An Optimization Tool Developed for Fluted Tube Heat Exchangers

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by
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Authorization to Submit Thesis

This thesis of Kevin Terrill, submitted for the degree of Master of Science with a Major in Mechanical Engineering and titled "An Optimization Tool Developed for Fluted Tube Heat Exchangers," has been reviewed in final form. Permission, as indicated by the signatures and dates below is now granted to submit final copies for the College of Graduate Studies for approval.

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Abstract

As the nuclear industry is moving towards compact reactors and power plants through the designs of small modular reactors (SMRs) and micro reactors, compact heat exchangers will minimize the size of the reactor plant designs. Replacing straight tubes with fluted tube in a vertical shell and tube heat exchanger will significantly decrease heat exchanger size. Fluted tubes enhance heat transfer through passive geometric techniques and increased surface area but at the cost of increasing pressure drop. An optimization code has been developed to find ideal design parameters that maximize heat transfer per pressure drop. The code compares 50 different standard sizes of fluted tubes through varying a key non dimensional scaling parameter that relates the tube to tube pitch and flow area. The non dimensional parameter was developed to allow scaling of flow areas to a direct comparison of different size tubes.

Acknowledgments

I would like to show my appreciation for the University of Idaho Idaho Falls campus and staff. They have made graduate school a great opportunity to improve myself through easy access to resources that will make me successful in the future. The professors and mentors at the Idaho Falls campus have always supported me and made me feel valued. I want to acknowledge Dr. Christensen as a key role model for me to emulate through me education and professional career. Not only have I learned a great deal from his vast thermal hydraulics knowledge, but also the importance of developing professional connections that will help me navigate the academia and research environment for the duration of my career.

Dedication

This thesis is dedicated to my family, friends and mentors who have supported me through my entire education process.

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List of Variables

A	Heat Transfer Area	m^2
A_{eff}	Effective Flow Area	m^2
θ	Flute Helix Angle	Degree
Cp	Fluid Specific Heat	$\frac{J}{kg\,^\circ K}$
D_{bi}	Inside Bore Diameter	m
D_{bo}	Outside Bore Diameter	m
D_{ei}	Inside Envelope Diameter	m
D_{eo}	Outside Envelope Diameter	m
D_{hyd}	Hydraulic Diameter	m
D_{oi}	Outer Area Diameter	m
D_{vi}	Inside Volumetric Diameter	m
D_{vo}	Outside Volumetric Diameter	m
e	Flute Height	m
$\boldsymbol{f}$	Friction Factor	
FB	Base of Each Flute Circumference	m
$\boldsymbol{h}$	Heat Transfer Coefficient	$\frac{W}{m^2\,^\circ K}$
k	Fluid Thermal Conductivity	$\frac{W}{m\,^\circ K}$
K_{wall}	Tube Conductivity	$\frac{W}{m\,^\circ K}$
L	Length of Tube/Shell	m
$\dot{m}$	Mass Flow Rate	$\frac{kg}{s}$
μ	Fluid Viscosity	$\frac{N-sec}{m^2}$
Ns	Number of Flute Starts	
Nu	Nusselt Number	
$\boldsymbol{p}$	Tube to Tube Pitch	m
ΔP_l	Pressure Drop	Pa
P_{tube}	Flute Pitch	m

Pr	Prandtl Number	
r	Trough-to-Circumference	
Re	Reynolds Number	
R^*	Radius Ratio Between Shell and Tube	
ρ	Fluid Density	$\frac{kg}{m^3}$
TL	Sum of Trough Lengths	m
ΔT_{lm}	Log-Mean Temperature Difference	$^\circ K$
Tw	Wall Thickness	m
U	Overall Heat Transfer Coefficient	$\frac{W}{m^2\,^\circ K}$
V	Velocity	$\frac{m}{s}$

Chapter 1: Introduction

1.1 Background

The ability to control heat transfer in a system that requires thermal heating and cooling is done through heat exchangers. Heat transfer is essential part of many chemical and power production processes and these industries are continually researching improved heat exchanger designs. The current research focus is investigating the ability to create compact heat exchangers. Furthering this research, an optimization tool has been developed to design a compact fluted tube heat exchanger and it is presented in this paper

The energy production industry is currently investigating compact heat exchangers due to the rapid transition to clean energy. This transfer is forcing a switch to carbon free energy through nuclear, natural gas and renewable energy. When looking at the nuclear industry, the development of the new Generation IV reactors will require improved heat exchangers designs to meet size and heat transfer demands. The new power plant are adapting a smaller design because of the smaller reactor design known as small modular reactors. Due to the more compact designs, reduced capital cost, and inherent safety parameters, the smaller reactors are a reliable option to provide power throughout the world. These new reactor designs use different reactor coolant fluid types for improved heat transfer and reactor safety. The use of different coolants such as; molten salt, helium, sodium, etc. typically require intermediate heat exchanger to transfer the heat generated in the reactor to a secondary loops to produce energy.

Designing a compact heat exchanger is done through taking advantage of heat transfer enhancement methods. These methods are used to improve current heat exchanger designs to minimize their sizes by maximizing the heat transfer per surface area of the heat exchanger[1][2][3]. Adapting current designs like the shell and tube heat exchangers can provide simple solutions to creating compact heat exchangers. The shell and tube heat exchanger are commonly used because they are dynamic sizes and large heat transfer capacity. Changing straight tubes with fluted tubes is a proven method of enhancing heat transfer[3] [4]. The design of using fluted tube for a shell and tube heat exchanger is a practical solution for a compact heat exchanger.

There are different styles and variation of fluted tubes. Designs vary from the number of fluted channels on the tube to the number of spirals each flute will make per length. Especially with the recent advancement in 3D-printing technology, the complexity of fluted tubes will increase significantly. Each design and variation will effect the heat transfer and pressure drop through the tube. Distinct correlations must be used to accurately represent the geometry of the tube [4]. The optimal design will need to compare these different variation of fluted tube under the same correlation to find a balance in heat transfer, pressure drop and heat exchanger size. There are different areas to optimize for a heat exchanger based on the need of the facility it is being used and limiting boundary conditions. The objective of this paper is to present an optimization tool developed to compare different sizes and shaping of fluted tubes against each other to find different optimal designs.

1.2 Scope of Work

Compact heat exchangers is an area of interest in several industries. Improving heat transfer methods to limit heat exhanger size and maximizing performance will always be in demand. A fluted tube, shell and tube heat exchanger is a simple design that enhances the heat transfer by adapting a currently used heat exchanger. There are several different geometries for fluted tubes that may alter heat transfer, but the target of the optimization script is to compare similar fluted tube geometries that utilize the same correlation against each other. By varying a few parameters such as a pitch, tube length, single tube mass flow rate, and fluids conditions, the optimal design can be found. The parametric study conducted on the tubes allows for heat transfer and pressure drop trends to be analyzed.

1.3 Thesis Outline

The purpose of the thesis is the development of a computational script to size and optimize fluted tube heat exchangers. Chapter 1: Focuses on the overall problem and background of the need for compact heat exchangers. Chapter 2: A detailed look at heat exchanger design principals and the fluted tube correlations used in the optimization script. Chapter 3: An overview of the two different codes used to run parametric studies to find the optimal designs for fluted tube heat exchangers. Chapter 4: Results from the parametric studies conducted on three distinct case studies are presented. Chapter 5: Conclusion of the optimization scripts and the resulting case studies.

Chapter 2: Theory

2.1 Heat Exchangers

Heat exchangers are key to chemical processing and power generation systems that are dependent on thermal loads. These systems require effective heat transfer to create the proper conditions to maximize efficiency for their production. The overall thermal efficiency is controlled by the percentage of heat that can be kept in the system or transferred between two systems. Advanced heat exchangers help maximize the heat transfer to conserve heat in the system. There are several different forms of heat exchangers needed in a thermal system such as steam generators, condensers, coolers, and intermediate heat exchangers, etc. All of these are different forms of heat exchangers that are designed for a specific reason and are important in different aspects of a thermal loop. Each form of heat transfer and heat exchanger uses a distinct style of heat transfer enhancement methods to maximize performance depending on the desired results.

A specific area of interest are the use and functionality of single phase intermediate heat exchangers. Intermediate heat exchangers are used to improve the system efficiency though minimizing waste heat in a single system or to transfer heat between two systems. Waste heat is the excess heat needed to be taken out of the system before the fluid goes through the compressor/pump. The waste heat is completely lost to the system and effects the overall thermal efficiency of the system. Reintroducing this excess heat into the system will minimize the waste heat removed by the condenser and the required heat to be added into the system. This style of intermediate heat exchanger are often know as a recuperative heat exchangers, especially when used to transfer heat within the same system. There are several different designs that incorporate the reuse of waste heat into the system. Figure 2.1 represents a basic design that is commonly used and will be the general guideline for the recuperative heat exchangers designed in the optimization code.

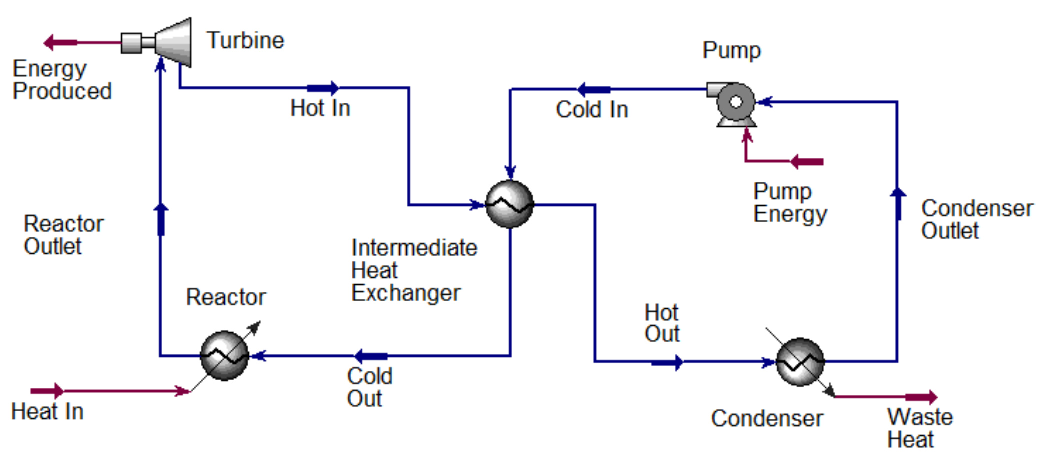


Figure 2.1: An Intermediate Heat Exchanger system designed to reintroduce the heat back into the system. Commonly referred to as a recuperative heat exchanger

A key use of an intermediate heat exchanger is to transfer heat between a primary and secondary loop. The generation IV reactor designs incorporate several intermediate heat exchangers designed to transfer the heat from the reactor primary coolant loop to a second or third loop. This is done when the primary loop cannot produce energy due to lack of coolant turbine technology or safety concerns with radiation effects throughout the system. This situation requires a second or third loop to minimize radiation contamination and damage on the turbine machinery. This is a common design for generation IV reactors that are exploring new coolants in the reactor design.

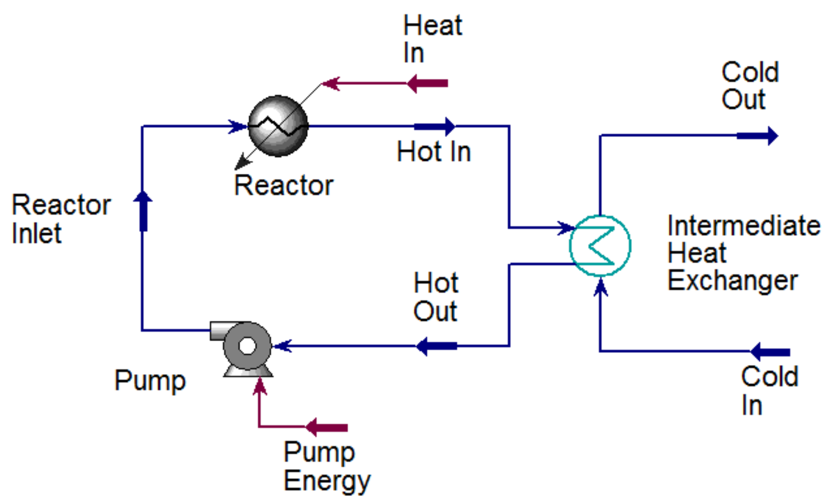


Figure 2.2: An Intermediate Heat Exchanger used to transfer heat between two systems

Intermediate heat exchangers are also used when the thermal load needs to be split between processes. The ability to separate the heat allows multiple processes to run off of the main heat generation loop. There are generation IV reactors that operate at higher temperatures allowing for the use of process heat in desalination, oil production, oil refining, biomass-based ethanol production, and hydrogen production[5]. Building reactors close to these other processes allows the heat to be used for multiple processes.

2.2 Heat Exchanger Design

Heat exchangers are designed based on the needs of the system. The fluid types, flow parameters and size constraints will influence the type and design of heat exchanger needed for the system. There are a number of variables within a single type of heat exchanger that will influence the design. Each type of heat exchanger may have different geometries or layout to meet the systems needs. There are two different approaches to heat exchanger design: First, the sizing of the heat exchanger is set and the performance needs to be calculated. Second, the required heat transfer is known but the size needs to be designed. The optimization script uses the second method to optimize the heat exchanger for a specific heat load. This approach allows the variation of the physical parameters of the heat exchanger to see the effects on the performance. The heat exchanger design approach for the developed script is to size a shell and fluted tube intermediate heat exchanger for a desired performance.

2.2.1 Flow and Geometric Parameters

When using the sizing approach to heat exchanger design, there are overarching parameters that are set for the system and individual parameters that can be varied within the heat exhanger. Each parameter in the design of the heat exchanger will influence the overall heat transfer coefficient, pressure drop and heat exchanger size. Overarching parameters in the heat exchanger system are the system mass flow rates, inlet temperatures and outlet temperatures. These parameters are designed to be constant parameters and cannot be changed throughout the optimizations. The user defines each of these parameters before running the optimization script. Although the overarching parameters cannot be changed during the script, the mass flow rate per tube is different than the system mass flow rate. The mass flow rate per tube can be manipulated by the heat exchanger design optimization through changing the number of tubes.

Tube spacing, layout, length and tube mass flow rate are all varying parameters that are considered during the design of the heat exchanger. All of these parameters may be altered, within predetermined boundary conditions. When dealing with varying parameters in the script it is important to set strict boundary conditions to ensure realistic designs. Each of these parameters influence the flow area of the tube and fluid behavior. The script analysis the sizing on a per tube basis The inside of the tube flow will only vary depending on the variation of the tube tested. The shell flow area per tube spacing is effected by tube to tube pitch and tube layout. Constricting the flow area will only effect a single side of the heat exchanger. These changes are key to run parametric studies to find the optimal design for a given system.

2.2.2 Design Guidelines

There are general guidelines to properly design a heat exchanger to ensure that realistic parameters are used. Majority of these guidelines are set within the correlations used to ensure that calculated results are accurate. The optimization script has built in limitations to ensure that the results are accurate according to the correlations. There are design parameters and aspects that are not considered in the script. The system pressure drop limitations, material constraints and manufacturing limitations will all need to be considered by the user. The only fabrication limitation considered is the physical spacing of the tubes in the tube layout. All other fabrication limitations such as tube length, tube sheets possibility, and material machinability will need to be considered by the user.

The system operating pressure and the allowed pressure drop for the heat exchanger will vary depending on the fluid type and the systems performance. The allowable pressure drop needs to be decided by the user for each system. The results of the optimization will be filtered to remove heat exchangers with a pressure drop that exceeds the specified limits. In a two loop system with an intermediate heat exchanger, the higher pressure of the two systems should passes through the tubes instead of the shell. The script does not restrict the user and will calculate the higher pressure system within either the tube or shell, but for cost and material constraints will typically forces the tubes to operate at a higher pressure. In addition, different materials have different pressure thresholds and the material used must be able to hold the pressure of the heat exchanger. The thermal conductivity of the tube will need to be updated for the desired material.

2.2.3 Heat Transfer Enhancement and Pressure Drop

There are several methods to enhance the heat transfer in the heat exchanger but at the cost of increasing the pressure drop. There are passive and active heat transfer enhancement techniques and both can be used to maximize heat transfer. Passive techniques involve modifying geometry through extended surfaces, disturbing flow channels, coiling tubes, and through rough surface finishes. Active methods involve external power to distribute the flow including adding vibration, injection, suction, and jet impingement[1]. Fluted tubes incorporate several passive heat transfer techniques through the use of extended surface and disrupting the fluid boundary layers[6]. The only heat transfer enhancement considered in the design is the passive attributes provided through fluted tubes. These added heat transfer enhancements will cause a high pressure drop in the heat exchanger.

The higher pressure drop of a system requires additional cost for the pump or compressor operation. A significantly high pressure drop will lower system performance and overall efficiency. The system efficiency is calculated by the work done to the system compares to the energy produced from the system. Overcoming a high pressure drop will force more energy entering into the system. All heat transfer enhancement methods effect the pressure drop but will vary on severity based on the different methods used. Fluted tubes have been known to increase the friction factor and overall pressure drop by a conservative amount. There are several factors in heat exchanger design that will influence the overall pressure drop, such as tube length, friction factor, fluid $\frac{m}{s}$ (V) and flow area. All of these will influence the pressure drop differently. This fundamental ideal is important when dealing with the optimization parameter of maximizing heat transfer while minimizing the pressure drop.

2.2.4 Optimization Parameters

The optimal design of a heat exchanger will change depending on the desired heat exchanger quality. The general areas of optimization are maximizing the overall heat transfer coefficient, minimizing shell diameter, minimizing tube length, or minimizing pressure drop. Instead of optimizing for a specific parameter, each optimization parameters will be filtered from a general parametric studies that varies the geometric and tube flow parameters. This method provides different designs for each optimization parameter and will also allow the user to find general trends on the heat exchanger performance. For example, maximizing the overall heat transfer coefficient will generally gives the smallest heat transfer area required but at the cost of a high pressure drop. Restricting the highest pressure drop allowed in the system may limit the results for the best overall heat transfer coefficient calculated.

2.3 Fluted Tube Geometry

The geometry of fluted tubes analyzed have outside protruding flutes on the tube and an uneven surface on the inside as shown in Figure 2.3 and 2.4. The extended surfaces on the outside of the tube act as fins and are spiralled throughout the length of the tube. These extrusions are referred to as flutes defining the tube as a fluted tube. The number of rotations down the tube is dependent on the flutes spiral angle determined in the tube design. The inside has a disturbed surface resulted from the flutes spiraled throughout the tube (This is significant difference between fluted tubes and finned tubes). This geometry effects several aspect of heat transfer compared to a straight tube. The extended surface increases the convective heat transfer surface. The geometry also effects the flow parameters through both inside and outside of the fluted tubes. Both of these features act as a passive method to increase heat transfer.

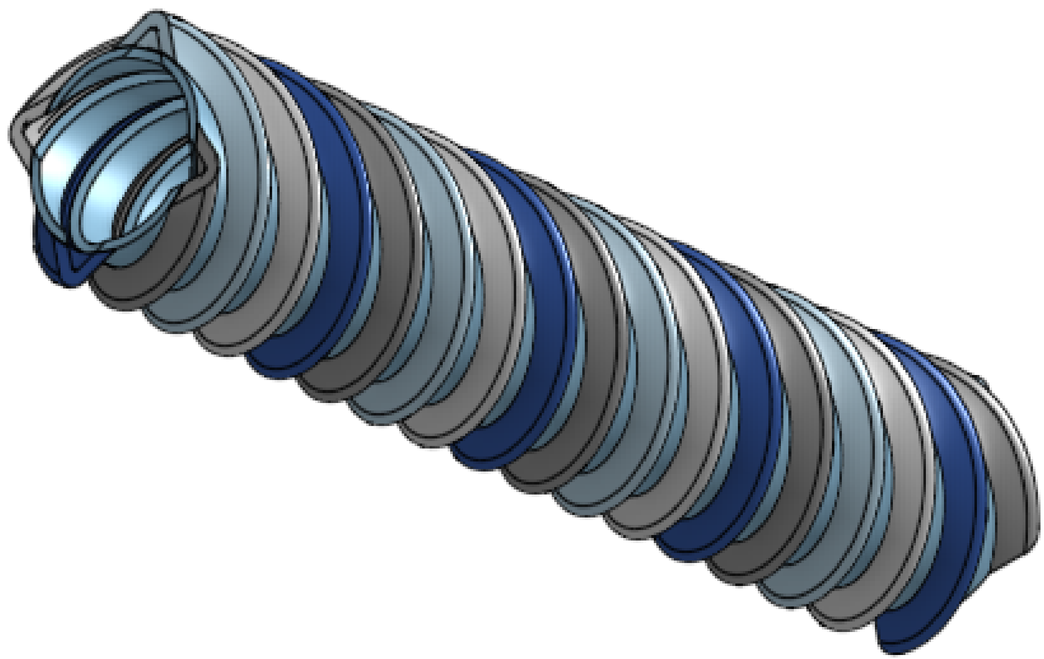


Figure 2.3: Fluted Tube

There are several different variations of this style of fluted tubes. The main parameters that are varied between different tubes are as follows Number of Flute Starts (Ns), m (D_{bo}), m (TL), m (D_{ei}), m (D_{bi}), etc. Each variation of these tubes have a distinct characteristic that will effect the overall heat transfer coefficient and pressure drop based of its geometry and surface area. Understanding the geometry of the fluted tube is essential to find the effective surface area needed for the correlation. There are a few key parameters needed to be calculated to find the effective heat transfer area. These geometric parameters have been established from [4]. The original list of fluted tubes measured and tested can be found in Appendix B.

The calculated surface areas for fluted tubes are separated between the outside and inside surface area. There are measured dimensions of the fluted tube that are used to calculated the effective surface area. The m (D_{vi}) is the effective heat transfer surface area calculated for the inside of the tube. The calculated D_{vi} is different for each variation of the tube and needs to be calculated before use in the optimization script.

The effective heat transfer surface area incorporates the different measured dimensions of the fluted

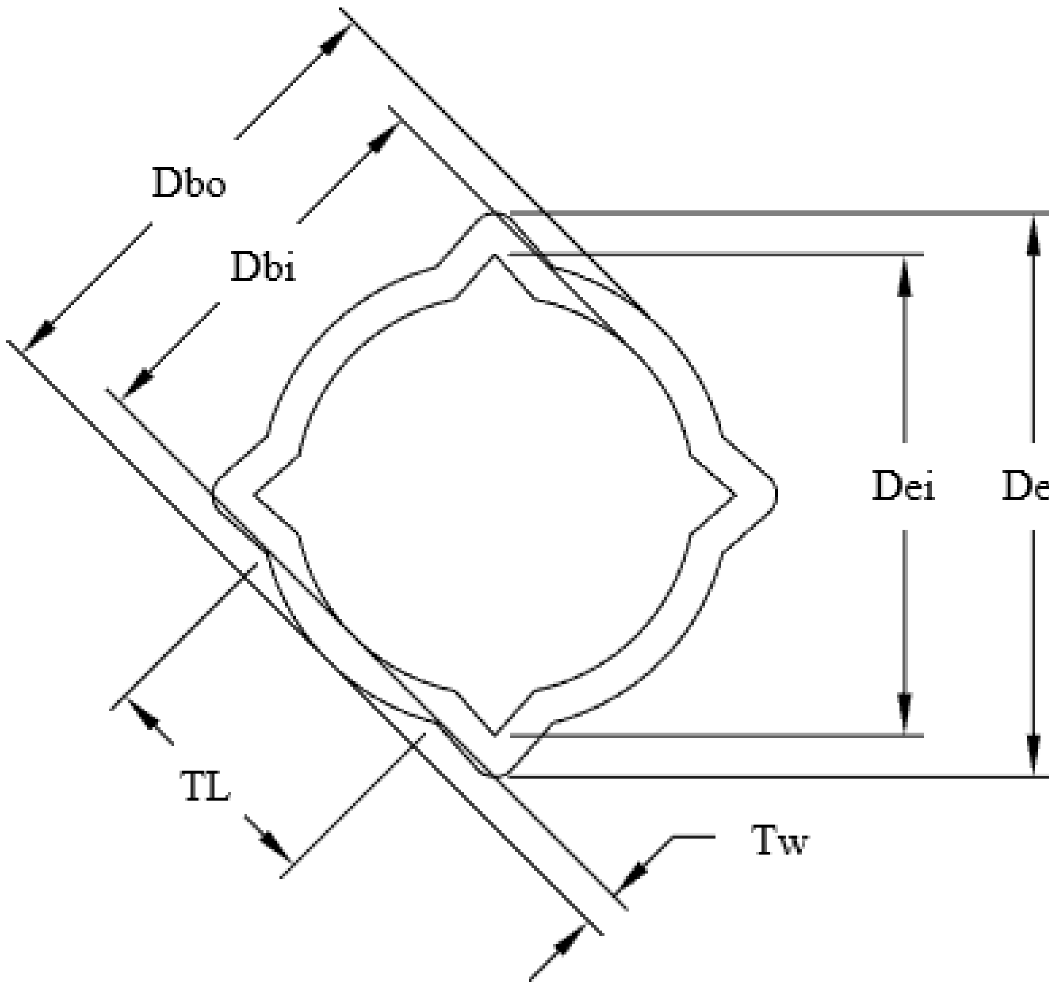


Figure 2.4: Fluted Tube Dimensions

tubes. This begins with the Trough-to-Circumference (r) which is used to understand the ratio of area not occupied by the flutes to the inner diameter. The m (FB) calculates the effective surface area for each flute.

$$\mathrm{r} = \frac{\mathrm{TL}}{\pi \cdot D_{bi}} \tag{2.1}$$

$$\mathrm{FB} = \frac{(1 - \mathrm{r}) \cdot \pi \cdot D_{bi}}{\mathrm{Ns}} \tag{2.2}$$

The volumetric diameter of the tube is calculated by using the defined r, FB and other measured dimensions. The D_{vi} incorporates the fluted geometry by relating the area taken by the flutes to a standard tube when calculating effective surface area. The effective surface area on the outside area is directly related to the inside. Instead of incorporating the geometry of the flutes to calculate the outside volumetric diameter, a relationship between the D_{vi} and the m (D_{vo}) is used. This relationship relates

D_{vi} and D_{vo} by adding the m (Tw) to the inside diameter

$$D_{vi} = \sqrt{{D_{bi}}^2 + \frac{\mathrm{Ns} \cdot (D_{ei} - D_{bi}) \cdot \mathrm{FB}}{\pi}} \tag{2.3}$$

$$D_{vo} = D_{vi} + 2 \cdot \mathrm{Tw} \tag{2.4}$$

The inside and outside fluted cross sectional areas are needed to calculate the effective heat transfer surface area across the length of the tube. The m (e) and Degree (θ) are parameters needed to understand the flow area across the length of the tube. Calculating the e and θ incorporates a number of different calculated and measured geometries. The height of the flute looks at the relation between the m (D_{eo}), D_{bi}, and Tw. This relation calculates e at the distance from the top height of the flute to the base of the outside diameter of the tube.

$$\mathrm{e} = \frac{D_{eo} - (D_{bi} + 2 \cdot \mathrm{Tw})}{2} \tag{2.5}$$

There are two different pitches referenced throughout the paper. The m (P_{tube}) and the m (p). The $P_{tube}$is the distance between each flute down the length of the tube. The parameter is important in calculating the surface area of the tube. The p is the pitch between each tube looking at the tube layout across the entire heat exchanger. The θ is the angle at which the flute rotates down the length of the tube. The higher angle θ results in a tighter rotation and a lower p value. A lower θ results in a larger rotation down the tube and a larger p value.

$$\theta = arctan(\frac{\pi \cdot D_{vo}}{\mathrm{Ns} \cdot p}) \tag{2.6}$$

These make up the dimensional geometry needed for the correlations used to calculate the heat transfer and pressure drop across fluted tubes. There are also non-dimensional parameters that related to the fluted parameters that are used in the correlations that need to be defined.

$$e^* = \frac{\mathrm{e}}{D_{vi}} \tag{2.7}$$

$$P_{tube}{}^* = \frac{P_{tube}}{D_{vi}} \tag{2.8}$$

$$\theta^* = \frac{\theta}{90} \tag{2.9}$$

All the dimensional and non-dimension dimensions and parameters are key to calculating the correlations used for heat transfer. A comprehensive list of different variation of fluted tubes originally test can be found in the Appendix B and Appendix A includes all measured and calculated dimensions ready to be used.

2.3.1 Tube Layout and Shell Flow Parameters

A common design option for shell and tube heat exchangers is the layout of the tubes. The spacing between tubes is important when sizing the heat exchanger and determining the flow parameters. Changing the p will alter the flow parameters through the shell such as velocity and Reynolds Number (Re). These parameters are dependent on these parameters and will directly effect the pressure drop, mass flow rate, and heat transfer through the system. The tube layout and pitch determines the shell side heat transfer because it is all dependent on m^2 (A_{eff}), m (D_{hyd}) and Radius Ratio Between Shell and Tube (R^*). These are the key geometry parameters that represent the flow area on the outside of an individual tube.

These variables are dependent on the m (D_{oi}). The D_{oi} represents the diameter used to calculate a single tube and shell cross sectional flow area and is directly related to the pitch and the tube orientation.

There are two common tube layouts that will be considered through this analysis and generally covers the standard tube layouts. The layout of the tubes will influence the pitch that will influence the pitch. The first is each tube is spaced at a 45° and 90° degrees forming a square outline and the second is a separation of 30° and 60° degrees forming a diamond outline.

The 45° and 90° orientation is shown in Figure 2.5 where the tube pitch is from the center of one tube to another one and creates a square between four tubes. In this orientation, the D_{oi} and pitch are related through the following equations.

$$D_{oi} = p\sqrt{\frac{4}{\pi}} \tag{2.10}$$

The control volume in the square orientation is know to be a simple design. The tube spacing does not minimize the space within the shell due to physical constraints. The diamond shape shown in Figure 2.6 allows for more of the central space to be used creating a channels for the shell flow. The geometry orientation assumes a constant pitch between each tubes in the layout so that all flow areas are the same. This allows the calculations for a single flow area to apply for all flow channels in the heat exchanger. Calculating the D_{oi} for the diamond orientation will incorporate the different angle of spacing than the square orientation. The angle allows for a tighter distance between the tube, but it should be noted that the pitch between all tubes remain constant.

$$D_{oi} = p\sqrt{\frac{2}{\pi} \cdot 3^{\frac{1}{2}}} \tag{2.11}$$

The pitch can be varied to determine a different D_{oi} for both tube orientation. The changing of the pitch will changes the A_{eff}. The D_{oi} is one of the main parameters varied in the optimization script and is done through a developed non dimensional scaling factors addressed further in the paper. All of these variables are depended on the D_{oi} to describe the geometric attributes of the fluted tube and shell that will be used in the correlation for heat transfer and pressure drop. The equations describe the A_{eff}, and D_{hyd} relationships to the geometry of the tube.

$$D_{hyd} = D_{oi} - D_{vo} \tag{2.12}$$

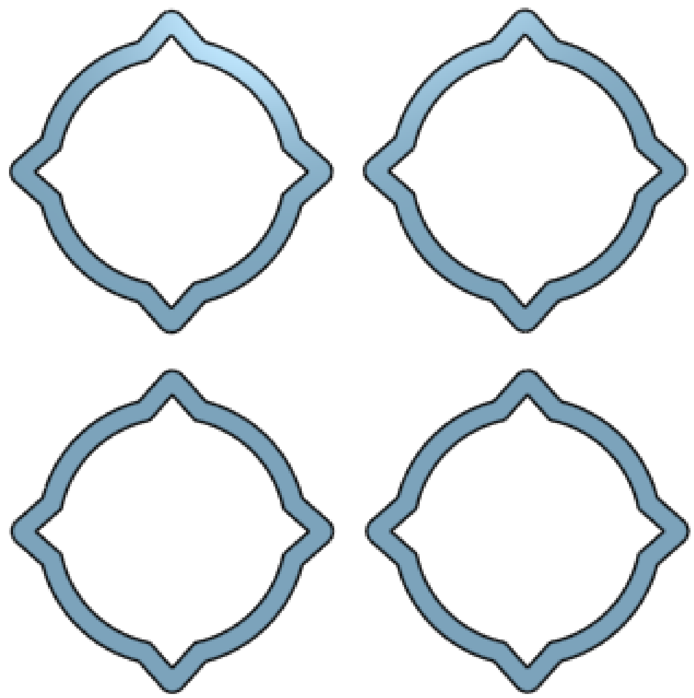


Figure 2.5: Tube layout at 45° and 90° in a heat exchanger known as a Square Orientation

$$A_{eff} = \frac{\pi}{4}({D_{oi}}^2 - {D_{vo}}^2) \tag{2.13}$$

$$R^* = \frac{D_{vo}}{D_{oi}} \tag{2.14}$$

These are all of the parameters needed to correctly define the surface area of the fluted tube. These variables will be used to calculate the overall heat transfer coefficient through the derived correlations.

2.3.2 Single Phase Fluted Tube Correlation

The fluted tube heat transfer and pressure drop correlations used for single phase heat exchanger are taken from experimental work[7]. This work was done by gathering experimental for different variations of the fluted tube found in Appendix B. The correlations were developed for counter current flow using the geometry specified for each tube, $\frac{kg}{s}$ ($\dot{m}$) and average temperatures thermal fluid properties.

Since the geometry and heat transfer surface have been defined, the thermal fluid properties are needed to accurately use the correlations. The fluid properties are taken by averaging the inlet and outlet temperatures of the fluid. The properties needed for the correlations $\frac{J}{kg\,{}^\circ K}$ (Cp), $\frac{kg}{m^3}$ (ρ), $\frac{N-sec}{m^2}$ (μ), and $\frac{W}{m\,{}^\circ K}$ (k). These properties are needed for both the tube and shell side. The thermal properties are used to calculate the Prandtl Number (Pr). The same equation is used to calculate the Pr for both sides of the heat exchanger because it relates to the thermal properties of the fluid, not the geometry of the tubes. The average temperature approach is only used if the fluid properties do not

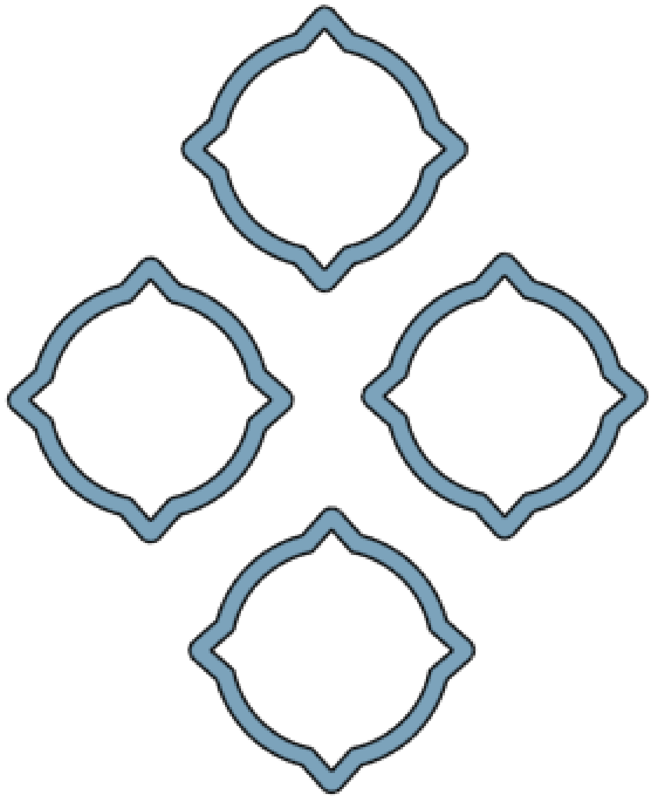


Figure 2.6: Tube layout at 30° and 60° in a heat exchanger known as a Diamond Orientation

significantly vary. If they do, a segmented approach will be needed to calculate the fluid properties.

$$\mathrm{Pr} = \frac{\mathrm{Cp} \cdot \mu}{\mathrm{k}} \tag{2.15}$$

The different geometry, flow parameters, and different fluid properties require two different sets of equations in the correlation used for each side of the heat exchanger. The first set of equations describe the overall heat transfer coefficient for the inside of the tube. The second set of equations will calculate the overall heat transfer coefficient for the fluid flow on the outside of the fluted tube. Both correlations will use the developed effective surface area defined previously. The flow parameters are calculated by individual tube assuming even distribution across the heat exchanger and the velocity at which the fluid is moving through the pipe.

Starting with the tube side of the correlations, the flow parameters needs to be calculated. The tube velocity and Re will characterize the per tube fluid flow.

$$\mathrm{V} = \frac{4 \cdot \dot{m}}{\Pi \cdot \rho \cdot {D_{vi}}^2} \tag{2.16}$$

$$\mathrm{Re} = \frac{\mathrm{V} \cdot \rho \cdot D_{vi}}{\mu} \tag{2.17}$$

The Reynolds number characterizes the flow regime and will determine which equations to use throughout the correlations. The Re will characterize the flow as laminar, transition or turbulent. This determines which Nusselt Number (Nu) correlation correctly reflects the heat transfer of the fluid and flow.

$$\mathrm{Nu} = 0.014\mathrm{Re}^{0.842} \cdot \mathrm{e}^{*-0.067} \cdot p^{*-0.293} \cdot \theta^{*-0.705} \cdot \mathrm{Pr}^{0.4} \tag{2.18}$$

$$500 \leq \mathrm{Re} \leq 5000$$

$$\mathrm{Nu} = 0.064\mathrm{Re}^{0.773} \cdot \mathrm{e}^{*-0.242} \cdot p^{*-0.108} \cdot \theta^{*0.599} \cdot \mathrm{Pr}^{0.4} \tag{2.19}$$

$$5000 \leq \mathrm{Re} \leq 80,000$$

The correlation used were tested and derived under specific conditions and for a general range of fluted tube geometries. All the tubes used to derive the correlations are on the list in the Appendix A which developed a non dimensional parameter range that this set of correlation is valid. The correlations are valid for the following non-dimensional ranges:

$$0.11 \leq \mathrm{e}^* \leq 0.42$$

$$0.41 \leq p^* \leq 7.29$$

$$0.28 \leq \theta^* \leq 0.65$$

$$2.5 \leq \mathrm{Pr}^* \leq 7.0$$

The Friction Factor (f) is used to calculate the pressure drop through the heat exchanger. This is also dependent on the flow characterization defined from the Reynolds number.

$$f = \frac{64}{\mathrm{Re} - 45.0}(0.554\mathrm{e}^{*0.384} \cdot p^{*(-1.454+2.083e^*)} \cdot \theta^{*-2.426}) \tag{2.20}$$

$$100 \leq \mathrm{Re} \leq 1,500$$

$$f = 1.209\mathrm{Re}^{-0.261} e^{*(1.26-0.050p^*)} \cdot p^{*-1.660+2.033e^*} \cdot \theta^{*-2.699+3.670e^*} \tag{2.21}$$

$$\mathrm{Re} \geq 3,000$$

Note that there is a gap in the Reynolds number when calculating the friction factor

$$1,500 \leq \text{Re} \leq 3,000$$

Linear interpolation is required for any Reynolds number between the values calculated in equation 2.20 and equation 2.21. There are also limits on the geometric limits of the tubes have been tested.

$$0.11 \leq \text{e}^{*} \leq 0.42$$

$$0.41 \leq p^{*} \leq 7.29$$

$$0.28 \leq \theta^{*} \leq 0.65$$

The heat transfer coefficient and Pa (ΔP_l) for the tube side of the heat exchanger are calculated through the following equations. The pressure drop is dependent on the overall m (L).

$$h_i = \frac{\text{Nu} \cdot \text{k}}{D_{vi}} \tag{2.22}$$

$$\Delta P_l = f\frac{\text{L} \cdot \rho \cdot \text{V}^2}{D_{vi} \cdot 2} \tag{2.23}$$

These are all the equations used to calculate the heat transfer and and pressure drop through the tube side of the heat exchanger. The shell side of the heat exchanger equations change due to geometry differences and flow characteristics.

$$\text{V} = \frac{\dot{m}}{\rho \cdot A_{eff}} \tag{2.24}$$

$$\text{Re} = \frac{\text{V} \cdot \rho \cdot D_{hyd}}{\mu} \tag{2.25}$$

An important difference in the shell correlations is the influence of the friction factor in the Nu. This requires the friction factor to be calculated first.

$$f = \frac{96 \cdot R^{*0.035}}{\text{Re}}[1 + 101.7 \cdot \text{Re}^{0.52} \cdot \text{e}^{*1.65+2.00\theta^{*}} \cdot R^{*5.77}] \tag{2.26}$$

$$\text{Re} \leq 800$$

$$f = 4[1.7372 ln(\frac{\text{Re}}{1.964 \cdot ln(\text{Re}) - 3.8215})]^{-2}(1 + 0.0925 * R^{*}) \cdot (1 + 222\text{Re}^{0.09}\text{e}^{*2.40}p^{*-0.49}\theta^{*-0.38}R^{*2.22}) \tag{2.27}$$

$$800 \leq \text{Re} \leq 40,000$$

Although there are two friction factor equations for different ranging Re, the Nu correlation will need to come from the high Reynolds number calculated from equation 2.27

$$\mathrm{Nu} = [\frac{\frac{f}{8} \cdot \mathrm{Re} \cdot \mathrm{Pr}}{1 + 9.77\sqrt{\frac{f}{8}}(\mathrm{Pr}^{2/3} - 1)}](\mathrm{Re}^{-0.20}\mathrm{e}^{*-0.32}p^{*-0.28}R^{*-1.64}) \tag{2.28}$$

$$700 \leq \mathrm{Re} \leq 40,000$$

The shell side heat transfer coefficient and pressure drop follow the same equations as the tube side with a small variation from the D_{vi} to D_{hyd}.

$$h_o = \frac{\mathrm{Nu} \cdot \mathrm{k}}{D_{hyd}} \tag{2.29}$$

$$\Delta P_l = f\frac{\mathrm{L} \cdot \rho \cdot \mathrm{V}^2}{D_{hyd} \cdot 2} \tag{2.30}$$

The $\frac{W}{m^2\,{}^{\circ}K}$ (U) is calculated to find the size of the heat exchanger by finding the m^2 (A) required for the given load. This coefficient incorporates the conductive heat transfer through the fluted tube requiring the use of the $\frac{W}{m\,{}^{\circ}K}$ (K_{wall}) and the convective heat transfer through the use of h_i and h_o. The total area is calculated by using the ${}^{\circ}K$ (ΔT_{lm}), heat duty and the overall heat transfer coefficient.

$$\mathrm{U} = \frac{1}{(\frac{D_{vo}}{D_{vi}})\frac{1}{h_i} + \frac{D_{vo}}{2 \cdot K_{wall}}ln(\frac{D_{vo}}{D_{vi}}) + \frac{1}{h_o}} \tag{2.31}$$

$$\mathrm{A} = \frac{Q}{\mathrm{U} \cdot \Delta T_{lm}} \tag{2.32}$$

$$\mathrm{L}_t = \frac{\mathrm{A}}{\pi \cdot D_{vo}} \tag{2.33}$$

Where L_t is the total length of the tubes combined. The L is calculated by taking the total length divided by the number of tubes. Taking the tube to tube pitch, number of tubes, and tube length will size the heat exchanger

2.3.3 Optimization correlation

The optimization tool developed is designed to run a parametric study by changing the fluted tube style, tube to tube pitch and tube length for any specific thermal load. This is a sizing tool used to find the optimal design through the specified parameters. Each variation of all these parameter will result in different overall heat transfer coefficient and pressure drops. The script then compares the results to find the optimal design. In addition to finding the optimal design, the data can be used to find trends and behaviors through analyzing and plotting the performance data.

Due to the different sizes of fluted tube, comparing tubes against each other requires constant scaling parameters to ensure the same ratios are compared. This is the main challenge in comparing different fluted tube. Comparing a single pitch across different tube sizes can drastically change the mass flow rate and heat transfer. Utilizing a non dimensional tube to tube pitch (p^*) will ensure that there is scaled distances between different tubes.

$$p^* = \frac{p}{D_{eo}} \tag{2.34}$$

The non-dimensional pitch will relate the actual pitch and flow area of the fluted tube. The pitch in relationship with D_{eo} will allow for consistent dimensional scaling to relate to previous pitch boundary condition. The smallest pitch is set to D_{eo} to ensure enough spacing between tubes. This relationship shows that the smallest non dimensional pitch is one within the specified boundary condition. Although scaling pitches allows for an improved comparison model, a scaling of flow areas is also incorporated to ensure the flow area ratios are scaled as well.

As the pitch is related to the outside area of the tube, a non-dimensional parameter has been created to scale the inside and outside tube flow areas. This relation is used to understand scaling ratio between the two areas.

$$A^* = \frac{A_{eff}}{{D_{vi}}^2 * \frac{\pi}{4}} \tag{2.35}$$

It has been previously shown that the pitch is directly related to the A_{eff}. This relationship signifies that the non-dimensional pitch and the non-dimensional area are also both related to each other. The non-dimensional pitch and non-dimensional area relationship can be derived for the two different tube layouts.

When the non-dimensional area scaling factor is to a value of one, this signifies the flow area inside of the tube is equal to the flow area on the outside of the tube. This shows that the ratio of the shell and tube areas are connected to the pitch. The non-dimensional area parameter does not relate to any boundary conditions, but does show the difference of flow areas.

$$p^*_{square} = \frac{(\frac{\pi}{4} \cdot {D_{vi}}^2 \cdot A^* + \frac{\pi}{4} {D_{vo}}^2)^{\frac{1}{2}}}{D_{eo}} \tag{2.36}$$

$$p^*_{diamond} = \frac{(\frac{2}{3^{\frac{1}{2}}} \frac{\pi}{4} \cdot {D_{vi}}^2 \cdot A^* + \frac{\pi}{4} {D_{vo}}^2)^{\frac{1}{2}}}{D_{eo}} \tag{2.37}$$

This relationship will force the decision of which non dimensional scaling parameter should be driven. A range for either the A^* or the p^* can be used to scale different sizes of fluted tubes against each other. As the user defines to scale either A^* or the p^*, this will cause a change in the A_{eff} at each step. These changes will influence all the flow parameters be either restricting or loosening the flow area. This area determines the Reynolds number and the flow characterization and ultimately the overall heat transfer coefficient and pressure drop.

Tube length is a major factor with pressure drop and mass flow rate through the entire heat exchanger. The longer the tube the fewer number of tubes are required to meet the thermal load. The less tubes available with a constant mass flow rate in the system would cause a larger mass flow rate per tube.

Any two of the following parameters need to be specified by the user to run the optimization tool; tube length, mass flow rate per tube, or number of tubes.

2.3.4 Conservation of Energy

The heat exchanger designed during optimization must ensure that energy is conserved. The non-dimensional area relationship derived for optimization directly relates to the heat transfer on each side of the heat exchanger. The heat transfer properties and flow area must be balanced with the cold and hot side of the heat exchanger.

$$\dot{Q}_H = \dot{m}_H C_{p,H} \Delta T_H \tag{2.38}$$

$$\dot{Q}_C = \dot{m}_C C_{p,C} \Delta T_C \tag{2.39}$$

$$\dot{m}_C C_{p,C} \Delta T_C = \dot{m}_H C_{p,H} \Delta T_H \tag{2.40}$$

Steady state conditions that have a defined temperature change will mandate that the tube mass flow rate has the following relationship.

$$\frac{\dot{m}_H}{\dot{m}_C} = \frac{C_{p,C} \Delta T_C}{C_{p,H} \Delta T_H} \Rightarrow \frac{\dot{m}_H}{\dot{m}_C} = \frac{\rho_H A_H V_H}{\rho_C A_C V_C} \Rightarrow m^* = \rho^* A^* V^* \tag{2.41}$$

The flow area relationship defined by A^* in the optimization script will determine the tube mass flow rate ratio to conserve energy. Changing the non-dimensional area and mass flow rate will maintain the velocity ratio for the tubes during the optimization process. This relationship allows for a direct comparison of different tubes for the specified temperatures and fluids of the system.

Chapter 3: Method and Procedure

3.1 Optimization

To find an optimal solution of the size and performance of the fluted tube heat exchanger, a coding script is written to iterate through different parameters. The script designed will calculate the heat exchanger size needed for each changing parameter to operate at the desired performance. This will find the best performing heat exchanger within the design specifications.

The optimization script was developed and is scripted into two different softwares. The script is written into MATLAB ® and Aspen HYSYS. In general, the best performing heat exhanger is the one with the best heat transfer per surface area required with the minimal amount of pressure drop. There are different optimization parameters that the code calculate such as shell diameter or pressure drop. In order to understand the capability and limitation of the programs, all the assumptions and user defined variables are specified.

3.1.1 Boundary Conditions

There are boundary conditions and assumptions set to ensure accurate design and results. These design limitations are there to ensure the correlation are used within tolerances of their test parameters and that realists design are being calculated. Although the correlation does not specify a p limitation, the smallest p that the script allows is the D_{eo} to ensure there is no interference between tubes. Although there are fluted tubes that would allow nesting of the flutes to achieve a smaller pitch, there is not a consistent ratio that remain true for all cases and to maintain simplicity and consistency for comparing different style of fluted tubes. As shown in Figure 3.1, the single fluted tube can be placed in a tighter formation compared to the four fluted tube.

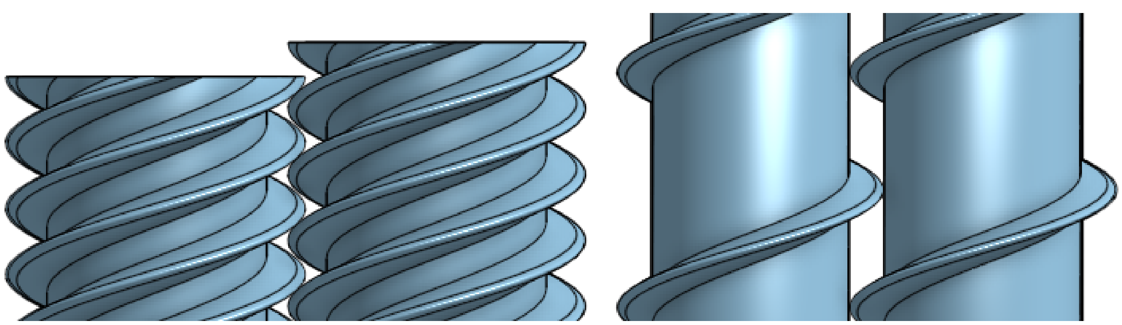


Figure 3.1: Comparison of a single flute tube spacing and a four fluted tube spacing

The correlations were developed under a specific range of fluted tubes design parameters. There is a list of 55 different style of tubes in the appendix that all fall within the fluted tube correlations specifications. It is possible to add additional tubes to the list, but all calculations to determine the D_{vi}, D_{vo} and non dimensional parameters will need to be done and added to the program by the user. The equations and specific measurements needed to calculate the tube geometry are defined in Equations 2.1 to 2.9. Independent verification is required to ensure the new tubes fall within the correlation specifications identified. There are a few parameters on the shell side that deal with the pitch and D_{hyd} that are checked in the program and will notify the user in the results if the calculations are within the correlations.

3.1.2 Assumptions and Limitations

There are general assumptions made for calculating the heat transfer coefficient and pressure drop that may effect the performance of an actual heat exchanger. These assumptions are placed to simplify the optimization code and to ensure the comparison between fluted tubes is the main focus of the script.

The flow characterization throughout the heat exchanger will influence the overall performance. The flow distribution is assumed to be even throughout the entire heat exchangers. This will ensure the mass flow rate per tube will remain constant for all tubes. Larger heat exchanger often require baffles to ensure distribution of flow and heat transfer is happening throughout the heat exchanger. Baffles are not incorporated into these calculations for simplicity because there is no specification on what situations will need the baffles. The script calculates the size of the heat exchanger based on the thermal load it needs to transfer, thus there will be a large variation of sizes calculated based on different thermal loads. The larger heat exchangers, in theory, may need to be placed in a vertical orientation or use baffles to ensure even distribution of the flow.

Along with the flow remaining constant and evenly distributed throughout the heat exchanger, the tube flow area and shell flow areas remain constant. The tubes are assumed to be manufactured to perfection without any deviation from the given geometric dimensions. The size of the tubes are relatively small and slight manufacturing defects can effect the area significantly. This also includes the tube spacer sheets used in the shell side. The tube spacers are what control the pitch spacing in the shell and hold the tubes steadying during operations. The number of tubes spacer sheets needed depends in the size of the heat exchanger, but tube spacers in the middle of the heat exchanger would restrict the flow. These are not incorporated in the calculations because it can be assumed that the tubes need to be non-fluted in the ares to be attached to the tube sheet there that the extra area will act as the fluted dimensions in these sections.

When looking at the pressure loss in the system, the only pressure drop considered is from the flow length through the fluted tube and shell. There are other pressure drops in the heat exchanger that are not considered. Entrance and exit pressure losses are not considered in the script. It should be noted that although small length tubes generally will have smaller pressure drops as shown in Equation 2.23 and 2.30 that the equations for pressure drop depend on the length of the tube. When optimizing the pressure drop, shorter length does reduce pressure drop through the tube, but would increase the pressure drop at the entering and exiting locations of the heat exhanger due to expansion and compression of the fluid and minor loss.

There are several factors that influence the effectiveness of a heat exhanger over time. Dealing with problems that involve time such as fouling or oxidation effects are not considered. It is noted that one of the most common issues with heat exchangers is fouling which can especially dangerous with fluted tubes because of more narrow channels and areas for particle buildup. These effects can cause a decrease in heat transfer and overall performance of the heat exchanger. Additionally the system is assumed to be perfectly insulated and that there is no heat loss outside of the heat exchanger. The effects of fouling oxidation and heat loss are not addressed in the script. The optimization is done under the assumption of standard operation.

3.2 Programming

The script is programmed to calculate the overall heat transfer coefficient and the pressure drop for a specific length, pitch and mass flow rate for each tube. It then compares each of the tubes against each other at the same non-dimensional area or pitch to find the optimal design. The script has been developed through two different softwares, each with different user inputs and distinct advantages and disadvantages.

3.2.1 MATLAB® Code

MATLAB® is a computation tool that incorporates a various number of programming functions to accomplish complicated parametric studies. There are programming capabilities that allow for variable definition, iteration through loops and plotting data. It also has a GUI interface that allows for an independent program to be established with a user interface and the optimization script contained in the background.

The MATLAB® code has been developed for single fluid intermediate heat exchangers. This code does not handle a heat exchanged between two different fluids, rather it is built so that the fluid and overall mass flow rate are the same throughout the loop. The code only handles recuperative heat exchanger that keep the same fluid for each side of the heat exchanger as shown in Figure 2.1. The code is developed to compare three different fluids against each other for the same system. Instead of two different fluids in the same loop, the code will compare the same system with three different operational fluids. The general restrictions is that the fluid needs to be single phase throughout the heat exchanger and they need to operate at similar mass flow rates and temperatures.

3.2.1.1 MATLAB® Layout

The GUI interface allows the user to define temperatures and thermal load for the system. The focus of the GUI is for an easy user experience and should be clear on what the user needs to specify. The interface is broken into three different sections each with a distinct purpose. Each interface is placed in the order that the user needs to input the required information.

The first interface allows the user to change anything in the green fields to set up the system environment. This section is used to define the heat transfer specifications, temperatures, and tube conductivity. This is where the user controls the non-dimensional parameters to setup the parametric study. All fields in green are edit fields and requires the user input to setup the desired system. The MATLAB® code allows the user to specify the tube mass flow rate, tube length and area ratios.

The second interface is the tube selection part. To run the script for a single tube, a single row needs to selected and highlighted. To run for all the tubes available, there are programmed buttons to run different cases studies for all tubes explained in the third user interface section. The fluted tube

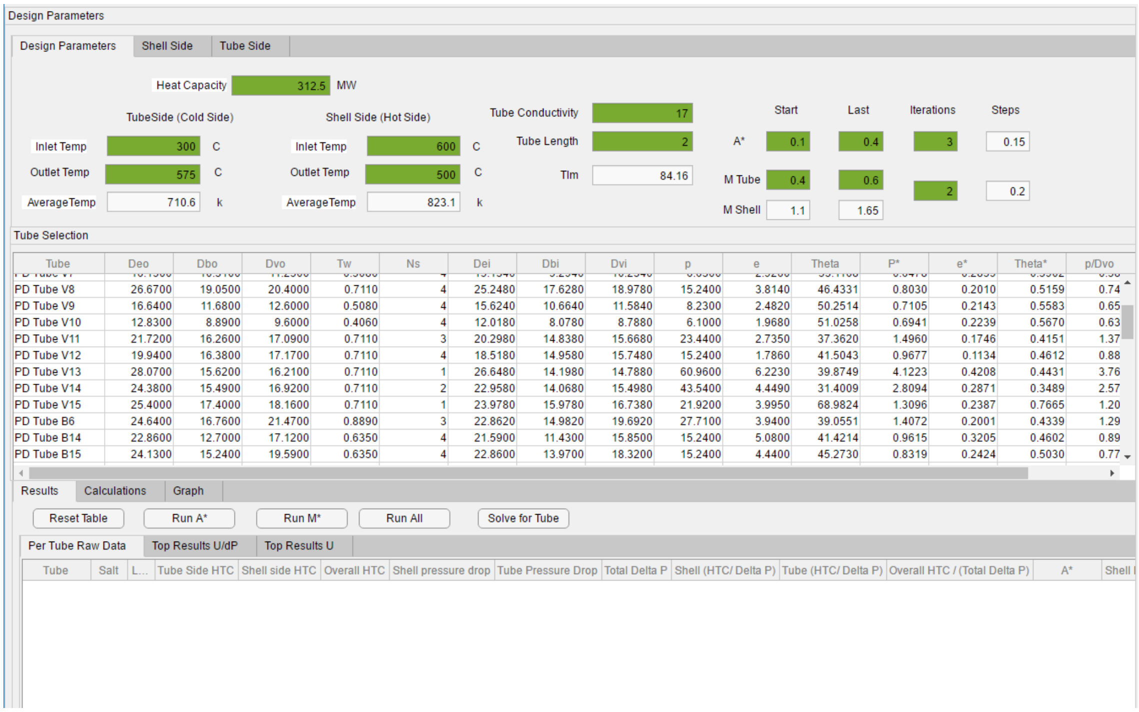


Figure 3.2: Overview of the GUI interface for the MATLAB® code

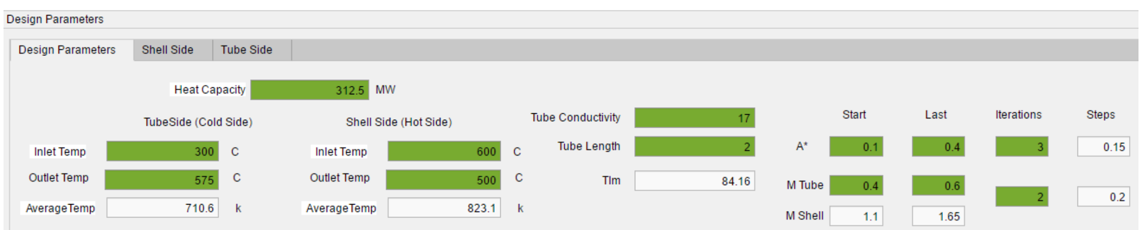


Figure 3.3: Initial Conditions Setup overview of the MATLAB® code. Green edit fields requires the user input.

geometry information is held in a separate CSV file and is imported into the code during its operation. This CSV file includes all measured and calculated geometry for each of the 55 standard fluted tubes tested. In order to add additional tube, all tube information needs to be added in the next empty row in the spreadsheet. No alteration to the code needs to be made because the program will run until it finds an empty row.

The last section of the interface is used to display the results and also holds the operational buttons used to run the script. There are several parametric studies the user is allowed to run, each specified by the different buttons. The user can run the following studies; varying the pitch for all tubes, the tube mass flow rate, or a combination of them both. The solve for tube button allows the user to size the heat exchanger at all the initial conditions for a single tube selected. The results of all tubes are displayed on the table below the control buttons. The results for different parametric studies will stack in the results section until the user selects the reset table button. There are different optimization results displayed on

Tube Selection

Tube	Deo	Dbo	Dvo	Tw	Ns	Dei	Dbi	Dvi	p	e	Theta	P*	e*	Theta*	p/Dvo	e/Dvo
PD Tube VI	18.8000	11.2300	12.6000	0.5080	3	17.7840	10.2140	11.5840	13.2600	3.7920	44.8585	1.1447	0.3273	0.4984	1.0524	0.3004
PD Tube V2	19.8100	15.4900	16.2300	0.5080	5	18.7940	14.4740	15.2140	6.2200	2.1590	58.6190	0.4088	0.1419	0.6513	0.3832	0.1331
PD Tube V4	21.0800	13.9700	14.7600	0.5080	3	20.0640	12.9540	13.7440	13.2600	3.5570	49.3742	0.9648	0.2588	0.5486	0.8984	0.2409
PD Tube V5	14.2200	9.2700	10.2600	0.4060	3	13.4080	8.4580	9.4480	13.2600	2.4830	39.0170	1.4035	0.2628	0.4335	1.2924	0.2412
PD Tube V6	17.1500	11.1300	11.9900	0.5080	3	16.1340	10.1140	10.9740	11.7300	3.0090	46.9477	1.0689	0.2742	0.5216	0.9783	0.2510
PD Tube V7	16.1500	10.3100	11.2500	0.5080	4	15.1340	9.2940	10.2340	6.6300	2.9260	53.1168	0.6478	0.2859	0.5902	0.5893	0.2596
PD Tube V8	26.6700	19.0500	20.4000	0.7110	4	25.2480	17.6280	18.9780	15.2400	3.8140	46.4331	0.8030	0.2010	0.5159	0.7471	0.1868
PD Tube V9	16.6400	11.6800	12.6000	0.5080	4	15.6240	10.6640	11.5840	8.2300	2.4820	50.2514	0.7105	0.2143	0.5583	0.6532	0.1968
PD Tube V10	12.8300	8.8900	9.6000	0.4060	4	12.0180	8.0780	8.7880	6.1000	1.9680	51.0258	0.6941	0.2239	0.5670	0.6354	0.2052
PD Tube V11	21.7200	16.2600	17.0900	0.7110	3	20.2980	14.8380	15.6680	23.4400	2.7350	37.3620	1.4960	0.1746	0.4151	1.3716	0.1597

Figure 3.4: Tube selection and table population for the user interface for the MATLAB® code

separate tabs in the results section. In each of these tables, the top five sizes for the specific optimization areas are displayed. These tables are designed for a quick reference of the best tubes but additional filtering may need to be done to find the best design. To properly process the data, export the data as a CSV. Copy the imported data to a specific filtration spreadsheets used to plot the results data to find trends and see improved optimization tables.

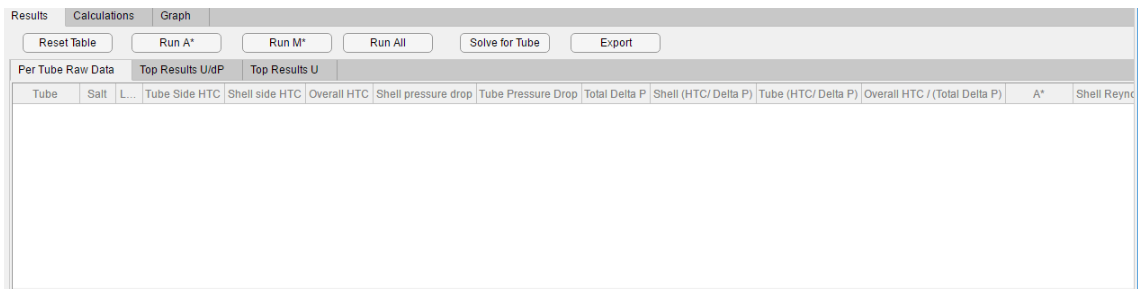


Figure 3.5: User controlled buttons and results section of the MATLAB® code

3.2.1.2 Thermal Properties

The fluid thermal properties are unique in this script and take significant effort to change the script to different fluids. The script was first created to compare different molten salts against each other in a fluted tube heat exchanger to find an optimal design against tubes and fluids. The script uses thermal properties equations to calculate the thermal properties of each fluid which are dependent on the average fluid temperature. The script can be adjusted to run any fluid and even compare the design with different fluid against each other but the thermal property equations will need to be update in the code. The user would have to pull the thermal property temperature equations and update the current equations for the desired properties. Than the user will actual make changes to the code behind the GUI interface. For example, a comparison of different sub-cooled water can be compared against each other. The fluid properties are calculated at the average temperatures to ensure the correlations are calculated at the same procedure as the test data was gathered. This requires that all the properties equations be calculated with the temperature as the only variable.

The program uses the average temperature to determine the fluid thermal properties. This assumes that the changes of properties is low with the variation of temperature. If the thermal properties of the fluid behavior significantly change, a segmented approach to determine the fluid properties is needed.

```
%Fluid 1` Properties Tube
app.TubeCpSalt1.Value = 0.917;
app.TubeViscositySalt1.Value = 152.368* exp(-averageTemp/56.0314)+0.05994*exp(-averageTemp/235.787)+0.00297
app.TubeDensitySalt1.Value = 2541.737 + -0.53018*averageTemp;
app.TubekSalt1.Value = (0.437196 - 0.00012*averageTemp)/1000;

%Fluid 2` Properties Tube
app.TubeCpSalt2.Value = 0.917;
app.TubeViscositySalt2.Value = 131.0731* exp(-averageTemp/62.36328)+0.00446;
app.TubeDensitySalt2.Value = 2581.09 + -0.43206*averageTemp;
app.TubekSalt2.Value = (0.38949 - 0.000082*averageTemp)/1000;

%Fluid 3` Properties Tube
app.TubeCpSalt3.Value = 0.917;
app.TubeViscositySalt3.Value = 0.12055* exp(-averageTemp/204.709)+497613*exp(-averageTemp/29.9169)+0.00341;
app.TubeDensitySalt3.Value = 2878.32 - 0.9263*averageTemp;
app.TubekSalt3.Value = (0.51447 - 0.00023*averageTemp)/1000;
```

Figure 3.6: Example of the code that controls the thermal properties. This section would need to be updated to run the script for any other fluids

3.2.1.3 Optimization Design

The parametric study is done through nested loops that iterate through the different parameters and tube variations. The code was written follows the object orientation style of programming. There are several functions that run a specific part of the code but can be quickly added or adapted to other programs if needed. This style of coding allows for new parametric studies to be created quickly and without altering any of the main code. This also allows for simple nested loops to be implemented that will not interfere with the other parts of the code when changes need to be made. This is a useful attribute to look at different optimization variables, but does require the understanding of the MATLAB® coding language.

The code will pull the top five tube and design parameters in separate tables that show the best tubes for each optimization parameter. The tables built into MATLAB® are designed as a quick reference but does not contain an in depth detailing on the optimization data. The results section of the code allows all the raw data to be exported into an Excel® spreadsheet. The spreadsheet will allow the user to define additional distinct design limitation for the data to be separated and optimized to the allowable system tolerances. This allows for more control of the desired and allowed pressure drop.

3.2.1.4 Code advantages and limitations

There are distinct advantages with the MATLAB® code. The code is robust due to the separation between user interface and the scripting code. Once the coding has been set for the desired thermal properties and optimization parameters, a comprehensive case study is ran. Even when changes are made to the code, this alteration will only run different parametric studies, the bulk of the correlation code will not need to be changed. This code also allows a comparison of the same fluids at different pressures or different fluids at the same thermal conditions. Automatic 2D and 3D plotting are implemented in the code and additional graphs can easily be added to look at different parameters and optimization with slight alteration of the code. These plots allow for a quick understanding of the heat exchanger trends across all changing variables.

Although the code is able to be changed and adapted to different fluid, the current setup makes it difficult and error prone to add fluids. This methods forces the user to have access and understanding of the code in order to update it. The code can be updated so that the user can manually entering the average temperature fluid properties. This method can be tedious for three different fluids especially if the user wants to consistently change the temperature for each study.

3.2.2 Aspen HYSYS Code

Aspen HYSYS is used to design and calculate complicated thermal and chemical processes. It is known for utilizing a vast database that gives thermal fluid properties for a large list of fluids under any condition. The script written for the HYSYS program will calculate the required heat transfer coefficient and area the heat exchanger must have to meet the thermal load. Since HYSYS is system software, the sizing of the heat exchanger will be specified after an ideal thermal cycle has been developed. The benefit of Aspen HYSYS is the user may design any cycle and the code will find the optimal size flute tube heat exchanger to match the cycle.

The user interface for the HYSYS modeling is more complicated than the previous MATLAB® code. There is no GUI interface that allows for a simple user interface, rather the user needs to define the loop and specify the parametric study within the HYSYS software. A general understanding of Aspen HYSYS is required to run the script in HYSYS. There are a few cases where a predefined loop has been established and the user will only need to update the thermal conditions. This method will allow for more users to run the script with a minimal understanding of HYSYS.

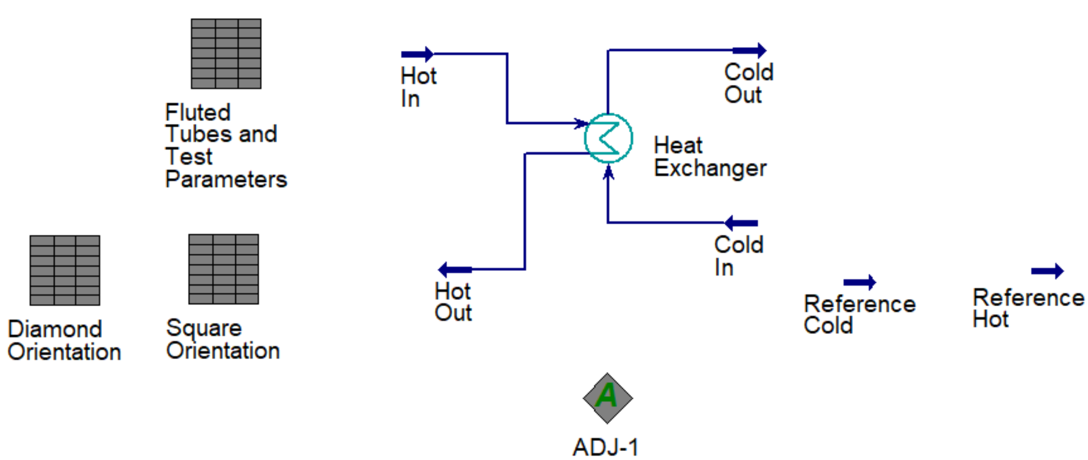


Figure 3.7: Overview of the HSYS loop design and user interface

The user input required to run the script will depend on if the thermal cycle is already setup. If the desired thermal cycle is not set, the user will need to design the custom loop in the HYSYS model. Ensure the model is solved. A solved loop is vital because the heat exhanger script cannot be operated until all streams are fully defined and the heat exchanger is error free. After the loop has been designed and the parameters set, the case study controls the parametric study to calculate the heat transfer coefficient and pressure drop for all tubes and parameters.

There is a dedicated spreadsheet to define the initial conditions including the non-dimensional parameters. This spreadsheet also contains all the information on all 55 initial tubes. In this spreadsheet, a single number associated with a tube defines which selection is in the active state for the calculations. This number starts at one and will go for as many tubes are defined in the spreadsheet. To add an addition tube for testing, all the measured and calculated dimensions need to be placed at the next empty row at the bottom of the spreadsheet. After the tube information has been added, an additional number must be added to the case study to make sure it is included in the optimization calculations.

The thermal properties are calculated using reference streams at the average temperature of the fluid per heat exchanger side. This approach is only used if the fluid properties do not significantly vary over the specified temperature range. If they do, a segmented approach will be needed to calculate the fluid properties.

Tube	Deo	Dbo	Dvo	Tw	Ns	Dei	Dbi	Dvi	p	e
PD Tube VI	18.80	11.23	12.60	0.5100	3.000	17.78	10.21	11.58	13.26	3.790
PD Tube V2	19.81	15.49	16.23	0.5100	5.000	18.79	14.47	15.21	6.220	2.160
PD Tube V4	21.08	13.97	14.76	0.5100	3.000	20.06	12.95	13.74	13.26	3.560
PD Tube V5	14.22	9.270	10.26	0.4100	3.000	13.41	8.460	9.450	13.26	2.480
PD Tube V6	17.15	11.13	11.99	0.5100	3.000	16.13	10.11	10.97	11.73	3.010
PD Tube V7	16.15	10.31	11.25	0.5100	4.000	15.13	9.290	10.23	6.630	2.930
PD Tube V8	26.67	19.05	20.40	0.7100	4.000	25.25	17.63	18.98	15.24	3.810
PD Tube V9	16.64	11.68	12.60	0.5100	4.000	15.62	10.66	11.58	8.230	2.480
PD Tube V10	12.83	8.890	9.600	0.4100	4.000	12.02	8.080	8.790	6.100	1.970
PD Tube V11	21.72	16.26	17.09	0.7100	3.000	20.30	14.84	15.67	23.44	2.740
PD Tube V12	19.94	16.38	17.17	0.7100	4.000	18.52	14.96	15.75	15.24	1.790
PD Tube V13	28.07	15.62	16.21	0.7100	1.000	26.65	14.20	14.79	60.96	6.220
PD Tube V14	24.38	15.49	16.92	0.7100	2.000	22.96	14.07	15.50	43.54	4.450
PD Tube V15	25.40	17.40	18.16	0.7100	1.000	23.98	15.98	16.74	21.92	4.000
PD Tube B6	24.64	16.76	21.47	0.8900	3.000	22.86	14.98	19.69	27.71	3.940
PD Tube B14	22.86	12.70	17.12	0.6400	4.000	21.59	11.43	15.85	15.24	5.080
PD Tube B15	24.13	15.24	19.59	0.6400	4.000	22.86	13.97	18.32	15.24	4.440
PD Tube B16	25.02	12.95	17.80	0.8900	3.000	23.24	11.17	16.02	15.24	6.030
PD Tube B17	25.02	13.08	19.36	0.8900	3.000	23.24	11.30	17.58	23.45	5.970
PD Tube B18	22.91	13.64	16.99	0.8900	4.000	21.13	11.86	15.21	9.240	4.640
PD Tube B19	18.42	9.780	13.62	0.5100	3.000	17.40	8.760	12.60	14.51	4.320
PD Tube 821	16.89	9.780	12.21	0.5100	4.000	15.87	8.760	11.19	7.620	3.560

Figure 3.8: Overview of the table insert for all the tubes geometric measured and calculated parameters in HYSYS

The program uses spreadsheets to integrate the user interface and the optimization script. The user interface is where the user controls tube selection, non-dimensional parameters and the initial conditions. This spreadsheet is linked to the other spreadsheets that calculate the pressure drop and heat transfer coefficient for both tube layout orientation. The two different layouts are separated into their own spreadsheets in the same model in order to make the coding work properly. This allows both layouts to be tested in either the same or separate parametric studies.

There are two separate calculations methods used in the HYSYS script that allow the user to decide what parameters they want to control. The first parametric study method established in the HYSYS model allows the user to select the tube length and control the overall mass flow rate of the system. Instead of having a user defined mass flow rate per tube, the code uses a tool in HYSYS that adjusts the mass flow rate of the tube to find the solve the heat exchanger design for a given length. This method allows the user to specify the total length of tube desired and the code will determine the number of tubes required and its corresponding mass flow rate.

	A	B	C
1	Tube Selection Nu...	7.000	Limit 1 - 55
2		26.67	19.05
3	A*	2.000	Minimal 1
4	Tube Length	1.000 m	Meters
5	Tlm	139.6 C	
6	Shell Duty	37.02 kW	
7	Tube Duty	-37.02 kW	
8	Kwall	14.40	w/mk
9		96.14	
10		96.14	
11			

Figure 3.9: Overview of the initial conditions setup found in the dedicated spreadsheet titled "Fluted Tubes and Test Parameters"

	A	B	C	D	E	F	G	H	I	J	K
3	Deo	26.67	2.667e-002	Hot Fluid		Cold Fluid			P	2.635e-002	
4	Dbo	19.05	1.905e-002	Hot In	600.0 C	Cold In	20.00 C		A*	1.300	
5	Dvo	20.40	2.040e-002	Hot Out	30.00 C	Cold Out	26.15 C		r*	0.6859	
6	Tw	0.7100	7.100e-004	Average	315.0 C	Average	23.07 C		Dhyd	9.340e-003	
7	Ns	4.000							Aeff	3.676e-004	
8	Dei	25.25	2.525e-002	Cp	5.196 kJ/kg-K	Cp	4.313 kJ/kg-K		P*	0.9880	
9	Dbi	17.63	1.763e-002	p	2.436 kg/m3	p	1009 kg/m3				
10	Dvi	18.98	1.898e-002	u	3.184e-005 kg/s-m	u	9.309e-004 kg/s-m				
11	p	15.24	1.524e-002	k	0.2458 W/m-K	k	0.6081 W/m-K				
12	e	3.810	3.810e-003	M	1.250e-002 kg/s		1.396 kg/s				
13	Theta (°)	46.43			1.300e-004 kg/s		1.452e-002 kg/s				
14	p* (Dvi)	0.8000									
15	e* (Dvi)	0.2000			Tube Side Hot Side				Shell Side		
16	Theta*	0.5200		Pr	0.6730			Pr	6.603		
17	p/Dvo	0.7500		V	0.1887			V	3.915e-002		
18	e/Dvo	0.1900		Re	274.0			Re	396.3		
19				Nu	2.543	"500 re 5000"		F	0.9503	"re < 800 "	
20				Nu	4.279	"5000 re 80000"		F	0.5082	"800 < Re < 40000"	
21				F	0.5139	"100 re 1500"		Nu	23.87	" 700 < Re < 40000	
22				F	0.7807	-0.8173		ho	1554		
23				F	0.1875	"3000 re "		Delta P	7.868e-002 kPa		
24				hi	32.94						
25				Delta P	1.175e-003 kPa	kPa					

Figure 3.10: Overview of the spreadsheet calculating the pressure drop and heat transfer coefficent in HSYS

The second method parametric study method established in the HYSYS allows the user to define the systems overall mass flow rate and the number of tube that the flow will be distributed across all the tubes. This method will allow the user to directly influence the shell diameter by limiting the number of tubes in the heat exchanger. This method does not allow the user to specify the overall length of the heat exchanger, but it will be calculated for each tube variation at the specified scaled area variation.

There are case studies in HSYS that allows the user to vary the key non-dimensional scalar parameters to find the optimal solution in the given range. The case study has four main areas that the user will need to be aware of when running the test. The case study first requires the needed parameters to change and record during the course of it operations. These will be set to change tube length, A^* and the fluted tube selection. It will record the corresponding pressure drop, heat transfer coefficient, number of tubes, pitch, P^* and correlation check to ensure that the pitch and R^* are within the boundary conditions. If the conditions are true it will give the P^* value and if it is false it will give the number 333 to show an error.

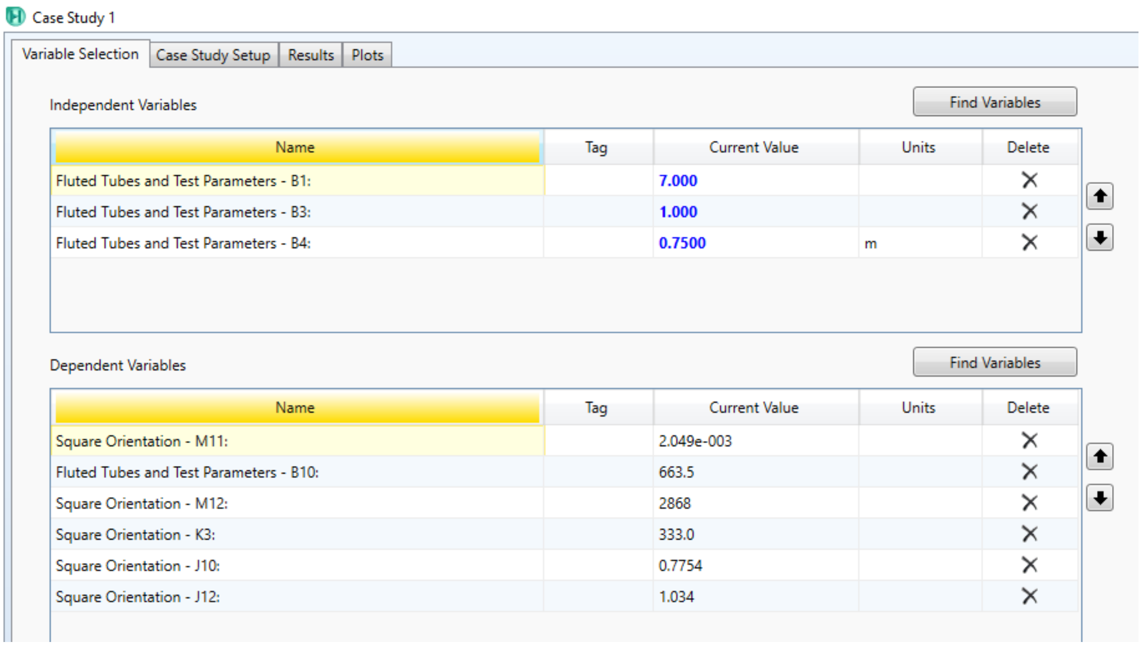


Figure 3.11: Overview of the case study used in the HYSYS program that controls the parametric study

The first step to running a case study is ensuring the initial parameters are set. The user will need to define the scaling ranges and iterations for each parameter. This step allows the user to define the first parameter, and the tube selection range. Currently there are 55 different tubes that are being compared. If the user decided to add an additional tube, this number will need to be increased to include the additional tube. The scalar non-dimensional area range is defined next on the case study. The last parameter set is the variation of the desired tube length. This will allow the user to see the effects of tube length on pressure drop and tube mass flow rate.

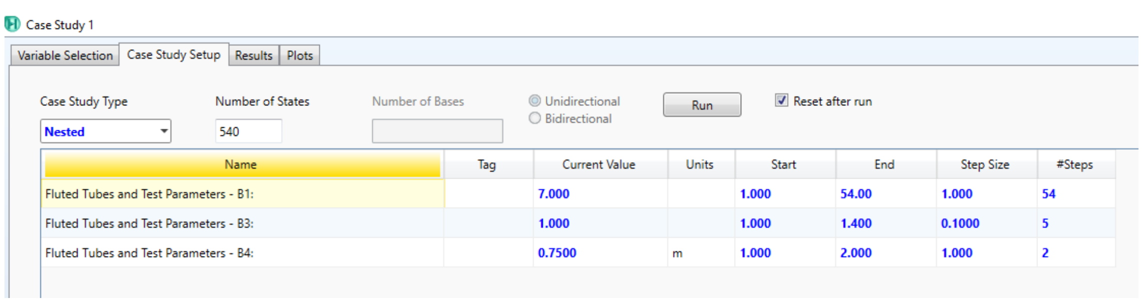


Figure 3.12: Overview of the user defined parameter rangers required to run the parametrix script in HYSYS

3.2.2.1 Code advantages and limitations

The distinct advantage of HYSYS is the large database of thermal fluid properties available to the user. HYSYS allows the user to select the fluid type available in its library, use the Aspen library or create user defined properties through preformed equations or test data entry. The latest method will use interpolation to find a correlation between the test data and the HYSYS system conditions. All the properties will be given at the system defined conditions and the user will be able to run multiple case studies without updating the thermal property limits.

The limitation of the HYSYS script is access to the software and an understanding of its system modeling. There is no GUI interface that allows for executable file format to be created. Although the main spreadsheet may be used as a user interface, the system must be solved to run the case study which may require an understanding of HYSYS. The program availability is limited to those who have access to the program an a general understanding of the system modeling.

3.2.3 Export, Data Processing and Optimization

The script is used to run parametric studies for the sizing of fluted tube heat exchangers, but both programs have limited filtering capabilities. In order to find the optimal design, pressure drop and heat exchanger size within the system tolerances, an improved data filtration method is required. Both of the programs have the capabilities to export all raw data into CSV or Excel® format. Excel® has larger data manipulation capabilities that will allow the user to filter the data to find optimal designs but requires access to the software.

An Excel® template has been established to guide the user to properly filter the data. Both of the data from Matlab® and HYSYS can use the same method and template. The main difference between the two codes is the order each variable appears in the spread spreadsheet.The spreadsheet has the step by step instruction on how to filter the data correctly. Additional limits can be used during filtration to include allowable pressure drop, shell diameter and tube length. The spreadsheet dedicated sheet in the spreadsheet for each of the optimization parameters.

Chapter 4: Analysis

4.1 Case Studies

The scripts were made to conduct an analysis of the possibility of using fluted tubes in a number of different scenarios. In order to test the effectiveness of the fluted tube heat exchanger and optimization script, several case studies were conducted to test the performance of fluted tube heat exchangers and the capability of the script. The optimization tool will also determine the best fluted tube variation for each situation in the case studies. The different case studies that will be analyzed in this chapter will highlight benefits of each optimization programs. The case studies are theoretical studies and are not planned for actual construction and operation.

4.2 Molten Salt Case Study

The Molten Chloride Salt Fast Reactor (MCSFR) developed by Elysium Industries is a prominent advanced reactor design. The MCSFR uses molten chloride salt as a coolant and has unique salts that are able to perform at lower temperatures than are typically researched. It's coolant salt has a lower melting temperature of approximately 250 °C. There are three distinct and different compositions of the salts that are desired to be used. The script will compare the three different salt compositions against each other to find the optimization design and determine the best performing salt for the situation.

These are the design conditions and parameters of the reactor system where the fluted tube heat exchanger would be used. The coolant pressures is at atmospheric which is typical for all molten salt reactors. The reactor inlet and outlet temperatures are 500 °C and 600 °C which are the hot side inlet and outlet temperatures. The MCSFR's has a 2,500 MWth thermal capacity and is split equally between eight 312.5 MWth intermediate heat exchangers.

This study will look at a single 312.5 Mwth heat exchanger and compares different coolant properties based on the three compositions of $NaCl - KCl - ZnCl_2^2$. Each of the salts thermal properties will calculated at the same average temperatures in both side of the heat exchangers. The code will compare the performance of each salt against each other to determine the best heat exchanger design and which salt composition performs the best under different optimizations parameters. The Matlab® program was used to conduct this parametric study because of the desire to compare the three different salts for the same system. Several different parametric studies were done with the data collected from the optimization script to be able to find the performance of each salt, the volume of the heat exchanger based on increasing the secondary loop outlet temperature and finding the optimization of different parameters of the heat exchanger.

4.2.1 Assumptions

The following high-level assumptions were made to bound the analyses, although during optimization some of the bounding temperatures were exceeded to illustrate performance at the far extremes of the reactor temperature ranges. Intermediate loop temperatures based on maintaining the cold side inlet approximately 50 °C above the freezing temperature and pinch point thumb-rule of 25 °C with the temperature in at 300 °C and temperature out is 575 °C as a standard. The tube thermal conductivity is $K_{HastelloyC-276} = 17.857\frac{W}{(m-K)}$ and the minimal tube spacing of the defined limit of the program is adequate to prevent rubbing from vibration. The heat exchangers are assumed to be well insulated. Fouling and surface roughness of the tubes were not considered. It was assumed that the primary (hot) coolant would flow through the inside of the tubes because the enhanced heat exchange surface of the flutes would better support the larger temperature difference of the intermediate (cold) coolant for this specific case. The primary and intermediate loop coolants for each set of calculations was assumed to be the same composition of salt so that leaks during operation would not change the characteristics of the coolant by mixing.

4.2.2 Results

The results are presented in two sections: The first analysis the different salts thermal properties and overall performances. The second is an evaluation and optimization of the heat exchanger size. The parameters altered for the following analysis is the different sizes of fluted tubes at different lengths, mass flow rate and pitch spacing. All of the available fluted tubes - a total of 55 - were analyzed for each of the three salts at tube lengths ranging from two, four, six, eight and ten meters. The impacts of the different sizing and salt parameters are analyzed by the gathered result data and processed to find trends on coolant and size behavior.

4.2.2.1 Thermal Properties Of All Salts

Parameter	**Salt 1**	**Salt 2**	**Salt 3**
Dynamic Viscosity (Pa s)	4.859E-03	4.703E-03	5.573E-03
Density (kg/m3)	2.106E+03	2.225E+03	2.116E+03
Thermal Conductivity k (W/(m K)	0.3360	0.3669	0.3226

Table 4.1: Salt Properties - Primary Loop (Hot) Side of Heat Exchanger

Parameter	**Salt 1**	**Salt 2**	**Salt 3**
Dynamic Viscosity (Pa s)	6.384E-03	5.934E-03	7.181E-03
Density (kg/m3)	2.165E+03	2.274E+03	2.220E+03
Thermal Conductivity k (W/(m K)	0.3498	0.3700	0.3488

Table 4.2: Salt Properties - Intermediate Loop (Cold) Side of HEX

Due to the constant temperature for the reactor conditions, the thermal properties of the salt remain constant. The thermal properties were calculated with the same equations found in Figure 3.6 with the programmed temperature based equations. Dynamic viscosity, density and thermal conductivity of $NaCl - KCl - ZnCl_2$ are temperature dependent and can be programmed in the code. The results

Parameter	Salt 1	Salt 2	Salt 3
Heat Capacity cp (J/(kg K))	917	913	900

Table 4.3: Salt Properties - Heat Capacity (Constant over Temperature Range)

Parameter	Salt 1	Salt 2	Salt 3
Average Temperature Dynamic Viscosity (Pa s)	6.4E-03	5.934E-03	7.181E-03
Integral Average Dynamic Viscosity (Pa s) @ Integral Average	7.2E-03	7.5E-03	7.7E-03
Average Temperature Density (kg/m3)	2.165E+03	2.274E+03	2.220E+03
Integral Average Density (kg/m3)	2.165E+03	2.274E+03	2.220E+03
Average Temperature Thermal Conductivity k (W/(m K)	0.3498	0.3700	0.3488
Integral Average Thermal Conductivity k (W/(m K)	0.3498	0.3700	0.3488

Table 4.4: Salt Properties - Intermediate Loop (Cold) Side of HEX - comparing the step integral average approach to the average temperature approach for thermal properties

of the properties are as shown in Table 4.1 and 4.2. Because the temperature range remains consistant for majority of the testing, these properties will remain constant as well. Thermal conductivity is independent of temperature across the range but varies with salt composition as shown in Table 4.3. All of the properties were calculated using the equations provided in Reference [8]. Additionally, table 4.4 shows the comparison between the two different approach methods of taking the average temperature against an integral approach. The table shows a small variation in the performance of the Dynamic viscosity for all three salts, and no changes for the rest of the fluid thermal properties.

The three different salts show there is a distinct difference between each one that will effect the thermal performance. The full effect of the different thermal properties are shown and highlighted in the optimization analysis of the fluted tubes. The heat exchanger performance is affected by which coolant is used through the system. The initial conditions in the Matlab® code was configured to keep all variables constant and to only vary the tube length for the three salts. The data from the script was used and processed to find the performance of each of the three salts being compared. The heat exchanger pressure drop and volume where analyzed to be able to compare the performance of each salts.

The first comparison is the pressure drop each fluted tube experienced in the shell side of the heat exchanger for each salt. Each of the vertical lines formed in the Figure 4.1 represent a distinct tube for all three salts. The total number of lines represents all 55 tubes analyzed at five different lengths. This shows the general trend of the pressure drop compared to the different salts.

The general trend shows that the pressure drop is lowest for Salt 1. Comparison of the average increase in the pressure drop for the same tube at the same length relative to each other, are:

- Salt 1 to Salt 2: 35.35%
- Salt 1 to Salt 3: 72.37%

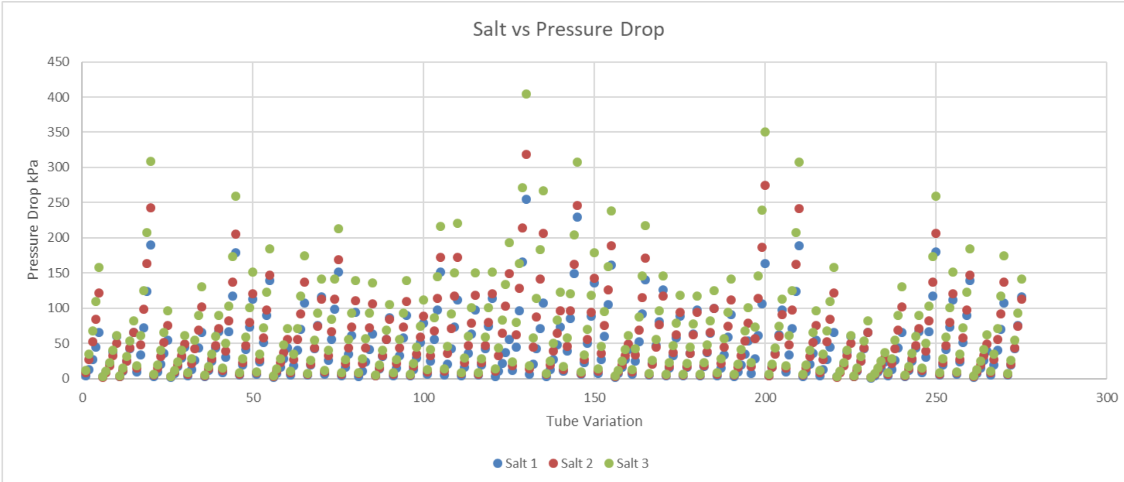


Figure 4.1: Comparison of the different coolant salts to the pressure drop through the shell side of the heat exchanger

- Salt 2 to Salt 3: 26.94%

The impact of salt composition on volume was also analyzed for all of the tubes. Each of the vertical lines formed in the Figure 4.2 represent a distinct tube for all three salts. The total number of lines represents all 55 tubes analyzed at five different lengths. This shows the general trend of the volume behavior for each salts.

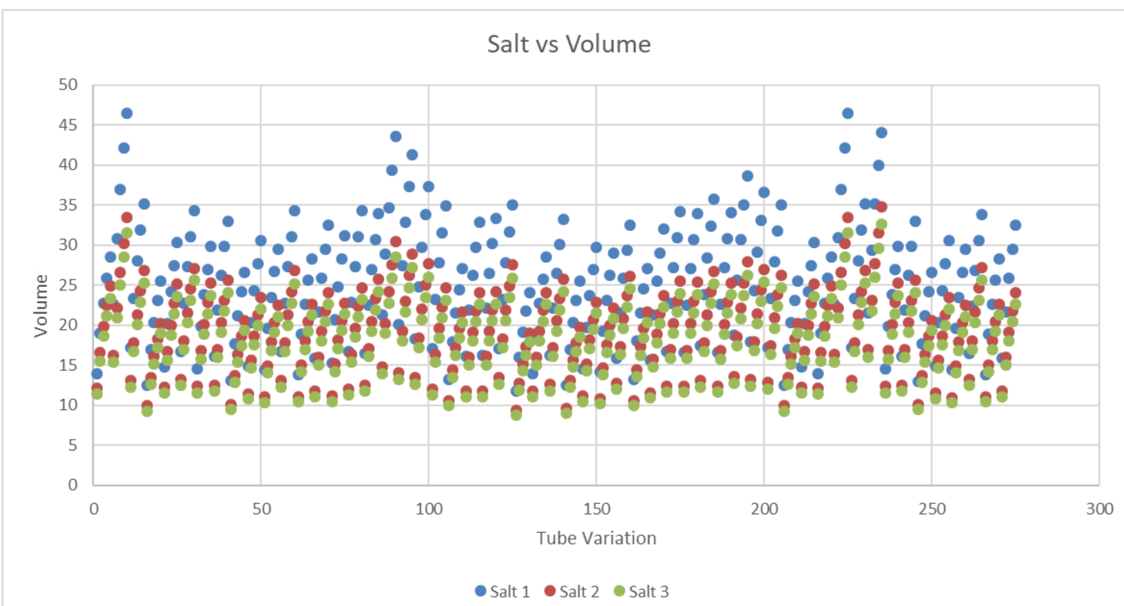


Figure 4.2: Comparison of the different coolant salts to the heat exchanger volume of the heat exchanger

The general trend here shows that Salt 1 has the highest volume size. The comparison of the average increase in the volume of the heat exchangers are:

- Salt 2 to Salt 1: 30.05%
- Salt 3 to Salt 1: 38.39%
- Salt 3 to Salt 2: 6.41%

Generally, salt 1 will provides a lower pressure drop but will require a larger heat exchanger per a given length. This can be contributed to the lower thermal conductivity properties found in Table 4.3. Salt 3 is effective in in decreasing the required volume needed at the cost of a high pressure drop in the system. Salt 2, generally sits in the middle between both salt 1 and salt 2 for volume and pressure drop, but on average there is a smaller volume difference between sale 3 and salt 2 at just 6.41%

4.2.2.2 Heat Exchanger Size Analyses

Parametric studies were conducted in the following areas to develop trend and test heat exchanger performance: Through comparing the volume and shell side pressure drop by varying the non-dimensional flow parameters, and tube length. Comparing the effects of secondary side outlet temperature against the heat exchanger volume at a constant non-dimensional parameters, flow rates and tube length. Effects of the non-dimensional area parameter and pressure loss.

The first parametric study is to look at the effect pressure drop for varying tube length against the volume of the heat exchanger. The analyses only looks at the core of the heat exchanger and does not consider the pressure loss due to expansion or contraction when the fluid exits or enters the heat exchanger. This will allow for a general trend of what the pressure drop will look like for the fluted tube heat exchanger.

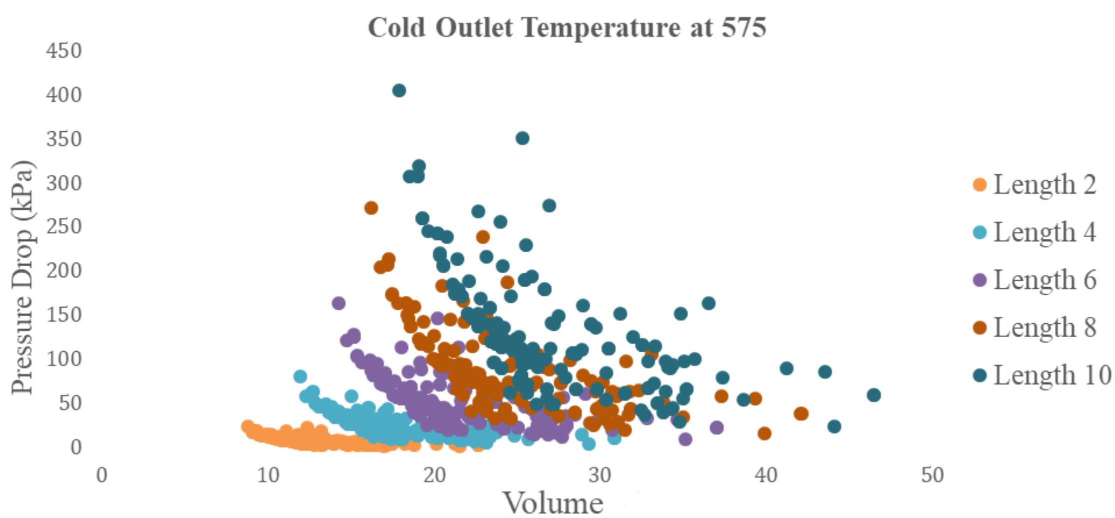


Figure 4.3: The shell side pressure drop for the tube length versus the volume of the heat exchanger

The plot of pressure drops in the shell volume is shown in Figure 4.3. As seen in the figure that the general shape tube is the same for each of the tube length. The five different branches of data in the figure are connected to a specific tube length, the shortest (2 meters) branch at the lowest volumes and pressures and the longest (10 meters) family at the highest volumes and pressures. Although the shorter length of the heat exchanger appears to have improved tube pressure drop performance, this does not consider the pressure drop from expansion. The shorter tube length will typically require a larger number of tubes required potentially increasing the entry and exit pressure losses.

Temperature Increase °C	**Average Volume Increase**
525 to 575	29.78%
525 to 595	62.97%
525 to 598	82.80%
575 to 595	33.25%
595 to 598	12.17%

Table 4.5: The required heat exchanger volume increase needed for the desired output temperature change. The base case analyzed has an output temperature of 575 °C

An analyses was conducted on the secondary side outlet temperature effects on the required heat exchanger volume. Reducing the outlet temperature significantly reduces the required volume for all variation of the heat exchanger, but at the cost of less heat available to downstream power production and/or industrial processes. This will lower the overall thermal efficiency of the heat exchanger. Increasing outlet temperature to 595°C and 598 °C requires a significant increase in volume and also an increase in the pressure drop. The small temperature increase comes at a large cost compared to the 575 °C base parametric study.

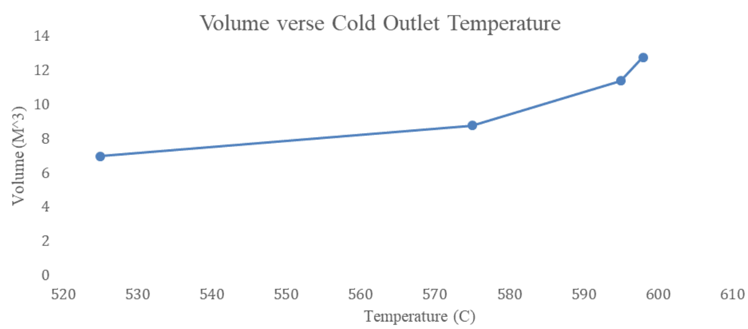


Figure 4.4: Graph of the heat exchanger volume increase as the secondary temperature output is increased.

Increasing the desired outlet temperature causes a very significant changes in volume. Table 4.5 shows the average heat exchanger volume increase needed to reach the output temperature.

The Table 4.5 is represented in graph form in Figure 4.4. This shows the general trend of an increasing volume as the outlet temperature is increased. It also shows as the outlet temperature reaches 575 °C, the volume increase is relatively small compared to the sharp increase in volume for temperatures above 575°C .

An analyses was also conducted by looking at the effects of A^* on the pressure drop of the shell side

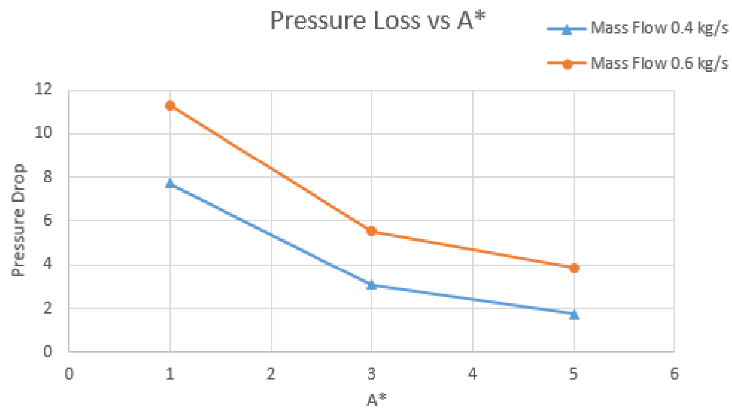


Figure 4.5: The general shell side pressure drop relation to the non-dimensional A^*. Shows the rate the pressure drop is decreased by increasing the flow area

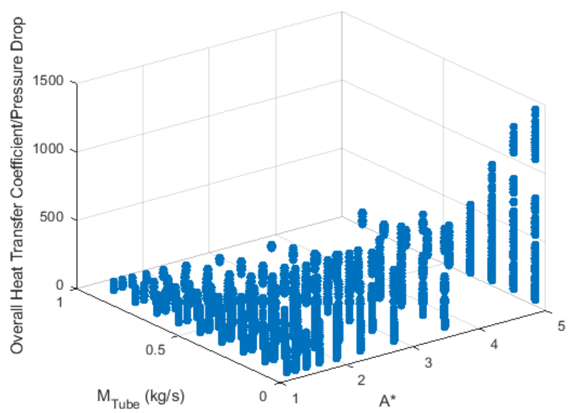


Figure 4.6: 3-Dimensional scatter plot of a parametric study conducted for varying mass flow rate and A* to the resulting overall heat transfer coefficient per pressure drop

of the heat exchanger. The larger flow area denoted by the large non-dimension A^* should signify a lower pressure drop. The goal behind this study is to find the rate at which the shell side pressure drop is effected by the pitch spacing. Even at different flow rates, the general trend of pressure drop is decreased by the larger shell flow areas as shown in Figure 4.5.

These parametric studies allow for a better understanding of the performance of the heat exchanger under different circumstances. This will give general feedback on the overall performance of the heat exchanger and the parameters that need to be adjusted to find the optimal design. These studies also allow the user to understand the scaling boundary conditions that should be applied to the program before running the script.This is shown when plotting the two changing parameters of mass flow rate and A^* to the overall heat transfer heat transfer per pressure drop in a 3-dimensional plot 4.6.

4.2.2.3 Optimization

Optimization of the molten salt heat exchanger design options were performed by running parametric studies changing all the allowable variables. The main parameters that are held constant is the temperature specified in the initial conditions. The changing parameters for this study are the tube length, number of tubes, and the non-dimensional A^*. Running this parametric studies gives the sizing of the heat exchanger for all parameters variations. The results may be filtered to find the optimal design. This case looks specifically to optimize the following parameters; pressure drop, number of tube, and shell diameter.

Fluted Tube ID	**Salt Type**	**Tube Layout**	**Pressure Drop**	**Number of Tubes**	**Shell Diameter**	**Volume**
PD Tube B6	Salt 3	30	0.716 kPa	10835	5.25 m	43.29 m^3
PD Tube B6	Salt 2	30	0.716 kPa	10319	5.12 m	41.18 m^3
PD Tube B6	Salt 1	30	0.716 kPa	10201	5.1 m	40.86 m^3
PD Tube B6	Salt 3	45	0.716 kPa	10420	4.80 m	36.19 m^3
PD Tube B6	Salt 2	45	0.716 kPa	9924	4.68 m	34.40 m^3

Table 4.6: The Top 5 Optimized Fluted Tubes for a Molten salt heat exchanger for pressure drop

Fluted Tube ID	**Type**	**Tube Layout**	**Pressure Drop**	**Number of Tubes**	**Heat Transfer/-Pressure Drop**	**Shell Diameter**
PD Tube V13	Salt 1	45	20.52 kPa	3681	96.5 $\frac{W}{Km^2kPa}$	2.14 m
HT Tube G13	Salt 1	45	20.56 kPa	3681	96.3 $\frac{W}{Km^2kPa}$	2.14 m
PD Tube V13	Salt 3	45	20.69 kPa	3907	90.2 $\frac{W}{Km^2kPa}$	2.21 m
HT Tube G13	Salt 3	45	20.7 kPa	3906	90.0 $\frac{W}{Km^2kPa}$	2.21 m
HT Tube G13	Salt 3	30	20.02 kPa	4106	88.7 $\frac{W}{Km^2kPa}$	2.43 m

Table 4.7: The Top 5 Optimized Fluted Tubes for a Molten salt heat exchanger for heat transfer coefficient per pressure drop

Typically there are specific areas that are optimized based on the needed quality that works for the system. This analysis looks at four different areas of optimization of pressure drop, heat transfer coefficient per pressure drop, shell diameter size and number of tubes.

The max pressure drop limit set for the heat exchanger in this system is 150 kPa. When filtering the data, this parameter used to remove any results that exceeds this pressure drop. These are the different optimization for a fluted tubes shell and tube heat exchanger. Each table shows the top five fluted tube heat exchanger design for the specified optimization.

Pressure drop in a heat exchanger is an area of interest to improve the performance and cost of operating the loop. Often times using a larger heat exchanger at a lower flow will decrease the amount of

Fluted Tube ID	Salt Type	Tube Layout	Pressure Drop	Number of Tubes	Shell Diameter
PD Tube B22	Salt 2	45	143.50 kPa	2759	1.12 m
HT Tube G6	Salt 1	45	141.38 kPa	2888	1.14 m
PD Tube V6	Salt 1	45	141.25 kPa	2889	1.14 m
HT Tube G5	Salt 2	30	147.81 kPa	3401	1.14 m
PD Tube V5	Salt 2	30	147.92 kPa	3403	1.14 m

Table 4.8: The Top 5 Optimized Fluted Tubes design for a Molten salt heat exchanger for shell diameter

Fluted Tube ID	Salt Type	Tube Layout	Pressure Drop	Number of Tubes	Shell Diameter
PD Tube B19	Salt 1	45	127.16 kPa	2682	1.25 m
PD Tube B19	Salt 2	45	119.06 kPa	2721	1.26 m
PD Tube B22	Salt 2	45	143.50 kPa	2759	1.12 m
PD Tube B25	Salt 1	45	141.15 kPa	2841	1.31 m
PD Tube B19	Salt 3	45	128.78 kPa	2851	1.29 m

Table 4.9: The Top 5 Optimized Fluted Tubes for a Molten salt heat exchanger for number of tubes

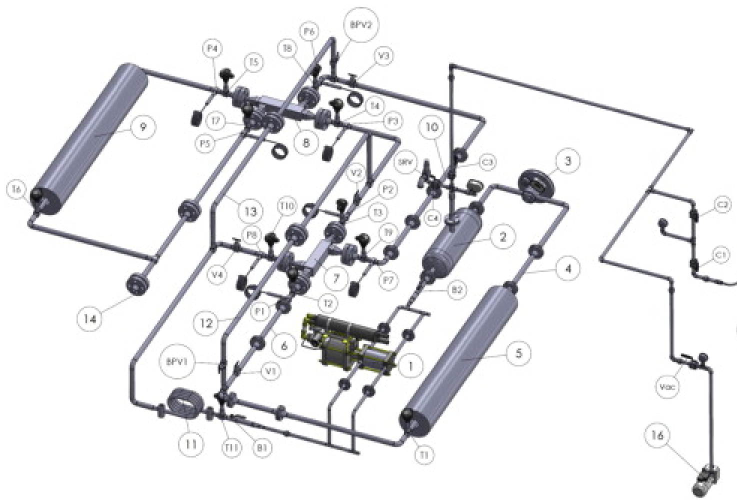
pressure drop. This comes at the high cost of materials and manufacturing. The top five heat exchanger design parameters for the lowest pressure drop of the system is show in Table 4.6. The overall best performing heat exchanger is typically the highest heat transfer per pressure drop of the system. The top five tubes for heat transfer per pressure drop where pressure drop is found in Table 4.7. The sizing of the heat exchanger may be a constraint depending on the location within the facility and the manufacturing cost to build the fluted tube heat exchanger. Welding fluted tubes to a flow sheet cost are increased compared to a straight tube. Table 4.8 and Table 4.9 are optimized to show the top five fluted tubes for smallest shell diameter and the number of tubes used respectfully.

4.3 Test Loop Case Study

Test loops are vital to gathering experimental data under required circumstances. The design of test loops often require heat exchangers for condensing or cooling needed before the fluid goes to the compressor or the pump. Additionally, test loops will often use intermediate heat exchanger to minimize waste heat and the required energy input of the system. Two separate test loops are analyzed to find the optimal heat exchanger design for the cooling portion of the loops. Each of the two test loops analyzed have a single phase gas as their primary fluids. The first loop is a high temperature helium test loop designed to test intermediate heat exchangers. The second is a nitrogen test loop designed to test fouling in printed circuit heat exchangers.

4.3.1 Helium Test Loop

The helium test loop is operated to test and analyze intermediate heat exchanger. The analysis is conducted on the helium test loop at the High-Temperature Helium Test Facility at the Ohio State University [9]. The loop operates with heaters, gas booster, intermediate heat exchanger and a cooler. A comprehensive list of all the different parts are depicted in Figure 4.7. The cooling section of the test loop is where the proposed fluted tube heat exchanger design would be integrated. The hot helium fluid passes through the tube side of the heat exchanger and water passes through the shell.



Balloon No.	Test Facility Component
1	Gas booster
2	Surge volume tank
3	Pressure reducing valve
4	Venturi flowmeter 1
5	Pre-heater
6	Venturi flowmeter 2
7	PCHE1
8	PCHE2
9	Main-heater
10	Venturi flowmeter 3
11	Cooler
12	Bypass 1
13	Bypass 2
14	Blind flange for future use
15	Helium gas cylinder
16	Vacuum pump
T1-T11	Thermowells
P1-P8	Pressure taps
V1-V4	Needle valves
BPV1, BPV2	Needle valves in bypass lines
B1, B2	Ball valves at gas booster inlet and exit
C1-C4	Ball valves in helium charging line
Vac	Ball valve in vacuum line
SRV	Safety relief valve

Figure 4.7: Overview of High-Temperature Helium Test Facility at the Ohio State University. The cooler is component 11 in the test loop where the designed heat exchanger would be placed.

The test loop operates at the temperatures up to 800 °C and 3 MPa. The designed flow rates are 45 kg/h and 1.396 kg/s for the hot side and cold side respectively. The helium side will enter the cooler from 476 °C and will exit at 30 °C. The cooling water enters the system at 20 °C and exits at 25 °C. The hot side operates at 3 MPa with an allowable pressure drop of 30.0 kPa. The shell side operates at a pressure of 300 kPa and has an allowable pressure drop of 15.0 kPa. The case study is to run a parametric study on the cooler heat exchanger to find the optimal designs for manufacturing a fluted tube heat exchanger.

The heat exchanger size of the cooler will be smaller due to the nature of a test loop compared to an industrial size heat exchanger. The required heat transfer load for the test loop is small causing majority of the heat exchangers to fall within a few meter in length for a single tube. This style of heat exchanger is a tube in tube heat exchanger. The parametric study conducted on the helium test loop is to investigate the different outer tube dimensions through changing the A^* scalar parameter and iterating through the different tube variation. Due to the low heat transfer load the loop only requires a single fluted tube to supply sufficient cooling.

From the two different methods to calculate the parametric study in HYSYS, this study was conducted by specifying the A^* value and the number of tubes. The code will calculate the length for each of the variations calculated. There are two separate parameters looked at to find the performance of the heat exchanger. The thermal fluid properties comparison and the optimization parameters are used to analyze the different lengths and sizes of the heat exchanger. The optimization areas investigated are heat transfer per pressure drop, total pressure drop and tube length.

4.3.1.1 Thermal Properties

The large difference in fluid thermal properties causes a large temperature difference in the helium versus a small temperature change in the water. The difference in temperature changes is shown by the large difference between the thermal conductivity and heat capacity of each fluid.

Parameter	**Helium**	**Water**
Dynamic Viscosity (Pa s)	2.946E-05	9.456E-04
Density (kg/m3)	2.721	1,009
Thermal Conductivity k (W/(m K)	0.2274	0.6071
Heat Capacity (kJ/kg-K)	5.196	4.314

Table 4.10: Helium and water properties for the test loop conditions

4.3.1.2 Optimization

The optimization for the helium test loop is investigated in three different areas; between the heat transfer per pressure drop, smallest total pressure drop, and the smallest tube length.

Fluted Tube ID	**Heat Transfer/-Pressure Drop**	**Length**	**Pressure Drop**	**Shell Diameter**
PD Tube B6	722.9	3.6 m	1.49 kPa	0.0505 m
PD Tube V15	666.4	2.5 m	2.79 kPa	0.0428 m
PD Tube V8	624.5	2.9 m	2.25 kPa	0.0485 m
PD Tube V12	623.7	2.6 m	2.97 kPa	0.0404 m
PD Tube G12	623.7	2.6 m	2.97 kPa	0.0404 m

Table 4.11: The Top 5 Optimized Fluted Tubes for a Helium - Water Heat Exchanger for heat transfer per pressure drop

The total pressure drop is always an area of concern when dealing with a a test loop. There are set limits in the code that determines if the pressure drop is too high for the system. The helium test loop has a set pressure drop limitation to 30.0 kPa and 15.0 kPa for the hot side and cold side respectfully. The data is filtered to ensure that the results will not include results outside the design parameters. The optimization for the minimal pressure drop is shown in Table 4.12 and for the smallest tube length is shown in Table 4.13

There are several different options that are viable to operate as the cooler. The best operating fluted tube in this case is the tube with the highest heat transfer per pressure drop because it is also the tube with the lowest overall pressure drop. The PD Tube B6 fluted tube variation looks to be the best performance at a higher A^*. The shortest tube within the pressure threshold is also a valid option. It is

Fluted Tube ID	Heat Transfer/-Pressure Drop	Length	Pressure Drop	Shell Diameter
PD Tube B6	722.9	3.6 m	1.49 kPa	0.0505 m
PD Tube B6	509.8	3.5 m	2.15 kPa	0.0469 m
PD Tube V8	624.5	2.9 m	2.25 kPa	0.0485 m
PD Tube G8	593.8	2.8 m	2.37 kPa	0.0482 m
PD Tube G15	371.8	2.6 m	2.41 kPa	0.0428 m

Table 4.12: The Top 5 Optimized Fluted Tubes for a Helium - Water Heat Exchanger for pressure drop

Fluted Tube ID	Heat Transfer/-Pressure Drop	Length	Pressure Drop	Shell Diameter
PD Tube V2	158.6	1.86 m	17.46 kPa	0.0312 m
PD Tube G2	158.6	1.86 m	17.46 kPa	0.0312 m
PD Tube V2	219.0	1.88 m	12.52 kPa	0.0328 m
PD Tube G2	219.0	1.88 m	12.52 kPa	0.0328 m
PD Tube V2	321.24	1.91 m	8.40 kPa	0.0352 m

Table 4.13: The Top 5 Optimized Fluted Tubes for a Helium - Water Heat Exchanger for tube length

significantly smaller at only half of the length of the other optimal solutions for pressure drop and heat transfer per pressure drop.

4.3.2 Nitrogen Test Loop

The Nitrogen test loop is designed to look at fouling in compact heat exchangers. The test loop operates at high pressure and moderate temperatures for extended periods of time. Each heat exchanger is tested for 400 hours to look at the effects of fouling due to fluid impurities. After the 400 hour test, impurities are introduced to the system to force fouling and channel blockage. The main purpose for the loop construction is the testing of compact printed circuit heat exchangers. This style of heat exchanger has small channels that may be blockage with small impurities that can drastically effect its performance. The test loop is similar to the layout of the helium test loop with the system components. The parametric study was conducted to size the heat exchanger that will cool the nitrogen before entering the compressor. The main focus of this case study is to compare the sizing of a single tube fluted tube heat exchanger and a single tube straight tube heat exchanger. This will check the size benefits of utilizing the fluted tubes.

The main nitrogen fluid test loop operates at temperatures at 200 °C and 4 MPa. The nitrogen enters the cooler at 100 °C and exits at 20 °C at a 4 MPa and the mass flow rate at 5.05 kg/hr. The water enters the heat exchanger at 15 °C and leaves at 20.5 °C at a flow rate of 19 kg/hr and at a pressure of 200 kPa.

4.3.2.1 Thermal Properties

The first analysis conducted looks at the thermal properties between the Nitrogen and the water. The difference in thermal properties and the predetermined flow rates will directly effect the size of the heat exchanger. Table 4.14 shows the thermal properties of each fluid.

Parameter	Nitrogen	Water
Dynamic Viscosity (Pa s)	2.052E-05	8.205E-04
Density (kg/m3)	40.54	1,005
Thermal Conductivity k (W/(m K)	0.03	0.6164
Heat Capacity (kJ/kg-K)	1.104	4.313

Table 4.14: Nitrogen and Water thermal properties for the nitrogen test loop

The shell of the heat exchanger is the cold side of the heat exchanger with the cooling water. The tube side is determined as the hot side with the nitrogen passing through the tubes. This orientation holds the standard of the high pressure system passing through the tubes and the lower pressure through the shell.

4.3.2.2 Optimization

The parametric study is conducted on the cooler for the nitrogen heat exchanger. The data is used and processed to find the optimal design for tube length, pressure drop and heat transfer per pressure drop. The study only looks at single fluted tube in tube and tube heat exchangers because of the low heat transfer load required in the cooler. The length of the tube will range from 1 to 5 Meters in length.

Fluted Tube ID	Layout	Length	Total Pressure Drop	Pitch	Shell Diameter
PD Tube V15	Diamond	1.95 m	4.5 Pa	0.0363 m	0.0409 m
PD Tube V15	Square	1.97 m	4.49 Pa	0.0368 m	0.0416 m
PD Tube V15	Diamond	2.04 m	4.37 Pa	0.0396 m	0.0447 m
PD Tube V15	Square	1.88 m	4.76 Pa	0.0337 m	0.0381 m
PD Tube V15	Diamond	1.85 m	4.93 Pa	0.0325 m	0.0367 m

Table 4.15: The Top 5 Optimized Fluted Tubes for a Nitrogen - Water Heat Exchanger for Heat Transfer

The parameter optimized is the heat transfer coefficient per pressure drop, overall pressure drop and tube length of the heat exchanger. These are the general areas of interest for optimization. The pressure drop limit set are 120 kPa and 10 kPa for the tube side and shell side respectively. The results from the parametric study were filtered to ensure the optimal designs will fall within these thresholds.

Fluted Tube ID	Layout	Length	Total Pressure Drop	Pitch	Shell Diameter
PD Tube V15	Diamond	2.04 m	4.37 Pa	0.0396 m	0.0447 m
PD Tube V15	Square	1.97 m	4.49 Pa	0.0368 m	0.0416 m
PD Tube V15	Diamond	1.95 m	4.53 Pa	0.0362 m	0.0409 m
PD Tube B6	Diamond	3.23 m	4.66 Pa	0.0466 m	0.0456 m
PD Tube B6	Square	3.15 m	4.72 Pa	0.0434 m	0.0489 m

Table 4.16: The Top 5 Optimized Fluted Tubes for a Nitrogen - Water Heat Exchanger for Pressure Drop

Fluted Tube ID	Heat Transfer / Pressure Drop	Length	Total Pressure Drop	Shell Diameter
HT Tube G10	Square	1.013 m	36.695 Pa	0.0157 m
PD Tube V10	Square	1.014 m	36.695 Pa	0.0157 m
HT Tube G10	Diamond	1.046 m	32.858 Pa	0.0168 m
PD Tube V10	Diamond	1.048 m	32.858 Pa	0.0168 m
HT Tube G10	Square	1.077 m	31.287 Pa	0.0179 m

Table 4.17: The Top 5 Optimized Fluted Tubes for a Nitrogen - Water Heat Exchanger for tube length

The pressure drop per heat transfer and total pressure drop were optimized and the top results for each parameter are found in Table 4.15 and Table 4.16. The small mass flow rates and the larger A^* values calculates a very small pressure drop. The tube selection and parameters are similar for the optimized pressure drop and heat transfer per pressure drop parameters.

The optimization of tube length largely reduces the tube length, but it does increase the pressure drop by a factor of about 8. The benefit of the slower mass flow rate system is that the overall pressure drop is still well within the pressure drop limits. The optimization of the tube selection for the smallest length is shown in 4.17.

There is a large pressure drop change compared between different tube lengths. Although the shortest tube is well within the pressure drop limits, it does increase it by almost a factor of ten and only reduces the tube length in half. The best design for this test loop could be considered as the smallest tube selection because it still as a significantly low pressure drop for half of the size of the other options.

4.3.2.3 Fluted Tube Comparison Against Straight Tube

The original loop is designed to use a straight tube in tube heat exchanger. This parametric study allowed for a direct comparison between the size of the heat exchanger. The straight tube and tube heat exchanger was not optimized and a single solution was calculated to size a heat exchanger. The analyses

Parameter	**Values**
Length	3.45 m
Inner Tube ID	0.0176 m
Inner Tube OD	0.0213
Outside Tube ID	0.0300
Tube Pressure Drop	3.19 kPa
Shell Pressure Drop	1.98 kPa

Table 4.18: Nitrogen - Water straight tube and tube heat exchanger design

does compare the straight tube heat exchanger against the optimized fluted tube heat exchanger. The length and pressure drop were calculated and will be compared to the fluted tube optimization.

The straight tube heat exchanger is presented in Table 4.18. Comparing the optimized fluted tube design to the results of the straight train can significantly improve the size. The fluted tubes can significantly reduce the length and overall volume of the cooler. Using the smallest fluted tube design, the heat exchanger can be reduced to the third of the size of the straight tube design but a significant increase to pressure drop. For a more conservative design, the fluted tube can reduce the length by a third at a slight increase to pressure drop.

Chapter 5: Summary and Conclusions

5.1 Discussion

The case studies show the capabilities of the fluted tube heat exchangers in different situations. There is a real potential to use fluted tubes over straight tubes to reduce the size of the traditional shell and tube heat exchanger. The large variation of fluted tubes allows for this style of heat exhanger be used in multiple different situations. The versatile nature of the heat exchanger allows it to replace current heat exchanger designs in more of a compact design for different industries.

The script allows for a detailed analysis of the performance and pressure drop for each case study. The code provides the data to find the optimal solutions and to develop design trends for the sizing of the heat exchanger in specific heat exchanger designs. This information is useful comparing the cost versus benefits of the heat exchanger design. This information allows for a clear display of the added cost and savings that can be made by making small changes to the general heat exchanger design. This script allows optimal heat exchanger designs to be created for situation specific needs. The non-dimensional scaling parameters is an effective method in the script to compare different sizes of fluted tubes against each other and to find an optimal heat exchanger design.

5.2 Future Work

Improvement to the user experience is key to a successful of the optimization script. The current script is effective in running parametric studies with the given parameters to find the optimal solution desired for the system but required manual filtering of the results. Changing the filtering process to an automated process will create a positive user experience and eliminate user errors. The integrated filtering capabilities to derive heat exchanger design trends in a single operation would allow for simple user experience

Fluted tubes have been shown to be an effective method of creating single phase compact heat exchanger. Research has shown that fluted tubes provide improved results for evaporating and condensing applications [7]. There has been several studies on the added benefit the geometry gives to condensing water[10] [11]. Adding the capability to size a condenser or an evaporator heat exchanger would increase the versatility of the current script. Adding two phase regimes to the code would require the input of fluted tube two phase correlations.

5.3 Conclusion

The script is effective in calculating the optimal design for compact fluted tube heat exchanger. The script calculates thousands of different heat exchanger designs to find the optimal configuration and to develop performance trends. Both programs have distinct advantages and run the optimization script as intended. Each program utilizing the non-dimensional scaling parameters to compare different fluted tube sizes and tube layouts to find the optimal design. Each of the two programs have been shown to be effective in their designed areas. The scaling parameter also allows a clear way to ensure that the different tubes are scaled properly. This script can be used for any industry looking at using a fluted tube to replace the straight tube to decrease the heat exchanger size.

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Appendix A: Tube Dimension Matrix

Tube Dimension Matrix, Page 1

	Deo	Dbo	Dvo	Tw	Ns	Dei	Dbi	Dvi
PD Tube V1	18.8	11.23	12.6	0.508	3	17.784	10.214	11.584
PD Tube V2	19.81	15.49	16.23	0.508	5	18.794	14.474	15.214
PD Tube V4	21.08	13.97	14.76	0.508	3	20.064	12.954	13.744
PD Tube V5	14.22	9.27	10.26	0.406	3	13.408	8.458	9.448
PD Tube V6	17.15	11.13	11.99	0.508	3	16.134	10.114	10.974
PD Tube V7	16.15	10.31	11.25	0.508	4	15.134	9.294	10.234
PD Tube V8	26.67	19.05	20.4	0.711	4	25.248	17.628	18.978
PD Tube V9	16.64	11.68	12.6	0.508	4	15.624	10.664	11.584
PD Tube V10	12.83	8.89	9.6	0.406	4	12.018	8.078	8.788
PD Tube V11	21.72	16.26	17.09	0.711	3	20.298	14.838	15.668
PD Tube V12	19.94	16.38	17.17	0.711	4	18.518	14.958	15.748
PD Tube V13	28.07	15.62	16.21	0.711	1	26.648	14.198	14.788
PD Tube V14	24.38	15.49	16.92	0.711	2	22.958	14.068	15.498
PD Tube V15	25.4	17.4	18.16	0.711	1	23.978	15.978	16.738
PD Tube B6	24.64	16.76	21.47	0.889	3	22.862	14.982	19.692
PD Tube B14	22.86	12.7	17.12	0.635	4	21.59	11.43	15.85
PD Tube B15	24.13	15.24	19.59	0.635	4	22.86	13.97	18.32
PD Tube B16	25.02	12.95	17.8	0.889	3	23.242	11.172	16.022
PD Tube B17	25.02	13.08	19.36	0.889	3	23.242	11.302	17.582
PD Tube B18	22.91	13.64	16.99	0.889	4	21.132	11.862	15.212
PD Tube B19	18.42	9.78	13.62	0.508	3	17.404	8.764	12.604
PD Tube B21	16.89	9.78	12.21	0.508	4	15.874	8.764	11.194
PD Tube B22	17.53	12.32	14.07	0.508	4	16.514	11.304	13.054
PD Tube B24	20.96	12.19	15.73	0.635	5	19.69	10.92	14.46
PD Tube B25	19.76	11.23	14.02	0.635	5	18.49	9.96	12.75
PD TUBE G1	18.796	11.2268	12.2428	0.508	3	17.78	10.2108	11.2268
PD TUBE G2	19.812	15.494	17.399	0.508	5	18.796	14.478	16.383
PD TUBE G4	21.082	13.97	16.2814	0.508	3	20.066	12.954	15.2654
PD TUBE G5	14.224	9.271	10.1854	0.4064	3	13.4112	8.4582	9.3726
PD TUBE G6	17.145	11.1252	12.1158	0.508	3	16.129	10.1092	11.0998
PD TUBE G7	16.1544	10.3124	12.0396	0.508	4	15.1384	9.2964	11.0236

Tube Dimension Matrix, Page 2

	Deo	Dbo	Dvo	Tw	Ns	Dei	Dbi	Dvi
PD TUBE G8	26.67	19.05	19.9644	0.508	4	25.654	18.034	18.9484
PD TUBE G9	16.637	11.684	12.7254	0.508	4	15.621	10.668	11.7094
PD TUBE G10	12.827	8.89	9.8044	0.4064	4	12.0142	8.0772	8.9916
HT Tube G1	18.796	11.2268	12.5984	0.508	3	17.78	10.2108	11.5824
HT Tube G2	19.812	15.494	16.2306	0.508	5	18.796	14.478	15.2146
HT Tube G4	21.082	13.97	14.7574	0.508	3	20.066	12.954	13.7414
HT Tube G5	14.224	9.271	10.2616	0.4064	3	13.4112	8.4582	9.4488
HT Tube G6	17.145	11.1252	11.9888	0.508	3	16.129	10.1092	10.9728
HT Tube G7	16.1544	10.3124	11.2522	0.508	4	15.1384	9.2964	10.2362
HT Tube G8	26.67	17.907	20.3962	0.7112	4	25.2476	16.4846	18.9738
HT Tube G9	16.637	11.684	12.5984	0.508	4	15.621	10.668	11.5824
HT Tube G10	12.827	8.89	9.6012	0.4064	4	12.0142	8.0772	8.7884
HT Tube G11	21.717	16.256	17.0942	0.7112	3	20.2946	14.8336	15.6718
HT Tube G12	19.939	16.383	17.1704	0.7112	4	18.5166	14.9606	15.748
HT Tube G13	28.067	15.621	16.2052	0.7112	1	26.6446	14.1986	14.7828
HT Tube G14	24.384	15.494	16.9164	0.7112	2	22.9616	14.0716	15.494
HT Tube G15	25.4	17.399	18.161	0.7112	1	23.9776	15.9766	16.7386
PD Tube J.T1	21.9456	13.1318	16.7386	0.8128	4	20.32	11.5062	15.113
PD Tube J.T2	21.7678	13.1318	15.621	0.8128	4	20.1422	11.5062	13.9954
PD Tube J.T3	20.5232	12.4968	13.716	0.8128	4	18.8976	10.8712	12.0904
PD Tube J.T4	20.4216	12.4968	13.3604	0.8128	4	18.796	10.8712	11.7348
PD Tube J.T5	20.1168	12.319	13.9446	0.8128	4	18.4912	10.6934	12.319
PD Tube J.E1	19.0754	11.2522	15.9258	1.4478	4	16.1798	8.3566	13.0302
PD Tube J.E2	30.099	16.9418	24.0792	1.4478	6	27.2034	14.0462	21.1836

Tube Dimension Matrix, Page 3

	p	e	Theta	p* (Dvi)	e* (Dvi)	Theta*	p/Dvo	e/Dvo
PD Tube V1	13.26	3.792	44.8585	1.1447	0.3273	0.4984	1.0524	0.3004
PD Tube V2	6.22	2.159	58.6190	0.4088	0.1419	0.6513	0.3832	0.1331
PD Tube V4	13.26	3.557	49.3742	0.9648	0.2588	0.5486	0.8984	0.2409
PD Tube V5	13.26	2.483	39.0170	1.4035	0.2628	0.4335	1.2924	0.2412
PD Tube V6	11.73	3.009	46.9477	1.0689	0.2742	0.5216	0.9783	0.2510
PD Tube V7	6.63	2.926	53.1168	0.6478	0.2859	0.5902	0.5893	0.2596
PD Tube V8	15.24	3.814	46.4331	0.8030	0.2010	0.5159	0.7471	0.1868
PD Tube V9	8.23	2.482	50.2514	0.7105	0.2143	0.5583	0.6532	0.1968
PD Tube V10	6.1	1.968	51.0258	0.6941	0.2239	0.5670	0.6354	0.2052
PD Tube V11	23.44	2.735	37.3620	1.4960	0.1746	0.4151	1.3716	0.1597
PD Tube V12	15.24	1.786	41.5043	0.9677	0.1134	0.4612	0.8876	0.1037
PD Tube V13	60.96	6.223	39.8749	4.1223	0.4208	0.4431	3.7606	0.3840
PD Tube V14	43.54	4.449	31.4009	2.8094	0.2871	0.3489	2.5733	0.2627
PD Tube V15	21.92	3.995	68.9824	1.3096	0.2387	0.7665	1.2070	0.2203
PD Tube B6	27.71	3.94	39.0551	1.4072	0.2001	0.4339	1.2906	0.1835
PD Tube B14	15.24	5.08	41.4214	0.9615	0.3205	0.4602	0.8902	0.2967
PD Tube B15	15.24	4.44	45.2730	0.8319	0.2424	0.5030	0.7779	0.2269
PD Tube B16	15.24	6.03	50.7308	0.9512	0.3764	0.5637	0.8562	0.3390
PD Tube B17	23.45	5.97	40.8451	1.3338	0.3396	0.4538	1.2113	0.3084
PD Tube B18	9.24	4.64	55.2993	0.6074	0.3050	0.6144	0.5438	0.2728
PD Tube B19	14.51	4.32	44.5078	1.1512	0.3427	0.4945	1.0653	0.3172
PD Tube B21	7.62	3.56	51.5293	0.6807	0.3180	0.5725	0.6241	0.2912
PD Tube B22	13.85	2.6	38.5854	1.0610	0.1992	0.4287	0.9844	0.1851
PD Tube B24	7.26	4.38	53.7004	0.5021	0.3029	0.5967	0.4615	0.2788
PD Tube B25	5.26	4.27	59.1579	0.4125	0.3349	0.6573	0.3752	0.3042
PD TUBE G1	13.2588	4.27	44.0374	1.1810	0.3803	0.4893	1.0830	0.3091
PD TUBE G2	6.223	4.27	60.3497	0.3798	0.2606	0.6706	0.3577	0.1241
PD TUBE G4	13.2588	4.27	52.1296	0.8686	0.2797	0.5792	0.8144	0.2184
PD TUBE G5	13.2588	4.27	38.8151	1.4146	0.4556	0.4313	1.3017	0.2431
PD TUBE G6	11.7348	4.27	47.2342	1.0572	0.3847	0.5248	0.9686	0.2484
PD TUBE G7	6.6294	4.27	54.9662	0.6014	0.3874	0.6107	0.5506	0.2426

Tube Dimension Matrix, Page 4

	p	e	Theta	p* (Dvi)	e* (Dvi)	Theta*	p/Dvo	e/Dvo
PD TUBE G8	15.24	4.27	45.8153	0.8043	0.2253	0.5091	0.7634	0.1908
PD TUBE G9	8.2296	4.27	50.5315	0.7028	0.3647	0.5615	0.6467	0.1946
PD TUBE G10	6.096	4.27	51.6330	0.6780	0.4749	0.5737	0.6218	0.2008
HT Tube G1	13.2588	3.7846	44.8575	1.1447	0.3268	0.4984	1.0524	0.3004
HT Tube G2	6.223	2.159	58.6077	0.4090	0.1419	0.6512	0.3834	0.1330
HT Tube G4	13.2588	3.556	49.3718	0.9649	0.2588	0.5486	0.8985	0.2410
HT Tube G5	13.2588	2.4765	39.0239	1.4032	0.2621	0.4336	1.2921	0.2413
HT Tube G6	11.7348	3.0099	46.9331	1.0694	0.2743	0.5215	0.9788	0.2511
HT Tube G7	6.6294	2.921	53.1247	0.6476	0.2854	0.5903	0.5892	0.2596
HT Tube G8	15.24	4.3815	46.4278	0.8032	0.2309	0.5159	0.7472	0.2148
HT Tube G9	8.2296	2.4765	50.2492	0.7105	0.2138	0.5583	0.6532	0.1966
HT Tube G10	6.096	1.9685	51.0477	0.6936	0.2240	0.5672	0.6349	0.2050
HT Tube G11	23.4442	2.7305	37.3638	1.4959	0.1742	0.4152	1.3715	0.1597
HT Tube G12	15.24	1.778	41.5050	0.9677	0.1129	0.4612	0.8876	0.1036
HT Tube G13	60.96	6.223	39.8666	4.1237	0.4210	0.4430	3.7618	0.3840
HT Tube G14	43.5356	4.445	31.3980	2.8098	0.2869	0.3489	2.5736	0.2628
HT Tube G15	121.92	4.0005	25.0780	7.2838	0.2390	0.2786	6.7133	0.2203
PD Tube J.T1	13.843	4.4069	43.5216	0.9160	0.2916	0.4836	0.8270	0.2633
PD Tube J.T2	11.7348	4.318	46.2742	0.8385	0.3085	0.5142	0.7512	0.2764
PD Tube J.T3	8.9662	4.0132	50.2286	0.7416	0.3319	0.5581	0.6537	0.2926
PD Tube J.T4	6.6294	3.9624	57.7162	0.5649	0.3377	0.6413	0.4962	0.2966
PD Tube J.T5	6.096	3.8989	60.8993	0.4948	0.3165	0.6767	0.4372	0.2796
PD Tube J.E1	16.9418	3.9116	36.4383	1.3002	0.3002	0.4049	1.0638	0.2456
PD Tube J.E2	11.7348	6.5786	47.0540	0.5540	0.3106	0.5228	0.4873	0.2732

Appendix B: Fluted Tube Log

Twist Tube Log

Tube No No	Tube OD	Wall	Mat'l	Grade	Flute OD	Flute ID	Starts	Flutes /FOOT	RATIO
141					(Copy Pictures)				
156.1	.025	.012	AL		.255	.135	3	88	1.17
156.2	.250	.012	AL		.255	.135	3	88	1.17
156.3	.250	.012	AL		.255	.135	3	88	1.17
117	.313	.012	AL		.343	.220	3	42	
50	.375	.012	AL		.400	.245	3	38	1.03
143	.375	.012	AL		.380	.230	4	50	1.11
154.01	.375	.012	AL		.380	.215		32	1.15
154.01	.375	.012	AL		.380	.215		32	1.15
45	.500	.016	AL		.505	.320	4	42	1.00
46	.500	.016	AL		.490	.283	4	44	1.08
47	.500	.016	AL		.495	.300	4	38	1.00
48	.500	.016	AL		.500	.345	4	30	.92
118.01	.500	.016	AL		.470		4	17	1.16
155.01	.500	.016	AL		.512	.318	4	4	1.11
162	.500	.016	AL		.518	.310	4	4.5	1.18
165.1	.500	.016	AL				1Spirale3/16 pi		
165.2	.500	.016	AL				1Spirale3/6 pit		
170	.500	.035	AL		.522	.243	3	42	1.14
36	.625	.020	AL		.615	.495	4	21	.85
37	.625	.020	AL		.630	.445	4	21	.92
38	.625	.020	AL		.625	.400	5	64	1.25
39	.625	.020	AL		.625	.405	4	31	1.00
40	.625	.020	AL		.640	.375	4	32	1.08
41	.625	.020	AL		.640	.395	4	36	1.08
42	.625	.020	AL		.645	.420	4	38	1.08
43	.625	.020	AL		.635	.471	3	20	.89
44	.625	.020	AL		.655	.384	4	29	1.11
145	.625	.020	AL		.625	.409	5	60	1.17
164.2	.625	.020	AL		.660	.398	4	42	1.23
177	.625	.020	AL		.620	.394	5	58	1.20
213	.625	.020	AL		.595	.325	4	47	1.21
226	.625	.035	AL		.680	.340	3	31	1.15
108	.750	.020	AL		.759	.560	4	31	.95
152	.750	.020	AL		.665	.381	5	36	1.10
178	.750	.020	AL		.750	.450	4	39	1.25
191	.750	.020	AL		.785	.505	4	36	1.12
192	.750	.020	AL		.780	.493	4	42	1.26
193	.750	.020	AL		.775	.474	4	48	1.40
203	.750	.020	AL		.785	.475	4	48	1.39
35	.875	.020	AL		.920	.620	4	21	1.11
49	1.000	.035	AL		1.055	.665	4	20.5	1.07
127	1.250	.035	AL		1.245	.675	4	16	1.13
153	1.250	.035	AL		1.232	.712	4	17.5	1.14
7			Copy Pictures						
184	.625	.028	CS	1010 513/5	.660	.325	3	31	1.20
195	.625	.028	CS		.660	.340	3	33	1.18
196	.625	.028	CS		.665	.325	3	31	1.14
197	.625	.028	CS		.650	.319	3	37	1.26
198	.625	.028	CS		.625	.333	4	40	1.15
199	.625	.028	CS		.625	.338	4	48	1.26
201	.625	.028	CS	1010 513/2	.630	.281	3	41	1.26
227	.625	.028	CS		.670	.350	3	31	1.15
168	.625	.035	CS		1.515	1.030	4	48	
202	.625	.035	CS	1010 513/2	.635	.284	3	40	1.14

e*, p*, θ*

Twist Tube Log

Tube No No	Tube OD	Wall	Mat'l	Grade	Flute OD	Flute ID	Starts	Flutes /FOOT	RATIO
113	.875	M	CU		.865	.480	4	32	1.30
126	.875	M	CU		.480	.860	4	36	1.29
179	.875	M	CU		.865	.498	4	29	1.15
43.01	.875	.028	CU		.990	.610	3	21	1.18
15.01	.875	.032	CU		.985	.635	3	16	1.08
44.01	.875	.032	CU				3		
223	.875	.032	CU	3/4M	.865	.480	4	36	1.31
160	1.000	M	CU		1.140	.665	4	25	1.26
189	1.125	M	CU		1.140	.670	4	22.5	1.19
45.01	1.125	.032	CU		1.200	.760	4	18	1.10
214	1.125	.035	CU	1"M	1.130	.665	4	25	1.26
222	1.125	.035	CU	1"M	1.135	.665	4	25	1.26
22.01	1.375	.032	CU		1.200	.705	4	36	1.33
48.01	1.375	.035	CU		1.188	.688	4	25	1.24
118	1.375	.042	CU	1-1/4 M	1.335	.780	4	25	1.31
125	1.375	.042	CU	1-1/4 M	1.375	.850	4	19	1.17
205	1.375	.049	CU		1.435	.830	4	24	1.30
180	1.375	.055	CU	1-1/4 L	1.455	.955	4	18	1.10
175	1.375	.058	CU	1-3/8 ACR	1.425	.830	4	25	1.30
49.01	1.625	.042	CU		1.455	.920	4	18	1.20
50.01	1.625	.042	CU				5		
51.01	1.625	.042	CU		1.200	.705	5	36	1.33
106	1.625	.042	CU	1-1/2 M	1.645	.935	4	15	1.16
208	1.625	.042	CU	1-1/2 M	1.685	1.000	4	13	1.13
225	1.625	.049	CU	1-1/2M	1.590	.900	4	16	1.23
52.01		.032	CU		.985	.635	3	16	1.08
159.1	.156	.010	SS		.160	.070	3	11	1.06
159.2	.156	.010	SS		.160	.072	3	7	.89
53.01	.250	.012	SS		.281		3	5	
105	.250	.016	SS		.280	.135	3	78	1.20
55	.250	.020	SS		Copy Picture				
140	.250	.020	SS		(Copy Pictures)				
56	.313	.016	SS		.353	.172	3	66	1.20
137	.313	.016	SS		.353	.172	3	66	1.20
57	.313	.020	SS		.355	.165	3	76	1.27
130	.313	.020	SS		.355	.165	3		1.27
138	.375	.012	SS		.400	.230	3	48	1.19
155.5	.375	.012	SS		.360	.167	4	48	1.00
186	.375	.012	SS		.380	.212	4	60	1.20
144	.375	.020	SS		.430	.210	3	45	1.20
163	.375	.020	SS		.425		3	51	1.23
61	.500	.016	SS		Flattend Tube				
62	.500	.016	SS		Copy Picture				
109	.500	.016	SS		Flattend Tube				
139	.500	.016	SS		(Copy Pictures)				
151	.500	.016	SS		.497	.278	4	54	1.16
124	.505	.020	SS		.555	.318	3	24	.98
135	.625	.014	SS		.651	.453	5	36	1.11
147	.625	.014	SS		.650		5	36	1.14
148	.625	.014	SS		.640		5	42	1.19
149	.625	.014	SS		.618		5	51	1.28
161	.625	.014	SS		.620	.385	5	4	1.23
174	.625	.014	SS		.596	.357	4	45	1.24
164.1	.625	.016	SS		.660	.398	4	40	1.23
200	.625	.016	SS		.600	.355	4	45	1.26

Twist Tube Log

Tube No	Tube OD	Wall	Mat'l		Grade	Flute OD	Flute ID	Starts	Flutes /FOOT	RATIO
22	.625	.020	SS			.675	.482	4	24	1.06
28	.625	.020	SS			.600	.320	4	54	1.26
119	.625	.020	SS			.643	.418	5	46	
120	.625	.020	SS			.680	.440	4	27	1.07
207	.625	.020	SS	316L		.625	.348	4	47	1.26
100	.625	.028	SS			.699	.327	3	20	1.06
101	.625	.028	SS			.705	.337	3	18	1.00
102	.625	.028	SS			.701	.352	3	16.8	1.00
103	.625	.028	SS			.553	.304	3	21	1.00
104	.625	.028	SS			.555	.306	3	20	1.00
[illegible]	.625	.028	SS	316L		.675	.325	3	34	
132	.625	.028	SS			.720	.390	3	22.5	1.10
134	.625	.028	SS			.600	.300	4	48	1.26
142	.625	.028	SS	316L		.690	.340	3	35	1.32
210	.625	.028	SS			.620	.288	3	41	1.32
19	.625	.030	SS			.715	.390	3	21	1.10
20	.625	.030	SS			.700	.456	3	15	1.01
52	.750	.018	SS			.715	.410	4	35	1.26
133	.750	.020	SS			.715	.420	5	49	1.26
183	.750	.020	SS			.750	.445	4	34	1.10
230	.750	.028	SS	Sea-Cure		.745	.389	4	42	1.26
114	.750	.035	SS			.812	.361	3	27	1.12
115	.750	.035	SS			.820	.390	3	19.636	1.09
116	.750	.035	SS			.822	.437	3	16.8	.99
121	.750	.035	SS			.805	.380	3	24	1.09
123	.750	.035	SS			.805	.380	3	24	1.09
187	.750	.035	SS			.787	.357	3	32	1.20
229	.750	.035	SS	294C		.825	.371	3	28	1.20
14	.875	.025	SS			.900	.500	4	20	
15	.875	.025	SS			.950	.600	4	20	1.11
18	.875	.025	SS			.890	.520	4	32	1.29
24	.875	.025	SS			.812	.455	5	40	1.31
25	.875	.025	SS			.765	.433	5	58	1.58
30	.875	.025	SS			.890	.520	4	27.5	1.23
6	.875	.035	SS			.970	.660	3	11	
16	.875	.035	SS			.985	.510	3	20	1.20
17	.875	.035	SS			.985	.515	3	13	1.06
158.1	.875	.049	SS		404	.980	.440	3	20.25	1.07
158.2	.875	.049	SS		404	.965	.410	3	23	1.10
158.3	.875	.049	SS		404	.965	.410	3	27.5	1.24
85	.970	.535	SS			.930	.515	4	24	1.17
31	1.000	.028	SS			1.070	.640	4	18	1.10
33	1.000	.028	SS			.855	.470	5	50	1.50
34	1.000	.028	SS			.890	.492	5	39	
69	1.000	.028	SS			1.185	.750	3	11.5	1.06
70	1.000	.028	SS			1.100	.750	4	19	1.07
71	1.000	.028	SS			1.100	.725	4	24.5	1.24
86	1.000	.028	SS			.930	.515	4	24	
90	1.000	.028	SS			1.093	.740	4	18	1.10
92	1.000	.028	SS			.970	.635	5	30	1.09
93	1.000	.028	SS			1.093	.740	4	18	1.10
107	1.000	.028	SS			.945	.540	4	30	1.22
169	1.000	.028	SS			.880	.494	5	32	1.14
176	1.000	.028	SS			.985	.569	4	37	1.45
204	1.000	.028	SS	SEA CURE		.970	.525	4	32	1.30

Twist Tube Log

Tube No	Tube OD	Wall	Mat'l	Grade	Flute OD	Flute ID	Starts	Flutes /FOOT	RATIO
121	1.125	.028	SS		1.205	.728	4	16	1.08
122	1.125	.028	SS		1.200	.717	4	19	1.13
122.2	1.375		SS	294C	1.160	.687	4	20	1.13
166	1.375		SS	6XN	1.185	.765	4	14.5	1.04
167	1.375		SS	6XN	1.180	.755	4	4.5	1.04
51	1.500	.035	SS		1.485	.930	5	22	1.29
54	1.500	.035	SS		1.455	.915	5	29	1.43
157.2	1.500	.035	SS		1.475	.940	5	22	1.21
157.3	1.500	.035	SS		1.500	1.020	5	17	1.08
215	1.500	.035	SS	304	1.475	.940	5	21	1.21
131	2.500	.016	SS		Pressed	Can	5.200 Pitch		
146		.035	SS		3.070	3.000	.85		Pressed
111.01	.188	.012			.200	.098	3	6	
112.01	.250	.012			.285	.005	3	6.5	
112.01	.250	.012			.285	.005	3	6.5	
113.01	.250	.012			.268	.005	3	7	
114.01	.250				.250	.002	3	18	
115.01	.250				.268		3	24	
116.01	.500	.016			.468	.245	4	50	
117.01	.500	.016			.422	.195	4	4	1.09
119.01	.620				.596			6.5	1.45
120.01	.620				.590			7	1.49
121.01	.620				.603			6.5	1.43
122.01	.620				.610			6	1.43
123.01	.620				.623			6	1.40
124.01	.620				.609			6.5	1.42
125.01	.620				.605				
126.01	.620				.616			6.5	1.43
127.01	.620				.616			6.5	1.38
128.01	.620				.616			6.5	1.38
129.01	.620				.615			6.5	1.39
130.01	.620				.590			6.5	1.39
131.01	.620				.615			6.25	
132.01	.620				.614			6.25	1.39
133.01	.620				.617			6.5	
134.01	.625	.020			.615	.297	3	20	1.07
135.01	.625	.020			.630	.330	3	30	
136.01	.625	.020			.655	.364	4	40	
138.01	.625	.020			.620		5	6	1.43
139.01	.625	.020			.618		4	5	1.43
141.01	.625	.030			.712	.003	3	15	
142.01	.625	.030			.710	.005	3	22	
145.01	.625				.725			7	
146.01	.625				.725			26	
147.01	.625				.788			13	
148.01	.625				.736			13	
149.01	.625				.785			9	
150.01	.750	.018			.675	.400	5	60	1.41
151.01	.750	.020			.669	.403	4	21	
152.01	.750	.020			.675	.004	5	3.8	1.20
153.01	.750				.817			11	
158.01	.875	.025			.963	.630	4	21	
165.01	.875	.025			.865	.490	4	10	1.14
166.01	.875	.025			.865	.005	4	10'	1.14
167.01	.875	.035			1.000	.002	3	13	

Twist Tube Log

Tube No	Tube OD	Wall	Mat'l	Grade	Flute OD	Flute ID	Starts	Flutes /FOOT	RATIO
185	.750	.028	CS	1010 513/5	.765	.410	4	37	1.20
155.1	.750	.035	CS		.830	.443	3	19.5	1.04
155.2	.750	.035	CS		.840	.472	3	18.5	1.05
155.3	.750	.035	CS		.845	.490	3	18	1.02
155.4	.750	.035	CS		.850	.490	3	18 3/4	1.04
188	.750	.035	CS		.840	.475	3	17	1.04
129	1.000	.028	CS		1.070	.625	4	17.5	1.06
136	1.000	.049	CS		1.115	.600	3	16	1.07
150	3/4	M	CU		.860			20	.89
21.01	.056	.032	CU		1.150	.665	4	26	1.29
172	.375	.012	CU		.410	.196	3	56	1.16
182	.375	.025	CU		.387	.187	3	53	1.03
213	.497	.023	CU		.487	.256	4	67	1.22
[illegible]	.497	.023	CU		.490	.259	4	60	1.15
220	.497	.023	CU		.480	.249	4	74	1.26
221	.497	.023	CU		.485	.255	4	73	1.26
32.01	.500	M	CU		.500	.240	3	42	1.18
29.01	.500	.016	CU		.490	.280	4	60	1.17
30.01	.500	.016	CU		.490	.280	4	54	1.19
173	.500	.016	CU		.510	.305	4	64	1.34
1	.500	.025	CU		.512	.245	3	42	1.15
31.01	.500	.025	CU		.500	.240	3		1.18
33.01	.500	.025	CU		.544	.280	3	48	1.27
34.01	.500	.025	CU				3		
35.01	.500	.025	CU				3		1.18
181	.500	.025	CU		.500	.240	3	46	1.20
217	.500	.025	CU		.485	.260	4	65	1.22
228	.500	.025	CU		.512	.248	3		1.14
37.01	.500		CU		.484	.003	4	57	1.29
154.1	.625	M	CU		.685	.450	3		
154.2	.625	M	CU		.692	.410	3	30	1.07
154.3	.625	M	CU		.666	.490	3	24	1.09
154.4	.625	M	CU		.666	.500	3	20	1.09
154.5	.625	M	CU		.666	.500	3	20	1.00
154.6	.625	M	CU		.666	.520	3	30	1.09
190	.625	M	CU		.655	.349	3	33	1.14
194	.625	M	CU		.603	.335	4	46	1.11
171	.625	.020	CU		.628	.349	4	44	1.15
38.01	.625	.025	CU		.688	.425	3	16	1.01
2	.625	.028	CU		.628		4	44	1.15
4	.625	.028	CU		.655	.349	3	33	1.15
36.01	.625	.028	CU		.625	.355	4	48	1.16
39.01	.625	.028	CU		.705	.405	3	30	1.15
40.01	.625	.028	CU		.665	.422	3	48	1.20
206	.625	.028	CU		.625	.348	4	44	1.12
209	.625	.028	CU	1/2 H	.585	.293	4	55	1.23
212	.625	.028	CU	1/2 H	.620	.333	4	58	1.32
216	.625	.028	CU	1/2H	.623	.340	4	51	1.22
3	.625	.032	CU		.630		4	48	1.15
211	.625	.040	CU	1/2 L	.628	.304	3	43	1.20
42.01	.750	.028	CU		.740	.387	4	11	1.35
224	.750	.028	CU		.750	.385	4	40	1.34
110	.875	M	CU		.865	.480	4	32	1.30
111	.875	M	CU		.865	.480	4	32	1.30
112	.875	M	CU		.865	.480	4	32	1.30