube} = 0,616 \text{ [W/m-K]}$$

$$k_{o,tube} = 0,614 \text{ [W/m-K]}$$

$$k_{ave,tube} = \left| \frac{k_{o,tube} + k_{i,tube}}{2} \right|$$

$$k_{ave,shell} = \left| \frac{k_{o,shell} + k_{i,shell}}{2} \right|$$

$$w_{i,tube} = 0,887 \text{ [m/s]}$$

$$w_{o,tube} = 0,443 \text{ [m/s]}$$

$$w_{ave,tube} = \frac{w_{i,tube} + w_{o,tube}}{2}$$

$$v_{i,tube} = \mathbf{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$v_{o,tube} = \mathbf{KinematicViscosity}(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$V_{\text{ave;tube}} = \frac{V_{\text{i;tube}} + V_{\text{o;tube}}}{2}$$

Convert Units

$$Cp_{\text{o;tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{\text{o;tube;kJ/kgK}}$$

Parametric Table: Table 1

	Re_o [-]	$h_{o,tube}$ [W/m ² -K]
Run 1	4000	226,6
Run 2	4778	270,7
Run 3	5556	314,8
Run 4	6333	358,9
Run 5	7111	402,9
Run 6	7889	447
Run 7	8667	491,1
Run 8	9444	535,1
Run 9	10222	579,2
Run 10	11000	623,3

Parametric Table: Table 2

	Re_{shell} [-]	P_{shell} [Pa]
Run 1	4000	734,8
Run 2	5111	778
Run 3	6222	821,3
Run 4	7333	864,5
Run 5	8444	907,7
Run 6	9556	950,9
Run 7	10667	994,2
Run 8	11778	1037
Run 9	12889	1081
Run 10	14000	1124

Parametric Table: Table 3

	Re_{tube} [-]	P_{tube} [Pa]
Run 1	4000	65,12
Run 2	5222	98,45
Run 3	6444	131,8
Run 4	7667	165,1
Run 5	8889	198,5
Run 6	10111	231,8
Run 7	11333	265,1
Run 8	12556	298,5
Run 9	13778	331,8
Run 10	15000	365,1

Parametric Table: Table 4

	m [-]	e_1 [%]
Run 1	10	71,09
Run 2	18,89	65,61
Run 3	27,78	63,52
Run 4	36,67	62,43
Run 5	45,56	61,76
Run 6	54,44	61,3
Run 7	63,33	60,97
Run 8	72,22	60,72

Parametric Table: Table 4

	m [-]	e₁ [%]
Run 9	81,11	60,53
Run 10	90	60,37

Parametric Table: Validation 1

	m_{shell} [kg/s]	P_{shell} [Pa]
Run 1	0,003923	9,314
Run 2	0,005988	21,7
Run 3	0,007974	38,48
Run 4	0,01248	94,26
Run 5	0,01773	190,2
Run 6	0,02392	346,3
Run 7	0,03077	573

Parametric Table: Validation 2

	Re_o [-]	h_{o;tube} [W/m ² -K]
Run 1	4000	3,939
Run 2	6000	6,089
Run 3	8000	7,896
Run 4	10000	9,486

Parametric Table: Validation 3

	Re_{shell} [-]	P_{shell} [Pa]
Run 1	4007	20,21
Run 2	5008	29,87
Run 3	6009	41,09
Run 4	7010	53,81
Run 5	8011	67,97
Run 6	9012	83,52
Run 7	10014	100,4
Run 8	11015	118,7
Run 9	12016	138,2
Run 10	13017	158,9

Parametric Table: Validation 4

	Re_{tube} [-]	P_{tube} [Pa]	P_{tube;check} [Pa]
Run 1	4013	49,35	12,66
Run 2	5000	72,52	19,65
Run 3	5988	99,59	28,18
Run 4	6975	130,5	38,24
Run 5	7963	165	49,84
Run 6	8950	203,2	62,96
Run 7	9938	245	77,62
Run 8	10925	290,3	93,81
Run 9	11913	339,1	111,5

Parametric Table: Validation 4

	Re_{tube} [-]	P_{tube} [Pa]	$P_{tube;check}$ [Pa]
Run 10	12900	391,3	130,8

Parametric Table: Validation 5

	m_{shell} [kg/s]	eff [%]
Run 1	0,5	53,79
Run 2	0,9875	65,74
Run 3	1,475	73,82
Run 4	1,963	80,01
Run 5	2,45	85,02

Parametric Table: Simulation

	$T_{i;shell}$ [K]	$T_{i;tube}$ [K]	$h_{i;shell}$ [W/m ² -K]	$h_{i;tube}$ [W/m ² -K]	$h_{ave;shell}$	$h_{ave;tube}$
Run 1	396,2	353,2	132	371,2	106,7	285,9
Run 2	394,2	354,3	132,3	371	106,8	285,8
Run 3	392,2	355,4	132,6	370,9	107	285,7
Run 4	390,2	356,5	132,9	370,7	107,1	285,6
Run 5	388,2	357,6	133,1	370,5	107,2	285,5
Run 6	386,2	358,7	133,4	370,3	107,4	285,4
Run 7	384,2	359,8	133,6	370,1	107,5	285,3
Run 8	382,2	360,9	133,9	369,9	107,6	285,2
Run 9	380,2	362	134,1	369,7	107,7	285,1
Run 10	378,2	363,2	134,4	369,5	107,8	285

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$a = 2,4$ [-]
 $b = 1,8$ [-]
 $clearance = 0,00495$ [m]
 $C_{p,ave,tube} = 4,201$ [kJ/kg-K]
 $C_{p,i,shell,kJ/kgK} = 4,248$ [kJ/kg-K]
 $C_{p,i,tube,kJ/kgK} = 4,197$ [kJ/kg-K]
 $C_{p,o,shell,kJ/kgK} = 4,222$ [kJ/kg-K]
 $C_{p,o,tube,kJ/kgK} = 4,205$ [kJ/kg-K]
 $C_h = 1,362E+06$ [W/K]
 $C_{if} = 2,000E+06$ [R]
 $C_{min} = 1,362E+06$ [W/K]
 $c_{pump} = 0,00288$ [R/Wh]
 $\delta T_{tube} = 26,8$ [K]
 $d_{inlet,shell} = 0,1701$ [m]
 $d_{i,shell} = 0,568$ [m]
 $d_{o,shell} = 0,6$ [m]
 $e = 2,718$ [-]
 $enthalpy_{i,shell} = 516,6$ [kJ/kg]
 $enthalpy_{o,shell} = 441,6$ [kJ/kg]
 $e_1 = 49,88$ [%]
 $fluid_{shell,side\$} = 'Steam'$
 $f_a = 1,37$ [-]
 $f_{tube} = 0,002954$ [-]
 $h_{ave,shell} = 4554$ [W/m²-K]
 $h_{i,shell} = 6602$ [W/m²-K]
 $h_{o,shell} = 2505$ [W/m²-K]
 $i = 1,21$ [%]
 $j = 1,88$ [R/W]
 $k_{ave,tube} = 0,6699$ [W/m-K]
 $k_{i,tube} = 0,667$ [W/m-K]
 $k_{o,tube} = 0,6728$ [W/m-K]
 $L_b = 0,3$ [m]
 $L_{tube,check} = 533875$ [m]
 $\mu_{ave,shell} = 0,0002464$ [Pa-s]
 $\mu_{bulk} = 0,0003215$ [Pa-s]
 $\mu_{i,shell} = 0,000226$ [Pa-s]
 $\mu_{o,shell} = 0,0002668$ [Pa-s]
 $\mu_{wall} = 0,0002744$ [Pa-s]
 $m_1 = 1$ [-]
 $m_{i,tube} = 71,77$ [kg/s]
 $m_{o,tube} = 63,78$ [kg/s]
 $m_{tube} = 572,7$ [kg/s]
 $v_{1,i} = 6297$ [-]
 $v_{i,lam} = 3570$ [-]
 $v_{i,turb} = 5187$ [-]
 $v_{o,shell} = 3477$ [-]
 $n_1 = 1$ [-]
 $n_p = 58$ [%]
 $N_{tu} = 0,8889$ [-]
 $pitch = 0,024$ [m]
 $Pr_{o,tube} = 1,964$ [-]
 $P_{bulk,shell} = 305,5$ [kPa]
 $P_{i,tube} = 150$ [kPa]
 $P_{o,tube} = 135$ [kPa]
 $P_{tube} = 1,097E+10$ [Pa]
 $P_{tube,i} = 2,205E+06$ [Pa]
 $P_{wall} = 170$ [kPa]
 $q = 780$ [W/h]
 $Q_{max} = 5,858E+07$ [W]

$A_{tube} = 39,63$ [m²]
 $C = 0,386$ [-]
 $C_{p,ave,shell} = 4,235$ [kJ/kg-K]
 $C_{p,i,shell} = 4248$ [J/kg-K]
 $C_{p,i,tube} = 4197$ [J/kg-K]
 $C_{p,o,shell} = 4222$ [J/kg-K]
 $C_{p,o,tube} = 4205$ [J/kg-K]
 $C_c = 2,403E+06$ [W/K]
 $c_i = 10000$ [R]
 $C_{max} = 2,403E+06$ [W/K]
 $c_p = 0,86$ [-]
 $C_s = 1,500E+06$ [R]
 $d_e = 0,05307$ [m]
 $d_{inlet,tube} = 0,1168$ [m]
 $d_{i,tube} = 0,0067$ [m]
 $d_{o,tube} = 0,01$ [m]
 $eff = 86,46$ [%]
 $enthalpy_{i,tube} = 335,1$ [kJ/kg]
 $enthalpy_{o,tube} = 377,1$ [kJ/kg]
 $e_2 = 49,88$ [%]
 $fluid_{tube,side\$} = 'Water'$
 $f_{shell} = 0,007447$ [-]
 $G_s = 16295$ [kg/m²-s]
 $h_{ave,tube} = 2,358E+07$ [W/m²-K]
 $h_{i,tube} = 3,712E+07$ [W/m²-K]
 $h_{o,tube} = 1,003E+07$ [W/m²-K]
 $i\% = 11$ [%]
 $k_{ave,shell} = 0,6809$ [W/m-K]
 $k_{i,shell} = 0,6826$ [W/m-K]
 $k_{o,shell} = 0,6791$ [W/m-K]
 $l = 121,9$ [-]
 $L_{tube} = 4,26$ [m]
 $m = 0,14$ [-]
 $\mu_{ave,tube} = 0,0003341$ [Pa-s]
 $\mu_{bulk,shell} = 0,0002363$ [Pa-s]
 $\mu_{i,tube} = 0,0003541$ [Pa-s]
 $\mu_{o,tube} = 0,0003142$ [Pa-s]
 $\mu_{wall,shell} = 0,00001231$ [Pa-s]
 $m_{i,shell} = 46,55$ [kg/s]
 $m_{o,shell} = 54,46$ [kg/s]
 $m_{shell} = 320,7$ [kg/s]
 $n = 1$ [-]
 $v_{1,o} = 2537$ [-]
 $v_{i,shell} = 8629$ [-]
 $v_{o,lam} = 1735$ [-]
 $v_{o,turb} = 1851$ [-]
 $n_b = 4$ [-]
 $n_t = 442$ [-]
 $opp_{time} = 20$ [yr]
 $Pr_{i,tube} = 2,228$ [-]
 $P_{bulk} = 138$ [kPa]
 $P_{i,shell} = 306$ [kPa]
 $P_{o,shell} = 305$ [kPa]
 $P_{shell} = 1,129E+06$ [Pa]
 $P_{tubes} = 424956$ [Pa]
 $P_{tube,o} = 72399$ [Pa]
 $P_{wall,shell} = 30$ [kPa]
 $Q_{in} = 24050$ [kW]
 $Q_{out} = 24050$ [kW]

$Q_{\text{shell}} = 2,413\text{E}+07$ [W]
 $Re_i = 1,942\text{E}+06$ [-]
 $Re_{\text{shell}} = 3,242\text{E}+06$ [-]
 $\rho_{\text{ave,shell}} = 947,6$ [kg/m³]
 $\rho_{i,\text{shell}} = 940,7$ [kg/m³]
 $\rho_{o,\text{shell}} = 954,6$ [kg/m³]
 $RUN_{\text{EB\$}} = \text{'True'}$
 $RUN_{\text{HEAT\$}} = \text{'True'}$
 $R_c = 0,5668$ [-]
 $S_1 = 0,024$ [m]
 $s_m = 0,03515$ [m²]
 $tube_{\text{thk}} = 0,00165$ [m]
 $T_{\text{bulk,shell}} = 391,2$ [K]
 $T_{i,\text{tube}} = 353,2$ [K]
 $T_{o,\text{tube}} = 363,2$ [K]
 $T_{\text{wall,shell}} = 373,2$ [K]
 $U_{\text{tube}} = 250,7$ [W/m²-K]
 $v_{i,\text{tube}} = 3,643\text{E}-07$ [m²/s]
 $w_{\text{ave,shell}} = 34$ [m/s]
 $w_{i,\text{shell}} = 15$ [m/s]
 $w_{o,\text{shell}} = 53$ [m/s]
 $x = 1,172$ [-]
 $z = 130000$ [-]

$Q_{\text{tube}} = 2,403\text{E}+07$ [W]
 $Re_o = 589864$ [-]
 $Re_{\text{tube}} = 1,309\text{E}+08$ [-]
 $\rho_{\text{ave,tube}} = 968,6$ [kg/m³]
 $\rho_{i,\text{tube}} = 971,8$ [kg/m³]
 $\rho_{o,\text{tube}} = 965,3$ [kg/m³]
 $RUN_{\text{Eff\$}} = \text{'True'}$
 $RUN_{\text{PRE\$}} = \text{'True'}$
 $shell_{\text{thk}} = 0,016$ [m]
 $S_2 = 0,018$ [m]
 $tt = 7300$ [h/yr]
 $T_{\text{bulk}} = 361,2$ [K]
 $T_{i,\text{shell}} = 396,2$ [K]
 $T_{o,\text{shell}} = 378,4$ [K]
 $T_{\text{wall}} = 375,7$ [K]
 $U_{\text{shell}} = 21,57$ [W/m²-K]
 $v_{\text{ave,tube}} = 3,449\text{E}-07$ [m²/s]
 $v_{o,\text{tube}} = 3,255\text{E}-07$ [m²/s]
 $w_{\text{ave,tube}} = 32,5$ [m/s]
 $w_{i,\text{tube}} = 55$ [m/s]
 $w_{o,\text{tube}} = 10$ [m/s]
 $y = 0,8182$ [-]

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$A_{\text{shell}} = 0,03142 \text{ [m}^2\text{]}$
 $\text{clearance} = 0,005 \text{ [m]}$
 $C_{p,\text{ave,tube}} = 3,05 \text{ [kJ/kg-K]}$
 $C_{p,i,\text{tube}} = 4,194 \text{ [kJ/kg-K]}$
 $C_{p,o,\text{shell}} = 1004 \text{ [J/kg-K]}$
 $C_{p,o,\text{tube}} = 1907 \text{ [J/kg-K]}$
 $c_p = 0,86 \text{ [-]}$
 $d_{h,\text{tube}} = 0,015 \text{ [m]}$
 $d_{\text{inlet,tube}} = 0,015 \text{ [m]}$
 $d_{i,\text{tube}} = 0,015 \text{ [m]}$
 $d_{o,\text{tube}} = 0,017 \text{ [m]}$
 $\text{fluid}_{\text{tube,side\$}} = \text{'Water'}$
 $f_{\text{tube}} = 0,07769 \text{ [-]}$
 $k_{\text{ave,shell}} = 0,02412 \text{ [W/m-K]}$
 $k_{i,\text{shell}} = 0,02401 \text{ [W/m-K]}$
 $k_{o,\text{shell}} = 0,02423 \text{ [W/m-K]}$
 $L_b = 0,125 \text{ [m]}$
 $m = 0,14 \text{ [-]}$
 $\mu_{\text{ave,tube}} = 0,0002747 \text{ [Pa-s]}$
 $\mu_{\text{bulk,shell}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{tube}} = 0,0005379 \text{ [Pa-s]}$
 $\mu_{o,\text{tube}} = 0,0000114 \text{ [Pa-s]}$
 $\mu_{\text{wall,shell}} = 0,0003545 \text{ [Pa-s]}$
 $m_{i,\text{tube}} = 0,7605 \text{ [kg/s]}$
 $m_{o,\text{tube}} = 0,3803 \text{ [kg/s]}$
 $m_{\text{tube}} = 0,03435 \text{ [kg/s]}$
 $n_t = 37 \text{ [-]}$
 $P_{i,\text{shell}} = 150 \text{ [kPa]}$
 $P_{o,\text{shell}} = 85 \text{ [kPa]}$
 $P_{\text{shell}} = 0,06409 \text{ [Pa]}$
 $P_{\text{tubes}} = 0,0001599 \text{ [Pa]}$
 $P_{\text{tube,o}} = 0,0004679 \text{ [Pa]}$
 $Re_o = 10000 \text{ [-]}$
 $Re_{\text{tube}} = 273,6 \text{ [-]}$
 $\rho_{\text{ave,tube}} = 485,9 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{tube}} = 971,8 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{tube}} = 0,06239 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{Eff\$}} = \text{'False'}$
 $RUN_{\text{PRE\$}} = \text{'True'}$
 $SP_{\text{heat,ratio,tube}} = 1 \text{ [-]}$
 $s_m = 0,003125 \text{ [m}^2\text{]}$
 $T_{i,\text{tube}} = 324,1 \text{ [K]}$
 $T_{o,\text{tube}} = 348 \text{ [K]}$
 $v_{i,\text{tube}} = 5,447\text{E-}07 \text{ [m}^2\text{/s]}$
 $w_{i,\text{shell}} = 0,002622 \text{ [m/s]}$
 $w_{o,\text{shell}} = 0,1204 \text{ [m/s]}$
 $x = 0,04948 \text{ [-]}$

$b = 1,176 \text{ [-]}$
 $C_{p,\text{ave,shell}} = 1,005 \text{ [kJ/kg-K]}$
 $C_{p,i,\text{shell}} = 1,006 \text{ [kJ/kg-K]}$
 $C_{p,i,\text{tube,calc}} = 4,182 \text{ [kJ/kg-K]}$
 $C_{p,o,\text{shell,kJ/kgK}} = 1,004 \text{ [kJ/kg-K]}$
 $C_{p,o,\text{tube,kJ/kgK}} = 1,907 \text{ [kJ/kg-K]}$
 $d_e = 0,008764 \text{ [m]}$
 $d_{\text{inlet,shell}} = 0,1 \text{ [m]}$
 $d_{i,\text{shell}} = 0,1 \text{ [m]}$
 $d_{o,\text{shell}} = 0,2 \text{ [m]}$
 $\text{fluid}_{\text{shell,side\$}} = \text{'Air'}$
 $f_{\text{shell}} = 0,03678 \text{ [-]}$
 $G_s = 10,99 \text{ [kg/m}^2\text{-s]}$
 $k_{\text{ave,tube}} = 0,3392 \text{ [W/m-K]}$
 $k_{i,\text{tube}} = 0,6562 \text{ [W/m-K]}$
 $k_{o,\text{tube}} = 0,02216 \text{ [W/m-K]}$
 $L_{\text{tube}} = 0,5 \text{ [m]}$
 $\mu_{\text{ave,shell}} = 0,00001903 \text{ [Pa-s]}$
 $\mu_{\text{bulk}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{shell}} = 0,00002038 \text{ [Pa-s]}$
 $\mu_{o,\text{shell}} = 0,00001767 \text{ [Pa-s]}$
 $\mu_{\text{wall}} = 0,0003545 \text{ [Pa-s]}$
 $m_{i,\text{shell}} = 0,00997 \text{ [kg/s]}$
 $m_{o,\text{shell}} = 0,003986 \text{ [kg/s]}$
 $m_{\text{shell}} = 0,00002613 \text{ [kg/s]}$
 $n_b = 4 \text{ [-]}$
 $\text{pitch} = 0,02 \text{ [m]}$
 $P_{i,\text{tube}} = 101,3 \text{ [kPa]}$
 $P_{o,\text{tube}} = 10 \text{ [kPa]}$
 $P_{\text{tube}} = 1,018 \text{ [Pa]}$
 $P_{\text{tube,i}} = 29,15 \text{ [Pa]}$
 $Re_i = 4000 \text{ [-]}$
 $Re_{\text{shell}} = 5450 \text{ [-]}$
 $\rho_{\text{ave,shell}} = 1,161 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{shell}} = 1,269 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{shell}} = 1,054 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{EB\$}} = \text{'False'}$
 $RUN_{\text{HEAT\$}} = \text{'False'}$
 $SP_{\text{heat,ratio,shell}} = 1,4 \text{ [-]}$
 $s_2 = 0,02 \text{ [m]}$
 $T_{i,\text{shell}} = 340 \text{ [K]}$
 $T_{o,\text{shell}} = 281 \text{ [K]}$
 $v_{\text{ave,tube}} = 0,00009163 \text{ [m}^2\text{/s]}$
 $v_{o,\text{tube}} = 0,0001827 \text{ [m}^2\text{/s]}$
 $w_{i,\text{tube}} = 0,2 \text{ [m/s]}$
 $w_{o,\text{tube}} = 0,1 \text{ [m/s]}$
 $z = 1 \text{ [-]}$

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

```

a = 1,176 [-]
b = 1,176 [-]
clearance = 0,005 [m]
Cpi,shell = 1,006 [kJ/kg-K]
Cpo,shell = 1004 [J/kg-K]
Cpo,tube = 1907 [J/kg-K]
cp = 0,86 [-]
dinlet,shell = 0,1 [m]
di,shell = 0,1 [m]
do,shell = 0,2 [m]
fluidshell,side$ = 'Air'
fa = 1,567 [-]
have,tube = 1116 [W/m2-K]
hi,tube = 2231 [W/m2-K]
ho,tube = 0,9299 [W/m2-K]
kave,tube = 0,3392 [W/m-K]
ki,tube = 0,6562 [W/m-K]
ko,tube = 0,02216 [W/m-K]
Ltube = 0,5 [m]
μave,shell = 0,00001903 [Pa-s]
μbulk = 0,00001754 [Pa-s]
μi,shell = 0,00002038 [Pa-s]
μo,shell = 0,00001767 [Pa-s]
μwall = 0,0003545 [Pa-s]
mave,shell = 0,006978 [kg/s]
mi,shell = 0,00997 [kg/s]
mo,shell = 0,003986 [kg/s]
n = 0,492 [-]
v1,o = 6,887 [-]
vi,shell = 893,1 [-]
vo,lam = 6,557 [-]
vo,turb = 0,6271 [-]
nt = 37 [-]
Pri,tube = 3,438 [-]
Pi,shell = 150 [kPa]
Po,shell = 85 [kPa]
Rei = 20002 [-]
Reshell = 8000 [-]
ρave,shell = 1,161 [kg/m3]
ρi,shell = 1,269 [kg/m3]
ρo,shell = 1,054 [kg/m3]
RUNEB$ = 'False'
RUNHEAT$ = 'True'
SPheat,ratio,shell = 1,4 [-]
s1 = 0,02 [m]
Ti,shell = 340 [K]
To,shell = 281 [K]
Ushell = 0,2882 [W/m2-K]
vave,tube = 0,00009163 [m2/s]
vo,tube = 0,0001827 [m2/s]
wi,tube = 0,2 [m/s]
wo,tube = 0,1 [m/s]

Ashell = 0,03142 [m2]
C = 0,386 [-]
Cpave,shell = 1,005 [kJ/kg-K]
Cpi,tube = 4194 [J/kg-K]
Cpo,shell,kJ/kgK = 1,004 [kJ/kg-K]
Cpo,tube,kJ/kgK = 1,907 [kJ/kg-K]
dh,tube = 0,015 [m]
dinlet,tube = 0,015 [m]
di,tube = 0,015 [m]
do,tube = 0,017 [m]
fluidtube,side$ = 'Water'
have,shell = 68,67 [W/m2-K]
hi,shell = 136,5 [W/m2-K]
ho,shell = 0,8321 [W/m2-K]
kave,shell = 0,02412 [W/m-K]
ki,shell = 0,02401 [W/m-K]
ko,shell = 0,02423 [W/m-K]
Lb = 0,125 [m]
m = 0,14 [-]
μave,tube = 0,0002747 [Pa-s]
μbulk,shell = 0,00001754 [Pa-s]
μi,tube = 0,0005379 [Pa-s]
μo,tube = 0,0000114 [Pa-s]
μwall,shell = 0,0003545 [Pa-s]
mave,tube = 0,5704 [kg/s]
mi,tube = 0,7605 [kg/s]
mo,tube = 0,3803 [kg/s]
v1,i = 570,1 [-]
vi,lam = 559,2 [-]
vi,turb = 109,3 [-]
vo,shell = 10,79 [-]
nb = 4 [-]
pitch = 0,02 [m]
Pro,tube = 0,9809 [-]
Pi,tube = 101,3 [kPa]
Po,tube = 10 [kPa]
Reo = 33,79 [-]
Retube = 8000 [-]
ρave,tube = 485,9 [kg/m3]
ρi,tube = 971,8 [kg/m3]
ρo,tube = 0,06239 [kg/m3]
RUNEff$ = 'False'
RUNPRE$ = 'False'
SPheat,ratio,tube = 1 [-]
s2 = 0,02 [m]
Ti,tube = 324,1 [K]
To,tube = 348 [K]
Utube = 0,9671 [W/m2-K]
vi,tube = 5,447E-07 [m2/s]
wi,shell = 0,002622 [m/s]
wo,shell = 0,1204 [m/s]
y = 0,4325 [-]

```

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$A_{\text{shell}} = 0,03142 \text{ [m}^2\text{]}$
 $\text{clearance} = 0,005 \text{ [m]}$
 $cp_{i,\text{tube}} = 4,194 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{shell}} = 1004 \text{ [J/kg-K]}$
 $Cp_{o,\text{tube}} = 1907 \text{ [J/kg-K]}$
 $c_p = 0,86 \text{ [-]}$
 $d_{h,\text{tube}} = 0,015 \text{ [m]}$
 $d_{\text{inlet,tube}} = 0,015 \text{ [m]}$
 $d_{i,\text{tube}} = 0,015 \text{ [m]}$
 $d_{o,\text{tube}} = 0,017 \text{ [m]}$
 $\text{fluid}_{\text{tube,side\$}} = \text{'Water'}$
 $f_{\text{tube}} = 0,07769 \text{ [-]}$
 $k_{\text{ave,shell}} = 0,02412 \text{ [W/m-K]}$
 $k_{i,\text{shell}} = 0,02401 \text{ [W/m-K]}$
 $k_{o,\text{shell}} = 0,02423 \text{ [W/m-K]}$
 $L_b = 0,125 \text{ [m]}$
 $m = 0,14 \text{ [-]}$
 $\mu_{\text{ave,tube}} = 0,0002747 \text{ [Pa-s]}$
 $\mu_{\text{bulk,shell}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{tube}} = 0,0005379 \text{ [Pa-s]}$
 $\mu_{o,\text{tube}} = 0,0000114 \text{ [Pa-s]}$
 $\mu_{\text{wall,shell}} = 0,0003545 \text{ [Pa-s]}$
 $m_{\text{ave,tube}} = 0,5704 \text{ [kg/s]}$
 $m_{i,\text{tube}} = 0,7605 \text{ [kg/s]}$
 $m_{o,\text{tube}} = 0,3803 \text{ [kg/s]}$
 $m_{\text{tube}} = 0,03435 \text{ [kg/s]}$
 $n_t = 37 \text{ [-]}$
 $P_{i,\text{shell}} = 150 \text{ [kPa]}$
 $P_{o,\text{shell}} = 85 \text{ [kPa]}$
 $P_{\text{shell}} = 0,2309 \text{ [Pa]}$
 $P_{\text{tubes}} = 0,0001599 \text{ [Pa]}$
 $P_{\text{tube,o}} = 0,0004679 \text{ [Pa]}$
 $Re_o = 10000 \text{ [-]}$
 $Re_{\text{tube}} = 273,6 \text{ [-]}$
 $\rho_{\text{ave,tube}} = 485,9 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{tube}} = 971,8 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{tube}} = 0,06239 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{Eff\$}} = \text{'False'}$
 $RUN_{\text{PRE\$}} = \text{'True'}$
 $SP_{\text{heat,ratio,tube}} = 1 \text{ [-]}$
 $s_m = 0,003125 \text{ [m}^2\text{]}$
 $T_{i,\text{tube}} = 324,1 \text{ [K]}$
 $T_{o,\text{tube}} = 348 \text{ [K]}$
 $v_{i,\text{tube}} = 5,447\text{E-}07 \text{ [m}^2\text{/s]}$
 $w_{i,\text{shell}} = 0,2189 \text{ [m/s]}$
 $w_{o,\text{shell}} = 0,2285 \text{ [m/s]}$
 $x = 0,04948 \text{ [-]}$

$b = 1,176 \text{ [-]}$
 $cp_{i,\text{shell}} = 1,006 \text{ [kJ/kg-K]}$
 $cp_{i,\text{tube,calc}} = 4,182 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{shell,kJ/kgK}} = 1,004 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{tube,kJ/kgK}} = 1,907 \text{ [kJ/kg-K]}$
 $d_e = 0,008764 \text{ [m]}$
 $d_{\text{inlet,shell}} = 0,1 \text{ [m]}$
 $d_{i,\text{shell}} = 0,1 \text{ [m]}$
 $d_{o,\text{shell}} = 0,2 \text{ [m]}$
 $\text{fluid}_{\text{shell,side\$}} = \text{'Air'}$
 $f_{\text{shell}} = 0,03678 \text{ [-]}$
 $G_s = 10,99 \text{ [kg/m}^2\text{-s]}$
 $k_{\text{ave,tube}} = 0,3392 \text{ [W/m-K]}$
 $k_{i,\text{tube}} = 0,6562 \text{ [W/m-K]}$
 $k_{o,\text{tube}} = 0,02216 \text{ [W/m-K]}$
 $L_{\text{tube}} = 0,5 \text{ [m]}$
 $\mu_{\text{ave,shell}} = 0,00001903 \text{ [Pa-s]}$
 $\mu_{\text{bulk}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{shell}} = 0,00002038 \text{ [Pa-s]}$
 $\mu_{o,\text{shell}} = 0,00001767 \text{ [Pa-s]}$
 $\mu_{\text{wall}} = 0,0003545 \text{ [Pa-s]}$
 $m_{\text{ave,shell}} = 0,006978 \text{ [kg/s]}$
 $m_{i,\text{shell}} = 0,00997 \text{ [kg/s]}$
 $m_{o,\text{shell}} = 0,003986 \text{ [kg/s]}$
 $m_{\text{shell}} = 0,002182 \text{ [kg/s]}$
 $n_b = 4 \text{ [-]}$
 $\text{pitch} = 0,02 \text{ [m]}$
 $P_{i,\text{tube}} = 101,3 \text{ [kPa]}$
 $P_{o,\text{tube}} = 10 \text{ [kPa]}$
 $P_{\text{tube}} = 1,018 \text{ [Pa]}$
 $P_{\text{tube,i}} = 29,15 \text{ [Pa]}$
 $Re_i = 4000 \text{ [-]}$
 $Re_{\text{shell}} = 5450 \text{ [-]}$
 $\rho_{\text{ave,shell}} = 1,161 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{shell}} = 1,269 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{shell}} = 1,054 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{EB\$}} = \text{'False'}$
 $RUN_{\text{HEAT\$}} = \text{'False'}$
 $SP_{\text{heat,ratio,shell}} = 1,4 \text{ [-]}$
 $s_2 = 0,02 \text{ [m]}$
 $T_{i,\text{shell}} = 340 \text{ [K]}$
 $T_{o,\text{shell}} = 281 \text{ [K]}$
 $v_{\text{ave,tube}} = 0,00009163 \text{ [m}^2\text{/s]}$
 $v_{o,\text{tube}} = 0,0001827 \text{ [m}^2\text{/s]}$
 $w_{i,\text{tube}} = 0,2 \text{ [m/s]}$
 $w_{o,\text{tube}} = 0,1 \text{ [m/s]}$
 $z = 1 \text{ [-]}$

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$A_{\text{shell}} = 0,03142 \text{ [m}^2\text{]}$
 $\text{clearance} = 0,005 \text{ [m]}$
 $cp_{i,\text{tube}} = 4,194 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{shell}} = 1004 \text{ [J/kg-K]}$
 $Cp_{o,\text{tube}} = 1907 \text{ [J/kg-K]}$
 $c_p = 0,86 \text{ [-]}$
 $d_{h,\text{tube}} = 0,015 \text{ [m]}$
 $d_{\text{inlet,tube}} = 0,015 \text{ [m]}$
 $d_{i,\text{tube}} = 0,015 \text{ [m]}$
 $d_{o,\text{tube}} = 0,017 \text{ [m]}$
 $\text{fluid}_{\text{tube,side\$}} = \text{'Water'}$
 $f_{\text{tube}} = 491642 \text{ [-]}$
 $k_{\text{ave,shell}} = 0,02412 \text{ [W/m-K]}$
 $k_{i,\text{shell}} = 0,02401 \text{ [W/m-K]}$
 $k_{o,\text{shell}} = 0,02423 \text{ [W/m-K]}$
 $L_b = 0,125 \text{ [m]}$
 $m = 0,14 \text{ [-]}$
 $\mu_{\text{ave,tube}} = 0,0002747 \text{ [Pa-s]}$
 $\mu_{\text{bulk,shell}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{tube}} = 0,0005379 \text{ [Pa-s]}$
 $\mu_{o,\text{tube}} = 0,0000114 \text{ [Pa-s]}$
 $\mu_{\text{wall,shell}} = 0,0003545 \text{ [Pa-s]}$
 $m_{\text{ave,tube}} = 0,5704 \text{ [kg/s]}$
 $m_{i,\text{tube}} = 0,7605 \text{ [kg/s]}$
 $m_{o,\text{tube}} = 0,3803 \text{ [kg/s]}$
 $m_{\text{tube}} = 0 \text{ [kg/s]}$
 $n_t = 37 \text{ [-]}$
 $P_{i,\text{shell}} = 150 \text{ [kPa]}$
 $P_{o,\text{shell}} = 85 \text{ [kPa]}$
 $P_{\text{shell}} = 0 \text{ [Pa]}$
 $P_{\text{tubes}} = 0 \text{ [Pa]}$
 $P_{\text{tube,o}} = 0 \text{ [Pa]}$
 $Re_o = 10000 \text{ [-]}$
 $Re_{\text{tube}} = -1,707E-25 \text{ [-]}$
 $\rho_{\text{ave,tube}} = 485,9 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{tube}} = 971,8 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{tube}} = 0,06239 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{Eff\$}} = \text{'False'}$
 $RUN_{\text{PRE\$}} = \text{'True'}$
 $SP_{\text{heat,ratio,tube}} = 1 \text{ [-]}$
 $s_m = 0,003125 \text{ [m}^2\text{]}$
 $T_{i,\text{tube}} = 324,1 \text{ [K]}$
 $T_{o,\text{tube}} = 348 \text{ [K]}$
 $w_{\text{ave,shell}} = 2,758E-17 \text{ [m/s]}$
 $w_{i,\text{shell}} = 2,699E-17 \text{ [m/s]}$
 $w_{o,\text{shell}} = 2,818E-17 \text{ [m/s]}$
 $x = 0,04948 \text{ [-]}$

$b = 1,176 \text{ [-]}$
 $cp_{i,\text{shell}} = 1,006 \text{ [kJ/kg-K]}$
 $cp_{i,\text{tube,calc}} = 4,182 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{shell,kJ/kgK}} = 1,004 \text{ [kJ/kg-K]}$
 $Cp_{o,\text{tube,kJ/kgK}} = 1,907 \text{ [kJ/kg-K]}$
 $d_e = 0,008764 \text{ [m]}$
 $d_{\text{inlet,shell}} = 0,1 \text{ [m]}$
 $d_{i,\text{shell}} = 0,1 \text{ [m]}$
 $d_{o,\text{shell}} = 0,2 \text{ [m]}$
 $\text{fluid}_{\text{shell,side\$}} = \text{'Air'}$
 $f_{\text{shell}} = 349 \text{ [-]}$
 $G_s = 1,355E-15 \text{ [kg/m}^2\text{-s]}$
 $k_{\text{ave,tube}} = 0,3392 \text{ [W/m-K]}$
 $k_{i,\text{tube}} = 0,6562 \text{ [W/m-K]}$
 $k_{o,\text{tube}} = 0,02216 \text{ [W/m-K]}$
 $L_{\text{tube}} = 0,5 \text{ [m]}$
 $\mu_{\text{ave,shell}} = 0,00001903 \text{ [Pa-s]}$
 $\mu_{\text{bulk}} = 0,00001754 \text{ [Pa-s]}$
 $\mu_{i,\text{shell}} = 0,00002038 \text{ [Pa-s]}$
 $\mu_{o,\text{shell}} = 0,00001767 \text{ [Pa-s]}$
 $\mu_{\text{wall}} = 0,0003545 \text{ [Pa-s]}$
 $m_{\text{ave,shell}} = 0,006978 \text{ [kg/s]}$
 $m_{i,\text{shell}} = 0,00997 \text{ [kg/s]}$
 $m_{o,\text{shell}} = 0,003986 \text{ [kg/s]}$
 $m_{\text{shell}} = 2,690E-19 \text{ [kg/s]}$
 $n_b = 4 \text{ [-]}$
 $\text{pitch} = 0,02 \text{ [m]}$
 $P_{i,\text{tube}} = 101,3 \text{ [kPa]}$
 $P_{o,\text{tube}} = 10 \text{ [kPa]}$
 $P_{\text{tube}} = 0 \text{ [Pa]}$
 $P_{\text{tube,i}} = 0 \text{ [Pa]}$
 $Re_i = 4000 \text{ [-]}$
 $Re_{\text{shell}} = 6,721E-13 \text{ [-]}$
 $\rho_{\text{ave,shell}} = 1,161 \text{ [kg/m}^3\text{]}$
 $\rho_{i,\text{shell}} = 1,269 \text{ [kg/m}^3\text{]}$
 $\rho_{o,\text{shell}} = 1,054 \text{ [kg/m}^3\text{]}$
 $RUN_{\text{EB\$}} = \text{'False'}$
 $RUN_{\text{HEAT\$}} = \text{'False'}$
 $SP_{\text{heat,ratio,shell}} = 1,4 \text{ [-]}$
 $s_2 = 0,02 \text{ [m]}$
 $T_{i,\text{shell}} = 340 \text{ [K]}$
 $T_{o,\text{shell}} = 281 \text{ [K]}$
 $v_{o,\text{tube}} = 0,0001827 \text{ [m}^2\text{/s]}$
 $w_{\text{ave,tube}} = 0 \text{ [m/s]}$
 $w_{i,\text{tube}} = 0 \text{ [m/s]}$
 $w_{o,\text{tube}} = -1,109E-28 \text{ [m/s]}$
 $z = 1 \text{ [-]}$

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

```

a = 1,25 [-]
b = 1,5 [-]
Cpi,shell = 4178 [J/kg-K]
Cpo,tube = 4180 [J/kg-K]
Cc = 6269 [W/K]
Cmax = 6269 [W/K]
δTtube = 8,208 [K]
di,tube = 0,0254 [m]
do,tube = 0,0257 [m]
eff = 98,52 [%]
e2 = -1,092 [%]
fluidtube,side$ = 'Water'
have,shell = 3,156E+09 [W/m2-K]
hi,shell = 6,312E+09 [W/m2-K]
ho,shell = 4547 [W/m2-K]
kave,shell = 0,628 [W/m-K]
ki,shell = 0,624 [W/m-K]
ko,shell = 0,632 [W/m-K]
l = 0,57 [-]
Ltube,check = 21,92 [m]
μi,shell = 698 [Pa-s]
μo,tube = 0,0007758 [Pa-s]
mshell = 1,5 [kg/s]
n = 0,492 [-]
v1,o = 1224 [-]
vi,shell = 2,453E+09 [-]
vo,lam = 1207 [-]
vo,turb = 200,7 [-]
nb = 22 [-]
Ntu = 0,2126 [-]
Pro,tube = 5,281 [-]
Pi,tube = 100,3 [kPa]
Po,tube = 103,7 [kPa]
Qshell = 9276 [W]
Rei = 76696 [-]
ρi,shell = 994,8 [kg/m3]
RUNEB$ = 'False'
RUNHEAT$ = 'True'
Rc = 0,9998 [-]
Ti,tube = 303,6 [K]
To,tube = 304,4 [K]
Utube = 1221 [W/m2-K]
vi,tube = 7,940E-07 [m2/s]
wave,tube = 0,665 [m/s]
wo,tube = 0,443 [m/s]

Atube = 1,915 [m2]
C = 0,386 [-]
Cpi,tube = 4179 [J/kg-K]
Cpo,tube,kJkgK = 4,18 [kJ/kg-K]
Ch = 6268 [W/K]
Cmin = 6268 [W/K]
di,shell = 0,1544 [m]
do,shell = 0,1564 [m]
e = 2,718 [-]
e1 = 17,43 [%]
fluidshell,side$ = 'Water'
fa = 1,444 [-]
have,tube = 2,081E+09 [W/m2-K]
hi,tube = 4,162E+09 [W/m2-K]
ho,tube = 2964 [W/m2-K]
kave,tube = 0,615 [W/m-K]
ki,tube = 0,616 [W/m-K]
ko,tube = 0,614 [W/m-K]
Ltube = 2 [m]
μave,tube = 393 [Pa-s]
μi,tube = 786 [Pa-s]
m1 = 1 [-]
mtube = 1,5 [kg/s]
v1,i = 1,698E+09 [-]
vi,lam = 1,698E+09 [-]
vi,turb = 377,2 [-]
vo,shell = 1768 [-]
n1 = 1 [-]
nt = 12 [-]
Pri,tube = 5,332E+06 [-]
Pi,shell = 100,3 [kPa]
Po,shell = 115,5 [kPa]
Qmax = 59228 [W]
Qtube = 5579 [W]
Reo = 39480 [-]
ρi,tube = 996,1 [kg/m3]
RUNEff$ = 'True'
RUNPRE$ = 'False'
Ti,shell = 313 [K]
To,shell = 311,5 [K]
Ushell = 297,5 [W/m2-K]
vave,tube = 7,868E-07 [m2/s]
vo,tube = 7,795E-07 [m2/s]
wi,tube = 0,887 [m/s]
y = 0,5811 [-]

```

No unit problems were detected.

APPENDIX J: TWISTED TUBE SHELL-AND-TUBE HEAT EXCHANGER EES CALCULATIONS

Main

$$\text{RUN}_{\text{Eff\$}} = \text{'True'}$$

$$\text{RUN}_{\text{HEAT\$}} = \text{'True'}$$

$$\text{RUN}_{\text{PRE\$}} = \text{'True'}$$

$$\text{RUN}_{\text{EB\$}} = \text{'True'}$$

$$\text{fluid}_{\text{tube;side\$}} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Steam'}$$

$$T_{i;\text{tube}} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o;\text{tube}} = (93 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;\text{tube}} = 150 \text{ [kPa]}$$

$$P_{o;\text{tube}} = 135 \text{ [kPa]}$$

$$P = 0,16 \text{ [m]}$$

$$d_{i;\text{tube}} = 0,01575 \text{ [m]}$$

$$d_{o;\text{tube}} = 0,01905 \text{ [m]}$$

$$d_{\text{outlet};\text{tube}} = 0,2762 \text{ [m]}$$

$$d_{\text{inlet};\text{tube}} = 0,1168 \text{ [m]}$$

$$a = 0,007 \text{ [m]}$$

$$b = 0,035 \text{ [m]}$$

$$\text{Aspect}_{\text{ratio}} = \frac{a}{b}$$

$$T_{i;\text{shell}} = (123 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;\text{shell}} = 306 \text{ [kPa]}$$

$$P_{o;\text{shell}} = 305 \text{ [kPa]}$$

$$v_{i;\text{shell}} = 15 \text{ [m/s]}$$

$$v_{o;\text{shell}} = 13 \text{ [m/s]}$$

$$v_{i;\text{tube}} = 55 \text{ [m/s]}$$

$$v_{o;\text{tube}} = 20 \text{ [m/s]}$$

$$H = 4,26 \text{ [m]}$$

$$d_{o;\text{shell}} = 0,6 \text{ [m]}$$

$$d_{\text{outlet};\text{shell}} = 0,0907 \text{ [m]}$$

$$d_{\text{inlet};\text{shell}} = 0,1707 \text{ [m]}$$

$$\text{shell}_{\text{thk}} = 0,006 \text{ [m]}$$

$$\text{tube}_{\text{thk}} = \frac{d_{\text{o,tube}} - d_{\text{i,tube}}}{2}$$

$$d_{\text{i,shell}} = d_{\text{o,shell}} - 2 \cdot \text{shell}_{\text{thk}}$$

$$m_{\text{shell}} = \rho_{\text{i,shell}} \cdot V_{\text{i,shell}} \cdot \pi \cdot \left[\frac{d_{\text{inlet,shell}}}{2} \right]^2$$

$$m_{\text{tube}} = \rho_{\text{o,tube}} \cdot V_{\text{o,tube}} \cdot \pi \cdot \left[\frac{d_{\text{inlet,shell}}}{2} \right]^2$$

$$n_t = 442 \text{ [-]}$$

$$n_p = 1 \text{ [%]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$d = 1,468 \text{ [m]}$$

$$C_{\text{if}} = 2, \times 10^6 \text{ [R]}$$

$$c_i = 10000 \text{ [R]}$$

$$i = 11 \text{ [%]}$$

$$\text{opp}_{\text{time}} = 20 \text{ [yr]}$$

$$C_s = 1,5 \times 10^6 \text{ [R]}$$

$$n = 1 \text{ [-]}$$

$$n_{\text{pump}} = 58 \text{ [%]}$$

$$c_{\text{pump}} = 0,00288 \text{ [R/Wh]}$$

$$tt = 7300 \text{ [h/yr]}$$

$$j = 1,88 \text{ [R/W]}$$

$$q = 780 \text{ [W/h]}$$

$$c = 4,7 \times 10^8 \text{ [-]}$$

$$e = 2,718 \text{ [-]}$$

$$R_w = \frac{1}{\text{tube}_{\text{thk}} \cdot k_{\text{i,tube}} \cdot \frac{1}{A_{\text{tube}}}}$$

$$R_f = 0,0002 \text{ [m}^2\text{-K/W]}$$

$$\rho_{\text{i,tube}} = \rho(\text{fluid}_{\text{tube,side\$}} ; T = T_{\text{i,tube}} ; P = P_{\text{i,tube}})$$

$$\rho_{\text{o,tube}} = \rho(\text{fluid}_{\text{tube,side\$}} ; T = T_{\text{o,tube}} ; P = P_{\text{o,tube}})$$

$$\mu_{i,tube} = \mathbf{Visc}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$\mu_{o,tube} = \mathbf{Visc}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$k_{i,tube} = \mathbf{k}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$k_{o,tube} = \mathbf{k}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$cp_{i,tube;kJ/kgK} = \mathbf{Cp}(\text{fluid}_{tube,side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$cp_{o,tube;kJ/kgK} = \mathbf{Cp}(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\rho_{i,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\mu_{i,shell} = \mathbf{Visc}(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$\mu_{o,shell} = \mathbf{Visc}(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$k_{i,shell} = \mathbf{k}(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$k_{o,shell} = \mathbf{k}(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$cp_{i,shell;kJ/kgK} = \mathbf{Cp}(\text{fluid}_{shell,side\$} ; T = T_{i,shell} ; P = P_{i,shell})$$

$$cp_{o,shell;kJ/kgK} = \mathbf{Cp}(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$Cp_{i,tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot cp_{i,tube;kJ/kgK}$$

$$Cp_{o,tube} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot cp_{o,tube;kJ/kgK}$$

$$Cp_{i,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot cp_{i,shell;kJ/kgK}$$

$$Cp_{o,shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot cp_{o,shell;kJ/kgK} \quad \text{\$insertTab Twi} \quad \text{insertTab Twi} \quad \text{insertTab Twi} \quad \text{nsertTab Twis}$$

Twisted Tube Calculation

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Heat Transfer Coefficient

$$d_{h,tube} = 4 \cdot \left[\frac{\pi \cdot a \cdot b}{2 \cdot \pi \cdot a + 4 \cdot (b - a)} \right]$$

$$\text{Perimeter}_{tube} = 2 \cdot \pi \cdot a + 4 \cdot (b - a)$$

$$Re_{i,tube} = \frac{\rho_{i,tube} \cdot v_{i,tube} \cdot d_{i,tube}}{\mu_{i,tube}}$$

This section is not calculated for validation 1

$$Re_{o,tube} = \frac{\rho_{o,tube} \cdot v_{o,tube} \cdot d_{i,tube}}{\mu_{o,tube}}$$

End of section

$$Pr_{i,tube} = \frac{\mu_{i,tube} \cdot Cp_{i,tube}}{k_{i,tube}}$$

$$Pr_{o,tube} = \frac{\mu_{o,tube} \cdot Cp_{o,tube}}{k_{o,tube}}$$

$$f_{i,tube} = 9,461 \cdot \frac{1}{Re_{i,tube}^{0,521}} \cdot \left[\frac{P}{d_{h,tube}} \right]^{0,082} \cdot \left[\frac{b}{a} \right]^{0,33}$$

$$f_{o,tube} = 9,461 \cdot \frac{1}{Re_{o,tube}^{0,521}} \cdot \left[\frac{P}{d_{h,tube}} \right]^{0,082} \cdot \left[\frac{b}{a} \right]^{0,33}$$

$$h_{i,tube} = 0,034 \cdot \left[\frac{Re_{i,tube}^{0,5}}{v_{i,tube}} \right]^{0,8} \cdot Pr_{i,tube}^{0,33} \cdot \left[\frac{b}{a} \right]^{0,59} \cdot \left[\frac{d_{h,tube}}{P} \right]^{0,165} \cdot \frac{k_{i,tube}}{d_{h,tube}} \cdot v_{i,tube}^{0,8}$$

$$h_{o,tube} = 0,034 \cdot \left[\frac{Re_{o,tube}^{0,5}}{v_{o,tube}} \right]^{0,8} \cdot Pr_{o,tube}^{0,33} \cdot \left[\frac{b}{a} \right]^{0,59} \cdot \left[\frac{d_{h,tube}}{P} \right]^{0,165} \cdot \frac{k_{o,tube}}{d_{h,tube}} \cdot v_{i,tube}^{0,8}$$

$$h_{ave,tube} = \frac{h_{i,tube} + h_{o,tube}}{2}$$

$$Pr_{i,shell} = \frac{\mu_{i,shell} \cdot Cp_{i,shell}}{k_{i,shell}}$$

$$Pr_{o,shell} = \frac{\mu_{o,shell} \cdot Cp_{o,shell}}{k_{o,shell}}$$

$$Fr_{mi} = \frac{v_{i,shell}}{(g \cdot d)^{0,5}}$$

$$Fr_{mo} = \frac{v_{o,shell}}{(g \cdot d)^{0,5}}$$

$$d_{h,shell} = d_{i,shell}$$

$$A_{ci} = \frac{\pi \cdot d_{h,shell}^2 \cdot n_t}{4 \cdot n_p} \cdot 1 \quad [\%]$$

$$A_{co} = \frac{\pi \cdot d_{h,shell}^2 \cdot n_t}{4 \cdot n_p} \cdot 1 \quad [\%]$$

$$G_{s,i} = \rho_{i,shell} \cdot v_{i,shell}$$

$$G_{s,o} = \rho_{o,shell} \cdot v_{o,shell}$$

$$Re_i = \frac{m_{shell} \cdot d_{h,shell}}{A_{ci} \cdot \mu_{i,shell}}$$

$$Re_o = \frac{m_{shell} \cdot d_{h,shell}}{A_{co} \cdot \mu_{o,shell}}$$

$$f_i = 4,572 \cdot \frac{1}{Re_i^{0,521}} \cdot \left[\frac{d_{h,shell}}{P} \right]^{0,082} \cdot \left[\frac{b}{a} \right]^{0,33}$$

$$f_o = 4,572 \cdot \frac{1}{Re_o^{0,521}} \cdot \left[\frac{d_{h,shell}}{P} \right]^{0,082} \cdot \left[\frac{b}{a} \right]^{0,33}$$

$$\text{deltha}_{P,i,shell} = \frac{f_i \cdot \rho_{i,shell} \cdot v_{i,shell}^2 \cdot H}{2 \cdot d_{h,shell}}$$

$$\text{deltha}_{P,o,shell} = \frac{f_o \cdot \rho_{o,shell} \cdot v_{o,shell}^2 \cdot H}{2 \cdot d_{h,shell}}$$

$$h_{i,shell} = \frac{0,2379 \cdot \left[\frac{d_{h,tube} \cdot G_{s,i}}{\mu_{i,shell}} \right]^{0,8} \cdot Pr_{i,shell}^{0,33} \cdot \frac{1}{Fr_{mi}^{0,4347}} \cdot \left[1 + \frac{1}{(3,6 \cdot Fr_{mi})^{0,357}} \right] \cdot k_{i,shell}}{d_{h,tube}}$$

$$h_{o,shell} = \frac{0,2379 \cdot \left[\frac{d_{h,tube} \cdot G_{s,o}}{\mu_{o,shell}} \right]^{0,8} \cdot Pr_{o,shell}^{0,33} \cdot \frac{1}{Fr_{mo}^{0,4347}} \cdot \left[1 + \frac{1}{(3,6 \cdot Fr_{mo})^{0,357}} \right] \cdot k_{o,shell}}{d_{h,tube}}$$

$$h_{ave,shell} = \frac{h_{i,shell} + h_{o,shell}}{2}$$

$$U = \frac{1}{\frac{1}{h_{i,tube}} + \frac{1}{h_{o,tube}} + R_w + R_f}$$

Pressure Drop

This section is not calculated for validation 3

$$Re_{tube} = \frac{\rho_{o,tube} \cdot v_{o,tube} \cdot d_{h,tube}}{\mu_{o,tube}}$$

End of section

$$f_t = 0,143 \cdot Re_{tube}^{0,169}$$

This section is not calculated for validation 2

$$Re_{shell} = \frac{\rho_{o,shell} \cdot v_{o,shell} \cdot d_{h,shell}}{\mu_{o,shell}}$$

End of section

$$f_s = 0,143 \cdot \text{Re}_{\text{shell}}^{0,169}$$

$$P_{\text{tube}} = \frac{f_t \cdot H \cdot \rho_{o;\text{tube}} \cdot v_{o;\text{tube}}^2}{2 \cdot d_{h;\text{tube}}}$$

$$P_{\text{shell}} = \frac{f_s \cdot H \cdot \rho_{o;\text{shell}} \cdot v_{o;\text{shell}}^2}{2 \cdot d_{h;\text{shell}}}$$

Effectiveness

$$C_h = m_{\text{shell}} \cdot C_{p_{i;\text{shell}}}$$

$$C_c = m_{\text{tube}} \cdot C_{p_{o;\text{tube}}}$$

$$C_{\min} = \text{Min}(C_h; C_c)$$

$$C_{\max} = \text{Max}(C_h; C_c)$$

$$R_c = \frac{C_{\min}}{C_{\max}}$$

$$A_{\text{tube}} = \pi \cdot H \cdot d_{h;\text{tube}}$$

$$N_{\text{tu}} = \frac{U \cdot A_{\text{tube}}}{C_{\min}}$$

$$m = \rho_{o;\text{shell}} \cdot A_{ci} \cdot v_{i;\text{shell}}$$

$$Q_{\max} = |C_{\min} \cdot (T_{i;\text{shell}} - T_{i;\text{tube}})|$$

$$Q_{\text{tube}} = |C_c \cdot (T_{o;\text{tube}} - T_{i;\text{tube}})|$$

$$Q_{\text{shell}} = |C_h \cdot (T_{i;\text{shell}} - T_{o;\text{shell}})|$$

$$\text{eff} = \frac{\tanh\left[\frac{N_{\text{tu}}}{2} \cdot \left[1 + \frac{m}{1 \text{ [kg/s]}} \cdot R_c^n\right]^{\left[\frac{1}{n}\right]}\right]}{\left[\frac{N_{\text{tu}}}{2} \cdot \left[1 + \frac{m}{1 \text{ [kg/s]}} \cdot R_c^n\right]^{\left[\frac{1}{n}\right]}\right]} \cdot 100 \text{ [%]}$$

$$\delta_{T;\text{tube}} = \frac{T_{o;\text{shell}} - T_{o;\text{tube}} - (T_{i;\text{shell}} - T_{i;\text{tube}})}{\ln\left[\frac{T_{o;\text{shell}} - T_{o;\text{tube}}}{T_{i;\text{shell}} - T_{i;\text{tube}}}\right]}$$

$$L_{\text{tube;check}} = \frac{Q_{\text{tube}}}{U \cdot \text{Perimeter}_{\text{tube}} \cdot \delta_{T;\text{tube}}}$$

Energy Balance

$$\text{enthalpy}_{i;\text{shell}} = h(\text{Steam}; T = T_{i;\text{shell}}; P = P_{i;\text{shell}})$$

$$\text{enthalpy}_{o;\text{shell}} = h(\text{Steam}; T = T_{o;\text{shell}}; P = P_{o;\text{shell}})$$

$$\text{enthalpy}_{i;\text{tube}} = h(\text{Water}; T = T_{i;\text{tube}}; P = P_{i;\text{tube}})$$

$$\text{enthalpy}_{o;\text{tube}} = h(\text{Water}; T = T_{o;\text{tube}}; P = P_{o;\text{tube}})$$

$$Q_{\text{in}} = Q_{\text{out}}$$

$$Q_{\text{in}} = |m_{\text{shell}} \cdot (\text{enthalpy}_{i;\text{shell}} - \text{enthalpy}_{o;\text{shell}})|$$

$$Q_{\text{out}} = |m_{\text{tube}} \cdot (\text{enthalpy}_{o;\text{tube}} - \text{enthalpy}_{i;\text{tube}})|$$

Twisted Tube Inputs

Twisted Tube Petrochemical Inputs

$$\text{RUN}_{\text{Eff\$}} = \text{'True'}$$

$$\text{RUN}_{\text{HEAT\$}} = \text{'True'}$$

$$\text{RUN}_{\text{PRE\$}} = \text{'True'}$$

$$\text{RUN}_{\text{EB\$}} = \text{'True'}$$

$$\text{fluid}_{\text{tube;side\$}} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Steam'}$$

Variables

$$T_{i;\text{tube}} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o;\text{tube}} = (93 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;\text{tube}} = 150 \text{ [kPa]}$$

$$P_{o;\text{tube}} = 135 \text{ [kPa]}$$

$$P = 0,16 \text{ [m]}$$

$$d_{i;\text{tube}} = 0,01575 \text{ [m]}$$

$$d_{o;\text{tube}} = 0,01905 \text{ [m]}$$

$$d_{\text{outlet;tube}} = 0,2762 \text{ [m]} \quad 10 \text{ \#150 Nozzle}$$

$$d_{\text{inlet;tube}} = 0,1168 \text{ [m]} \quad 4 \text{ \#150 Nozzle}$$

$$a = 0,007 \text{ [m]}$$

$$b = 0,035 \text{ [m]}$$

$$\text{Aspect}_{\text{ratio}} = \frac{a}{b}$$

$$T_{i;\text{shell}} = (123 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i;\text{shell}} = 306 \text{ [kPa]}$$

$$P_{o,shell} = 305 \text{ [kPa]}$$

$$v_{i,shell} = 15 \text{ [m/s]}$$

$$v_{o,shell} = 13 \text{ [m/s]}$$

$$v_{i,tube} = 55 \text{ [m/s]}$$

$$v_{o,tube} = 20 \text{ [m/s]}$$

$$H = 4,26 \text{ [m]}$$

$$d_{o,shell} = 0,6 \text{ [m]}$$

$$d_{outlet,shell} = 0,0907 \text{ [m]} \quad 3 \text{ \#150 Nozzle}$$

$$d_{inlet,shell} = 0,1707 \text{ [m]} \quad 6 \text{ \#150 Nozzle}$$

$$shell_{thk} = 0,006 \text{ [m]}$$

$$tube_{thk} = \frac{d_{o,tube} - d_{i,tube}}{2}$$

$$d_{i,shell} = d_{o,shell} - 2 \cdot shell_{thk}$$

$$m_{shell} = \rho_{i,shell} \cdot v_{i,shell} \cdot \pi \cdot \left[\frac{d_{inlet,shell}}{2} \right]^2$$

$$m_{tube} = \rho_{o,tube} \cdot v_{o,tube} \cdot \pi \cdot \left[\frac{d_{inlet,shell}}{2} \right]^2$$

$$n_t = 442 \text{ [-]}$$

$$n_p = 1 \text{ [%]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$d = 1,468 \text{ [m]}$$

$$C_{if} = 2, \times 10^6 \text{ [R]}$$

$$c_i = 10000 \text{ [R]}$$

$$i = 11 \text{ [%]}$$

$$opp_{time} = 20 \text{ [yr]}$$

$$C_s = 1,5 \times 10^6 \text{ [R]}$$

$$n = 1 \text{ [-]}$$

$$n_{pump} = 58 \text{ [%]}$$

$$c_{pump} = 0,00288 \text{ [R/Wh]}$$

$$tt = 7300 \text{ [h/yr]}$$

$$j = 1,88 \text{ [R/W]}$$

$$q = 780 \text{ [W/h]}$$

$$c = 4,7 \times 10^8 \text{ [-]}$$

$$e = 2,718 \text{ [-]}$$

$$R_w = \frac{1}{\frac{\text{tube}_{\text{thk}}}{k_{i,\text{tube}} \cdot \frac{1}{A_{\text{tube}}}}}$$

$$R_f = 0,0002 \text{ [m}^2\text{-K/W]}$$

$$\rho_{i,\text{tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{i,\text{tube}} ; P = P_{i,\text{tube}})$$

$$\rho_{o,\text{tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$\mu_{i,\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{i,\text{tube}} ; P = P_{i,\text{tube}})$$

$$\mu_{o,\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$k_{i,\text{tube}} = k(\text{fluid}_{\text{tube;side\$}} ; T = T_{i,\text{tube}} ; P = P_{i,\text{tube}})$$

$$k_{o,\text{tube}} = k(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$cp_{i,\text{tube};\text{kJ/kgK}} = \text{Cp}(\text{fluid}_{\text{tube;side\$}} ; T = T_{i,\text{tube}} ; P = P_{i,\text{tube}})$$

$$cp_{o,\text{tube};\text{kJ/kgK}} = \text{Cp}(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$\rho_{i,\text{shell}} = \rho(\text{fluid}_{\text{shell;side\$}} ; T = T_{i,\text{shell}} ; P = P_{i,\text{shell}})$$

$$\rho_{o,\text{shell}} = \rho(\text{fluid}_{\text{shell;side\$}} ; T = T_{o,\text{shell}} ; P = P_{o,\text{shell}})$$

$$\mu_{i,\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{i,\text{shell}} ; P = P_{i,\text{shell}})$$

$$\mu_{o,\text{shell}} = \text{Visc}(\text{fluid}_{\text{shell;side\$}} ; T = T_{o,\text{shell}} ; P = P_{o,\text{shell}})$$

$$k_{i,\text{shell}} = k(\text{fluid}_{\text{shell;side\$}} ; T = T_{i,\text{shell}} ; P = P_{i,\text{shell}})$$

$$k_{o,\text{shell}} = k(\text{fluid}_{\text{shell;side\$}} ; T = T_{o,\text{shell}} ; P = P_{o,\text{shell}})$$

$$cp_{i,\text{shell};\text{kJ/kgK}} = \text{Cp}(\text{fluid}_{\text{shell;side\$}} ; T = T_{i,\text{shell}} ; P = P_{i,\text{shell}})$$

$$cp_{o,\text{shell};\text{kJ/kgK}} = \text{Cp}(\text{fluid}_{\text{shell;side\$}} ; T = T_{o,\text{shell}} ; P = P_{o,\text{shell}})$$

Convert Units

$$Cp_{i,\text{tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot cp_{i,\text{tube};\text{kJ/kgK}}$$

$$Cp_{o,\text{tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot cp_{o,\text{tube};\text{kJ/kgK}}$$

$$Cp_{i,\text{shell}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot cp_{i,\text{shell};\text{kJ/kgK}}$$

$$Cp_{o;shell} = \left| 1000 \cdot \frac{J/kg-K}{kJ/kg-K} \right| \cdot cp_{o;shell;kJ/kgK}$$

IT V1

Twisted Tube Validation 1

Tube Side Heat Transfer Coefficient vs. Tube Side Reynolds Number

$$RUN_{Eff\$} = 'False'$$

$$RUN_{HEAT\$} = 'True'$$

$$RUN_{PRE\$} = 'False'$$

$$RUN_{EB\$} = 'False'$$

$$fluid_{tube;side\$} = 'Water'$$

$$fluid_{shell;side\$} = 'Steam'$$

Variables

$$P = 0,192 \text{ [m]}$$

$$pitch = 0,02778 \text{ [m]}$$

$$d_{o,tube} = 0,02049 \text{ [m]}$$

$$tube_{thk} = 0,001734 \text{ [m]}$$

$$d_{i,tube} = d_{o,tube} - 2 \cdot tube_{thk}$$

$$a = 0,0198 \text{ [m]}$$

$$b = 9,4 \text{ [m]}$$

$$L_{tube} = 9,926 \text{ [m]}$$

$$H = 9,926 \text{ [m]}$$

$$A_{tube} = 10170,8 \text{ [m}^2\text{]}$$

$$R_w = \frac{1}{\frac{tube_{thk}}{k_{i,tube} \cdot \frac{1}{A_{tube}}}}$$

$$R_f = 0,0002 \text{ [m}^2\text{-K/W]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$d = d_{o,shell}$$

$$n_t = 1591 \text{ [-]}$$

$$d_{o,shell} = 0,203 \text{ [m]}$$

$$d_{i,shell} = 0,12 \text{ [m]}$$

$$n_p = 1 \text{ [%]}$$

$$T_{i,tube} = (30 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,tube} = (53,1 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i,tube} = 115 \text{ [kPa]}$$

$$P_{o,tube} = 85 \text{ [kPa]}$$

$$\rho_{i,tube} = 1160 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,tube} = \rho(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$k_{i,tube} = 0,21 \text{ [W/m-K]}$$

$$k_{o,tube} = k(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$\mu_{i,tube} = \text{Visc}(\text{fluid}_{tube;side\$} ; T = T_{i,tube} ; P = P_{i,tube})$$

$$\mu_{o,tube} = \text{Visc}(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$cp_{i,tube} = 1920 \text{ [J/kg-K]}$$

$$Cp_{o,tube;kJ/kgK} = Cp(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$T_{i,shell} = (89,2 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,shell} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i,shell} = 120 \text{ [kPa]}$$

$$P_{o,shell} = 85 \text{ [kPa]}$$

$$k_{i,shell} = 670 \text{ [W/m-K]}$$

$$k_{o,shell} = k(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\rho_{i,shell} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\mu_{i,shell} = 0,236 \text{ [Pa-s]}$$

$$\mu_{o,shell} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$cp_{i,shell} = 4196,3 \text{ [J/kg-K]}$$

$$Cp_{o,shell;kJ/kgK} = Cp(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$v_{i,shell} = 0,085 \text{ [m/s]}$$

$$v_{o,shell} = 0,05 \text{ [m/s]}$$

$$v_{i,tube} = 1,26 \text{ [m/s]}$$

$$v_{o,tube} = 1,15 \text{ [m/s]}$$

$$m_{\text{shell}} = 1,88 \text{ [kg/s]}$$

$$m_{\text{tube}} = 1,57 \text{ [kg/s]}$$

Convert Units

$$Cp_{o,\text{tube}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{o,\text{tube};\text{kJ/kgK}}$$

$$Cp_{o,\text{shell}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot Cp_{o,\text{shell};\text{kJ/kgK}}$$

TT V2

Twisted Tube Validation 2

Shell Side Pressure Drop vs. Shell Side Reynolds Number

$$RUN_{\text{Eff\$}} = \text{'False'}$$

$$RUN_{\text{HEAT\$}} = \text{'False'}$$

$$RUN_{\text{PRE\$}} = \text{'True'}$$

$$RUN_{\text{EB\$}} = \text{'False'}$$

$$\text{fluid}_{\text{tube;side\$}} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Steam'}$$

Variables

$$d_{o,\text{tube}} = 0,019 \text{ [m]}$$

$$a = 0,235 \text{ [m]} - 2 \cdot \text{tube}_{\text{thk}}$$

$$b = 0,135 \text{ [m]} - 2 \cdot \text{tube}_{\text{thk}}$$

$$L_{\text{tube}} = 1,5 \text{ [m]}$$

$$H = 1,5 \text{ [m]}$$

$$\text{tube}_{\text{thk}} = 0,0015 \text{ [m]}$$

$$n_t = 7 \text{ [-]}$$

$$d_{o,\text{shell}} = 80 \text{ [m]}$$

$$v_{i,\text{tube}} = 3 \text{ [m/s]}$$

$$v_{o,\text{tube}} = 2 \text{ [m/s]}$$

$$T_{i,\text{tube}} = (17 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,\text{tube}} = (23 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{o,\text{tube}} = 30 \text{ [kPa]}$$

$$d_{h,tube} = 4 \cdot \left[\frac{\pi \cdot a \cdot b}{2 \cdot \pi \cdot a + 4 \cdot (b - a)} \right]$$

$$\mu_{o,tube} = \mathbf{Visc} \left(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube} \right)$$

$$\rho_{o,tube} = \rho \left(\text{fluid}_{tube,side\$} ; T = T_{o,tube} ; P = P_{o,tube} \right)$$

$$\rho_{i,shell} = -0,676 \text{ [kg/m}^3] \cdot T_{i,shell} \cdot \frac{1}{1 \text{ [K]}} + 885,243 \text{ [kg/m}^3]$$

$$\rho_{o,shell} = -0,676 \text{ [kg/m}^3] \cdot T_{o,shell} \cdot \frac{1}{1 \text{ [K]}} + 885,243 \text{ [kg/m}^3]$$

$$P_{o,shell} = 15 \text{ [kPa]}$$

$$v_{i,shell} = 1,4 \text{ [m/s]}$$

$$v_{o,shell} = 0,9 \text{ [m/s]}$$

$$T_{i,shell} = (60 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,shell} = (50 + 273,15) \cdot 1 \text{ [K]}$$

$$d_{h,shell} = d_{o,shell}$$

$$\mu_{o,shell} = \mathbf{Visc} \left(\text{fluid}_{shell,side\$} ; T = T_{o,shell} ; P = P_{o,shell} \right)$$

ITV3

Twisted Tube Validation 3

Tube Side Pressure Drop vs. Tube Side Reynolds Number

$$RUN_{Eff\$} = \text{'False'}$$

$$RUN_{HEAT\$} = \text{'False'}$$

$$RUN_{PRE\$} = \text{'True'}$$

$$RUN_{EB\$} = \text{'False'}$$

$$\text{fluid}_{tube,side\$} = \text{'Water'}$$

$$\text{fluid}_{shell,side\$} = \text{'Steam'}$$

Variables

$$d_{o,tube} = 0,019 \text{ [m]}$$

$$a = 0,235 \text{ [m]} - 2 \cdot \text{tube}_{thk}$$

$$b = 0,135 \text{ [m]} - 2 \cdot \text{tube}_{thk}$$

$$L_{tube} = 1,5 \text{ [m]}$$

$$H = 1,5 \text{ [m]}$$

$$tube_{thk} = 0,0015 \text{ [m]}$$

$$n_t = 7 \text{ [-]}$$

$$d_{o,shell} = 80 \text{ [m]}$$

$$v_{i,tube} = 3 \text{ [m/s]}$$

$$v_{o,tube} = 2 \text{ [m/s]}$$

$$T_{i,tube} = (17 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,tube} = (23 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{o,tube} = 30 \text{ [kPa]}$$

$$d_{h,tube} = 4 \cdot \left[\frac{\pi \cdot a \cdot b}{2 \cdot \pi \cdot a + 4 \cdot (b - a)} \right]$$

$$\mu_{o,tube} = \mathbf{Visc}(\text{fluid}_{tube;side\$}; T = T_{o,tube}; P = P_{o,tube})$$

$$\rho_{o,tube} = \rho(\text{fluid}_{tube;side\$}; T = T_{o,tube}; P = P_{o,tube})$$

$$\rho_{i,shell} = -0,676 \text{ [kg/m}^3] \cdot T_{i,shell} \cdot \frac{1}{1 \text{ [K]}} + 885,243 \text{ [kg/m}^3]$$

$$\rho_{o,shell} = -0,676 \text{ [kg/m}^3] \cdot T_{o,shell} \cdot \frac{1}{1 \text{ [K]}} + 885,243 \text{ [kg/m}^3]$$

$$P_{o,shell} = 15 \text{ [kPa]}$$

$$v_{i,shell} = 1,4 \text{ [m/s]}$$

$$v_{o,shell} = 0,9 \text{ [m/s]}$$

$$T_{i,shell} = (60 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,shell} = (50 + 273,15) \cdot 1 \text{ [K]}$$

$$d_{h,shell} = d_{o,shell}$$

$$\mu_{o,shell} = \mathbf{Visc}(\text{fluid}_{shell;side\$}; T = T_{o,shell}; P = P_{o,shell})$$

TT V4

Twisted Tube Validation 4

Heat Exchanger Effectiveness vs. Mas Flow Rate

$$RUN_{Eff\$} = \text{'True'}$$

$$RUN_{HEAT\$} = \text{'False'}$$

$$RUN_{PRE\$} = \text{'False'}$$

$$RUN_{EB\$} = \text{'False'}$$

fluid_{tube;side\$} = 'Water'

fluid_{shell;side\$} = 'Water'

Variables

d_{i,tube} = 0,01 [m]

tube_{thk} = 0,005 [m]

d_{o,tube} = 0,02 [m]

d_{i,shell} = 0,05 [m]

H = 0,45 [m]

a = 0,01 [m]

b = 0,02 [m]

n = 1 [-]

n_t = 1 [-]

n_p = 1 [%]

ρ_{o,tube} = 998,2 [kg/m³]

Cp_{o,tube} = 4,182 [J/kg-K]

k_{i,tube} = 387,6 [W/m-K]

μ_{o,tube} = 0,001003 [Pa-s]

T_{i,tube} = (380 + 273,15) · 1 [K]

T_{o,tube} = (371 + 273,15) · 1 [K]

T_{i,shell} = (330 + 273,15) · 1 [K]

T_{o,shell} = (343 + 273,15) · 1 [K]

P_{o,shell} = 85 [kPa]

P_{i,shell} = 85 [kPa]

This section is not calculated for validation 4

m_{shell} = 0,0006 [kg/s]

End of section

m_{tube} = 0,001 [kg/s]

Cp_{i,shell;kJ/kgK} = Cp (fluid_{shell;side\$} ; T = T_{i,shell} ; P = P_{i,shell})

ρ_{i,shell} = ρ (fluid_{shell;side\$} ; T = T_{i,shell} ; P = P_{i,shell})

ρ_{o,shell} = ρ (fluid_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})

$$\text{Perimeter}_{\text{tube}} = 2 \cdot \pi \cdot a + 4 \cdot (b - a)$$

$$d_{h,\text{tube}} = 4 \cdot \left[\frac{\pi \cdot a \cdot b}{2 \cdot \pi \cdot a + 4 \cdot (b - a)} \right]$$

$$d_{h,\text{shell}} = d_{i,\text{shell}}$$

$$A_{\text{ci}} = \frac{\pi \cdot d_{h,\text{shell}}^2 \cdot n_t}{4 \cdot n_p} \cdot 1 \quad [\%]$$

$$A = \pi \cdot H \cdot d_{h,\text{tube}}$$

$$v_{i,\text{shell}} = \frac{m_{\text{shell}}}{\rho_{i,\text{shell}} \cdot A}$$

$$\delta T_{\text{shell}} = \frac{T_{i,\text{tube}} - T_{o,\text{shell}} - (T_{o,\text{tube}} - T_{i,\text{shell}})}{\ln \left[\frac{T_{i,\text{tube}} - T_{o,\text{shell}}}{T_{o,\text{tube}} - T_{i,\text{shell}}} \right]}$$

$$U_{\text{kW/m}^2\text{K}} = \frac{m_{\text{shell}} \cdot C_{p_{i,\text{shell};\text{kJ/kgK}}} \cdot (T_{o,\text{shell}} - T_{i,\text{shell}})}{\delta T_{\text{shell}} \cdot A_{\text{ci}}}$$

Convert Units

$$C_{p_{i,\text{shell}}} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot C_{p_{i,\text{shell};\text{kJ/kgK}}}$$

$$U = \left| 1000 \cdot \frac{\text{W/m}^2\text{-K}}{\text{kW/m}^2\text{-K}} \right| \cdot U_{\text{kW/m}^2\text{K}}$$

TT V5

Twisted Tube Validation 5

Tube Side Heat Transfer Coefficient vs. Twist Pitch Length

$$\text{RUN}_{\text{Eff\$}} = \text{'False'}$$

$$\text{RUN}_{\text{HEAT\$}} = \text{'True'}$$

$$\text{RUN}_{\text{PRE\$}} = \text{'False'}$$

$$\text{RUN}_{\text{EB\$}} = \text{'False'}$$

$$\text{fluid}_{\text{tube;side\$}} = \text{'Water'}$$

$$\text{fluid}_{\text{shell;side\$}} = \text{'Steam'}$$

Variables

This section is not calculated for validation 5

$$P = 0,192 \quad [\text{m}]$$

End of section

$$\text{pitch} = 0,02778 \text{ [m]}$$

$$d_{o,\text{tube}} = 0,02049 \text{ [m]}$$

$$\text{tube}_{\text{thk}} = 0,001734 \text{ [m]}$$

$$d_{i,\text{tube}} = d_{o,\text{tube}} - 2 \cdot d_{i,\text{tube}}$$

$$a = 0,0198 \text{ [m]}$$

$$b = 9,4 \text{ [m]}$$

$$L_{\text{tube}} = 9,926 \text{ [m]}$$

$$H = 9,926 \text{ [m]}$$

$$A_{\text{tube}} = 10170,8 \text{ [m}^2\text{]}$$

$$R_w = \frac{\frac{1}{\text{tube}_{\text{thk}}}}{k_{i,\text{tube}} \cdot \frac{1}{A_{\text{tube}}}}$$

$$R_f = 0,0002 \text{ [m}^2\text{-K/W]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$d = d_{o,\text{shell}}$$

$$n_t = 1591 \text{ [-]}$$

$$d_{o,\text{shell}} = 0,203 \text{ [m]}$$

$$d_{i,\text{shell}} = 0,12 \text{ [m]}$$

$$n_p = 1 \text{ [%]}$$

$$T_{i,\text{tube}} = (30 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,\text{tube}} = (53,1 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i,\text{tube}} = 115 \text{ [kPa]}$$

$$P_{o,\text{tube}} = 85 \text{ [kPa]}$$

$$\rho_{i,\text{tube}} = 1160 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,\text{tube}} = \rho(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$k_{i,\text{tube}} = 0,21 \text{ [W/m-K]}$$

$$k_{o,\text{tube}} = k(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$\mu_{i,\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{i,\text{tube}} ; P = P_{i,\text{tube}})$$

$$\mu_{o,\text{tube}} = \text{Visc}(\text{fluid}_{\text{tube;side\$}} ; T = T_{o,\text{tube}} ; P = P_{o,\text{tube}})$$

$$c_{p,i,tube} = 1920 \text{ [J/kg-K]}$$

$$C_{p,o,tube;kJ/kgK} = Cp(\text{fluid}_{tube;side\$} ; T = T_{o,tube} ; P = P_{o,tube})$$

$$T_{i,shell} = (89,2 + 273,15) \cdot 1 \text{ [K]}$$

$$T_{o,shell} = (80 + 273,15) \cdot 1 \text{ [K]}$$

$$P_{i,shell} = 120 \text{ [kPa]}$$

$$P_{o,shell} = 85 \text{ [kPa]}$$

$$k_{i,shell} = 670 \text{ [W/m-K]}$$

$$k_{o,shell} = k(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\rho_{i,shell} = 971,8 \text{ [kg/m}^3\text{]}$$

$$\rho_{o,shell} = \rho(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$\mu_{i,shell} = 0,236 \text{ [Pa-s]}$$

$$\mu_{o,shell} = \text{Visc}(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$c_{p,i,shell} = 4196,3 \text{ [J/kg-K]}$$

$$C_{p,o,shell;kJ/kgK} = Cp(\text{fluid}_{shell;side\$} ; T = T_{o,shell} ; P = P_{o,shell})$$

$$v_{i,shell} = 0,085 \text{ [m/s]}$$

$$v_{o,shell} = 0,05 \text{ [m/s]}$$

$$v_{i,tube} = 1,26 \text{ [m/s]}$$

$$v_{o,tube} = 1,15 \text{ [m/s]}$$

$$m_{shell} = 1,88 \text{ [kg/s]}$$

$$m_{tube} = 1,57 \text{ [kg/s]}$$

Convert Units

$$C_{p,o,tube} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot C_{p,o,tube;kJ/kgK}$$

$$C_{p,o,shell} = \left| 1000 \cdot \frac{\text{J/kg-K}}{\text{kJ/kg-K}} \right| \cdot C_{p,o,shell;kJ/kgK}$$

Parametric Table: Table 1

	$Re_{i,tube}$ [-]	$h_{i,tube}$ [W/m ² ·K]
Run 1	4000	386,2
Run 2	4778	445,2
Run 3	5556	502,3
Run 4	6333	557,8
Run 5	7111	611,9
Run 6	7889	664,9
Run 7	8667	716,9
Run 8	9444	767,9
Run 9	10222	818,1
Run 10	11000	867,5

Parametric Table: Table 2

	Re_{shell} [-]	P_{shell} [Pa]
Run 1	4000	844,7
Run 2	5111	992,5
Run 3	6222	1130
Run 4	7333	1258
Run 5	8444	1381
Run 6	9556	1498
Run 7	10667	1610
Run 8	11778	1718
Run 9	12889	1823
Run 10	14000	1925

Parametric Table: Table 3

	Re_{tube} [-]	P_{tube} [Pa]
Run 1	4000	312,3
Run 2	5222	326,7
Run 3	6444	338,5
Run 4	7667	348,6
Run 5	8889	357,4
Run 6	10111	365,3
Run 7	11333	372,4
Run 8	12556	378,9
Run 9	13778	384,9
Run 10	15000	390,5

Parametric Table: Table 4

	m [kg/s]	eff [%]
Run 1	10	80,96
Run 2	18,89	75,52
Run 3	27,78	72,44
Run 4	36,67	70,32
Run 5	45,56	68,73
Run 6	54,44	67,45
Run 7	63,33	66,4
Run 8	72,22	65,49

Parametric Table: Table 4

	m [kg/s]	eff [%]
Run 9	81,11	64,71
Run 10	90	64,02

Parametric Table: Table 5

	P [m]	$h_{i,tube}$ [W/m ² -K]	P_{tube} [Pa]
Run 1	0,16	168,7	189
Run 2	0,2	162,6	151,2
Run 3	0,25	156,7	120,9
Run 4	0,3	152,1	100,8
Run 5	0,4	145	75,59

Parametric Table: Table 6

	a [m]	b [m]	Aspect _{ratio} [-]	P [m]	$h_{i,tube}$ [W/m ² -K]	P_{tube} [Pa]	A_{tube} [m ²]	Q_{in} [kW]	Q_{out} [kW]
Run 1	0,007	0,035	0,2	0,16	1537	472,5	0,2642	31423	31278
Run 2	0,009	0,033	0,2727	0,1771	1052	395,3	0,3274	31423	31278
Run 3	0,011	0,031	0,3548	0,1943	775,6	345,8	0,3846	31423	31278
Run 4	0,013	0,029	0,4483	0,2114	600,9	312	0,4352	31423	31278
Run 5	0,015	0,027	0,5556	0,2286	482,6	288,2	0,4788	31423	31278
Run 6	0,017	0,025	0,68	0,2457	398,4	271,3	0,5149	31423	31278
Run 7	0,019	0,023	0,8261	0,2629	336,1	259,7	0,5429	31423	31278
Run 8	0,021	0,021	1	0,28	288,6	252,3	0,5621	31423	31278
Run 9	0,023	0,019	1,211	0,2971	251,7	248,7	0,5719	31423	31278
Run 10	0,025	0,017	1,471	0,3143	222,5	248,8	0,5714	31423	31278
Run 11	0,027	0,015	1,8	0,3314	199,1	253,1	0,5599	31423	31278
Run 12	0,029	0,013	2,231	0,3486	180,3	262,3	0,5363	31423	31278
Run 13	0,031	0,011	2,818	0,3657	165,4	278,2	0,4996	31423	31278
Run 14	0,033	0,009	3,667	0,3829	153,7	304,3	0,4486	31423	31278
Run 15	0,035	0,007	5	0,4	145,4	347,9	0,3818	31423	31278

Parametric Table: Validation 1

	$Re_{o,tube}$ [-]	$h_{o,tube}$ [W/m ² -K]
Run 1	2000	1063
Run 2	3000	1250
Run 3	4000	1403
Run 4	5000	1534
Run 5	6000	1650

Parametric Table: Validation 2

	Re_{shell} [-]	P_{shell} [Pa]
Run 1	540	1,269
Run 2	680	1,667
Run 3	796,7	2,01
Run 4	910	2,352
Run 5	1017	2,682

Parametric Table: Validation 2

	Re_{shell} [-]	P_{shell} [Pa]
Run 6	1127	3,029
Run 7	1217	3,317
Run 8	1313	3,631
Run 9	1407	3,938
Run 10	1500	4,249

Parametric Table: Validation 3

	Re_{tube} [-]	P_{tube} [Pa]
Run 1	14525	4,148
Run 2	17494	6,049
Run 3	20226	8,119
Run 4	23373	10,89
Run 5	26045	13,56
Run 6	28777	16,6
Run 7	31508	19,95
Run 8	34299	23,69
Run 9	37150	27,86
Run 10	40178	32,66
Run 11	41781	35,35

Parametric Table: Validation 4

	m_{shell} [kg/s]	eff [%]
Run 1	0,000204	75,43
Run 2	0,000402	68,13
Run 3	0,000602	64,13
Run 4	0,000804	61,4
Run 5	0,001	59,43

Parametric Table: Validation 5

	P [m]	$h_{i,tube}$ [W/m ² -K]
Run 1	0,0993	10710
Run 2	0,1493	5845
Run 3	0,1971	3868
Run 4	0,2471	2765
Run 5	0,2963	2113
Run 6	0,3954	1376

Parametric Table: Simulation

	$T_{i,shell}$ [K]	$T_{i,tube}$ [K]	$h_{i,shell}$ [W/m ² -K]	$h_{i,tube}$ [W/m ² -K]	$h_{ave,shell}$	$h_{ave,tube}$
Run 1	396,2	353,2	262,1	780,6	247,3	490,4
Run 2	394,2	354,3	260,2	781,8	246,5	491
Run 3	392,2	355,4	258,3	782,9	245,6	491,6
Run 4	390,2	356,5	256,3	784	244,8	492,1
Run 5	388,2	357,6	254,3	785,1	243,9	492,7

Parametric Table: Simulation

	$T_{i,shell}$ [K]	$T_{i,tube}$ [K]	$h_{i,shell}$ [W/m ² -K]	$h_{i,tube}$ [W/m ² -K]	$h_{ave,shell}$	$h_{ave,tube}$
Run 6	386,2	358,7	252,2	786,1	243	493,2
Run 7	384,2	359,8	250,1	787,2	242	493,7
Run 8	382,2	360,9	248	788,2	241,1	494,2
Run 9	380,2	362	245,8	789,2	240,1	494,7
Run 10	378,2	363,2	243,7	790,2	239,1	495,2

SOLUTION

Unit Settings: SI K kPa kJ mass deg

```

a = 0,007 [m]
Aci = 120 [m2]
Atube = 0,2642 [m2]
c = 4,700E+08 [-]
CPi,shell,kJ/kgK = 4,248 [kJ/kg-K]
CPi,tube,kJ/kgK = 4,197 [kJ/kg-K]
CPo,shell,kJ/kgK = 4,222 [kJ/kg-K]
CPo,tube,kJ/kgK = 4,208 [kJ/kg-K]
Ch = 1,372E+06 [W/K]
Cif = 2,000E+06 [R]
Cmin = 1,372E+06 [W/K]
Cs = 1,500E+06 [R]
δTtube = 24,6 [K]
delthaP,o,shell = 54672 [Pa]
dh,tube = 0,01974 [m]
dinlet,tube = 0,1168 [m]
di,tube = 0,01575 [m]
doutlet,tube = 0,2762 [m]
do,tube = 0,01905 [m]
eff = 100 [%]
enthalpyi,tube = 335,1 [kJ/kg]
enthalpyo,tube = 389,7 [kJ/kg]
fluidtube,side$ = 'Water'
Frmo = 3,426 [-]
fi,tube = 0,009102 [-]
fo,tube = 0,0143 [-]
ft = 1,534 [-]
Gs,i = 14111 [kg/m2-s]
H = 4,26 [m]
have,tube = 1227 [W/m2-K]
hi,tube = 972,7 [W/m2-K]
ho,tube = 1480 [W/m2-K]
j = 1,88 [R/W]
ki,tube = 0,667 [W/m-K]
ko,tube = 0,6743 [W/m-K]
m = 1,718E+06 [kg/s]
μi,tube = 0,0003541 [Pa-s]
μo,tube = 0,0003037 [Pa-s]
mtube = 440,9 [kg/s]
np = 1 [%]
nt = 442 [-]
opptime = 20 [yr]
Perimetertube = 0,156 [m]
Pri,tube = 2,228 [-]
Pro,tube = 1,896 [-]
Pi,tube = 150 [kPa]
Po,tube = 135 [kPa]
Ptube = 6,379E+07 [Pa]
Qin = 24080 [kW]
Qout = 24080 [kW]
Qtube = 2,412E+07 [W]
Rei,tube = 2,378E+06 [-]
Reo,tube = 999035 [-]
Retube = 1,252E+06 [-]
ρi,tube = 971,8 [kg/m3]
ρo,tube = 963,3 [kg/m3]
RUNEff$ = 'True'
RUNPRES$ = 'True'
Rf = 0,0002 [m2-K/W]

Aspectratio = 0,2 [-]
Aco = 120 [m2]
b = 0,035 [m]
CPi,shell = 4248 [J/kg-K]
CPi,tube = 4197 [J/kg-K]
CPo,shell = 4222 [J/kg-K]
CPo,tube = 4208 [J/kg-K]
Cc = 1,855E+06 [W/K]
ci = 10000 [R]
Cmax = 1,855E+06 [W/K]
cpump = 0,00288 [R/W/h]
d = 1,468 [m]
delthaP,i,shell = 65835 [Pa]
dh,shell = 0,588 [m]
dinlet,shell = 0,1707 [m]
di,shell = 0,588 [m]
doutlet,shell = 0,0907 [m]
do,shell = 0,6 [m]
e = 2,718 [-]
enthalpyi,shell = 516,6 [kJ/kg]
enthalpyo,shell = 442,1 [kJ/kg]
fluidshell,side$ = 'Steam'
Frmi = 3,953 [-]
fi = 0,08586 [-]
fo = 0,09356 [-]
fs = 2,584 [-]
g = 9,81 [m/s2]
Gs,o = 12409 [kg/m2-s]
have,shell = 497170 [W/m2-K]
hi,shell = 524247 [W/m2-K]
ho,shell = 470093 [W/m2-K]
i = 11 [%]
ki,shell = 0,6826 [W/m-K]
ko,shell = 0,6792 [W/m-K]
Ltube,check = 1,509E+09 [m]
μi,shell = 0,000226 [Pa-s]
μo,shell = 0,0002665 [Pa-s]
mshell = 322,9 [kg/s]
n = 1 [-]
npump = 58 [%]
Ntu = 8,022E-10 [-]
P = 0,16 [m]
Pri,shell = 1,407 [-]
Pro,shell = 1,656 [-]
Pi,shell = 306 [kPa]
Po,shell = 305 [kPa]
Pshell = 1,510E+06 [Pa]
q = 780 [W/h]
Qmax = 5,899E+07 [W]
Qshell = 2,416E+07 [W]
Rei = 7000 [-]
Reo = 5937 [-]
Reshell = 2,738E+07 [-]
ρi,shell = 940,7 [kg/m3]
ρo,shell = 954,5 [kg/m3]
RUNEB$ = 'True'
RUNHEAT$ = 'True'
Rc = 0,7394 [-]
Rw = 240 [m2-K/W]

```

$\text{shell}_{\text{thk}} = 0,006 \text{ [m]}$
 $\text{tube}_{\text{thk}} = 0,00165 \text{ [m]}$
 $T_{\text{i,tube}} = 353,2 \text{ [K]}$
 $T_{\text{o,tube}} = 366,2 \text{ [K]}$
 $V_{\text{i,shell}} = 15 \text{ [m/s]}$
 $V_{\text{o,shell}} = 13 \text{ [m/s]}$

$\text{tt} = 7300 \text{ [h/yr]}$
 $T_{\text{i,shell}} = 396,2 \text{ [K]}$
 $T_{\text{o,shell}} = 378,5 \text{ [K]}$
 $U = 0,004166 \text{ [W/m}^2\text{-K]}$
 $V_{\text{i,tube}} = 55 \text{ [m/s]}$
 $V_{\text{o,tube}} = 20 \text{ [m/s]}$

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

```

a = 0,0198 [m]
Aco = 17,99 [m2]
b = 9,4 [m]
cp,i,tube = 1920 [J/kg-K]
Cp,o,shell,kJ/kgK = 4,197 [kJ/kg-K]
Cp,o,tube,kJ/kgK = 4,182 [kJ/kg-K]
delthap,i,shell = 45050 [Pa]
dh,shell = 0,12 [m]
di,shell = 0,12 [m]
do,shell = 0,203 [m]
fluidshell,side$ = 'Steam'
Frmi = 0,06023 [-]
fi = 155,1 [-]
fo = 5,242 [-]
g = 9,81 [m/s2]
Gs,o = 48,59 [kg/m2-s]
have,shell = 189060 [W/m2-K]
hi,shell = 316945 [W/m2-K]
ho,shell = 61175 [W/m2-K]
ki,shell = 670 [W/m-K]
ko,shell = 0,667 [W/m-K]
Ltube = 9,926 [m]
μi,tube = 0,0007972 [Pa-s]
μo,tube = 0,0005192 [Pa-s]
mtube = 1,57 [kg/s]
nt = 1591 [-]
Perimetertube = 37,65 [m]
Pri,shell = 1,478 [-]
Pro,shell = 2,228 [-]
Pi,shell = 120 [kPa]
Po,shell = 85 [kPa]
Rei = 0,05313 [-]
Reo = 35,41 [-]
ρi,shell = 971,8 [kg/m3]
ρo,shell = 971,8 [kg/m3]
RUNEB$ = 'False'
RUNHEAT$ = 'True'
Rf = 0,0002 [m2-K/W]
tubethk = 0,001734 [m]
Ti,tube = 303,2 [K]
To,tube = 326,3 [K]
vi,shell = 0,085 [m/s]
vo,shell = 0,05 [m/s]

Aci = 17,99 [m2]
Atube = 10171 [m2]
cp,i,shell = 4196 [J/kg-K]
Cp,o,shell = 4197 [J/kg-K]
Cp,o,tube = 4182 [J/kg-K]
d = 0,203 [m]
delthap,o,shell = 526,7 [Pa]
dh,tube = 0,06213 [m]
di,tube = 0,00683 [m]
do,tube = 0,02049 [m]
fluidtube,side$ = 'Water'
Frmo = 0,03543 [-]
fi,tube = 0,5814 [-]
fo,tube = 0,5306 [-]
Gs,i = 82,6 [kg/m2-s]
H = 9,926 [m]
have,tube = 568,5 [W/m2-K]
hi,tube = 303,7 [W/m2-K]
ho,tube = 833,4 [W/m2-K]
ki,tube = 0,21 [W/m-K]
ko,tube = 0,644 [W/m-K]
μi,shell = 0,236 [Pa-s]
μo,shell = 0,000354 [Pa-s]
mshell = 1,88 [kg/s]
np = 1 [%]
P = 0,192 [m]
pitch = 0,02778 [m]
Pri,tube = 7,289 [-]
Pro,tube = 3,372 [-]
Pi,tube = 115 [kPa]
Po,tube = 85 [kPa]
Rei,tube = 12522 [-]
Reo,tube = 14924 [-]
ρi,tube = 1160 [kg/m3]
ρo,tube = 986,6 [kg/m3]
RUNEff$ = 'False'
RUNPRE$ = 'False'
Rw = 2,793E+07 [m2-K/W]
Ti,shell = 362,4 [K]
To,shell = 353,2 [K]
U = 3,580E-08 [W/m2-K]
vi,tube = 1,26 [m/s]
vo,tube = 1,15 [m/s]

```

No unit problems were detected.

SOLUTION

Unit Settings: SI K kPa kJ mass deg

a = 0,232 [m]	b = 0,132 [m]	d _{h,shell} = 80 [m]
d _{h,tube} = 0,3638 [m]	d _{o,shell} = 80 [m]	d _{o,tube} = 0,019 [m]
fluid _{shell,side} = 'Steam'	fluid _{tube,side} = 'Water'	f _s = 3,146 [-]
f _t = 1,416 [-]	H = 1,5 [m]	L _{tube} = 1,5 [m]
μ _{o,shell} = 0,0005465 [Pa-s]	μ _{o,tube} = 0,0009321 [Pa-s]	n _t = 7 [-]
P _{o,shell} = 15 [kPa]	P _{o,tube} = 30 [kPa]	P _{shell} = 1,855E+06 [Pa]
P _{tube} = 13329 [Pa]	Re _{shell} = 8,785E+07 [-]	Re _{tube} = 778709 [-]
ρ _{i,shell} = 660 [kg/m ³]	ρ _{o,shell} = 666,8 [kg/m ³]	ρ _{o,tube} = 997,5 [kg/m ³]
RUN _{EB} = 'False'	RUN _{ECO} = 'False'	RUN _{Eff} = 'False'
RUN _{HEAT} = 'False'	RUN _{PRE} = 'True'	tube _{thk} = 0,0015 [m]
T _{i,shell} = 333,2 [K]	T _{i,tube} = 290,2 [K]	T _{o,shell} = 323,2 [K]
T _{o,tube} = 296,2 [K]	V _{i,shell} = 1,4 [m/s]	V _{i,tube} = 3 [m/s]
V _{o,shell} = 0,9 [m/s]	V _{o,tube} = 2 [m/s]	

No unit problems were detected.

KEY VARIABLES

P_{o,tube} = 30 [kPa]

SOLUTION

Unit Settings: SI K kPa kJ mass deg

a = 0,232 [m]	A _{shell} = 119 [m ²]	A _{tube} = 0,0002835 [m ²]
b = 0,132 [m]	d _{h,shell} = 80 [m]	d _{h,tube} = 0,3638 [m]
d _{o,shell} = 80 [m]	d _{o,tube} = 0,019 [m]	enthaphy _{i,shell} = 2610 [kJ/kg]
enthaphy _{i,tube} = 72,22 [kJ/kg]	enthaphy _{o,shell} = 2608 [kJ/kg]	enthaphy _{o,tube} = 96,49 [kJ/kg]
fluid _{shell,side} = 'Steam'	fluid _{tube,side} = 'Water'	f _s = 1,374 [-]
f _t = 1,416 [-]	H = 1,5 [m]	L _{tube} = 1,5 [m]
μ _{o,shell} = 0,00001083 [Pa-s]	μ _{o,tube} = 0,0009321 [Pa-s]	m _{shell} = 10,52 [kg/s]
m _{tube} = 0,8495 [kg/s]	n _t = 7 [-]	P _{i,shell} = 15 [kPa]
P _{i,tube} = 26 [kPa]	P _{o,shell} = 15 [kPa]	P _{o,tube} = 30 [kPa]
P _{shell} = 0,8293 [Pa]	P _{tube} = 13329 [Pa]	Q _{in} = 20,62 [kW]
Q _{out} = 20,39 [kW]	Re _{shell} = 653408 [-]	Re _{tube} = 778709 [-]
ρ _{i,shell} = 0,09795 [kg/m ³]	ρ _{i,tube} = 998,7 [kg/m ³]	ρ _{o,shell} = 0,09825 [kg/m ³]
ρ _{o,tube} = 997,5 [kg/m ³]	RUN _{EB} = 'False'	RUN _{ECO} = 'False'
RUN _{Eff} = 'False'	RUN _{HEAT} = 'False'	RUN _{PRE} = 'True'
tube _{thk} = 0,0015 [m]	T _{i,shell} = 333,2 [K]	T _{i,tube} = 290,4 [K]
T _{o,shell} = 332,2 [K]	T _{o,tube} = 296,2 [K]	v _{i,shell} = 1,4 [m/s]
v _{i,tube} = 3 [m/s]	v _{o,shell} = 0,9 [m/s]	v _{o,tube} = 2 [m/s]

No unit problems were detected.

KEY VARIABLES

Q_{in} = 20,62 [kW]
P_{o,tube} = 30 [kPa]
Q_{out} = 20,39 [kW]

SOLUTION

Unit Settings: SI K kPa kJ mass deg

```

a = 0,01 [m]
Atube = 0,03455 [m2]
Cpi,shell = 2026 [J/kg-K]
Cc = 0,004182 [W/K]
Cmin = 0,004182 [W/K]
dh,shell = 0,05 [m]
di,tube = 0,01 [m]
enthaphyi,shell = 3131 [kJ/kg]
enthaphyo,tube = 3218 [kJ/kg]
H = 0,45 [m]
m = 0,00003326 [kg/s]
mtube = 0,001 [kg/s]
nt = 1 [-]
Pi,shell = 85 [kPa]
Po,tube = 168 [kPa]
Qout = 0,01827 [kW]
ρi,shell = 0,3069 [kg/m3]
RUNEB$ = 'False'
RUNHEAT$ = 'False'
tubethk = 0,005 [m]
To,shell = 616,2 [K]
UkW\m2K = 0,2326 [kW/m2-K]

A = 0,03455 [m2]
b = 0,02 [m]
Cpi,shell,kJkgK = 2,026 [kJ/kg-K]
Ch = 1,216 [W/K]
δTshell = 39,92 [K]
dh,tube = 0,02444 [m]
do,tube = 0,02 [m]
enthaphyi,tube = 3236 [kJ/kg]
fluidshell,side$ = 'Water'
ki,tube = 387,6 [W/m-K]
μo,tube = 0,001003 [Pa-s]
n = 1 [-]
Ntu = 1922 [-]
Pi,tube = 175 [kPa]
Qin = 0,01842 [kW]
Qshell = 18,23 [W]
ρo,shell = 0,2994 [kg/m3]
RUNECO$ = 'False'
RUNPRE$ = 'False'
Ti,shell = 601,2 [K]
To,tube = 644,2 [K]
Vi,shell = 0,05658 [m/s]

Aci = 0,001963 [m2]
c = 4,700E+08 [-]
Cpo,tube = 4,182 [J/kg-K]
Cmax = 1,216 [W/K]
δTtube = -38,77 [K]
di,shell = 0,05 [m]
eff = 0,06985 [%]
enthaphyo,shell = 3162 [kJ/kg]
fluidtube,side$ = 'Water'
Ltube,check = -8,636E-14 [m]
mshell = 0,0006 [kg/s]
np = 1 [%]
Perimetertube = 0,1028 [m]
Po,shell = 85 [kPa]
Qmax = 0,2175 [W]
Qtube = 0,03764 [W]
ρo,tube = 998,2 [kg/m3]
RUNEff$ = 'True'
Rc = 0,003441 [-]
Ti,tube = 653,2 [K]
U = 232,6 [W/m2-K]

```

No unit problems were detected.

KEY VARIABLES

```

Qin = 0,01842 [kW]
Po,tube = 168 [kPa]
Qout = 0,01827 [kW]

```

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$a = 0,0198 \text{ [m]}$
 $A_{co} = 17,99 \text{ [m}^2\text{]}$
 $b = 9,4 \text{ [m]}$
 $cp_{i,tube} = 1920 \text{ [J/kg-K]}$
 $Cp_{o,shell,kJ/kgK} = 4,197 \text{ [kJ/kg-K]}$
 $Cp_{o,tube,kJ/kgK} = 4,182 \text{ [kJ/kg-K]}$
 $deltha_{P,i,shell} = 45050 \text{ [Pa]}$
 $d_{h,shell} = 0,12 \text{ [m]}$
 $d_{i,shell} = 0,12 \text{ [m]}$
 $d_{o,shell} = 0,203 \text{ [m]}$
 $fluid_{shell,side\$} = \text{'Steam'}$
 $Fr_{mi} = 0,06023 \text{ [-]}$
 $f_i = 155,1 \text{ [-]}$
 $f_o = 5,242 \text{ [-]}$
 $g = 9,81 \text{ [m/s}^2\text{]}$
 $G_{s,o} = 48,59 \text{ [kg/m}^2\text{-s]}$
 $h_{ave,shell} = 189060 \text{ [W/m}^2\text{-K]}$
 $h_{i,shell} = 316945 \text{ [W/m}^2\text{-K]}$
 $h_{o,shell} = 61175 \text{ [W/m}^2\text{-K]}$
 $k_{i,shell} = 670 \text{ [W/m-K]}$
 $k_{o,shell} = 0,667 \text{ [W/m-K]}$
 $L_{tube} = 9,926 \text{ [m]}$
 $\mu_{i,tube} = 0,0007972 \text{ [Pa-s]}$
 $\mu_{o,tube} = 0,0005192 \text{ [Pa-s]}$
 $m_{tube} = 1,57 \text{ [kg/s]}$
 $n_t = 1591 \text{ [-]}$
 $Perimeter_{tube} = 37,65 \text{ [m]}$
 $Pr_{i,shell} = 1,478 \text{ [-]}$
 $Pr_{o,shell} = 2,228 \text{ [-]}$
 $P_{i,shell} = 120 \text{ [kPa]}$
 $P_{o,shell} = 85 \text{ [kPa]}$
 $Re_i = 0,05313 \text{ [-]}$
 $Re_o = 35,41 \text{ [-]}$
 $\rho_{i,shell} = 971,8 \text{ [kg/m}^3\text{]}$
 $\rho_{o,shell} = 971,8 \text{ [kg/m}^3\text{]}$
 $RUN_{EB\$} = \text{'False'}$
 $RUN_{HEAT\$} = \text{'True'}$
 $R_f = 0,0002 \text{ [m}^2\text{-K/W]}$
 $tube_{thk} = 0,001734 \text{ [m]}$
 $T_{i,tube} = 303,2 \text{ [K]}$
 $T_{o,tube} = 326,3 \text{ [K]}$
 $v_{i,shell} = 0,085 \text{ [m/s]}$
 $v_{o,shell} = 0,05 \text{ [m/s]}$

$A_{ci} = 17,99 \text{ [m}^2\text{]}$
 $A_{tube} = 10171 \text{ [m}^2\text{]}$
 $cp_{i,shell} = 4196 \text{ [J/kg-K]}$
 $Cp_{o,shell} = 4197 \text{ [J/kg-K]}$
 $Cp_{o,tube} = 4182 \text{ [J/kg-K]}$
 $d = 0,203 \text{ [m]}$
 $deltha_{P,o,shell} = 526,7 \text{ [Pa]}$
 $d_{h,tube} = 0,06213 \text{ [m]}$
 $d_{i,tube} = 0,00683 \text{ [m]}$
 $d_{o,tube} = 0,02049 \text{ [m]}$
 $fluid_{tube,side\$} = \text{'Water'}$
 $Fr_{mo} = 0,03543 \text{ [-]}$
 $f_{i,tube} = 0,5814 \text{ [-]}$
 $f_{o,tube} = 0,5306 \text{ [-]}$
 $G_{s,i} = 82,6 \text{ [kg/m}^2\text{-s]}$
 $H = 9,926 \text{ [m]}$
 $h_{ave,tube} = 568,5 \text{ [W/m}^2\text{-K]}$
 $h_{i,tube} = 303,7 \text{ [W/m}^2\text{-K]}$
 $h_{o,tube} = 833,4 \text{ [W/m}^2\text{-K]}$
 $k_{i,tube} = 0,21 \text{ [W/m-K]}$
 $k_{o,tube} = 0,644 \text{ [W/m-K]}$
 $\mu_{i,shell} = 0,236 \text{ [Pa-s]}$
 $\mu_{o,shell} = 0,000354 \text{ [Pa-s]}$
 $m_{shell} = 1,88 \text{ [kg/s]}$
 $n_p = 1 \text{ [%]}$
 $P = 0,192 \text{ [m]}$
 $pitch = 0,02778 \text{ [m]}$
 $Pr_{i,tube} = 7,289 \text{ [-]}$
 $Pr_{o,tube} = 3,372 \text{ [-]}$
 $P_{i,tube} = 115 \text{ [kPa]}$
 $P_{o,tube} = 85 \text{ [kPa]}$
 $Re_{i,tube} = 12522 \text{ [-]}$
 $Re_{o,tube} = 14924 \text{ [-]}$
 $\rho_{i,tube} = 1160 \text{ [kg/m}^3\text{]}$
 $\rho_{o,tube} = 986,6 \text{ [kg/m}^3\text{]}$
 $RUN_{Eff\$} = \text{'False'}$
 $RUN_{PRE\$} = \text{'False'}$
 $R_w = 2,793E+07 \text{ [m}^2\text{-K/W]}$
 $T_{i,shell} = 362,4 \text{ [K]}$
 $T_{o,shell} = 353,2 \text{ [K]}$
 $U = 3,580E-08 \text{ [W/m}^2\text{-K]}$
 $v_{i,tube} = 1,26 \text{ [m/s]}$
 $v_{o,tube} = 1,15 \text{ [m/s]}$

No unit problems were detected.