FINAL DESIGN REPORT

DEVELOPING A DIESEL EXHAUST HEAT RECOVERY SYSTEM

Submission Date: *December 6, 2010*

Sponsoring Company: *New Flyer Industries*

Project Advisor: *Dr. Kuhn*

Team #9: *Energy Bankers*

Team Members:

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UNIVERSITY OF MANITOBA

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Energy Bankers
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Dear Dr. Kuhn:

The Energy Bankers are pleased to submit our Final Design Report titled “Proposed Diesel Exhaust Heat Recovery System” for your review. This report includes an overview of our project’s definition, such as, background and problem statement, project objectives and target specifications of our design. The main focus of this report is on the “Details of the Design”. This section provides information on the design methodology, including heat transfer and fluid dynamic theories and optimization techniques that were applied to our final design; these theories and supporting calculations and optimization techniques justify the geometry and style of heat recovery system that we have recommended. In order to provide a complete consultation to our client, this report also includes a cost analysis and bill of materials.

Our client, New Flyer Industries, has requested our help in designing a system that will capture heat from the exhaust system of an [redacted] diesel engine currently used in their 40 foot buses. This report provides a detailed summary of the techniques and tools we have used to develop, select and analyse concepts that will address this design challenge. Additionally, the appendices of this report will provide details on the design calculations, optimization, concept analysis and selection, and detailed drawings of the design.

It is our intent that upon complete review of this report, you will have a complete understanding of the heat recovery system we have designed as well as how it will integrate into the current exhaust and coolant system being used in New Flyer’s 40 foot bus. Should you have any questions, please contact us at energy-bankers.com, and we would be happy to answer any questions you may have.

Sincerely,

Trevor J. Price
Technical Secretary
Energy Bankers

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Abstract

This report describes the design process used to develop a diesel exhaust heat recovery system for a New Flyer 40 ft bus and proposes a design that meets the principle design constraints.

A heat recovery system is required to replace the auxiliary heater currently used in New Flyer's 40 ft bus. The auxiliary heater is used to supplement in cabin heat and maintain engine temperature. The diesel fired auxiliary heater emits unfiltered exhaust gas and lowers fuel economy. These by-products of its function are environmentally damaging and un-economical.

In order to replace the function of the auxiliary heater with a heat recovery system, several constraints and performance specifications, identified by New Flyer, need to be respected. Specifically, the heat recovery system must produce 23 kW of heat and impose no more than 3386 Pa of back pressure onto the exhaust system. Also, it should not measure more that 812.8 mm long, 304.8 mm in diameter and weigh no more than 26 kg. Additionally, the new system should cost less than $3000. These space and weight constraints imposed by New Flyer, limited the variety and size of possible designs that would meet the performance expectations stated above. Furthermore, adhering to industry heat exchanger design standards, such as TEMA (Tubular Exchange Manufacturing Association), imposed additional restrictions on the design possibilities.

In order to address these design constraints while achieving the required performance, numerical and optimization software, such as MATLAB, EXCEL and SOLIDWORKS were used to optimize and check possible heat exchanger geometries that met both TEMA standards and the size constraints imposed by New Flyer.

In addition to providing a detailed analysis of the heat exchanger, details of integrating it into the bus's exhaust and coolant system are provided in this report. A cost analysis, including the cost of manufacturing the heat exchanger, installation and the addition of tubing and valves required to control the flow of coolant and exhaust gas through the new system.

Based on our analysis, we have designed a single pass, shell and tube heat exchanger that meets the performance specifications but is outside of the space constraints. The exchanger should perform at approximately 30% efficiency; providing 24 kW at peak torque and an additional back pressure of 2.72 kPa at maximum power. The overall length of the design is 766.6 mm

which is within the space constraints; however the overall diameter is 336.4 mm which is beyond the height constraint by 32.4 mm. The weight of the heat exchanger is 100.5 kg; therefore, it does not meet the weight constraint. However, the overall cost of our design is $2957.40 which is within budget.

1.0 Introduction: Problem Description

The following final design report outlines a proposed Exhaust Heat Recovery System that can be implemented on New Flyer's buses, specifically a forty foot bus with an 8.9 L ISL engine. New Flyer set the objective of capturing wasted heat from the exhaust system to increase the overall fuel efficiency of the bus. The report includes a detailed description of the design chosen, as well as the methodology used to develop a solution that meets New Flyer's design constraints. Consideration for how an Exhaust Heat Recovery System will be implemented into the current system is included, along with a cost analysis. The report also includes the team's project schedule for completing the Exhaust Gas Heat Recovery System Project including the final Gantt chart and work breakdown structure.

1.1 Customer Needs

New Flyer Industries would like to collect heat energy from the exhaust gas on their diesel buses and use that energy to reduce emissions by reducing or eliminating the use of an auxiliary heater. Current diesel engines, which are provided by [redacted] convert fuel to mechanical energy with approximately 35% of the energy wasted as exhaust heat [1] . Buses designed by New Flyer, specifically those for the cold winter climates of Canada and parts of the Northern United States of America, require an auxiliary heater to keep the engine coolant warm. The warm coolant is used to maintain in-cabin heat, maintain engine temperature and pre-heat the engine in cold weather. The auxiliary heater burns diesel fuel, which reduces the overall fuel economy and increases emissions. By capturing waste heat from the exhaust stream, there is an opportunity to replace the auxiliary heater and therefore reduce fuel consumption and emissions. [2].

In order to comply with the EPA/CEPA 2010 (Environmental Protection Agency/Canadian Environmental Protection Act) on-highway vehicle emissions regulations, a complicated exhaust system is already used on all [redacted] engines installed by New Flyer [1], [2]. As illustrated in

Figure 1, the exhaust system is composed of a particulate filter and a selective catalytic reduction (SCR) module. [redacted] has strict restrictions on the temperature drops before the filtration system, limiting the portion available for heat capture to the tail pipe. This information is proprietary; therefore, it will not be discussed further in this report. However, New Flyer set constraint limits to the design, such as back pressure and temperature drops that will allow the design to meet [redacted] specifications.

The auxiliary heater is also used to decrease engine warm up time by heating the coolant and circulating it through the engine block before the engine is started. If the auxiliary heater is to be replaced by an exhaust heat recovery system, there is a need for thermal energy storage. This thermal storage will need to be utilized in two ways: to maintain the engine at its operating temperature during idle periods and to warm up the engine before a cold start.

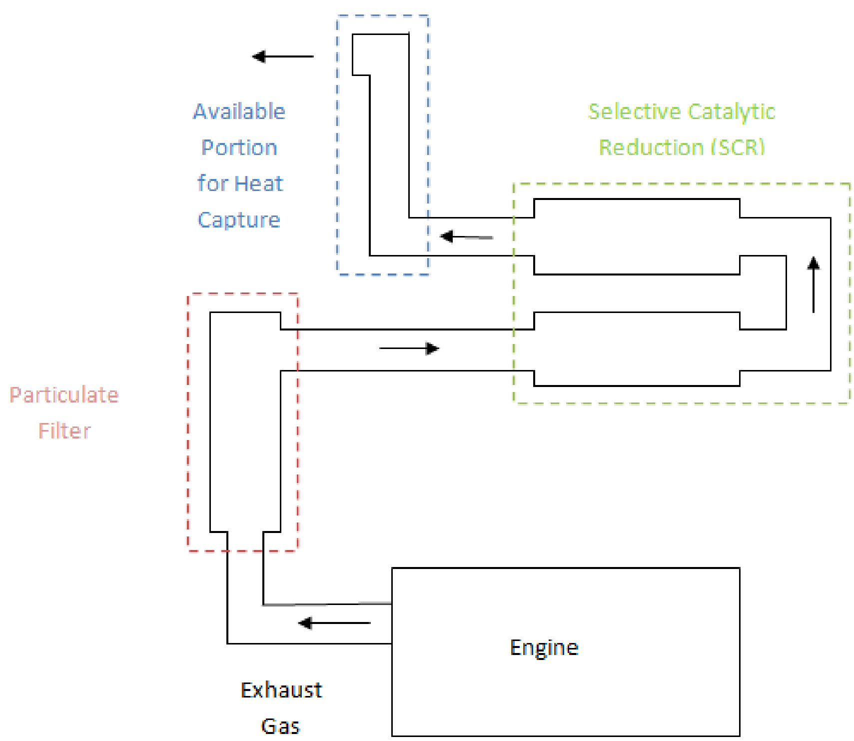


FIGURE 1: DIESEL FILTRATION SYSTEM

New Flyer's problem statement is to develop a means of capturing waste heat from the exhaust in order to replace the auxiliary heater on their diesel buses. Due to time constraints, a means of capturing waste heat is only included in this report; a means of storing thermal energy will need to be included at a later date pending the client's decision to continue with the project. Also, all data, calculations and specifications will be based on a 40 ft bus that uses a [redacted] engine, as per New Flyer [1].

1.2 Project Objectives

The objectives of the team's design resulting from New Flyer's problem statement and needs will be described in the following section. It will also include the heat recovery system portion of the design problem and the heat storage problem. The heat storage section is only included to give a clear understanding of the complete problem. No attempt is made in this report to propose a viable heat storage unit.

1.2.1 Heat Exchanger

The goal of this project is to eliminate the need for an auxiliary heater, resulting in improved system efficiency. This can be achieved by incorporating a heat exchanger that transfers heat from the exhaust system to the bus cabin and a latent heat storage system. The proposed system includes revision of the bus heating system to accommodate a heat exchanger, additional coolant system and control as well as exhaust modifications. A total estimate of procuring parts and manufacturing an installation of the proposed heat exchanger is also included.

Heat transfer from exhaust gas will be performed using a heat exchanger. The heat transfer must occur at a high rate, since a large amount of thermal transfer is required over a short distance, to maximize heat recovery. The selection of an appropriate type and size of heat exchanger that will accomplish this is discussed in the details of design section of this report.

1.2.2 Heat Storage

During the initial assessment of the project, the team presumed that the need for heat storage is an integral part of the exhaust gas heat recovery system in order to eliminate the existing diesel-fueled auxiliary heater. A possible solution considered for heat storage is a Phase Changing Material (PCM) heat accumulator. However, due to the complexity of the system and time limitation, the team and the client agreed to focus on the heat exchanger component of the system to replace the primary function of the auxiliary heater.

1.3 Target Specifications

The following section details the specifications and constraints that must be met and respected in order to develop a functional design that addresses New Flyer's problem statement. Specifications on cabin and engine heat, as well as physical and mechanical constraints, are included to clarify the project scope.

1.3.1 Cabin Heat

The required energy for cabin heat is 35 kW. Currently, 10 kW and 25 kW are supplied by the auxiliary heater and the engine cooling system, respectively. This calculation is based on a temperature rise of 90°F inside the cabin over 70 minutes and maintaining the temperature at 70°F [3].

1.3.2 Engine Heat

Currently, the auxiliary heater must be turned on and provide up to 25 kW (approximately 90% efficiency) to pre-heat the engine prior to ignition. Any heat storage system would have to meet or exceed this energy requirement.

1.3.3 Specifications and Constraints

The above target specifications are considered to be the minimal design constraints. The design must meet the specifications listed or else the goal of eliminating the auxiliary heater cannot be achieved. Once the specifications are met, no additional heat is required to be removed from the system.

In addition to the specifications listed earlier, there are several specifications that arise from the constraints given for the problem. Table I lists the details of each design constraint and what specification is required.

TABLE I: PHYSICAL CONSTRAINTS AND LIMITATIONS [1]

Constraint	Details	Specification
Maximum Allowable Back Pressure	The maximum allowable back pressure in the exhaust system is 8.2 in-Hg. The current system has approximately 7 in Hg allowing for the addition of up to 1 in-Hg.	1 in-Hg
Maximum Allowable System Weight	The design of the new system should not exceed the weight of the current auxiliary heater (19 kg), the coolant it holds (2 kg) and the exhaust piping it uses (5 kg).	26 kg
Engine Operating Temperature	The engine operating range is from 160 °F to 225 °F. The engine should reach the minimum as quickly as possible when starting.	160 °F - 225 °F
Maximum Vibrations	The maximum operating vibrations are up to a 50 Hz spectrum density.	Less than 50Hz
Available Space	The available space for the heat exchanger system is limited to 12 in (H) x 32 in (W) x 18 in (D). The heat exchanger must be located on the tailpipe after the emissions components of the exhaust system	12 in (H) x 32 in (W) x 18 in (D)
Materials	Must meet or exceed the corrosion resistance of 304 stainless steel.	---
Electrical	Additional electrical components must be compatible with a 24V system	24 V

2.0 Details of the Design

In the previous Concept Design Report, our team decided to further investigate three concepts as possibilities for our final design. Table 2 illustrates and describes each of these designs. For the purposes of this report, their reference numbers have been changed from the Concept Development Report to designs 1, 2, and 3.

TABLE 2: THE INITIAL THREE CONCEPTS UNDER CONSIDERATION FOR THE FINAL DESIGN.

	Design 1 The exhaust gas flows through the inner tube which is equipped with longitudinal fins. The cross section area of inner tube expands to compensate pressure drop. The coolant fluid flows over the inner tube through the annulus section.
	Design 2 The exhaust gas flows through the internally finned inner tube to enhance heat transfer rate on the exhaust side. The coolant fluid flows through the annulus section inside the outside shell.
	Design 3 Shell and tubes arrangement. The exhaust flows through smaller tubes and the coolant fluid flows over the tubes inside the shell.

Upon further study of heat transfer and fluid dynamic theories, it became evident that design 3 would provide the most effective heat transfer based on the surface area in contact with the exhaust gas (Appendix A: Concept Analysis and Selection). Similarly, design 3 would add the least amount of back pressure to the system since the pipes are in parallel flow.

This section will describe how heat transfer and fluid dynamic concepts were used to aid in the analysis and selection of this design. It will also include the tools that were used for numerical analysis and optimization of the design.

2.1 Design Methodology

The following section describes the design methodology used to select the heat exchanger geometry. All geometric standards are based on Tubular Exchange Manufacturers Association's (TEMA) standards as presented in the Heat Exchanger Design Handbook [4].

2.1.1 Outline of Design

A shell and tube heat exchanger consists of an outer shell, with a bundle of tubes located within the shell. A hot fluid passes through the tubes while a cold fluid flows through the shell side of the exchanger. Baffles are used on the shell side of the heat exchanger to force the fluid into a cross flow pattern across the tubes, increasing the overall heat transfer of the heat exchanger. Figure 2 outlines the basic components of a shell and tube heat exchanger. Shell and tube heat exchangers are commonly used in industry and are known for their robust design, and large tube side surface areas. [4]

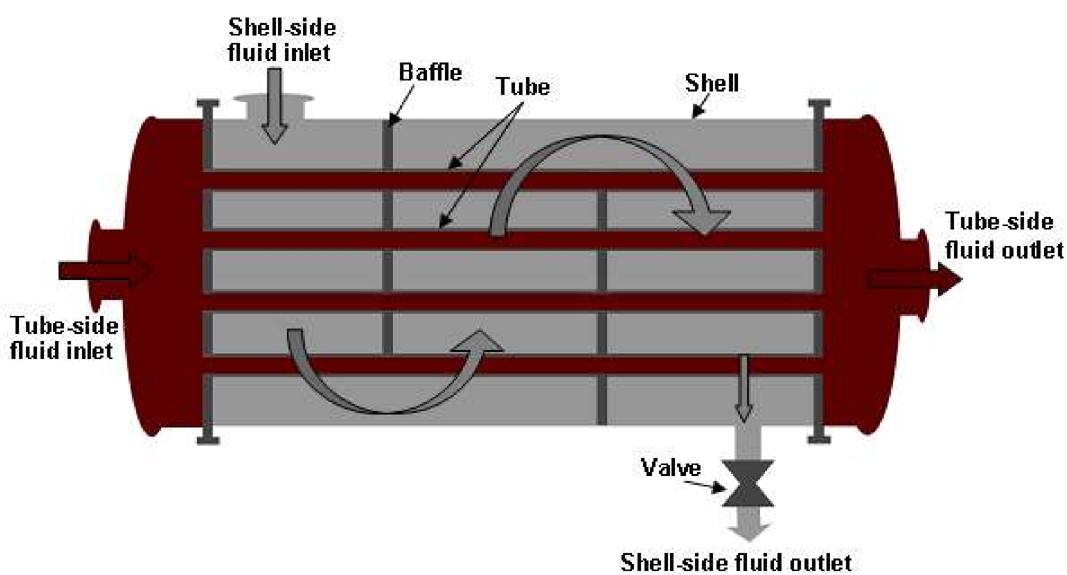


FIGURE 2 - SHELL AND TUBE HEAT EXCHANGER COMPONENTS [5]

In order to analyze a shell and tube heat exchanger, the geometries must be known, assumed or calculated. The following subsection details each of these geometries.

2.1.2 Shell and Tube Geometries

The tube and tube layout consist of the following geometries. Each of the geometries is given a symbol and a short description.

TABLE 3 - TUBE AND TUBE LAYOUT GEOMETRIES [4]

Tube and Tube Layout		
Description	**Symbol**	**Unit**
Inside Shell Diameter	D_s	mm
Tube Outside Diameter	D_t	mm
Tube Wall Thickness	L_{tw}	mm
Inside Tube Diameter	D_{ti}	mm
Tube wall material thermal conductivity	λ_{tw}	W/m K
Pitch Ratio	P_R	
Tube layout pitch	L_{tp}	mm
Tube layout characteristic angle	θ_{tp}	deg

The inside shell diameter, tube outside diameter, tube wall thickness, inside tube diameter and tube wall material thermal conductivity are implicit in their description and are shown in Figure 3. The pitch ratio is the ratio of the tube layout pitch over the outside tube diameter. The tube layout pitch determines the cross flow area between the tubes and is the center to center distance of the two adjacent tubes. The tube layout characteristic angle is the angle between columns of tubes in a tube bank. These geometries are summarized in Table 4.

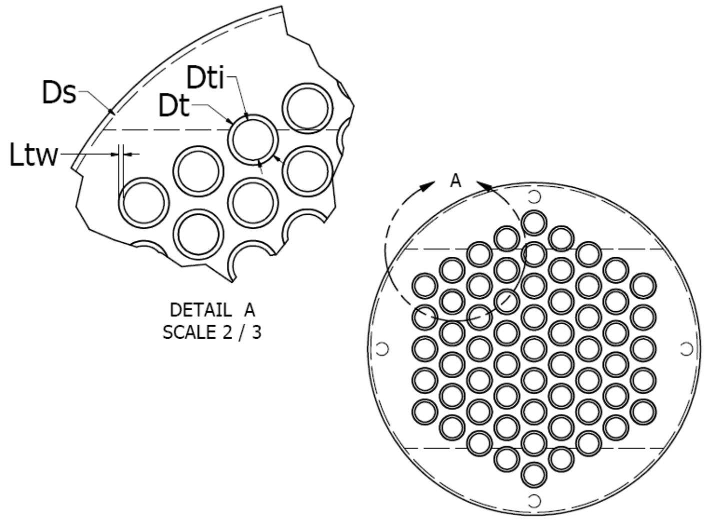
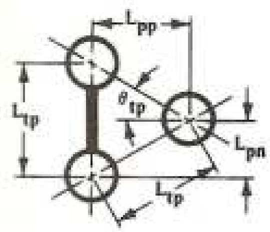
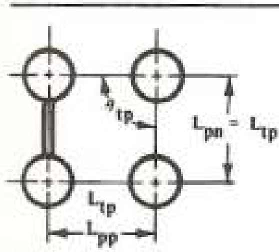
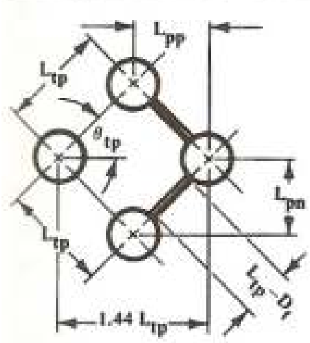


FIGURE 3 - TUBE GEOMETRY [Modeled by Fajar Firmansyah and Drawn by Travis Temple].

TABLE 4 - TUBE LAYOUT GEOMETRY [4]

Cross flow →	θ_{tp}	L_{tp}	L_{pp}
	30°	$0.5L_{tp}$	$0.866L_{tp}$
	90°	L_{tp}	L_{tp}
	45°	$0.707L_{tp}$	$0.707L_{tp}$

The tube length and associated tube lengths are described in Table 5. The baffled tube length is the overall nominal tube length minus the width of all the baffles used. If a single tube pass is used, the effective tube length is equal to the baffled tube length. Figure 4 illustrates each of the tube length geometries. L_{ts} stands for the tube sheet thickness and is calculated in the Summary of Results section of this report.

TABLE 5 - TUBE LENGTH GEOMETRIES

Tube Length		
Overall nominal tube length	L_{to}	mm
Baffled tube length	L_{ti}	mm
Effective tube length for heat transfer	L_{ta}	mm

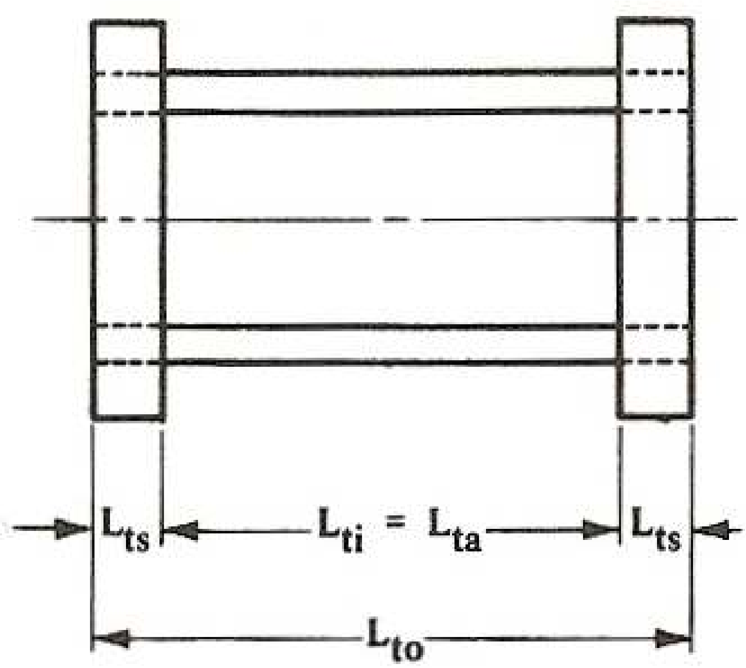


FIGURE 4 - TUBE LENGTH DEFINITIONS [4]

The baffle geometries are described in Table 6. The baffle cut is the height of the baffle as a percent of the inside diameter of the shell. From this percent the height of the baffle cut can be calculated and is shown in

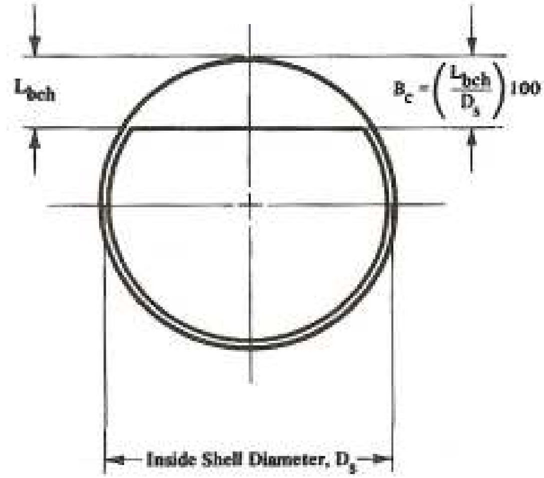


Figure 5. The central baffle spacing is the spacing between two adjacent baffles. If the central baffle spacing cannot be equally divided into the nominal tube length, different inlet and outlet baffle spacing are used. These parameters are shown in Figure 6.

TABLE 6 - BAFFLE GEOMETRIES

Baffle Geometry		
Baffle cut as percent of Ds	B_c	%
Central baffle spacing	L_{bc}	mm
Inlet baffle spacing (optional)	L_{bi}	mm
Outlet baffle spacing (optional)	L_{bo}	mm

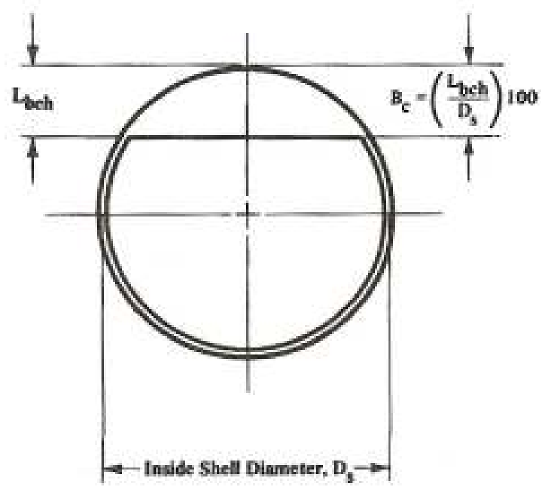


FIGURE 5- BAFFLE CUT [4]

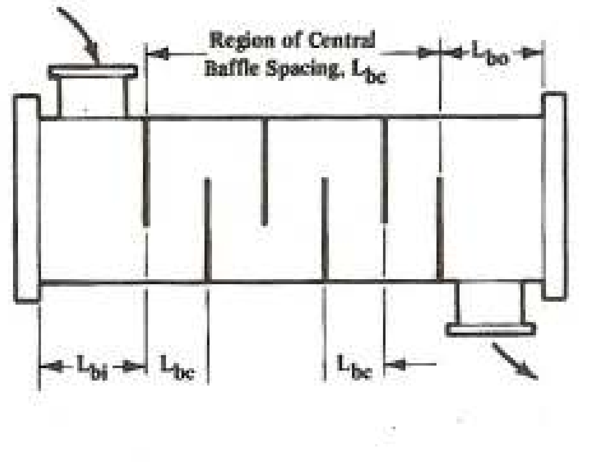


FIGURE 6 – BAFFLE DISTRIBUTION [4]

The tube bundle geometries are described in Table 7. Tube bundle designs include fixed tube (FX), which has the lowest construction cost, but the geometry has limited thermal expansion. The fixed tube design has been chosen. The UT stands for U-Tube construction, SRFH stands for split ring floating head, PFH stands for packed floating head and PTFH Pull through floating-head bundle. The details of the other designs can be found in the Heat Exchanger Design Handbook by Hewitt [4] or in the standards of the Tubular Exchanger Manufacturers Association [6]. The remaining geometries in Table 7 are used for laying out the tube bundle within the shell and also to calculate the heat transfer on the shell side of the heat exchanger. Figure 7 illustrates the baffle geometries

TABLE 7 - TUBE BUNDLE GEOMETRIES

Tube Bundle Geometry		
Total number of tubes	N_{tt}	
Number of tube passes	N_{tp}	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	code
Tube OD-to baffle hole clearance	L_{tb}	mm
Inside shell to baffle clearance	L_{sb}	mm
Inside shell-to-tube bundle bypass clearance	L_{bb}	mm
Center of outermost tube to center of shell	D_{ctl}	mm

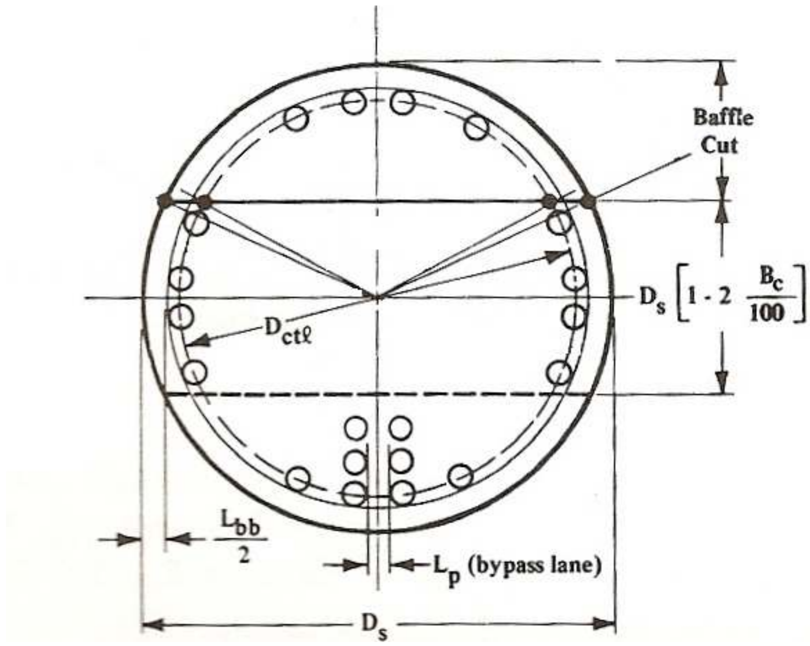


FIGURE 7 - BASIC BAFFLE GEOMETRIES [4]

In addition to the geometries of the design the inlet and outlet temperatures of the shell side and the tube side of the exchanger must also be known for the analysis. They are described as follows:

TABLE 8- TEMPERATURE DEFINITIONS

Temperatures		
Shell-side inlet temperature	T_{si}	°C
Shell-side outlet temperature	T_{so}	°C
Tube-side inlet temperature	T_{ti}	°C
Tube-side outlet temperature	T_{to}	°C

Finally the fluid properties are needed for both the shell side process and the tube side process of the exchanger. This includes the mass flow rates, as well as the thermo physical properties of the fluid. These include the density, thermal conductivity, specific heat, and dynamic viscosity. These thermal and fluid properties are taken at the mean fluid temperature of the shell side and the tube side of the heat exchanger. The properties are defined in Table 9 and Table 10.

TABLE 9 - SHELL SIDE PROCESS INFORMATION

Shell-side process information		
Shell fluid mass flow rate	M_s	kg/s
At shell fluid mean temperatures	$T_{s,av}$	°C
Density	ρ_s	kg/m3
Thermal Conductivity	Λ_s	W/m K
Specific Heat	$(cp)_s$	J/kg K
Dynamic Viscosity	η_s	cP=mPa/s

TABLE 10 - TUBE SIDE PROCESS INFORMATION

Tube-side process information		
Tube fluid mass flow rate	M_t	kg/s
At tube fluid mean temperature	$T_{t,av}$	K
Density	ρ_t	[kg/m^3]
Thermal Conductivity	λ_t	W/m K
Specific Heat	$(cp)_t$	J/kg K
Dynamic Viscosity	M	[kg/m s]

2.1.3 Heat Transfer Design Calculations

The following section outlines the methodology used to design a heat exchanger capable of producing 25 kilowatts of heat to the coolant system. The methodology was taken from the Heat Exchanger Design Handbook [4]. These equations are for an ideal heat exchanger and do not take into account for minor losses associated with each proposed geometry. Due to the fact that these are ideal calculations, a prototype should be developed and used for experimentation to verify the results.

The heat transferred from the shell and tube heat exchanger is calculated using Equation 1. This is the ideal heat transfer and correction factors can be applied to obtain a more accurate heat transfer.

$$Q = (U_i)\pi(D_{ti})(N_{tt})(L_{ta})(\Delta T_{lm}) \qquad \textbf{EQUATION 1}$$

where:

U_i = Overall internal heat transfer coefficient

ΔT_{lm}= Log mean temperature difference

The overall heat transfer coefficient is calculated using Equation 2. This is adapted from fundamentals of mass and heat transfer by Incropera et al. [7]. It is a ratio of the coefficient of convective heat transfer of the tube side, the shell side and the thermal resistance of the tube wall. It combines the two coefficients, h_t and h_s, so the overall heat transfer can be calculated using Equation 1.

$$U_i = \left[\frac{1}{h_t} + \frac{D_{ti}}{h_s D_{to}} + \frac{D_{ti}}{2\lambda_{tw}}\ ln\frac{D_{to}}{D_{ti}}\right]^{-1} \qquad \textbf{EQUATION 2}$$

where:

h_t = Tube side heat transfer coefficient

h_s = Shell side heat transfer coefficient

The log mean temperature is the logarithmic average of the temperature difference between the hot and cold streams of the heat exchanger. The heat exchanger used in this design is set up in counter flow and is calculated using **Error! Reference source not found.**.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} \qquad \textbf{EQUATION 3}$$

where:

$\Delta T_1 = T_{ti} - T_{so}$

$\Delta T_2 = T_{to} - T_{si}$

The effective tube length for heat transfer is the overall nominal tube length minus the total width of the baffles and tube sheet. Since this is an unknown parameter before the analysis begins, it can be estimated using Equation 4.

$$L_{ta} = L_{to} - 2(0.1)D_s \qquad \textbf{EQUATION 4}$$

To determine the heat transfer coefficient for the exhaust in the tube side of the exchanger Equation 5 is used.

$$h_t = \frac{Nu_t \lambda_t}{D_{ti}} \qquad \textbf{EQUATION 5}$$

where:

Nu_t = Tube side Nusselt number

The Nusselt number is a unitless factor that describes the heat transfer at the tube surface. It is calculated using Equation 6 for the tube side of the heat exchanger

$$Nu_t = 0.023\,(Re_t)^{4/5} {Pr_t}^{0.3} \qquad \textbf{EQUATION 6}$$

where:

Re_t= Tube side Reynolds number

The Reynolds number for the tube side is calculated using Equation 7. It is a measure of whether or not the flow of the fluid through the pipe or tube is laminar or turbulent.

$$R_e \leq 2300\ for\ laminar\ flow$$

$$R_e > 2300\ for\ turbulent\ flow$$

$$Re_t = \frac{4\ M_{tt}}{\pi D_{ti} \mu_t} \qquad \textbf{EQUATION 7}$$

where:

M_{tt} = mass flow rate per tube = M_t / N_{tt}

μ = Dynamic viscosity of air

Assuming the flow is evenly distributed through all of the tubes. Actual flows must be determined experimentally

This finishes defining Equation 5, the heat transfer coefficient for the tube side. The heat transfer coefficient for the shell side is calculated Equation 8.

$$h_s = j_i (Cp)_s \, \dot{m}_s (Pr_s)^{-\frac{2}{3}} \, \Phi^r$$ **EQUATION 8**

where:

j_i = Heat transfer coefficient

$\dot{m}_s$ = Max shell side cross flow velocity

Φ^r = Correction factor for viscosity gradient

The heat transfer coefficient is a factor based on the geometry of the shell and tube heat exchanger and is calculated using Equation 9.

$$j_i = (a_1) \left(\frac{1.33}{\frac{L_{tp}}{D_t}} \right)^a (Re_s)^{a_2}$$ **EQUATION 9**

where:

$$a = \frac{a_3}{(1 + 0.14(Re_s)^{a_4}}$$

The values for a_1, a_2, a_3, and a_4 are the correlation coefficients for the flow of fluid through the shell side of a shell and tube heat exchanger and are taken from Table 1-B in Appendix B [4].

The correction factor for viscosity gradient takes into account the viscosity at the tube wall versus the viscosity at the average bulk fluid temperature. The correction factor is calculated using Equation 13.

$$\Phi^r = \left(\frac{\eta_s}{\eta_{s,w}} \right)^{0.14}$$ **EQUATION 10**

where:

$\eta_{s,w}$ = the shell fluid viscosity at the tube wall

temperature

The maximum shell side cross flow mass velocity is the maximum velocity of the fluid flowing through the tubes. The shell fluid is in a cross flow configuration and due to contraction from the spacing of the tubes, the fluid velocity increases. The maximum shell side cross flow velocity is calculated using Equation 11.

$$\dot{m}_s = \frac{M_s}{S_m}(10^6) \qquad \textbf{EQUATION 11}$$

where:

S_m= cross flow area

(10^6) =Conversion from mm^2 to m^2

The cross flow area is calculated from the tube layout geometry using Equation 12.

$$S_m = L_{bc}\left[L_{bb} + \frac{D_{ctl}}{L_{tp,eff}}\left(L_{tp} - D_t\right)\right] \qquad \textbf{EQUATION 12}$$

where:

$L_{tp,eff}$=L_{tp} for 30° and 45° layouts

L_{tp}(0.707) for 45° staggered layout

The Reynolds number is calculated using a different equation than the one used for the tube side Reynolds number. It is calculated using Equation 13.

$$Re_s = \frac{D_{t\,s}}{\eta_s} \qquad \textbf{EQUATION 13}$$

Finally to fully define Equation 8 the Prandlt number has to be determined. It is calculated instead, using Equation 14.

$$Pr_s = \frac{c_{p,s}\eta_s}{\lambda_s}(10^{-3}) \qquad \textbf{EQUATION 14}$$

where:

(10^3) = is a correction factor from the input values

2.1.4 Pressure Drop Design Calculations

The total allowable back pressure allowed on the exhaust system is 8.2" Hg or 27.765 kPa. Due to the particulate filter and SCR in use in all New Flyer buses, currently the back pressure on the system is 7.2" Hg. This allows 1" Hg or 3386 Pa allowable back pressure to design the heat exchanger around while meeting all other constraints imposed by New Flyer. The following theories were used to accomplish these calculations.

Pressure difference on a horizontal flow system is found by,

$$\Delta P = h_l + h_{lm} \qquad \textbf{EQUATION 15}$$

where:

h_l = major system losses = $\rho\frac{V^2}{2}\left(f\left(\frac{L}{D}+\sum\frac{\mathrm{L_e}}{\mathrm{D}}\right)\right)$

h_{lm} = minor system losses = $\boldsymbol{\rho\frac{V^2}{2}\sum k}$

ρ = density of dry air at a specific temperature

V = Velocity of air

f = Friction factor

L = Length of pipe

D = Diameter of pipe

L_e/D = Equivalent length of pipe due to elbows, valves, etc...

K = Frictional loss coefficients

Density is found by using Table 1-C: Properties of dry air, found in Appendix C. L_e/D is found from Table 2-B in Appendix B. Length and diameter of pipe are estimated based on the space constraints from the problem statement and definition. The velocity and friction factor are calculated with the following equations.

The velocity is found by,

$$V = \frac{4\dot{m}}{\pi D_i^2 \rho}$$ **EQUATION 16**

where:

$\dot{m}$ = Mass flow rate of dry air

D_i = Inside diameter of pipe/tube

The mass flow rate is known from empirical data at a particular engine rpm found in Appendix D, Table 1-D; it is adjusted for temperature. The mass flow data was given to the Energy Bankers by New Flyer.

The friction factor is found by using the Colebrook Equation,

$$f = \left\{-2\log_{10}\left[\left(\frac{e/D_i}{3.7}\right) + \left(\frac{2.51}{R_e\sqrt{f}}\right)\right]\right\}^{-2}$$ **EQUATION 17**

where:

e = roughness of pipe/tube (material property)

R_e = reynolds number

Surface roughness is a material property that can be acquired through material property tables for most common materials or from the manufacturer of a less common material.

Lastly, the frictional loss coefficients are found using empirical data. Table 3-B in Appendix B [8] provides the necessary coefficient for gradual contractions based on the relationship between the inlet and outlet areas. This table was used to evaluate the pressure drop on the outlet of the heat exchanger exhaust.

Table 4-B in Appendix B [8] provides minor entrance losses that are dependent on entrance type. For the loss coefficients pertaining to an exit, Figure 1-B in Appendix B [8] gives the non-linear relationship associated with these losses. Table 6-B and Figure 1-B were required to estimate the minor losses associated with the entrance and exit of the tubes within the heat exchanger.

The components that will contribute to the total additional back pressure of the system are listed below.

1. The inlet exhaust butterfly valve used to bypass exhaust around the heat exchanger.
2. Elbows that may need to be installed to adapt the exhaust to the inlet of heat exchanger.
3. Contractions at the exhaust outlet of the heat exchanger.
4. The tubes that the exhaust travels through within the shell of the heat exchanger.

All of these need to be calculated individually and added together. Therefore, ΔP becomes

$$\Delta P = \Delta P_i + \Delta P_o + \Delta P_t$$

EQUATION 18

where:

ΔP_i = Back pressure due to the valves and elbows in the inlet exhaust pipe.

ΔP_o= Back pressure due to the heat exchanger outlet

ΔP_t=Back pressure due to the tubes in the heat exchanger.

2.1.5 Stress Analysis of Heat Exchanger

During a meeting on November 10 with New Flyer, they requested that the heat exchanger be designed for a burst pressure of 500 psi on the glycol or shell side of the heat exchanger [9]. This is outside the original scope of the project. The calculations in this section are only for stresses on the shell and nozzles resulting from an internal pressure as these are the weakest areas of the heat exchanger. For guidance on complete stress calculations, refer to TEMA standards.

The heat exchanger has been designed as per TEMA standards [6]. These standards do account for stresses resulting from thermal expansion and hydrostatic pressure. However, the design standards do not place a maximum allowable pressure associated with specific geometries. The standards do provide a number of equations to find the theoretical stress that will be applied to the shell side of the heat exchanger.

The following equations were used to calculate the theoretical stresses that may result from an internal pressure of 500 psi within the shell.

Shell stress due to surface pressure is found by,

$$S_{vp} = \sigma_p \left[-1.7988 + 3.5474 \left(\frac{d}{D}\right)^{0.6} \left(\frac{D}{T}\right)^{0.3} \left(\frac{t}{T}\right)^{-0.2} - 0.05716 \left(\frac{d}{D}\right)^{1.2} \left(\frac{D}{T}\right)^{0.3} \left(\frac{t}{T}\right)^{2.9} \right] \quad \textbf{EQUATION 19}$$

Shell stress due to membrane pressure is found by,

$$S_{vmp} = \sigma_p \left[1.2356 - 0.00161 \left(\frac{d}{D}\right) \left(\frac{D}{T}\right)^{0.6} \left(\frac{t}{T}\right)^{-2.4} + 0.633 \left(\frac{d}{D}\right) \left(\frac{D}{T}\right)^{0.5} \left(\frac{t}{T}\right)^{-0.8} \right] \quad \textbf{EQUATION 20}$$

where,

d = Mean diameter of corroded nozzle.

D = Mean diameter of corroded shell.

t = Corroded nozzle wall thickness.

T = Corroded shell wall thickness.

σ_p = Stress on shell resulting from pressure.

Stress on shell resulting from pressure is found by,

$$\sigma_p = {PD}/{2T}$$ **EQUATION 21**

where,

P = pressure inside the shell.

Using these concepts, the maximum stress due to surface and membrane pressure was found to be 62.8 ksi and 70.4 ksi respectively. The ultimate tensile strength of 304 Stainless Steel is 85 ksi; therefore, the heat exchanger should withstand a pressure of 500 psi. See Table 6-F in Appendix F for complete results.

It is recommended that the client have a finite element analysis (FEA) done on the heat exchanger design prior to manufacturing. FEA results should be used to ensure that no high stress concentrations exist in the model as these regions will fail prematurely.

2.1.6 Vibration Testing

In order to ensure that premature failure due to vibration does not occur on the heat exchanger, vibration testing will have to be performed. New Flyer follows a New York City Transit technical specification, which includes a procedure for vibration testing [10]. The heat exchanger design proposed in this report will have to pass this test before being implemented into New Flyer's buses.

2.1.7 Optimization

The theory presented in sections 2.1.3 and 2.1.4 introduces the fundamental equations required to calculate the heat transfer and back pressure of the new heat exchanger. However, these theories are only tools that can be used if the geometry of the heat exchanger is known. Furthermore, the heat transfer and fluid dynamic theories do not offer sufficient relationships to calculate the heat transfer of a heat exchanger at a desired pressure drop. In order to accomplish this, numerical analysis is required with an iterative approach.

To find an iterative solution, a program in MATLAB was written to perform iterations on the geometries of the heat exchanger and calculate the heat transfer and back pressure on each configuration. Specifically, heat transfer calculations were done at peak torque conditions and back pressure calculations were done at maximum power conditions. The methodology behind this approach was that the highest back pressure would occur at maximum power; similarly, since the bus rarely operates at governed speed and spends a small portion of its duty cycle at low or high idle, calculating the heat transfer at peak torque provided an average heat transfer value.

In addition to back pressure constraints, New Flyer set limited space constraints. As such these constraints were used to choose a range of sizes for the shell diameter and nominal length of the tubes. These ranges are shown below.

$$300 \text{ mm} \leq \text{Shell Diameter} \leq 500 \text{ mm}$$

$$450 \text{ mm} \leq \text{Tube Length} \leq 600 \text{ mm}$$

TEMA standards specify tube diameter dependent on shell diameter. Also, the pitch diameter that was defined in section 2.1.2 must follow the following constraints.

$$1.25 \leq \text{pitch ratio} \leq 1.5$$

These three criteria were used as boundary conditions of the iterative loops programmed into MATLAB. Within these loops, the back pressure and heat transfer were calculated for each geometric configuration. In order to check that the back pressure and heat transfer rate were within the design constraints the following IF statements were used to decipher what data should be kept or discarded. The criteria as it was specified in the IF statement is listed below.

If back pressure ≤ 3386 Pa and heat transfer > 23 kW, record data

Using these criteria yielded over 17,000 possible solutions the first time the program was used. After examining the data, we found that there were many solutions that met the design constraints with a smaller than expected geometry. Therefore, in an effort to find the smallest possible geometry, the constraints were tightened to the following parameters.

$$300 \text{ mm} \leq \text{Shell Diameter} \leq 340 \text{ mm}$$

$$450 \text{ mm} \leq \text{Tube Length} \leq 600 \text{ mm}$$

The program yielded 27 solutions using these constraints. Of these solutions, we recommended the most compact geometry to New Flyer since they desired a compact design. Its details are described in the following section, Summary of Results. Please see Appendix E for the complete MATLAB code.

2.2 Summary of Results

The two critical constraints of this design are maintaining the back pressure below 1 in of mercury (3.3 kPa), to maintain New Flyer's engine supplier specifications, and producing 25 kW of heat in order to successfully replace the auxiliary heater. 25 kW is the rated output of the current auxiliary heater. The actual output is around 23 kW of heat. To reduce the length of the heat exchanger the value of 23kW is used as the absolute minimum output heat of the exchanger. Even with this consideration, it was found that is still difficult to maintain a heat exchanger within the space constraints while meeting the critical constraints. After consulting with New Flyer, the proposed design would be acceptable. The following section details the results of the design.

An EXCEL spreadsheet was used to calculate the heat transfer and pressure drop based on the shell and tube geometries found using MATLAB. It was found that as the tube diameter decreased, the heat transferred and pressure drop increased.

Heat transfer and pressure drop depend on the operational state of the engine. These states are peak torque, maximum power, and governed speed. In addition to data on these three states, New Flyer gave us data on two additional states that were based on estimations; low idle and high idle. Each of these operational conditions have different mass flow rates of the exhaust stream and different temperatures which directly affect the amount of heat transferred from the heat exchanger and back pressure imposed. For this reason the back pressure and heat transferred is calculated at each of the operational conditions to ensure it is within specifications. The heat transferred is designed to only be above the minimum of 23 kilowatts during peak torque, maximum power and governed speed, where the back pressure is designed within specification under all conditions.

The final design geometries are summarized in Table 2. These outline the basic geometries used to define a shell and tube heat exchanger. A more detailed list of geometries and a complete set of the results, including the excel sheets used are found in Appendix F. Figure 8 illustrated the complete proposed heat exchanger design.

TABLE 11 - FINAL SHELL AND TUBE GEOMETRY

Shell and Tube Geometry			
Description	**Symbol**	**Value**	**Unit**
Inside Shell Diameter	D_s	330.00	mm
Tube Outside Diameter	D_t	25.00	mm
Tube Wall Thickness	L_{tw}	2.50	mm
Inside Tube Diameter	D_{ti}	20.00	mm
Tube layout pitch	L_{tp}	31.25	mm
Tube layout charastic angle	ϑ_{tp}	30.00	°
Overall nominal tube length	L_{to}	530.00	mm
Total length of heat exchanger	L_t	817.40	mm
Total number of tubes	N_{tt}	60.00	mm

All drawing dimensions are in millimeters

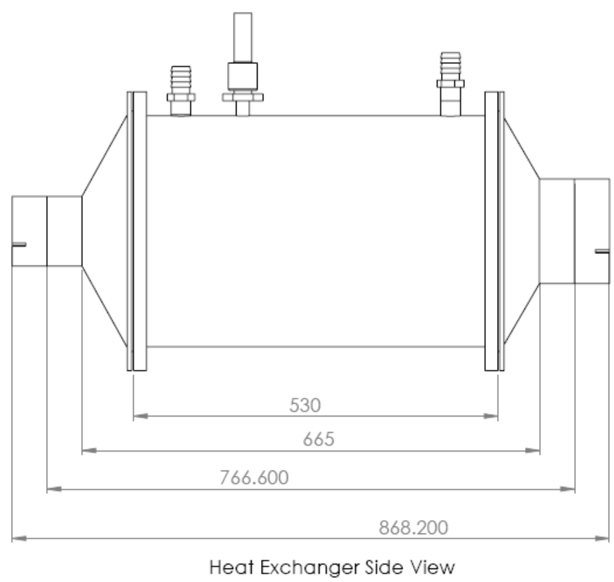


FIGURE 8 - OVERALL DIMENSIONS OF SHELL AND TUBE HEAT EXCHANGER

THE TOTAL LENGTH OF THE HEAT EXCHANGER IS 766.6 MM (30.18 IN), JUST INSIDE OUR TOTAL LENGTH CONSTRAINT OF 32 INCHES. IN ORDER TO MATE THE HEAT EXCHANGER TO THE CURRENT EXHAUST SYSTEM, TWO OPTIONAL COUPLERS CAN BE USED ON THE ENDS OF THE HEAT EXCHANGER. HOWEVER, THESE COUPLERS INCREASE THE OVERALL LENGTH TO 868.2 MM (34.18 IN). FIGURE 9 ILLUSTRATES THE DIMENSIONS OF THE TUBE BUNDLE THAT IS WITHIN THE SHELL.

Figure 10 and Figure 11 detail the cross section of the bundle geometry.

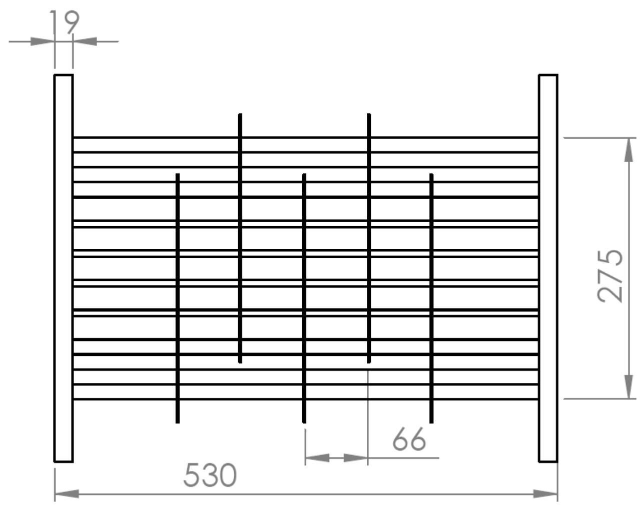


FIGURE 9 - TUBE BUNDLE GEOMETRIES

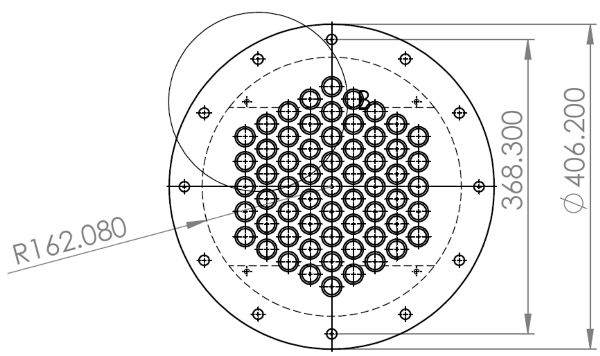


FIGURE 10 - TUBE BUNDLE CROSS VIEW

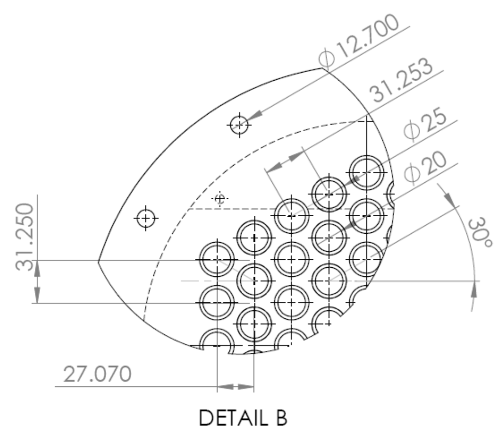


FIGURE 11 - TUBE BUNDLE DETAIL CROSS VIEW

A cut away view of the heat exchanger, Figure 12, illustrates the assembly of the shell and tube heat exchanger. A complete set of drawings for the shell and tube heat exchanger can be found in Appendix G, including a description of each of the components.

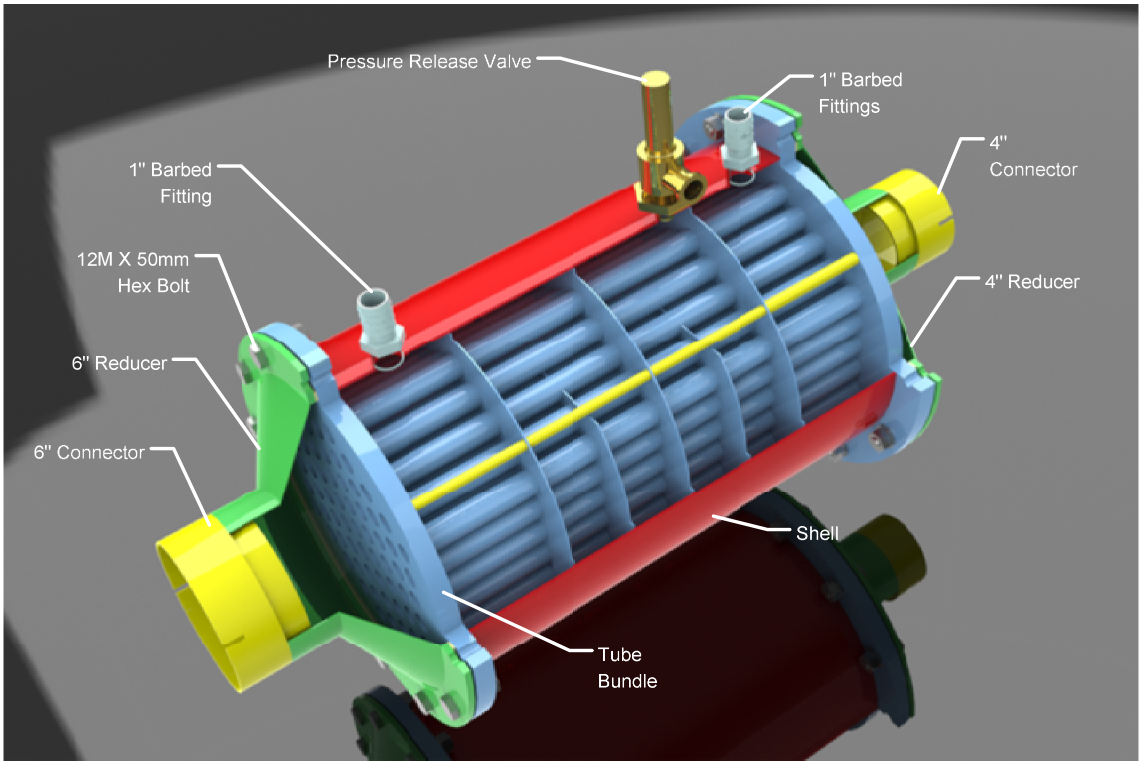


FIGURE 12 - CUT AWAY VIEW OF SHELL AND TUBE HEAT EXCHANGER

In order to comply with the corrosion resistance specification set by New Flyer, to meet [redacted] requirements, all of the exhaust side components of the shell and tube heat exchanger must meet or exceed the corrosion resistance of 304 stainless steel. Metals, other than 304 stainless were considered for this design in order to reduce weight, such as titanium. However, other suitable metals were found to increase the cost above the desired budget set by New Flyer. For this reason, all of the components for this design are to be manufactured from 304 stainless steel.

This shell and tube heat exchanger is capable of transferring 24 kW during the peak torque conditions and has a maximum pressure drop of 0.26 in Hg. This is well below the specified pressure drop, but additional equipment needs to be added to the exhaust stream to accommodate the heat exchanger. This is considered in the operation and control section of this report. The heat transfer and the pressure drop for each of the operating conditions are summarized in Table 12.

TABLE 12 - HEAT EXCHANGER PERFORMANCE SUMMARY

Heat Exchanger Performance Summary					
Condition	Engine Speed	Engine Power	Exhaust Gas Flow	Heat Transfer	Pressure Drop
	RPM	*hp*	*L/s*	*kW*	*in. Hg*
Governed Speed	2200	280.00	542	31.36	0.26
Maximum Power	2000	289.00	550	30.75	0.27
Peak Torque	1300	223.00	401	24.01	0.14
High Idle	1000		337.19	19.89	0.10
Low Idle	800		294.61	16.97	0.08

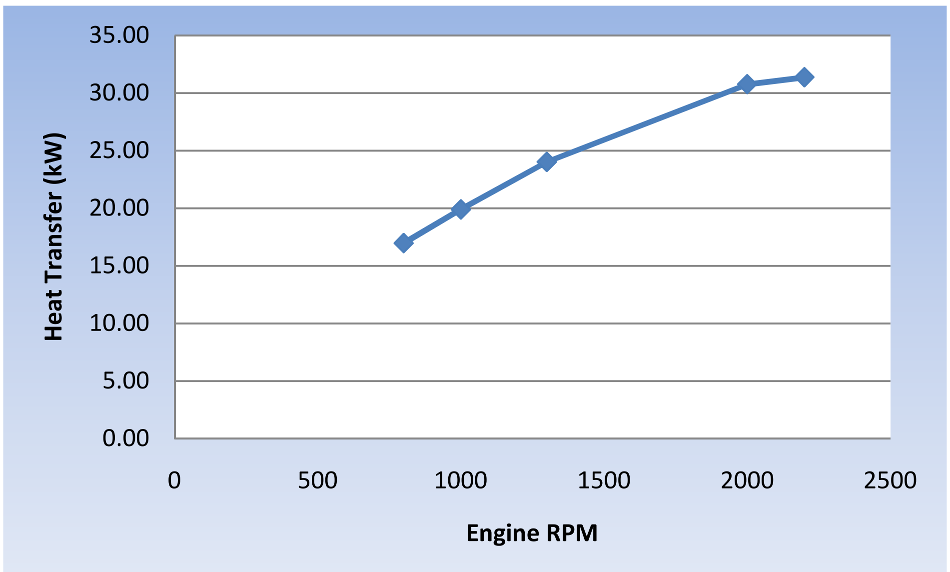


FIGURE 13 - HEAT TRANSFER VERSUS ENGINE RPM. DATA TAKEN FROM TABLE 12.

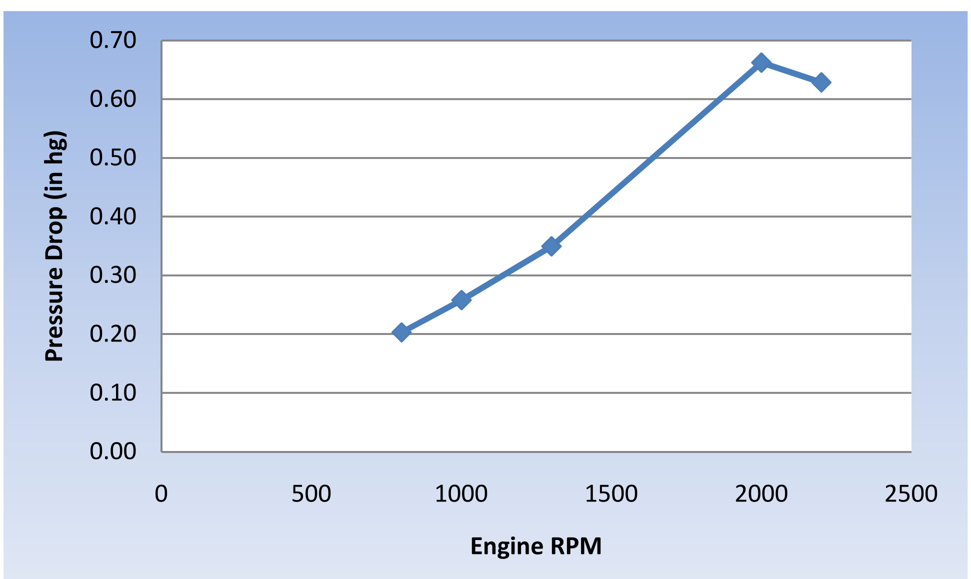


FIGURE 14 - PRESSURE DROP VERSUS ENGINE RPM. DATA TAKEN FROM TABLE 12.

2.3 Operation and Control

The following sections outline the operation and control of the exhaust and coolant system with respect to the proposed heat exchanger. This section also includes how the heat exchanger will be integrated into the current bus design. These sections are a recommendation of the controls and parts required, but are not a detailed description of the implementation of the exchanger. A description of how the system is to be controlled is included, excluding the method of connecting our control system to the current buses engine control unit. This connection may require re-design of New Flyer engine control system which is outside the scope of this project.

2.3.1 Exhaust Operation and Control

The proposed heat recovery system is only useful during the winter months when the auxiliary heater is normally used to supplement the bus cabin heat. In warmer months the extra heat is not required, and will need to be by-passed so that excess heat is not rejected into the buses coolant system, overheating the motor.

In order to bypass the heat exchanger a y-pipe will have to be installed in the exhaust stream to divert the flow. In addition to the y-pipe, a valve in each line will have to be incorporated to fully control the flow. A single valve would be a preferred configuration so that if a valve failure occurs, the exhaust stream would not be fully blocked. If a single valve is used, flow control of the system would be reduced. If a single valve is placed in the heat exchanger stream, the flow could be fully blocked off to the exchanger, but when the valve is open the heat exchanger would not receive the full mass flow rate of the exhaust, reducing the heat transferred. Conversely if a single valve is placed in the tailpipe stream, the flow can be fully directed at the heat exchanger. When the valve is open however, some flow may continue to the heat exchanger which is undesirable.

For these reasons it has been decided that a y-pipe with a valve in each fluid stream is the best possible choice for controlling the flow of the exhaust stream. A representation of such a valve is shown in Figure 15 through Figure 18.

Figure 15 shows the full assembly of a y-pipe exhaust control valve. It consists of a 304 stainless steel 4 inch T-pipe with a butterfly valve in two of the flow paths. The valves are connected by a control rod that moves both valves in unison, closing one path and opening the other. The remaining figures show the T-pipe faded out and the positions of the valves in each of its operational positions. Connecting the valves reduces the chances of having both valves closed at the same time. As well, both the valves can be controlled by a single actuator, reducing the cost and required equipment. The type of actuator is to be chosen by New Flyer as they see fit.

FIGURE 15 - EXHAUST CONTROL VALVE - FINAL ASSEMBLY

FIGURE 16 - EXHAUST CONTROL VALVE - POSITION 1

FIGURE 17 - EXHAUST CONTROL VALVE - INTERMEDIATE POSITION

FIGURE 18 - EXHAUST CONTROL VALVE - POSITION 2

Chrome Depot inc. based out of Mantua Ohio, currently offers this control valve in 4” and 5” sizes. This is the recommended option for New Flyer. A copy of the part number and general dimensions can be seen in Figure 19.

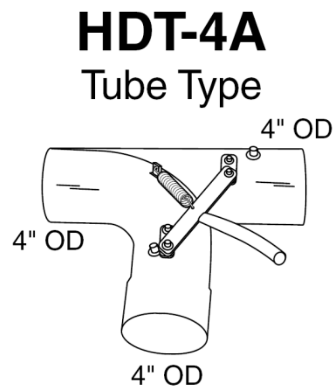


FIGURE 19 - CHROME DEPOT'S EXHAUST CONTROL VALVE [11]

The exhaust control valve would be included into the system after the selective catalytic reduction module, due to [redacted] specification (mentioned in the Constraints and Limitations Section) and before the proposed heat exchanger, shown in Figure 20.

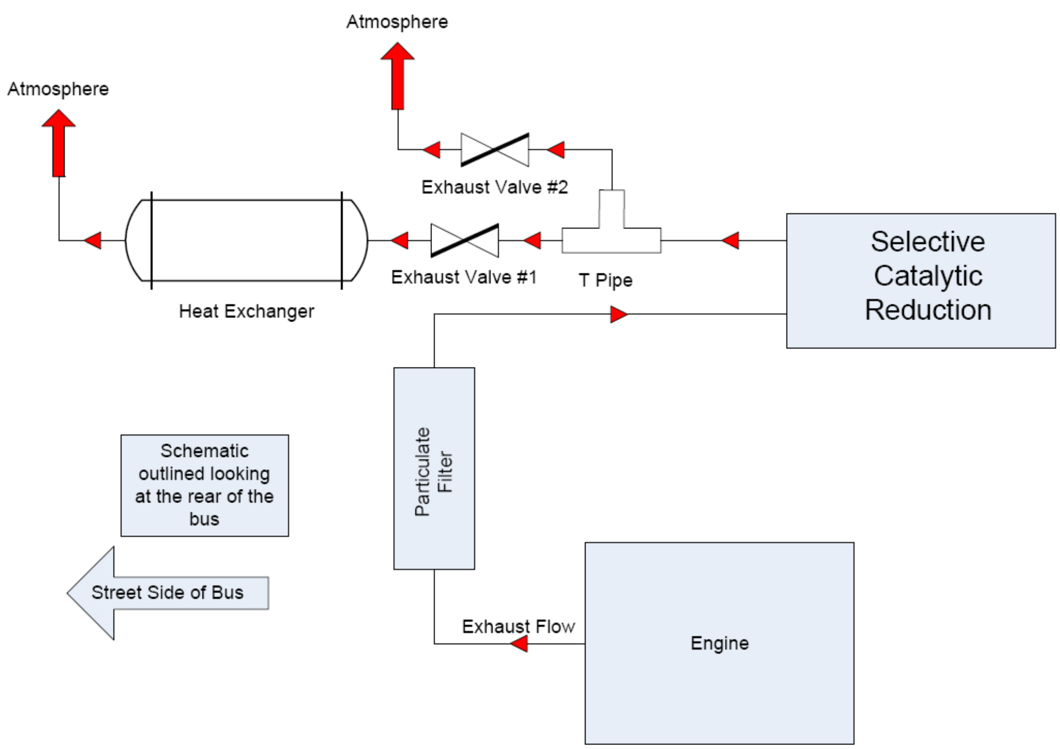


FIGURE 20 - EXHAUST SCHEMATIC

The addition of butterfly valves, connector pipes, and elbows impose backpressure on the system. To ensure that the design meets the back pressure design constraints equation 15 and Table 2-B in Appendix B, are used to calculate the back pressure of each component based on equivalent length in addition to the heat exchanger. A summary of the back pressure for the maximum power operating condition is shown in Table 13. This operating condition has the maximum back pressure, 2.72 kPa (0.8032 in. hg). A complete calculation for every operating condition is included in Appendix H.

TABLE 13 - COMPLETE EXHAUST SYSTEM BACK PRESSURE

Section	System Components	Equivalent Length of Componants	Number Components	Equivalent Length	Section Pressure Loss
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exhcanger					
	Straight Pipe	1	762	762	0.236834526
	Butterfly Valve	2032	1	2032	0.631558737
	Heat Exchanger	n/a	1	n/a	0.909146356
	90° Elbow	3048	1	3048	0.947338106
Section Summary					**2.724877725**
Total					**27.10587773**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.236834526
	Butterfly Valve	2032	1	2032	0.631558737
	90° Elbow	3048	1	3048	0.947338106
Section Summary					**1.815731369**
Total					**26.19673137**

2.3.2 Coolant Operation and Control

In order to completely replace the auxiliary heater, the heat recovery system will require a form of heat storage. The proposed design does not include this component and would not be able to replace the auxiliary heater under all possible operating conditions. It is, therefore, recommended that the heat exchanger be run in parallel with the auxiliary heater for testing and development until the heat exchanger's performance can be verified and a suitable heat storage system is derived.

The coolant system will need to be modified in order to accommodate both the auxiliary heater and the heat exchanger. This will allow for analysis and study of the heat exchanger's performance while the auxiliary heater will remain as a support system. Figure 21 shows the proposed system schematic.

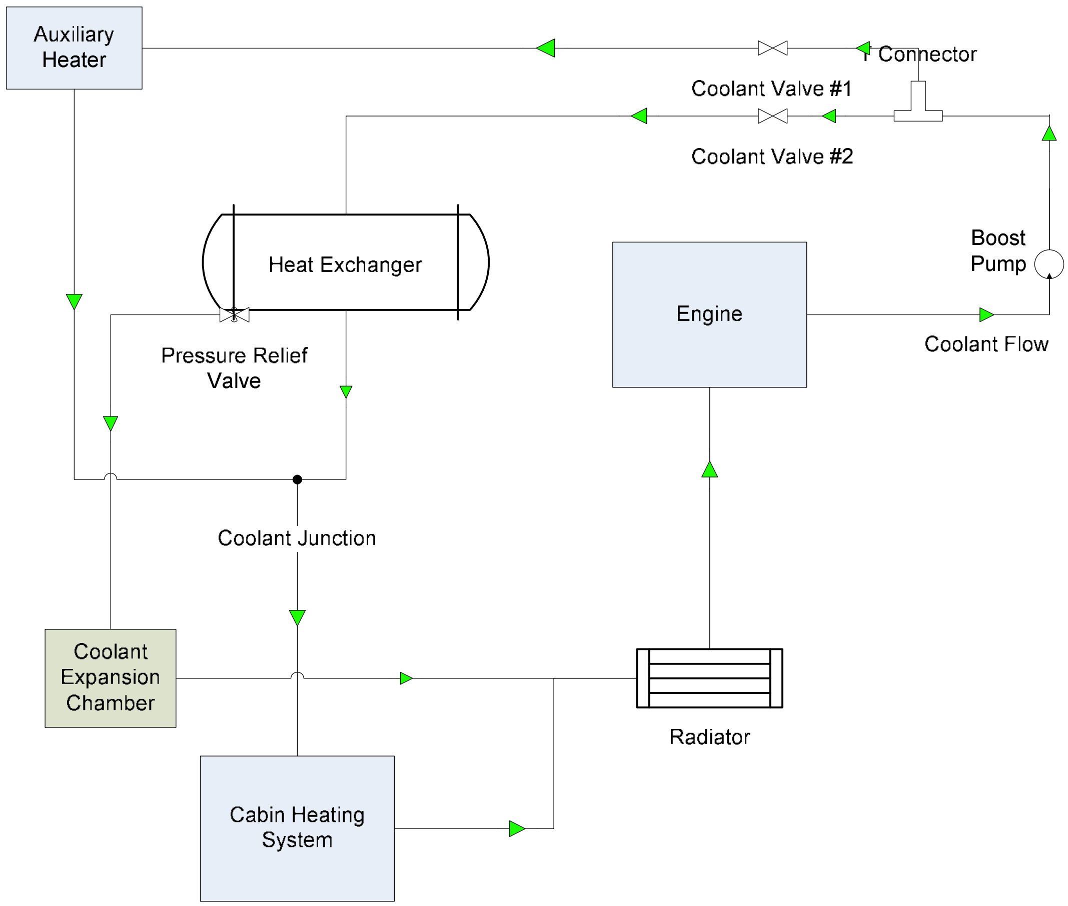


FIGURE 21 - COOLANT SCHEMATIC

A T-pipe should be placed in the coolant line leaving the boost pump with separate outlets leading to the auxiliary heater and the heat exchanger. A valve will be required on each line, immediately after the T-pipe, to direct the coolant flow to either unit. When Coolant Valve #1 is open, Coolant Valve #2 will be closed and all the coolant will be directed through the auxiliary heater. When their positions are switched and valve #2 is open and valve #1 closed, the coolant will be heated by the heat exchanger. The coolant pipes will then rejoin after the auxiliary heater and heat exchanger, before proceeding to the cabin heating circuit.

The valves should be linked together so that one is open while the other is closed. This will allow the operation of only one system at a time, and reduce the possibility of having both systems open or closed at the same time. This will also prevent coolant from flowing back into one of the systems after leaving the other.

After the coolant has been heated, either by the heat exchanger or the auxiliary heater, it will flow into the existing cabin heating circuit. This allows for the implementation of the heat recovery system, with very little modification to the existing heating systems. In the event that a valve malfunctions on the gas or coolant side, the pressure release valve is designed to open at 300 psi (2068.2 kPa) and vent into the coolant expansion chamber.

2.3.3 Combined Operation and control

The exhaust system and coolant system will need to run in unison. Figure 22 shows the two systems combined into one system. A simplified control logic diagram has been included to describe how the operation of the proposed system would work, refer to Figure 23 for these details.

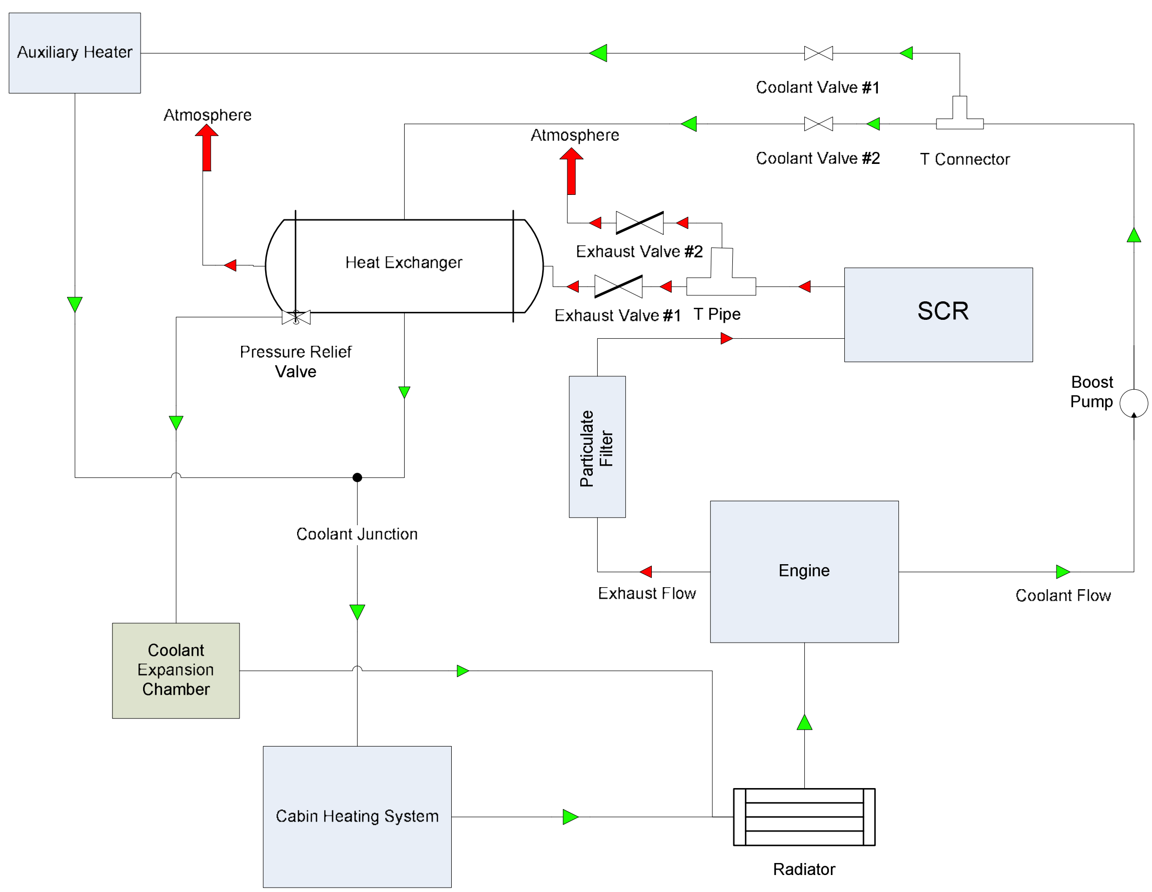


FIGURE 22 - COMBINED EXHAUST AND COOLANT SCHEMATIC

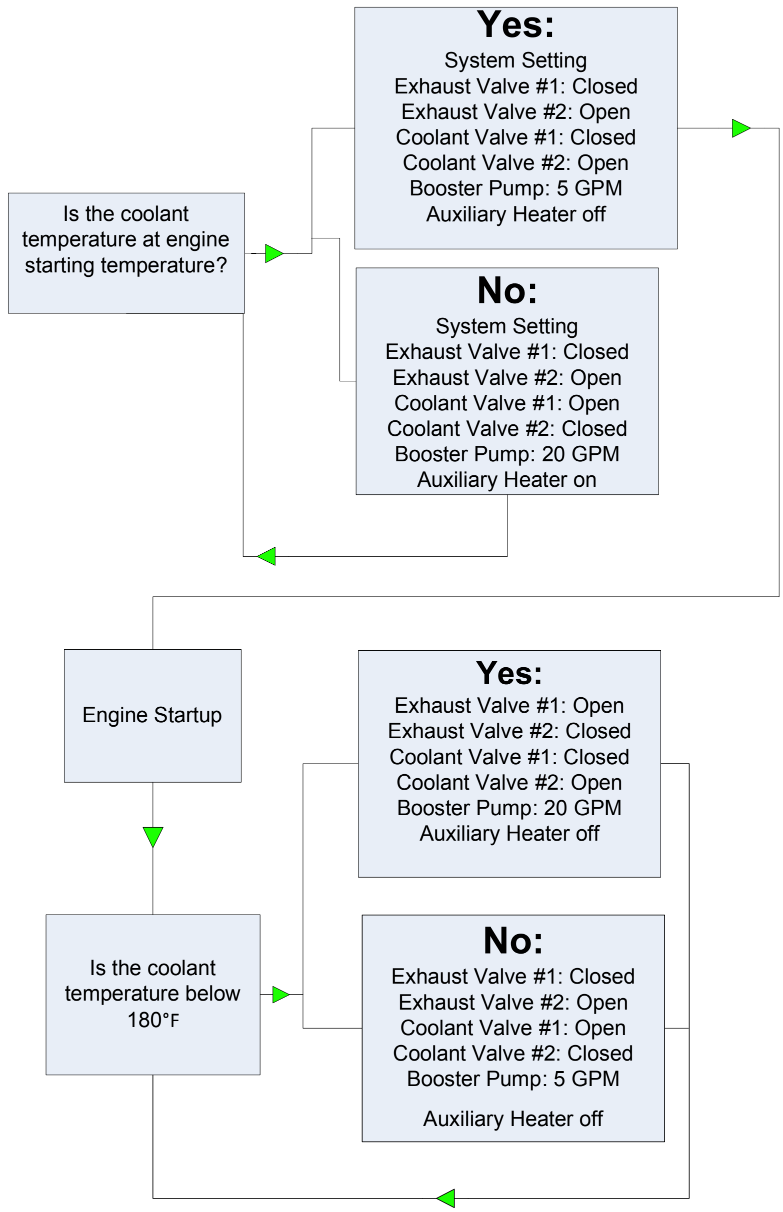


FIGURE 23 - EXHAUST AND COOLANT CONTROL LOGIC

The diagram includes the operation of the auxiliary heater, which has been included in order to maintain the function of preheating the engine.

2.3.4 Control System Safety

A shell and tube heat exchanger is considered a pressure vessel, and for that reason a pressure release valve has been incorporated to avoid over pressuring the system. In combination with this, it is suggested that New Flyer also incorporates a thermal couple on the heat exchanger to monitor the fluid temperature of the heat exchanger. In the case of overheating, from exhaust or valve failure, the thermal couple will send a warning light to the driver that the bus requires maintenance. The temperature at which the warning light is turned on should be set to the overheating temperature of the bus cooling system: 225 °F.

In additional to the control system safety and pressure release valve, it is also recommended that New Flyer incorporate a regularly scheduled maintenance for the heat exchanger along with their suggested bus maintenance schedule. A visual inspection for cracks and leaks of the heat exchanger should be carried out at every coolant change, every 150 000 miles. It is recommended that the coolant side of the heat exchanger be isolated from the system, pressurized up to 150 psi with air and checked to hold pressure. During this inspection, the exhaust valve assembly should be checked to ensure it is properly diverting the flow of exhaust. Additional bus maintenance is done at 300 000 miles when the bus undergoes a complete overhaul. During this time the heat exchanger should also be serviced. The reducers should be removed and the tubes inspected for fouling. If excessive fouling is found, the tubes should be cleaned. It is recommended that the Tubular Exchanger Manufacturers Association (TEMA) standards for cleaning and inspecting shell and tube heat exchangers are followed for these inspections.

2.4 Cost and Bill of Materials

New Flyer presented the team a maximum budget of $3000. This was determined from the current cost of the auxiliary heater plus the fuel savings predicted from its elimination. The cost of implementing the exhaust heat recovery unit includes the heat exchanger material, fabrication and coolant system re-routing.

Based on the recommendation by the client, the cost analysis of material and fabrication costs for the heat exchanger can be estimated using the total weight of the required material [12]. The relation between total weight and cost is expressed as follows:

$$\text{Total Cost} = Weight\ (lb) * \frac{\$4}{lb} * 3$$

The formula takes the total weight of the item to be manufactured, multiplied by the cost of the material and a fabrication factor. As mentioned previously in this report, in order to meet the corrosion resistance specification set by New Flyer and [redacted] 304 stainless steel shall be used to manufacture all the components of the proposed shell and tube heat exchanger. The estimated weight of each component was found using a SOLIDWORKS model. The price of 304 stainless steel is roughly $4 dollars per pound. It should also be noted that the installation cost is not included in the cost analysis based on the assumption, which was suggested by the client, that the installation of auxiliary heater is expected to be comparable. Table 14 lists the individual components of the heat exchanger, their weight and estimated costs.

TABLE 14: BILL OF MATERIALS: HEAT EXCHANGER (WEIGHTS BASED ON SS DENSITY OF 8000 KG/M^3)

Item No.	Part	Description	Quantity	Material	Weight
1	Tubes	25mm OD	60	304L SS	45 kg
2	Tube sheet	330 X 31.25 mm	2	304L SS	30.4 kg
3	Baffle	290 X 3.2 mm	6	304L SS	3.3 kg
4	Shell	340 mm OD	1	304L SS	13.2 kg
5	Reducer, Inlet	340 X 101 mm OD	1	304L SS	3.3 kg
6	Reducer, Outlet	340 X 150 mm OD	1	304L SS	3.2 kg
7	Slotted End Connection	101.6 mm OD	1	304L SS	0.4 kg
8	Slotted End Connection	152.4 mm OD	1	304L SS	0.6 kg
9	Gasket	335 ID X 345 OD	2	Metal Jacketed	0.5 kg
10	Tie Rods	6.35 mm	4	304L SS	0.6 kg
				Total Weight	100.5 kg
					221 lb

The coolant system also requires a select number of components. These components are listed in Table 15 along with their associated costs.

TABLE 15: BILL OF MATERIALS: COOLANT COMPONENTS

Item No.	Part	Description	Quantity	Unit cost	Total cost
1	Rubber Connector	Rubber pipe connectors. Use 6 inch lengths per connection.	3	$4/ft	$12.00
2	Hose Clamps	Hose clamps to keep rubber connectors in place. Use 2 per connection.	12	$1.70 each	$20.40
3	Copper Piping	Copper pipes to carry coolant to and from the heat exhanger. Use approx. 30 ft.	30	$3.60/ft	$108.00
4	Clamping Points	Estimate one clamping point every 2 ft of pipe.	15	$1 each	$15.00
5	Line Valves	Valves to isolate/divert flow to auxiliary heater or heat exchanger.	2	N/A	N/A
6	Pressure Relief Valve	Pressure release valve for the heat exchanger.	1	$150	$150.00
				Total	$305.40

It should be noted that the required amount of copper piping and clamping points will depend on the routing of the pipes. A conservative estimate of 10 ft per line was used for determining cost. Additionally, a pressure relief valve is required to release extraneous pressure through a vent line to a coolant expansion chamber. The cost of the pressure relief valve depends on the valve chosen, however, a valve similar to Aquatrol's Series 17 or Series 18, set to 300 psi, is recommended. The cost of the valve is estimated at $150.00. The coolant piping will also require 2 valves to control the flow path. New Flyer part number 093906 is a 24 V solenoid valve with mounting bracket which has performance characteristics suitable for this application. The price of this valve has not been included in the cost, since it may be possible to rearrange the current coolant system to reuse some of the valves already in place.

TABLE 16: COST SUMMARY

Budget		**Cost**	
Purchase Cost	$2000	Heat Exchanger	$2652.00
Fuel Saving	$1000	Coolant Components	$305.40
Total Budget	$3000	Total	$2957.4

3.0 Conclusion

The objective of this project was to develop a diesel exhaust heat recovery system for a 40 ft New Flyer bus to replace the main function of the auxiliary heater currently in use. This main function was to maintain engine temperature and cabin heat. The scope of this design involved meeting constraints on size, weight, back pressure and manufacturing and installation costs. In order to match the main function of the auxiliary heater, this heat recovery system would have to be capable of capturing enough heat from the exhaust to provide 23 kW into the coolant system.

The constraints listed above are all dependent on one another; the heat recovery system weight increases and the back pressure decreases with increased heat recovery system size. Similarly, the cost increases with an increase in size. Due to these dependencies, every step of the analysis for various designs had to consider these relationships. EXCEL, MATLAB and SOLIDWORKS provided the tools required to perform the analysis more efficiently and optimize our design to address as many constraints and target specifications as possible.

As a result, our team designed a heat recovery system in the form of a shell and tube heat exchanger. Its geometry has met the size constraint for length but not diameter; the overall dimensions are 766.6 mm long, and outside diameter of 336.4 mm. With these dimensions, the heat exchanger's weight is 100.5 kg which does not satisfy the constraint of 26 kg.

The heat exchanger does satisfy the heat transfer specification of 23 kW by providing 24 kW at peak torque; it imposes an additional 2.72 kPa of back pressure at Maximum Power, including necessary fittings for its operation and installation. This is within the allowable back pressure of 3.386 kPa.

Lastly, the budget given to us by New Flyer was $3000.00 for manufacturing and installation; our design will cost approximately $2957.40 to manufacture. New Flyer has advised us, to disregard the installation costs since they should be similar to the auxiliary heater installation costs. Therefore, our design is within budget.

The design did not completely satisfy the size or weight constraints imposed by New Flyer; however the target specifications have been achieved. Furthermore, New Flyer has seen our proposed design and is satisfied with its specifications and predicted performance. As such, our team feels that we have met the objectives for this project and by extension, we have satisfied the request by New Flyer to design a diesel exhaust heat recovery system for their 40 ft bus.

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Appendix A: Concept Analysis and Selection

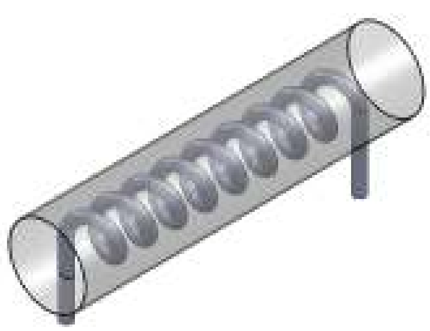
Concept Analysis and Selection

The starting point of our concept and screening analysis was to brainstorm as a team in order to conceptualize some designs that would meet the space constraints that were set by New Flyer. From the standpoint of the analysis, design, and manufacturability, a concentric tube style is the most basic heat exchanger design. On this basis, it serves as a reference point from which all the other concepts can be compared. It is from this reference point that the concept screening and scoring matrix were used to evaluate all other concepts; these will be discussed later in concept screening and scoring section of this report.

The process of screening and scoring concepts is iterative. As the analysis progressed, some concepts were eliminated and some were integrated. The remainder of this section discusses how this process of iterative selection was done using a sensitivity analysis, screening and scoring matrix, concept integration, product performance and implementation cost comparison. Table 1 illustrates each of the initial concepts and provides a brief explanation of how each one would function.

Table 1-A : Concepts Developed from First Brainstorming Session

	Design 1 Concentric tube type of heat exchanger. This design serves as a baseline for comparison with other possible designs. The exhaust gas flows through the inner tube and the coolant fluid flows through the annulus section between the inner tube and the shell.

	Design 2 The exhaust gas flows through the tube and the coolant fluid flows through the coiled pipe wrapped around exhaust pipe.
	Design 3 The exhaust gas flows through inner tube and the coolant fluid flows through the longitudinally finned annulus section.
	Design 4 The exhaust gas flows through the inner tube and the coolant flows over the inner tube. This arrangement allows adjustment in terms of the number of tube pass inside the shell.
	Design 5 The exhaust gas flows through the tail pipe tube and the coolant flows though coiled tube located inside the exhaust stream.

	Design 6 The exhaust gas flows through the inner tube and the coolant flows through the spiral-finned annulus section inside the shell.
	Design 7 The exhaust gas flows through the inner tube and the coolant flows through the flat plate-finned annulus section.

Concentric Heat Exchanger Technical & Cost Analysis

In order to fairly compare all other concepts to the concentric tube heat exchanger, a technical analysis was performed as per Table 2. From this analysis, the concentric heat exchanger would have to be 2 meters long. This is outside the space constraints set by New Flyer. The only way to get the required heat transfer in a shorter distance would be to increase the heat transfer coefficient of the exhaust. We identified that increasing the surface area contact of the exhaust gas while maintaining a turbulent flow was one of the primary factors that affect the heat transfer coefficient. Therefore, these criteria were weighted heavily when comparing the performance of other designs with the concentric concept.

Table 2-B: Concentric heat exchanger analysis.

Energy Balance Method

Exhaust outlet temperatur is calculated, water inlet and outlet temperatures are assumed to be the operating range of the engine

Required Heat	q	25.00	*[kw]*

Temperatures			
Exhaust in	$T_{h,i}$	519.00	*[°C]*
Exhaust out	$T_{h,o}$	424.69	*[°C]*
Water in	$T_{c,i}$	73.888	*[°C]*
Water out	$T_{c,o}$	101.66	*[°C]*

Required Length Calculation			
Overall Heat Transfer	U	101.02	*[W/m2 ·k]*
Delta T1	T_1	417.34	*[°C]*
Delta T2	T_2	350.80	*[°C]*
Log Mean Temp. Diff.	T_{lm}	383.11	*[°C]*
Required Surface area	A	0.65	*[m2]*
Required Length	L	**2.02**	*[m]*

Pressure Drop			
Fluid Velocity	u_m	66.85316755	*[m/s}*
Friction Factor	*f*	0.02	

NTU Method

Exhuast Inlet temperature is known, Water inlet temperature is assumed to be the minimal operating temperature of the engine

Required Heat	q	25.00	*[kw]*

Inputs			
Exhaust in	$T_{h,i}$	519.00	*[°C]*
Water out	$T_{c,i}$	73.89	*[°C]*

Required Length Calculation			
Exhaust Heat Capacity	C_{ex}	265.0813559	*[W/K]*
Water Heat Capacity	C_{H2O}	23953.13	*[W/K]*
Min Heat Capacity	C_{min}	265.08	*[W/K]*
Max Heat Capacity	C_{max}	23953.13474	*[W/K]*
Ratio	C_r	0.01	
Maximum Heat Transfer	q_{max}	117990.36	*[w]*
Effectivness	ε	0.211881712	
NTU		0.23839778	
Overall Heat Transfer	U	101.02	*[W/m2 ·k]*
Required Surface Area	A	0.625537059	*[m]*
Required Length	L	**1.96**	*[m]*

Pressure Drop

Pressure Drop	Δp	370.57	*[N/m2]*
Pressure Drop	Δp	4.8733	*[in. Hg]*

Fluid Velocity	u_m	66.85316755	*[m/s}*
Friction Factor	*f*	0.02	
Pressure Drop	Δp	358.87	*[N/m2]*
Pressure Drop	Δp	4.7194	*[in. Hg]*

Calculated Fluid Properties

Hot Fluid

Exhaust Gas Properties
Assuming equivilant to dry air

Input Values

	Symbol	*Value*	*Unit*
Inner Diameter	D	0.1016	*[m]*
Temperature	T	519	*[°C]*
Temperature	T	792	*[°K]*
Mass Flow Rate	ṁ	542	*[L/s]*
Mass Flow Rate	ṁ	0.241679968	*[kg/s]*

Properties from tables

	Symbol	*Value*	*Unit*
Dynamic Viscosity	µ	3.60128E-05	*[kg/m s]*
Prandlt Number	Pr	0.68952	
Density	ρ	0.445904	*[kg/m^3]*
Thermal Conductivity	k	0.057316	*[w/mk]*
Specific Heat Capacity	Cp	1096.828	*[J/kg k]*

Cold Fluid

Water Properties
Assuming equivilant to dry air

Input Values

	Symbol	*Value*	*Unit*
Inner Diameter	Di	0.1016	*[m]*
Outer Diameter	Do	0.1524	*[m]*
Temperature	T	90.5555	*[°C]*
Temperature	T	363.5555	*[°K]*
Mass Flow Rate	ṁ	5.9000	*[L/s]*
Mass Flow Rate	ṁ	5.6915	*[kg/s]*

Properties from tables

	Symbol	*Value*	*Unit*
Dynamic Viscosity	µ	0.0000031211	*[kg/m s]*
Prandlt Number	Pr	1.7422	
Density	ρ	964.6667	*[kg/m^3]*
Thermal Conductivity	k	0.6733	*[W/m k]*
Specific Heat Capacity	Cp	4208.5555	*[kj/kg k]*

Heat Transfer Coeficent			
	Sym bol	*Value*	*Unit*
Reynolds Number	ReD	84100.80955	
Nuselt Number	NuD	179.115033	
Heat Transfer Coeficent	h	101.0448546	*[W/m2·k]*

Calculated Properties			
	Sym bol	*Value*	*Unit*
Hydraulic Diameter	Dh	0.0508	*[m]*
Reynolds Number	ReD	45705263.0144	
Nuselt Number	NuD	38560.6545	
Heat Transfer Coeficent	h	511068.5481	*[W/m2·k]*

The cost of this design is difficult to quantify at this stage because critical cost criteria such as material selection, manufacturing processes, tolerances and design factors have not been selected. Once a final concept is chosen all these factors will be considered and implemented into a cost analysis that will include manufacturing and installation. For the purpose of concept selection, we are estimating that our reference design is the least expensive due it having the least complexity.

Concept Screening and Scoring

Table 3 displays our first concept screening matrix. Designs 3, 5 and 7 all had a net score of -5. Of these, we only decided to continue investigating 3, due to its better heat transfer rate with respect to the exhaust. Designs 1, 2, 3 and 4 do not meet all the criteria set out by our target specifications and constraints; however, each one meets a critical target specification or constraint, such as, rate of heat transfer from exhaust, manufacturability, cost and weight.

Table 3-A: First Concept Screening Analysis

	Concept Variants						
Criteria	2	3	4	5	6	7	1
Ease of installation	0	0	-	0	0	0	0
Size	+	0	-	+	0	0	0
Rate of heat transfer water	0	+	+	+	+	+	0
Rate of heat transfer gas	0	0	+	+	0	0	0
Weight	+	-	-	+	-	-	0
Corrosion resistance	0	0	-	-	-	-	0
Manufacturability	+	-	-	-	-	-	0
Adaptability	0	0	0	0	0	0	0
Ease of analysis	-	0	0	-	-	-	0
Water routing	0	0	0	0	0	0	0
Cost	+	-	-	-	-	-	0
Pressure drop Exhaust	0	0	-	-	0	0	0
Pressure drop Water	-	0	0	0	-	-	0
Pluses	4	1	2	4	1	1	0
Same	7	9	4	4	6	6	0
Minuses	2	3	7	5	6	6	0
Net	**2**	**-2**	**-5**	**-1**	**-5**	**-5**	**0**
Rank	**1**	**3**	**4**	**2**	**5**	**7**	
Continue?	Yes	Yes	Yes	Yes	No	No	

In order to move forward, we combined some of these designs and had a second brainstorming session to come up with some new ones. Table 4 illustrates the concepts developed during the team's second brainstorming session.

Table 4-A: Concepts Developed from Brainstorming Session 2.

	Design 8 The exhaust gas flows through the internally finned tube. The coolant flows through the annulus section inside the shell as well as the smaller pipe. This arrangement allows more interaction between the exhaust gas and the coolant fluid

	Design 9 The exhaust gas flows through the inner tube which is equipped with longitudinal fins. The cross section area of inner tube expands to compensate pressure drop. The coolant fluid flows over the inner tube through the annulus section.
	Design 10 The exhaust gas flows through the internally finned inner tube to enhance heat transfer rate on the exhaust side. The coolant fluid flows through the annulus section inside the outside shell.
	Design 11 Shell and tubes arrangement. The exhaust flows through smaller tubes and the coolant fluid flows over the tubes inside the shell.

Table 5 shows the concept screening matrix resulting from brainstorming session 2. It includes concept 11, a combination of concepts 4 and 6. Also, we combined concepts 2 and 6 into concept 10. We came up with new ideas 8 and 9. Concept 1 is used as a reference since it ranked the highest in the first screening matrix.

Table 5-A: Second Concept Screening Analysis

	Concept Variants				
Criteria	8	9	10	11	1
Ease of installation	0	0	-	0	0
Size	+	+	+	+	0
Rate of heat transfer water	+	+	+	+	0
Rate of heat transfer gas	+	+	+	+	0
Weight	-	-	-	-	0
Corrosion resistance	0	-	0	0	0
Manufacturibility	-	-	-	-	0
Adaptibility	0	0	0	0	0
Ease of analysis	+	+	+	+	0
Water routing	0	0	0	0	0
Cost	-	-	-	-	0
Pressure drop Exhaust	-	-	-	-	0
Pressure drop Water	0	0	+	+	0
Pluses	4	4	5	5	0
Sames	5	4	3	4	13
Minuses	4	5	5	4	0
Net	0	-1	0	1	0
Rank	2	3	2	1	
Continue?	Yes	Yes	Yes	Yes	

In the second screening matrix, concepts 8, 9, 10 and 11 ranked high enough to continue developing, thus they were included in Table 6, the scoring matrix. The sensitivity of the weights assigned to each of the criteria is based on the following system developed by our team.

$x<5$: Not critical with respect to performance.

$5 \le x \le 10$: Not critical with respect to performance; may impact the orientation or design of the current system.

$10<x<15$: Impacts performance of the design and feasibility of installation.

$15 \le x \le 20$: Critical criteria. Customer must have.

Table 6-A: Scoring Matrix

		Concepts							
		11		10		8		9	
Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Ease of installation	2.50%	2	0.05	4	0.1	3	0.08	2	0.05
Size	2.50%	5	0.13	5	0.13	5	0.13	5	0.13
Rate of heat transfer water	7.50%	4	0.3	3	0.23	4	0.3	4	0.3
Rate of heat transfer gas	20.00%	4	0.8	3	0.6	3	0.6	3	0.6
Weight	10.00%	2	0.2	4	0.4	3	0.3	3	0.3
Corrosion resistance	2.50%	2	0.05	3	0.08	3	0.08	3	0.08
Manufacturibility	10.00%	4	0.4	2	0.2	2	0.2	3	0.3
Adaptibility	2.50%	3	0.08	3	0.08	3	0.08	3	0.08
Ease of analysis	10.00%	4	0.4	3	0.3	2	0.2	3	0.3
Water routing	2.50%	4	0.1	4	0.1	3	0.08	4	0.1
Cost	7.50%	2	0.15	4	0.3	2	0.15	3	0.23
Pressure drop Exhaust	15.00%	3	0.45	3	0.45	3	0.45	4	0.6
Pressure drop Water	5.00%	3	0.15	5	0.25	4	0.2	4	0.2
Total Score		3.325		3.35		2.975		3.4	
Rank		3		2		4		1	
Continue?		Yes		Yes		No		Yes	

It can be seen from Table 6 that designs 9, 10 and 11 rank the highest. Therefore, further analysis was done on these three concepts during our final design stage.

Upon further analysis of designs 9, 10 and 11, the rating of each design was altered in Table 7 to more accurately account for their strengths and weaknesses. These changes altered the rank significantly; as a result, concept 11 and 9 scored the highest. However, concept 11 had a higher rated score for heat transfer, thus the team decided to pursue this concept into the final design stage of this project.

Table 7-A: Final scoring matrix.

		Concepts					
		11		10		9	
Criteria	**Weight**	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Ease of installation	2.50%	2	0.05	4	0.10	2	0.05
Size	2.50%	5	0.13	5	0.13	5	0.13
Rate of heat transfer water	7.50%	4	0.30	3	0.23	4	0.30
Rate of heat transfer gas	20.00%	4	0.80	3	0.60	3	0.60
Weight	10.00%	2	0.20	4	0.40	3	0.30
Corrosion resistance	2.50%	2	0.05	3	0.08	3	0.08
Manufacturibility	10.00%	4	0.40	2	0.20	3	0.30
Adaptibility	2.50%	3	0.08	3	0.08	3	0.08
Ease of analysis	10.00%	4	0.40	3	0.30	3	0.30
Water routing	2.50%	4	0.10	4	0.10	4	0.10
Cost	7.50%	2	0.15	4	0.30	3	0.23
Pressure drop Exhaust	15.00%	3	0.45	3	0.45	4	0.60
Pressure drop Water	5.00%	3	0.15	5	0.25	4	0.20
	Total Score	3.250		3.20		3.25	
	Rank	1		3		2	
	Continue?	Yes		Yes		Yes	

Appendix B: Thermal and Fluid Coefficients

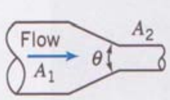
Table 1-B: Correlation coefficients for J_1.

Layout angle	Reynolds number	a_1	a_2	a_3	a_4
30°	10^5–10^4	0.321	−0.388	1.450	0.519
	10^4–10^3	0.321	−0.388		
	10^3–10^2	0.593	−0.477		
	10^2–10	1.360	−0.657		
	<10	1.400	−0.667		
45°	10^5–10^4	0.370	−0.396	1.930	0.500
	10^4–10^3	0.370	−0.396		
	10^3–10^2	0.730	−0.500		
	10^2–10	0.498	−0.656		
	<10	1.550	−0.667		
90°	10^5–10^4	0.370	−0.395	1.187	0.370
	10^4–10^3	0.107	−0.266		
	10^3–10^2	0.408	−0.460		
	10^2–10	0.900	−0.631		
	10	0.970	−0.667		

Table 2-B: Equivalent pipe lengths.

Fitting	**Equivalent Pipe Length** e.g. 30D = 30 x diameter of pipe
90 degree elbow	30D
45 degree elbow	20D
T straight through	16D
T through side	60D
Swept 90 bend	4-8D
Open gate valve	9D
Open globe valve	275D
Full bore non return valve	6D
Butterfly valve	20D

Table 3-B: Loss coefficients for gradual contractions.



	Included Angle, θ, Degrees						
A_2/A_1	10	15–40	50–60	90	120	150	180
0.50	0.05	0.05	0.06	0.12	0.18	0.24	0.26
0.25	0.05	0.04	0.07	0.17	0.27	0.35	0.41
0.10	0.05	0.05	0.08	0.19	0.29	0.37	0.43

Note: Coefficients are based on $h_{l_m} = K(\bar{V}_2^2/2)$.

Table 4-B: Minor entrance losses.

Entrance Type	Minor Loss Coefficient, K^{a}
Reentrant	0.78
Square-edged	0.5
Rounded	r/D: 0.02, 0.06, ≥ 0.15; K: 0.28, 0.15, 0.04

[a] Based on $h_{l_m} = K(\bar{V}^2/2)$, where $\bar{V}$ is the mean velocity in the pipe.

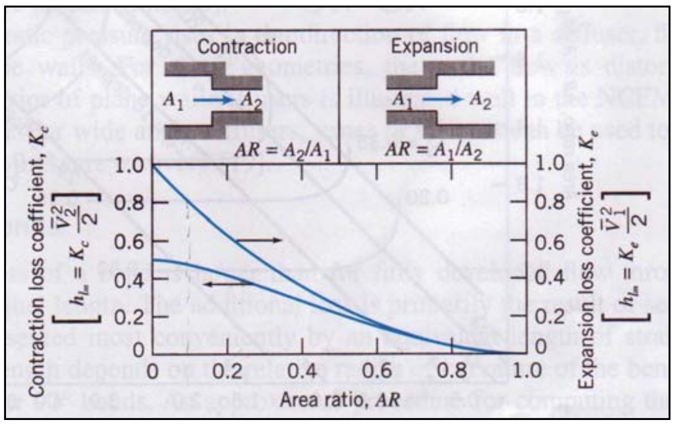


Figure 1-B: Loss coefficients due to sudden change in area.

Appendix C: Properties of Dry Air

Table 1-B: Properties of dry air.

Temperature (K)	Specific Heat Capacity - c_p - (kJ/kgK)	- c_v - (kJ/kgK)	Ratio of Specific Heats - k - (c_p/c_v)	Dynamic Viscosity - μ - 10^{-5} (kg/m s)	Thermal Conductivity 10^{-5} (kW/m K)	Prandtl Number	Kinematic Viscosity[1)] - ν - 10^{-5} (m^2/s)	Density1) - ρ - (kg/m^3)
175	1.0023	0.7152	1.401	1.182	1.593	0.744	0.586	2.017
200	1.0025	0.7154	1.401	1.329	1.809	0.736	0.753	1.765
225	1.0027	0.7156	1.401	1.467	2.020	0.728	0.935	1.569
250	1.0031	0.7160	1.401	1.599	2.227	0.720	1.132	1.412
275	1.0038	0.7167	1.401	1.725	2.428	0.713	1.343	1.284
300	1.0049	0.7178	1.400	1.846	2.624	0.707	1.568	1.177
325	1.0063	0.7192	1.400	1.962	2.816	0.701	1.807	1.086
350	1.0082	0.7211	1.398	2.075	3.003	0.697	2.056	1.009
375	1.0106	0.7235	1.397	2.181	3.186	0.692	2.317	0.9413
400	1.0135	0.7264	1.395	2.286	3.365	0.688	2.591	0.8824
450	1.0206	0.7335	1.391	2.485	3.710	0.684	3.168	0.7844
500	1.0295	0.7424	1.387	2.670	4.041	0.680	3.782	0.7060
550	1.0398	0.7527	1.381	2.849	4.357	0.680	4.439	0.6418
600	1.0511	0.7640	1.376	3.017	4.661	0.680	5.128	0.5883
650	1.0629	0.7758	1.370	3.178	4.954	0.682	5.853	0.5430
700	1.0750	0.7879	1.364	3.332	5.236	0.684	6.607	0.5043
750	1.0870	0.7999	1.359	3.482	5.509	0.687	7.399	0.4706
800	1.0987	0.8116	1.354	3.624	5.774	0.690	8.214	0.4412
850	1.1101	0.8230	1.349	3.763	6.030	0.693	9.061	0.4153
900	1.1209	0.8338	1.344	3.897	6.276	0.696	9.936	0.3922
950	1.1313	0.8442	1.340	4.026	6.520	0.699	10.83	0.3716
1000	1.1411	0.8540	1.336	4.153	6.754	0.702	11.76	0.3530
1050	1.1502	0.8631	1.333	4.276	6.985	0.704	12.72	0.3362
1100	1.1589	0.8718	1.329	4.396	7.209	0.707	13.70	0.3209
1150	1.1670	0.8799	1.326	4.511	7.427	0.709	14.70	0.3069
1200	1.1746	0.8875	1.323	4.626	7.640	0.711	15.73	0.2941
1250	1.1817	0.8946	1.321	4.736	7.849	0.713	16.77	0.2824
1300	1.1884	0.9013	1.319	4.846	8.054	0.715	17.85	0.2715
1350	1.1946	0.9075	1.316	4.952	8.253	0.717	18.94	0.2615
1400	1.2005	0.9134	1.314	5.057	8.450	0.719	20.06	0.2521
1500	1.2112	0.9241	1.311	5.264	8.831	0.722	22.36	0.2353
1600	1.2207	0.9336	1.308	5.457	9.199	0.724	24.74	0.2206
1700	1.2293	0.9422	1.305	5.646	9.554	0.726	27.20	0.2076
1800	1.2370	0.9499	1.302	5.829	9.899	0.728	29.72	0.1961
1900	1.2440	0.9569	1.300	6.008	10.233	0.730	32.34	0.1858

Appendix D: Engine Operational Data

Table 1-D: [redacted] engine performance data.

Maximum Rating Performance Data

	Governed Speed				Maximum Power				Peak Torque			
Engine Speed	2,200	RPM			2,000	RPM			1,300	RPM		
Output Power	280	hp	209	kW	289	hp	216	kW	223	hp	166	kW
Torque	668	lb-ft	906	N-m	759	lb-ft	1,029	N-m	900	lb-ft	1,220	N-m
Inlet Air Flow	595	ft3/min	281	L/s	543	ft3/min	256	L/s	352	ft3/min	166	L/s
Charge Air Flow	40.8	lb/min	18.5	kg/min	41	lb/min	19	kg/min	26	lb/min	12	kg/min
Exhaust Gas Flow	1,149	ft3/min	542	L/s	1,166	ft3/min	550	L/s	850	ft3/min	401	L/s
Exhaust Gas Temperature	967	deg F	519	deg C	912	deg F	489	deg C	899	deg F	482	deg C
Heat Rejection to Coolant	10,155	BTU/min	179	kW	9,175	BTU/min	161	kW	6,895	BTU/min	121	kW
Radiator Coolant Flow *	88	gpm	5.6	L/s	80	gpm	5	L/s	52	gpm	3.3	L/s
Turbo Comp. Outlet Pressure	50	in-Hg	170	kPa	56	in-Hg	189	kPa	51	in-Hg	172	kPa
Turbo Comp. Outlet Temperature	337	deg F	169	deg C	366	deg F	186	deg C	338	deg F	170	deg C
Fuel Consumption	109.6	lb/hr	49.7	kg/hr	107.7	lb/hr	48.9	kg/hr	78.5	lb/hr	35.6	kg/hr
Brake Mean Effective Pressure	187	psi	1,289	kPa	212	psi	1,462	kPa	251	psi	1,731	kPa

Appendix E: MATLAB Code

```
clear all;

%Tube inlet and outlet temperatures @peak torque in deg C%

Tti=409.7;
Tto=200;

%Shell inlet and outlet temperatures @ Peak torque in deg C%

Tsi=70;
Tso=85;

%Temperature difference between tube inlet/shell outlet and%
% tube outlet/shell inlet respectively @ peak torque%

DeltaT1=Tti-Tso;
DeltaT2=Tto-Tsi;

%Tube thickness and thermal conductivity respectively

Ltw=2.5;
LamdatW=16;

%Define variables for tube length as per excel spreadsheet V5

Lto=0;
Lti=0;
Lta=0;

%Define matrix for 30 deg pitch angle

Thirty=zeros(45,7);

%Counter

l=0;

%Following properties are for dry air at peak torque%

    Mtp=0.25;
    LamdaT=0.0453;
    PrT=0.68;

    %Dynamic viscosity of dry air kg/ms
    up=2.94e-5;

%Following properties are for glycol at peak torque%

    a1=0.321;
    a2=-0.388;
    a3=1.45;
    a4=0.519;
```

```
    %Prandlt number shell side
    PrS=7.36;
    %Correction factor
    CF=1.03;
    %Specific heat
    Cps=3576.89;
    %Mass flow rate of glycol in kg/s
    Ms=1.32;
    %Space between outer most tubes and shell in mm
    Lbb=50;
    %Dynamic viscosity for 50/50 glycol millipascals/s
    ug=0.88;

%Following properties are for dry air at governed speed%

     %Total mass flow rate kg/s at tube fluid mean temperature
     Mt=0.3226;
     %Mass flow rate at inlet of heat exchanger for 714.3 K
     Mi=0.2681;
     %Mass flow rate at outlet of heat exchanger for 473.15 K
     Mo=0.4055;

     %Density of dry air @ 320.575 C or 593.575 K
     density=0.5952;
     %Density of dry air @ 441.15 C or 714.3 K
     densityI=0.49466;
     %Density of dry air @ 200 C or 473.15 K
     densityO=0.7481;

     %Dynamic viscosity of dry air @ 593.575 K
     u=2.9954e-5;
     %Dynamic viscosity of dry air @ 714.3 K
     uI=3.3749e-5;
     %Dynamic viscosity of dry air @ 473.15 K
     uO=2.571e-5;

     %Minor losses in tubes due to entrance and exit
     kt=1.5;
     %Minor losses from outlet of heat exchanger based on a 60 deg transition
     kp=0.2;
     %Pressure per tube [Pa]
     Pdt=0;
     %Entrance back pressure due to butterfly valve
     Pdi=0;
     %Exit back pressure due to contraction
     Pdo=0;
     %Total back pressure caused by heat exchanger [Pa]
     PdT=0;
     %Nominal length of heat exchanger tubes [mm]
     L=0;
     %Total length of heat exchanger with transition
```

```
    LT=0;
    %Friction factors used in the colebrook equation. f1 and f3 are the output
    f1=0;
    f2=1;
    f3=0;
    f4=1;

%for loop to iterate tube length in [mm]
for k=450:10:600;

%For loop to optimize shell diameter and number of tubes in[mm]
    for i=300:5:340;

    %Tube ID in mm
        for j=20:1:51;
            L=k;
            Dsi=i;
            Lbc=.2*i;
            Dti=j;
            Dto=Dti+5;
            Di=101.6;
            Do=152.4;
            Ltp=((Dto-6.35)/(50.8-6.35))*(63.500-7.938)+7.938;

            %Pitch ratio and number of tubes
            PR=Ltp/Dto;
            Dctl=i-(Lbb+Dto);
            Ntt=(0.78*Dctl^2)/(0.866*Ltp^2);
            Nttr=round(Ntt);

            %Mass flow rate per tube kg/s
            Mtn=Mt/Nttr;

            %Total velocity entering heat exchanger from 4" exhaust in m/s
            VT4=Mi/((pi*((Di/1000)^2)/4)*densityI);

            %Velocity per tube in m/s
            Vt=Mtn/((pi*((Dti/1000)^2)/4)*density);

            %Velocity exiting the heat exchanger through 6" pipe
            VT6=Mo/((pi*((Do/1000)^2)/4)*densityO);

            %Reynolds number for flow through each tube
            Ret=(4*Mtn)/(pi*(Dti/1000)*u);

            %Reynolds number for flow through 4" exhaust pipe with
            %butterfly valve
            Re4=(4*Mi)/(pi*(Di/1000)*uI);

            %Reynolds number for flow exiting heat exchanger through 6"
            %pipe
            Re6=(4*Mo)/(pi*(Do/1000)*uO);
```

```
        %Friction factor
        f=0.316/(Ret^0.25);

        %Friction factor using colebrook equation for tubes
        %f1 is friction factor for tubes

        while f1 ~= f2;
            e=0.030e-3;
            f1 = ((-2)*log10(((e/Dti)/3.7)+(2.51/(Ret*sqrt(f2)))))^(-2);
            if f1 ~= f2;
                f2=f1;
            else
                f1=f2;
            end
        end

        %Friction factor using colebrook equation for 4" exhaust pipe
        %f3 is friction factor for exhaust pipe entering heat exchanger

        while f3 ~= f4;
            e=0.030e-3;
            f3 = ((-2)*log10(((e/Di)/3.7)+(2.51/(Re4*sqrt(f4)))))^(-2);
            if f3 ~= f4;
                f4=f3;
            else
                f3=f4;
            end
        end

        %Design factor of safety of the friction factor of 2 to allow
        %for flow that is not fully developed.

        f1=2*f2;
        f3=2*f4;

        %Back pressure per tube
        Pdt=((density*(Vt^2))/2)*(f1*(L/Dti)+kt);

        %Back pressure at inlet transition. Using equivalent length of
        %a globe valve = 79 + equivalent length of 1 90 deg elbow ~ 5 +
        %equivalent length of Tee line flow = 12
        Pdi=((densityI*(VT4^2))/2)*(f3*(20+6+16));

        %Back pressure at outlet transition going to 6" pipe.
        Pdo=((densityO*(VT6^2))/2)*kp;

        %Total back pressure caused by heat exchanger
        PdT=Pdt+Pdi+Pdo;

%%Heat transfer calculations at peak torque%%
```

```
        %Cross flow area at shell centerline within one baffle spacing
        sm=Lbc*(Lbb+(Dctl/Ltp)*(Ltp-Dto));

        %Max shell side cross flow velocity
        ms=(Ms/sm)*10^6;

        %Reynolds number shell side
        Resp=(Dto*ms)/ug;

        %Heat transfer factor a
        a=a3/(1+0.014*(Resp^a4));

        %Heat transfer factor
        htf=a1*((1.33/(Ltp/Dto))^a)*Resp^a2;

        %Heat transfer coefficient shell side
        Hs=htf*Cps*ms*(PrS^(-2/3))*CF;

        %Mass flow rate per tube
        Mtnp=Mtp/Nttr;

        %Reynolds number of dry air
        Retp=(4*Mtnp)/(pi*(Dti/1000)*up);

        %Nusselt number tube side
        NuT=0.023*(Retp^(4/5))*(PrT^(0.3));

        %Heat transfer coefficient tube side
        Ht=(NuT*LamdaT)/(Dti/1000);

        %Header thickness
        Lts=0.1*Dsi;

        %Effective tube length for heat transfer
        Lta=L-2*Lts;

        %Log mean temperature difference
        DeltaTLm=(DeltaT1-DeltaT2)/(log(DeltaT1/DeltaT2));

        %Overall heat transfer
        Ui=((1/Ht)+(Dti/(Hs*Dto))*((Dti/1000)/(2*LamdatW))*(log(Dto/Dti)))^(-1);

        %Amount of heat transfer
        Q=Ui*pi*(Dti/1000)*Nttr*(Lta/1000)*DeltaTLm;

        %if statement to check back pressure and number of tubes
        if (PdT<3386) && (Nttr>=60)&&(Q>=2.5e+4);
            l=l+1;

    %Total length of heat exchanger. Includes nominal tube length +
    %transitions at inlet and outlet + a 1" flange to attach pipe on
    %either side
```

```
        LT=((Dsi-Di)/2)*(cos(pi/3)/sin(pi/3))+((Dsi-Do)/2)*(cos(pi/3)/sin(pi/3))+L+50.8;

        % Matrices for Lpn,Lpp,tube wall thickness, Nominal tube length
        % Tube inside diameter, Shell inside diameter, Number of tubes
        % Pitch, pitch ratio, back pressure, heat transfer, friction factor

                Thirty(l,1)=0.5*Ltp;
                Thirty(l,2)=0.866*Ltp;
                Thirty(l,3)=Ltw;
                Thirty(l,4)=L;
                Thirty(l,5)=LT;
                Thirty(l,6)=Dti;
                Thirty(l,7)=Dsi;
                Thirty(l,8)=Nttr;
                Thirty(l,9)=PdT;
                Thirty(l,10)=Ltp;
                Thirty(l,11)=PR;
                Thirty(l,12)=Q;
                Thirty(l,13)=Resp;
                Thirty(l,14)=Hs;
                Thirty(l,15)=Ht;
                Thirty(l,16)=Ui;
                Thirty(l,17)=f;
                Thirty(l,18)=f1;
                Thirty(l,19)=f3;
                Thirty(l,20)=Pdt;
                Thirty(l,21)=Pdo;
                Thirty(l,22)=Pdi;

                f1=0;
                f3=0;
            else
                continue;
            end
        end
    end
end
```

Appendix F: Detailed Calculation and Geometries

Table 1-F : Peak Torque Calculations

Peak Torque

Input Data

Description	Symbol	Value	Unit
Tube and Tube Layout			
Inside Shell Diameter	Ds	330.00	mm
Tube Outside Diameter	Dt	25.00	mm
Tube Wall Thickness	Ltw	2.50	mm
Inside Tube Diameter	Dti	20.00	mm
Tube wall material thermal conductivity	λtw	16.00	W/m K
Pitch Ratio	PR	1.25	
Tube layout pitch	Ltp	31.25	mm
Tube layout characteristic angle	θtp	30.00	deg
Tube Length			
Overall nominal tube length	Lto	530.00	mm
Baffled tube length	Lti	464.00	mm
Effective tube length for heat transfer	Lta	464.00	mm
Baffle Geometry			
Baffle cut as percent of Ds	Bc	20.00	%
Central baffle spacing	Lbc	66.00	mm
Tube Bundle Geometry			
Estimated Total number of tubes	Ntt	59.97	
Total number of tubes	Ntt	60.00	
Number of tube passes	Ntp	1.00	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	PTFH	code
Tube OD-to baffle hole clearance	Ltb	0.60	mm
Inside shell to baffle clearance	Lsb	2.92	mm
Inside shell-to-tube bundle bypass clearance	Lbb	50.00	mm
Center of outermost tube to center of shell	Dctl	255.00	mm
Temperatures			
Shell-side inlet temperature	Tsi	70.00	°C
Shell-side outlet temperature	Tso	85.00	°C
Tube-side inlet temperature	Tti	409.70	°C
Tube-side ouet temperature	Tto	200.00	°C
Shell-side process information			
Shell fluid mass flow rate	Ms	20.00	gpm
Shell fluid mass flow rate	Ms	1.26	l/s
Shell fluid mass flow rate	Ms	1.32	kg/s
At shell fluid mean temperatures	Ts,av	77.50	°C
Density	ρs	1046.54	kg/m3
Thermal Conductivity	λs	0.43	W/m K
Specific Heat	(cp)s	3576.89	J/kg K
Dynamic Viscosity	ηs	0.88	cP=mPa/s
Tube-side process information			
Shell fluid mass flow rate	Mt	401.00	l/s
Tube fluid mass flow rate	Mt	0.25	kg/s
At tube fluid mean temperature	Tt,av	577.85	°K
Density	ρt	0.61	[kg/m^3]
Thermal Conductivity	λt	0.0453	W/m K
Specific Heat	(cp)t	1046.09	J/kg K
Dynamic Viscosity	μ	2.94E-05	[kg/m s]

Output Data

Description	Symbol	Value	Unit
Shell Side Data			
Cross-flow area Sm at the shell centerline	Sm	6666.00	mm2
Max Shell side cross flow velocity	ṁs	198.08	kg/m2 s
Reynolds Data - Shell Side	Res	5640.49	
Prandtl Number - Shell Side	Prs	7.30	
Delta T1	ΔT1	324.70	°C
Delta T2	ΔT2	130.00	°C
Log mean temperature difference	ΔTLM	212.70	°C
Effective Tube Length (Equivalent for Non U tube)	Lta	464.00	mm
Total heat transfer surface of the Exchanger	Ao	2.19	m2
Heat Transfer factor a	a	0.11	
Heat Transfer factor ji	ji	0.01	
Viscosity at wall (Maximum Viscosity for Water Glycol)	ηs,w	0.70	
Correction factor for viscosity gradient	$(\phi_s)^r$	1.03	
Heat Transfer Coefficient	hs	2199.68	W/m2 K
Tube Side Data			
Mass Flow Rate -Tube Side	Mt	0.25	kg/s
Mass Flow Rate per tube	Mtt	0.0041	kg/s
Reynolds Number - Tube Side	Ret	8849.06	
Prandtl Number - Tube Side	Pr	0.68	
Nusselt Number - Tube side	Nut	29.48	
Heat Transfer Coefficient	ht	66.71	[W/m2·k]
Overall Heat Transfer Coefficient			
Overall Heat Transfer Coefficient (without material resistance)	U	64.75	w/m2 k
Overall Heat Transfer Coefficient (with material resistance)	U	64.54	w/m2 k
Heat Transferred (without material resistance)	Q	24.09	KW
Heat Transferred (with material resistance)	Q	24.01	KW
Pressure Drop : Tube Side			
Friction Factor	ƒ	0.06	
Mass Flow Rate Per tube	Mtt	0.004090	kg/s
Total Inlet cross sectional area	Ati	0.00810732	m2
Cross sectional area of small tube	At	0.000314	m2
Total Inlet Fluid Velocity	VT	49.46147636	m/s
Fluid Velocity per Tube	Vt	21.27	m/s
Outlet Density	ρo	0.75	[kg/m^3]
Total outlet cross sectional area	Ato	0.018241469	m2
Mass flow rate at outlet	Mto	0.2999881	m/s
Total fluid velocity at outlet	Vto	21.98287838	m/s
Minor Loss due to entrance	kentr	1.50	
Minor loss due to exit transition (60°)	kexit	0.20	
Pressure Drop per tube	ΔPt	434.61	Pa
Back Pressure due to outlet transition	Pdo	36.15	Pa
Total back pressure caused by heat exchanger	ΔPT	470.76	Pa
Total back pressure caused by heat exchanger	ΔPT	0.139015045	in hg

Table 2-F :Maximum Power Calculations

Maximum Power

Input Data

Description	Symbol	Value	Unit
Tube and Tube Layout			
Inside Shell Diameter	Ds	330.00	mm
Tube Outside Diameter	Dt	25.00	mm
Tube Wall Thickness	Ltw	2.50	mm
Inside Tube Diameter	Dti	20.00	mm
Tube wall material thermal conductivity	λtw	16.00	W/m K
Pitch Ratio	PR	1.25	
Tube layout pitch	Ltp	31.25	mm
Tube layout characteristic angle	θtp	30.00	deg
Tube Length			
Overall nominal tube length	Lto	530.00	mm
Baffled tube length	Lti	464.00	mm
Effective tube length for heat transfer	Lta	464.00	mm
Baffle Geometry			
Baffle cut as percent of Ds	Bc	20.00	%
Central baffle spacing	Lbc	66.00	mm
Tube Bundle Geometry			
Estimated Total number of tubes	Ntt	59.97	
Total number of tubes	Ntt	59.00	
Number of tube passes	Ntp	1.00	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	FX	code
Tube OD-to baffle hole clearance	Ltb	0.60	mm
Inside shell to baffle clearance	Lsb	2.92	mm
Inside shell-to-tube bundle bypass clearance	Lbb	50.00	mm
Center of outermost tube to center of shell	Dctl	255.00	mm
Temperatures			
Shell-side inlet temperature	Tsi	70.00	°C
Shell-side outlet temperature	Tso	85.00	°C
Tube-side inlet temperature	Tti	415.65	°C
Tube-side ouet temperature	Tto	200.00	°C
Shell-side process information			
Shell fluid mass flow rate	Ms	20.00	gpm
Shell fluid mass flow rate	Ms	1.26	l/s
Shell fluid mass flow rate	Ms	1.32	kg/s
At shell fluid mean temperatures	Ts,av	77.50	°C
Density	ρs	1046.54	kg/m3
Thermal Conductivity	λs	0.43	W/m K
Specific Heat	(cp)s	3576.89	J/kg K
Dynamic Viscosity	ηs	0.88	cP=mPa/s
Tube-side process information			
Shell fluid mass flow rate	Mt	550.00	l/s
Tube fluid mass flow rate	Mt	0.33	kg/s
At tube fluid mean temperature	Tt,av	580.83	°K
Density	ρt	0.61	[kg/m^3]
Thermal Conductivity	λt	0.0454	W/m K
Specific Heat	(cp)t	1046.77	J/kg K
Dynamic Viscosity	μ	2.95E-05	[kg/m s]

Output Data

Description	Symbol	Value	Unit
Shell Side Data			
Cross-flow area Sm at the shell centerline	Sm	6666.00	mm2
Max Shell side cross flow velocity	ṁs	198.08	kg/m2 s
Reynolds Data - Shell Side	Res	5640.49	
Prandtl Number - Shell Side	Prs	7.30	
Delta T1	ΔT1	330.65	°C
Delta T2	ΔT2	130.00	°C
Log mean temperature difference	ΔTLM	214.94	°C
Effective Tube Length (Equivalent for Non U tube)	Lta	464.00	mm
Total heat transfer surface of the Exchanger	Ao	2.15	m2
Heat Transfer factor a	a	0.11	
Heat Transfer factor ji	ji	0.01	
Viscosity at wall (Maximum Viscosity for Water Glycol)	ηs,w	0.70	
Correction factor for viscosity gradient	$(\phi_s)^r$	1.03	
Heat Transfer Coefficient	hs	2199.68	W/m2 K
Tube Side Data			
Mass Flow Rate -Tube Side	Mt	0.33	kg/s
Mass Flow Rate per tube	Mtt	0.0057	kg/s
Reynolds Number - Tube Side	Ret	12237.06	
Prandlt Number - Tube Side	Pr	0.68	
Nusselt Number - Tube side	Nut	38.20	
Heat Transfer Coefficient	ht	86.81	[W/m2·k]
Overall Heat Transfer Coefficient			
Overall Heat Transfer Coefficient (without material resistance)	U	83.51	w/m2 k
Overall Heat Transfer Coefficient (with material resistance)	U	83.17	w/m2 k
Heat Transferred (without material resistance)	Q	30.87	KW
Heat Transferred (with material resistance)	Q	30.75	KW
Pressure Drop : Tube Side			
Friction Factor	ƒ	0.06	
Mass Flow Rate Per tube	Mtt	0.005675	kg/s
Total Inlet cross sectional area	Ati	0.00811	m2
Cross sectional area of small tube	At	0.000314	m2
Total Inlet Fluid Velocity	VT	67.8399	m/s
Fluid Velocity per Tube	Vt	29.67	m/s
Outlet Density	ρo	0.75	[kg/m^3]
Total outlet cross sectional area	Ato	0.01824	m2
Mass flow rate at outlet	Mto	0.41146	m/s
Total fluid velocity at outlet	Vto	30.1511	m/s
Minor Loss due to entrance	kentr	1.50	
Minor loss due to exit transition (60°)	kexit	0.20	
Pressure Drop per tube	ΔPt	841.14	Pa
Back Pressure due to outlet transition	Pdo	68.01	Pa
Total back pressure caused by heat exchanger	ΔPT	909.15	Pa
Total back pressure caused by heat exchanger	ΔPT	0.26847	in hg

Table 3-F : Governed Speed Calculations

Governed Speed

Input Data

Description	Symbol	Value	Unit
Tube and Tube Layout			
Inside Shell Diameter	Ds	330.00	mm
Tube Outside Diameter	Dt	25.00	mm
Tube Wall Thickness	Ltw	2.50	mm
Inside Tube Diameter	Dti	20.00	mm
Tube wall material thermal conductivity	λtw	16.00	W/m K
Pitch Ratio	PR	1.25	
Tube layout pitch	Ltp	31.25	mm
Tube layout characteristic angle	θtp	30.00	deg
Tube Length			
Overall nominal tube length	Lto	530.00	mm
Baffled tube length	Lti	464.00	mm
Effective tube length for heat transfer	Lta	464.00	mm
Baffle Geometry			
Baffle cut as percent of Ds	Bc	20.00	%
Central baffle spacing	Lbc	66.00	mm
Tube Bundle Geometry			
Estimated Total number of tubes	Ntt	59.97	
Total number of tubes	Ntt	59.00	
Number of tube passes	Ntp	1.00	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	FX	code
Tube OD-to baffle hole clearance	Ltb	0.60	mm
Inside shell to baffle clearance	Lsb	2.92	mm
Inside shell-to-tube bundle bypass clearance	Lbb	50.00	mm
Center of outermost tube to center of shell	Dctl	255.00	mm
Temperatures			
Shell-side inlet temperature	Tsi	70.00	°C
Shell-side outlet temperature	Tso	85.00	°C
Tube-side inlet temperature	Tti	441.15	°C
Tube-side ouet temperature	Tto	200.00	°C
Shell-side process information			
Shell fluid mass flow rate	Ms	20.00	gpm
Shell fluid mass flow rate	Ms	1.26	l/s
Shell fluid mass flow rate	Ms	1.32	kg/s
At shell fluid mean temperatures	Ts,av	77.50	°C
Density	ρs	1046.54	kg/m3
Thermal Conductivity	λs	0.43	W/m K
Specific Heat	(cp)s	3576.89	J/kg K
Dynamic Viscosity	ηs	0.88	cP=mPa/s
Tube-side process information			
Shell fluid mass flow rate	Mt	542.00	l/s
Tube fluid mass flow rate	Mt	0.32	kg/s
At tube fluid mean temperature	Tt,av	593.58	°K
Density	ρt	0.60	[kg/m^3]
Thermal Conductivity	λt	0.0462	W/m K
Specific Heat	(cp)t	1049.65	J/kg K
Dynamic Viscosity	µ	3.00E-05	[kg/m s]

Output Data

Description	Symbol	Value	Unit
Shell Side Data			
Cross-flow area Sm at the shell centerline	Sm	6666.00	mm2
Max Shell side cross flow velocity	ṁs	198.08	kg/m2 s
Reynolds Data - Shell Side	Res	5640.49	
Prandtl Number - Shell Side	Prs	7.30	
Delta T1	ΔT1	356.15	°C
Delta T2	ΔT2	130.00	°C
Log mean temperature difference	ΔTLM	224.40	°C
Effective Tube Length (Equivalent for Non U tube)	Lta	464.00	mm
Total heat transfer surface of the Exchanger	Ao	2.15	m2
Heat Transfer factor a	a	0.11	
Heat Transfer factor ji	ji	0.01	
Viscosity at wall (Maximum Viscosity for Water Glycol)	ηs,w	0.70	
Correction factor for viscosity gradient	$(\phi_s)^r$	1.03	
Heat Transfer Coefficient	hs	2199.68	W/m2 K
Tube Side Data			
Mass Flow Rate -Tube Side	Mt	0.32	kg/s
Mass Flow Rate per tube	Mtt	0.0055	kg/s
Reynolds Number - Tube Side	Ret	11620.25	
Prandlt Number - Tube Side	Pr	0.68	
Nusselt Number - Tube side	Nut	36.66	
Heat Transfer Coefficient	ht	84.71	[W/m2·k]
Overall Heat Transfer Coefficient			
Overall Heat Transfer Coefficient (without material resistance)	U	81.57	w/m2 k
Overall Heat Transfer Coefficient (with material resistance)	U	81.25	w/m2 k
Heat Transferred (without material resistance)	Q	31.48	KW
Heat Transferred (with material resistance)	Q	31.36	KW
Pressure Drop : Tube Side			
Friction Factor	ƒ	0.06	
Mass Flow Rate Per tube	Mtt	0.005468	kg/s
Total Inlet cross sectional area	Ati	0.00811	m2
Cross sectional area of small tube	At	0.000314	m2
Total Inlet Fluid Velocity	VT	66.8532	m/s
Fluid Velocity per Tube	Vt	29.24	m/s
Outlet Density	ρo	0.75	[kg/m^3]
Total outlet cross sectional area	Ato	0.01824	m2
Mass flow rate at outlet	Mto	0.40547	m/s
Total fluid velocity at outlet	Vto	29.7125	m/s
Minor Loss due to entrance	kentr	1.50	
Minor loss due to exit transition (60°)	kexit	0.20	
Pressure Drop per tube	ΔPt	798.54	Pa
Back Pressure due to outlet transition	Pdo	66.04	Pa
Total back pressure caused by heat exchanger	ΔPT	864.59	Pa
Total back pressure caused by heat exchanger	ΔPT	0.25531	in hg

Table 4-F : High Idle Calculations

High Idle

Input Data

Description	Symbol	Value	Unit
Tube and Tube Layout			
Inside Shell Diameter	Ds	330.00	mm
Tube Outside Diameter	Dt	25.00	mm
Tube Wall Thickness	Ltw	2.50	mm
Inside Tube Diameter	Dti	20.00	mm
Tube wall material thermal conductivity	λtw	16.00	W/m K
Pitch Ratio	PR	1.25	
Tube layout pitch	Ltp	31.25	mm
Tube layout characteristic angle	θtp	30.00	deg
Tube Length			
Overall nominal tube length	Lto	530.00	mm
Baffled tube length	Lti	464.00	mm
Effective tube length for heat transfer	Lta	464.00	mm
Baffle Geometry			
Baffle cut as percent of Ds	Bc	20.00	%
Central baffle spacing	Lbc	66.00	mm
Tube Bundle Geometry			
Estimated Total number of tubes	Ntt	59.97	
Total number of tubes	Ntt	59.00	
Number of tube passes	Ntp	1.00	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	FX	code
Tube OD-to baffle hole clearance	Ltb	0.60	mm
Inside shell to baffle clearance	Lsb	2.92	mm
Inside shell-to-tube bundle bypass clearance	Lbb	50.00	mm
Center of outermost tube to center of shell	Dctl	255.00	mm
Temperatures			
Shell-side inlet temperature	Tsi	70.00	°C
Shell-side outlet temperature	Tso	85.00	°C
Tube-side inlet temperature	Tti	374.00	°C
Tube-side ouet temperature	Tto	200.00	°C
Shell-side process information			
Shell fluid mass flow rate	Ms	20.00	gpm
Shell fluid mass flow rate	Ms	1.26	l/s
Shell fluid mass flow rate	Ms	1.32	kg/s
At shell fluid mean temperatures	Ts,av	77.50	°C
Density	ρs	1046.54	kg/m3
Thermal Conductivity	λs	0.43	W/m K
Specific Heat	(cp)s	3576.89	J/kg K
Dynamic Viscosity	ηs	0.88	cP=mPa/s
Tube-side process information			
Shell fluid mass flow rate	Mt	337.19	l/s
Tube fluid mass flow rate	Mt	0.21	kg/s
At tube fluid mean temperature	Tt,av	560.00	°K
Density	ρt	0.63	[kg/m^3]
Thermal Conductivity	λt	0.0442	W/m K
Specific Heat	(cp)t	1042.06	J/kg K
Dynamic Viscosity	µ	2.88E-05	[kg/m s]

Output Data

Description	Symbol	Value	Unit
Shell Side Data			
Cross-flow area Sm at the shell centerline	Sm	6666.00	mm2
Max Shell side cross flow velocity	ṁs	198.08	kg/m2 s
Reynolds Data - Shell Side	Res	5640.49	
Prandtl Number - Shell Side	Prs	7.30	
Delta T1	ΔT1	289.00	°C
Delta T2	ΔT2	130.00	°C
Log mean temperature difference	ΔTLM	199.03	°C
Effective Tube Length (Equivalent for Non U tube)	Lta	464.00	mm
Total heat transfer surface of the Exchanger	Ao	2.15	m2
Heat Transfer factor a	a	0.11	
Heat Transfer factor ji	ji	0.01	
Viscosity at wall (Maximum Viscosity for Water Glycol)	ηs,w	0.70	
Correction factor for viscosity gradient	$(\phi_s)^r$	1.03	
Heat Transfer Coefficient	hs	2199.68	W/m2 K
Tube Side Data			
Mass Flow Rate -Tube Side	Mt	0.21	kg/s
Mass Flow Rate per tube	Mtt	0.0036	kg/s
Reynolds Number - Tube Side	Ret	7965.57	
Prandlt Number - Tube Side	Pr	0.68	
Nusselt Number - Tube side	Nut	27.10	
Heat Transfer Coefficient	ht	59.86	[W/m2·k]
Overall Heat Transfer Coefficient			
Overall Heat Transfer Coefficient (without material resistance)	U	58.27	w/m2 k
Overall Heat Transfer Coefficient (with material resistance)	U	58.11	w/m2 k
Heat Transferred (without material resistance)	Q	19.95	KW
Heat Transferred (with material resistance)	Q	19.89	KW
Pressure Drop : Tube Side			
Friction Factor	ƒ	0.06	
Mass Flow Rate Per tube	Mtt	0.003607	kg/s
Total Inlet cross sectional area	Ati	0.00811	m2
Cross sectional area of small tube	At	0.000314	m2
Total Inlet Fluid Velocity	VT	41.5908	m/s
Fluid Velocity per Tube	Vt	18.19	m/s
Outlet Density	ρo	0.75	[kg/m^3]
Total outlet cross sectional area	Ato	0.01824	m2
Mass flow rate at outlet	Mto	0.25225	m/s
Total fluid velocity at outlet	Vto	18.4848	m/s
Minor Loss due to entrance	kentr	1.50	
Minor loss due to exit transition (60°)	kexit	0.20	
Pressure Drop per tube	ΔPt	327.72	Pa
Back Pressure due to outlet transition	Pdo	25.56	Pa
Total back pressure caused by heat exchanger	ΔPT	353.28	Pa
Total back pressure caused by heat exchanger	ΔPT	0.10432	in hg

Table 5-F : Low idle Calculations

Low Idle

Input Data

Description	Symbol	Value	Unit
Tube and Tube Layout			
Inside Shell Diameter	Ds	330.00	mm
Tube Outside Diameter	Dt	25.00	mm
Tube Wall Thickness	Ltw	2.50	mm
Inside Tube Diameter	Dti	20.00	mm
Tube wall material thermal conductivity	λtw	16.00	W/m K
Pitch Ratio	PR	1.25	
Tube layout pitch	Ltp	31.25	mm
Tube layout characteristic angle	θtp	30.00	deg
Tube Length			
Overall nominal tube length	Lto	530.00	mm
Baffled tube length	Lti	464.00	mm
Effective tube length for heat transfer	Lta	464.00	mm
Baffle Geometry			
Baffle cut as percent of Ds	Bc	20.00	%
Central baffle spacing	Lbc	66.00	mm
Tube Bundle Geometry			
Estimated Total number of tubes	Ntt	59.97	
Total number of tubes	Ntt	59.00	
Number of tube passes	Ntp	1.00	
Tube bundle type (FX, UT, SRFH, PFH, PTFH)	CB	FX	code
Tube OD-to baffle hole clearance	Ltb	0.60	mm
Inside shell to baffle clearance	Lsb	2.92	mm
Inside shell-to-tube bundle bypass clearance	Lbb	50.00	mm
Center of outermost tube to center of shell	Dctl	255.00	mm
Temperatures			
Shell-side inlet temperature	Tsi	70.00	°C
Shell-side outlet temperature	Tso	85.00	°C
Tube-side inlet temperature	Tti	340.00	°C
Tube-side ouet temperature	Tto	200.00	°C
Shell-side process information			
Shell fluid mass flow rate	Ms	20.00	gpm
Shell fluid mass flow rate	Ms	1.26	l/s
Shell fluid mass flow rate	Ms	1.32	kg/s
At shell fluid mean temperatures	Ts,av	77.50	°C
Density	ρs	1046.54	kg/m3
Thermal Conductivity	λs	0.43	W/m K
Specific Heat	(cp)s	3576.89	J/kg K
Dynamic Viscosity	ηs	0.88	cP=mPa/s
Tube-side process information			
Shell fluid mass flow rate	Mt	294.61	l/s
Tube fluid mass flow rate	Mt	0.19	kg/s
At tube fluid mean temperature	Tt,av	543.00	°K
Density	ρt	0.65	kg/m^3
Thermal Conductivity	λt	0.0431	W/m K
Specific Heat	(cp)t	1038.36	J/kg K
Dynamic Viscosity	µ	2.82E-05	kg/m s

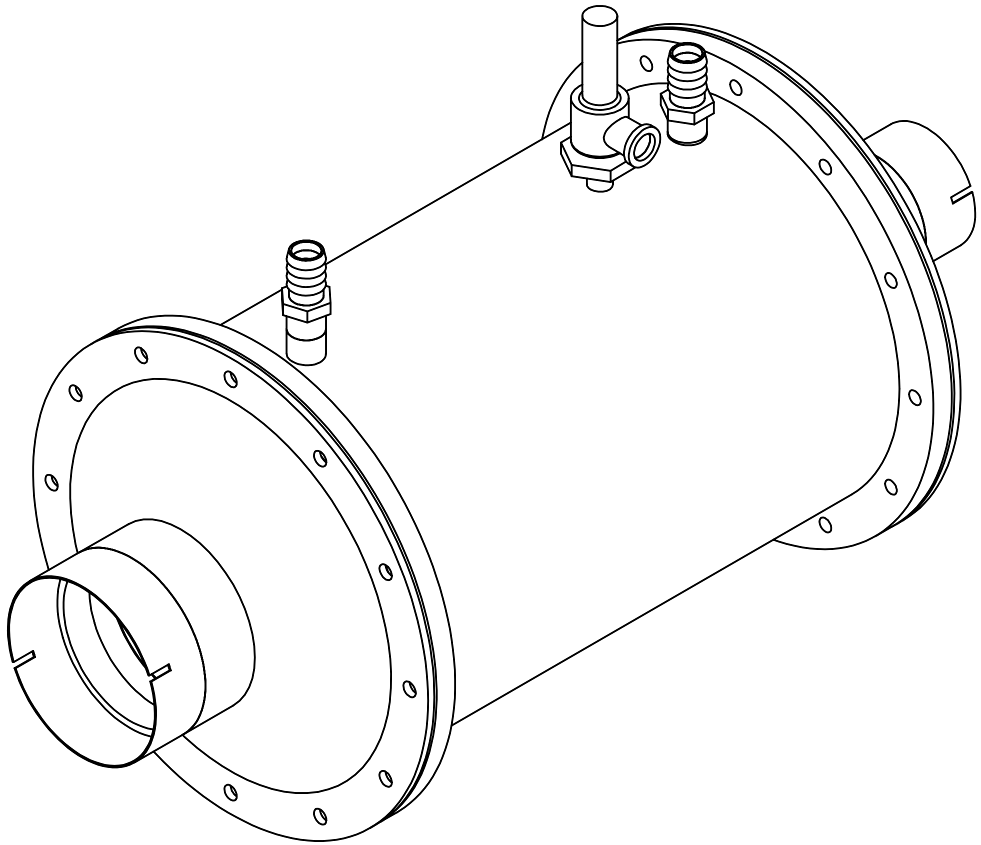
Output Data

Description	Symbol	Value	Unit
Shell Side Data			
Cross-flow area Sm at the shell centerline	Sm	6666.00	mm2
Max Shell side cross flow velocity	ṁs	198.08	kg/m2 s
Reynolds Data - Shell Side	Res	5640.49	
Prandtl Number - Shell Side	Prs	7.30	
Delta T1	ΔT1	255.00	°C
Delta T2	ΔT2	130.00	°C
Log mean temperature difference	ΔTLM	185.53	°C
Effective Tube Length (Equivalent for Non U tube)	Lta	464.00	mm
Total heat transfer surface of the Exchanger	Ao	2.15	m2
Heat Transfer factor a	a	0.11	
Heat Transfer factor ji	ji	0.01	
Viscosity at wall (Maximum Viscosity for Water Glycol)	ηs,w	0.70	
Correction factor for viscosity gradient	$(\phi_s)^r$	1.03	
Heat Transfer Coefficient	hs	2199.68	W/m2 K
Tube Side Data			
Mass Flow Rate -Tube Side	Mt	0.19	kg/s
Mass Flow Rate per tube	Mtt	0.0032	kg/s
Reynolds Number - Tube Side	Ret	7325.88	
Prandlt Number - Tube Side	Pr	0.68	
Nusselt Number - Tube side	Nut	25.34	
Heat Transfer Coefficient	ht	54.65	W/m2·k
Overall Heat Transfer Coefficient			
Overall Heat Transfer Coefficient (without material resistance)	U	53.32	w/m2 k
Overall Heat Transfer Coefficient (with material resistance)	U	53.19	w/m2 k
Heat Transferred (without material resistance)	Q	17.02	KW
Heat Transferred (with material resistance)	Q	16.97	KW
Pressure Drop : Tube Side			
Friction Factor	ƒ	0.06	
Mass Flow Rate Per tube	Mtt	0.003250	kg/s
Total Inlet cross sectional area	Ati	0.008107	m2
Cross sectional area of small tube	At	0.000314	m2
Total Inlet Fluid Velocity	VT	36.33877	m/s
Fluid Velocity per Tube	Vt	15.89	m/s
Outlet Density	ρo	0.75	[kg/m^3]
Total outlet cross sectional area	Ato	0.018241	m2
Mass flow rate at outlet	Mto	0.220398	m/s
Total fluid velocity at outlet	Vto	16.15056	m/s
Minor Loss due to entrance	kentr	1.50	
Minor loss due to exit transition (60°)	kexit	0.20	
Pressure Drop per tube	ΔPt	257.98	Pa
Back Pressure due to outlet transition	Pdo	19.51	Pa
Total back pressure caused by heat exchanger	ΔPT	277.50	Pa
Total back pressure caused by heat exchanger	ΔPT	0.081944	in hg

Table 6-F: Stress Analysis Vessel

Stress Calculations for Shell and Nozzle			
Inputs			
Pressure inside vessel	P	500	lb_f/in^2
Diameter of vessel	D	13.07	*in*
Diameter of nozzle	d	1	*in*
Corroded wall thickness	T	0.087	*in*
Corroded nozzle wall thickness	t	0.079	*in*
Ultimate Tensile of 304 SS	σ_u	85 000	lb_f/in^2
Calculated Stresses			
Stress due to pressure	σ_p	37557.5	lb_f/in^2
Stress on vessel due to surface pressure	S_{vp}	62798.8	lb_f/in^2
Stress on vessel due to membrane pressure	S_{vmp}	70371.3	lb_f/in^2

Appendix G: Shell and Tube Heat Exchanger Drawings



UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE			
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL: 304L SS		
				WEIGHT:		

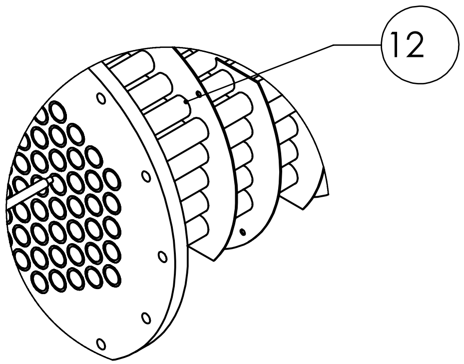
TITLE: Isometric view of Shell and tube Heat Exchanger

DWG NO. S&T Heat Exchanger Assy

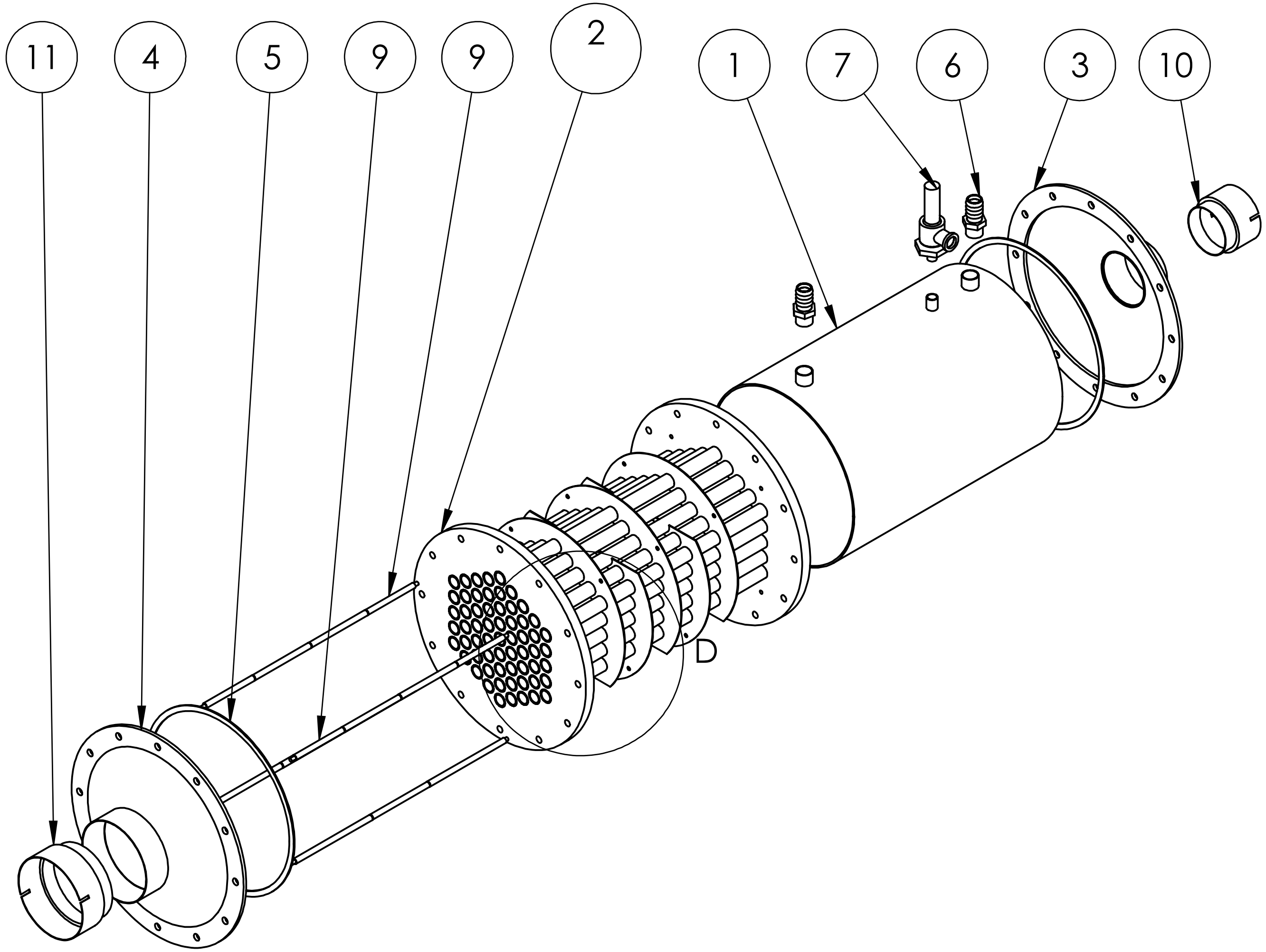
A4

SCALE:1:20

SHEET 1 OF 10



DETAIL D
SCALE 1 : 8



ITEM NO.	PART	QTY.
1	Shell	1
2	Tube Bundles	1
3	Inlet Reducer	1
4	Outlet Reducer	1
5	Gasket	2
6	Barb Fitting	2
7	Pressure Relief Valve	1
8	Tie Rod 1	2
9	Tie Rod 2	2
10	4 inch End Connection	1
11	6 inch End Connection	1
12	Tube	60

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: Heat Exchanger Exploded View

MATERIAL: 304L SS

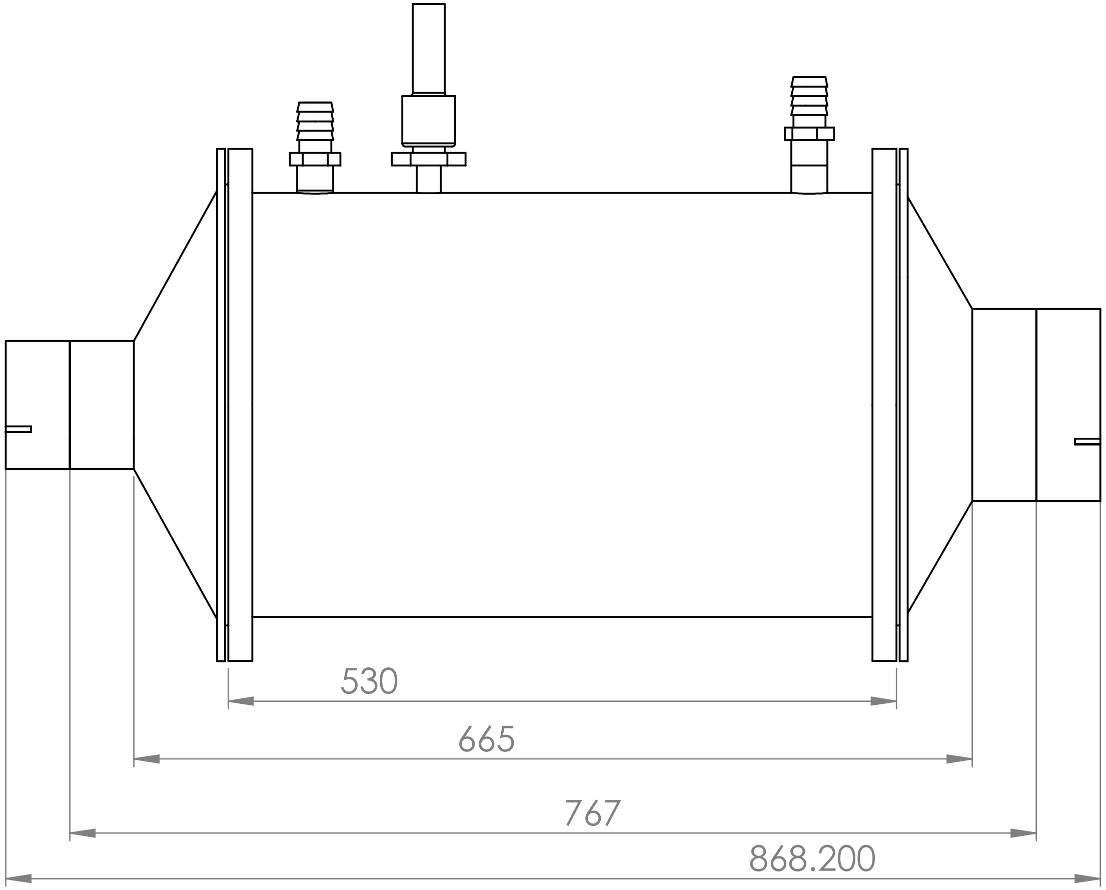
DWG NO. S&T Heat Exchanger Assy

A4

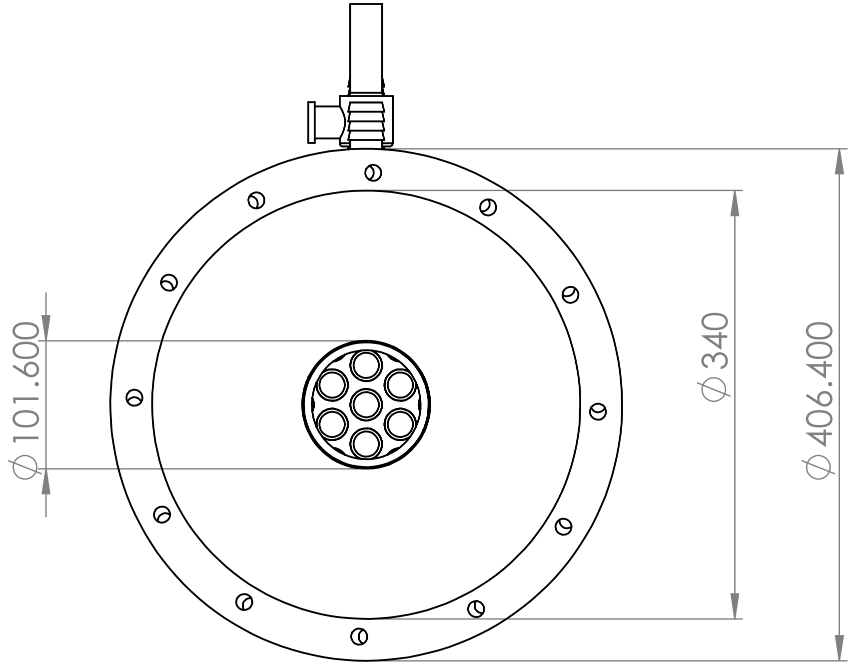
WEIGHT:

SCALE:1:20

SHEET 2 OF 10



Heat Exchanger Side View



Heat Exchanger Front View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: Heat Exchanger Overall Dimensions

MATERIAL: 304L SS

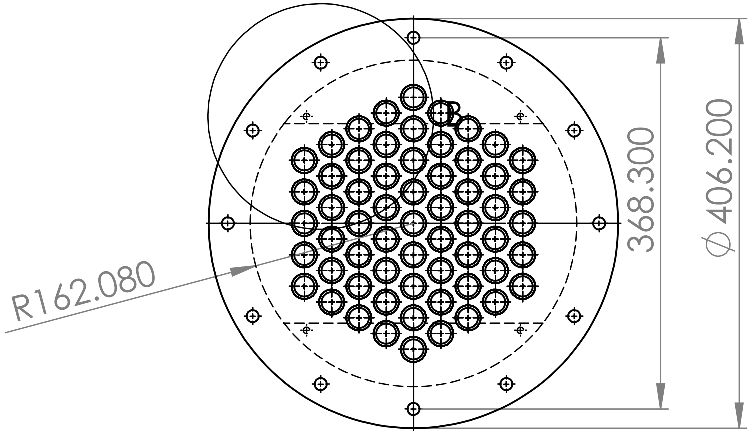
WEIGHT:

DWG NO. S&T Heat Exchanger Assy

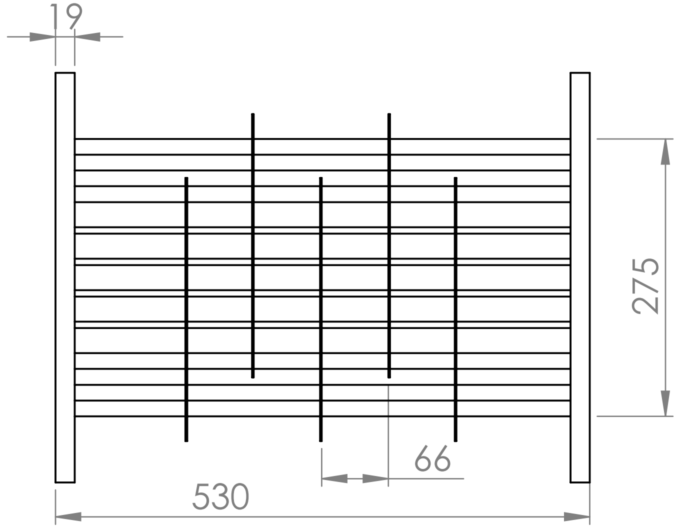
A4

SCALE:1:20

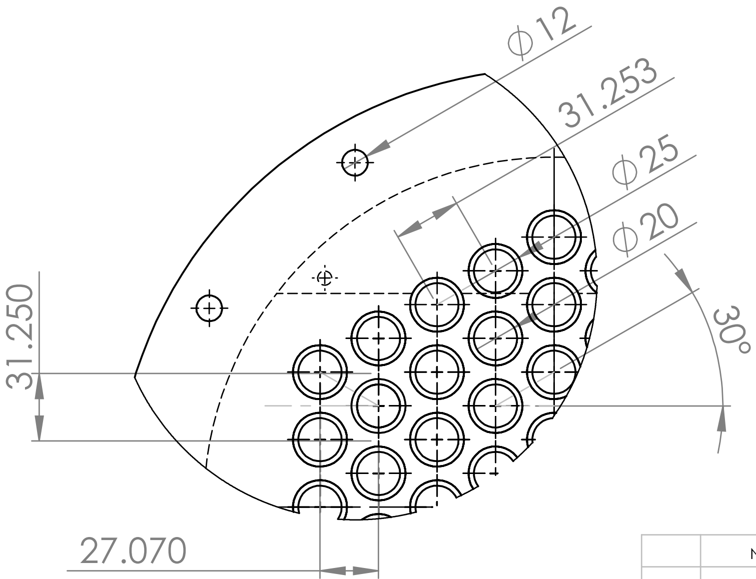
SHEET 3 OF 10



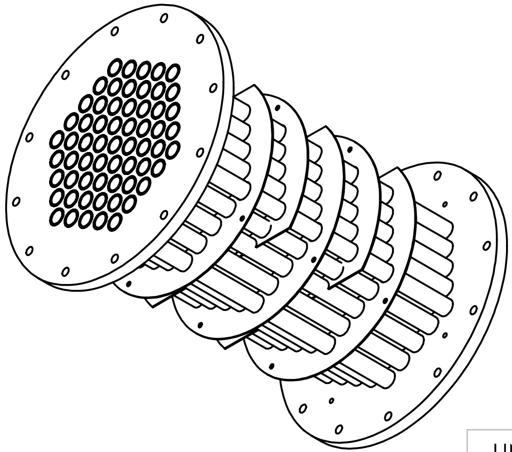
Tubebundle Cross-Section View



Tubebundle Side View



DETAIL B
SCALE 1 : 3.5



Tubebundle Isometric View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: Tube Bundle Geometries

MATERIAL: 304L SS

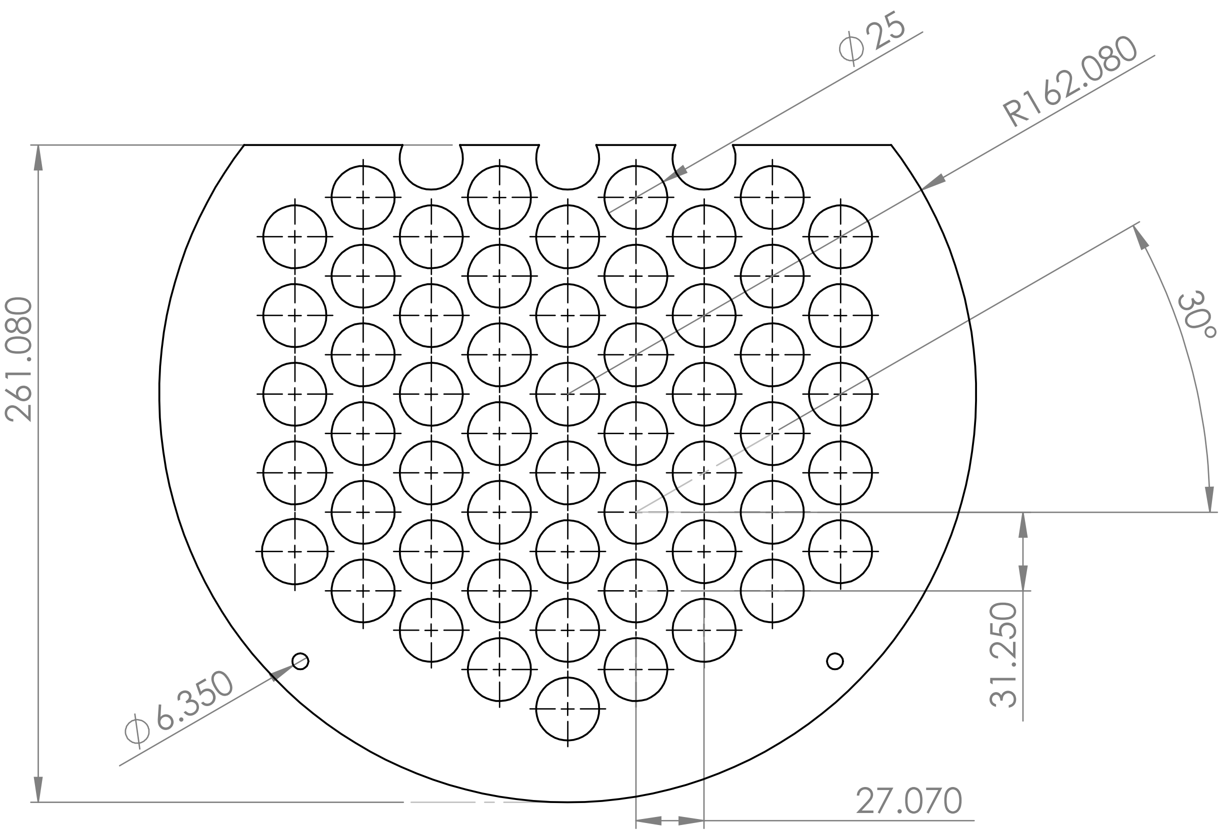
WEIGHT:

DWG NO. S&T Heat Exchanger Assy

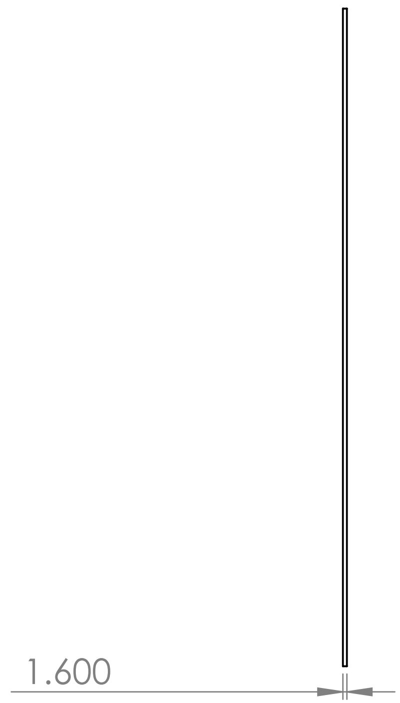
A4

SCALE:1:20

SHEET 4 OF 10



Baffle Cross-Section View



Baffle Side View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL: 304L SS

WEIGHT:

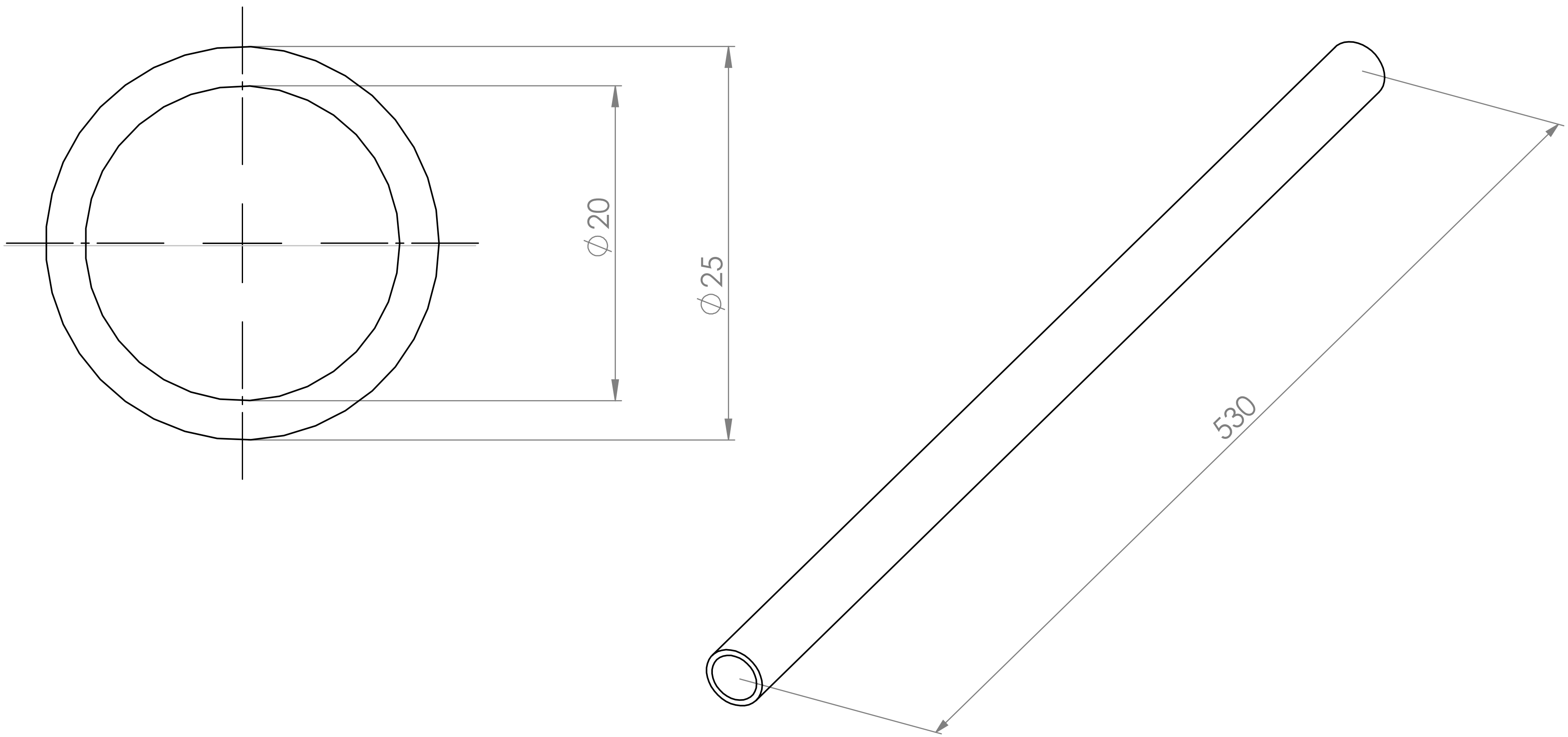
TITLE: Baffle Geometries

DWG NO. S&T Heat Exchanger Assy

A4

SCALE:1:20

SHEET 5 OF 10



UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

TubeDimensions

MATERIAL:

304L SS

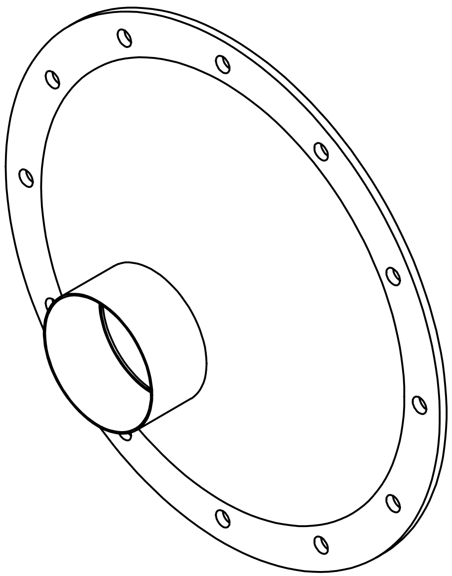
DWG NO. S&T Heat Exchanger Assy

A4

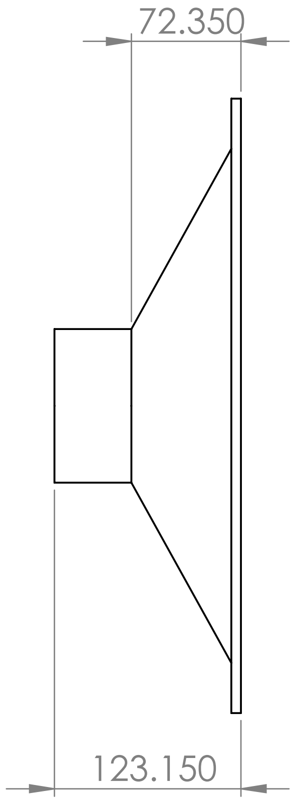
WEIGHT:

SCALE:1:20

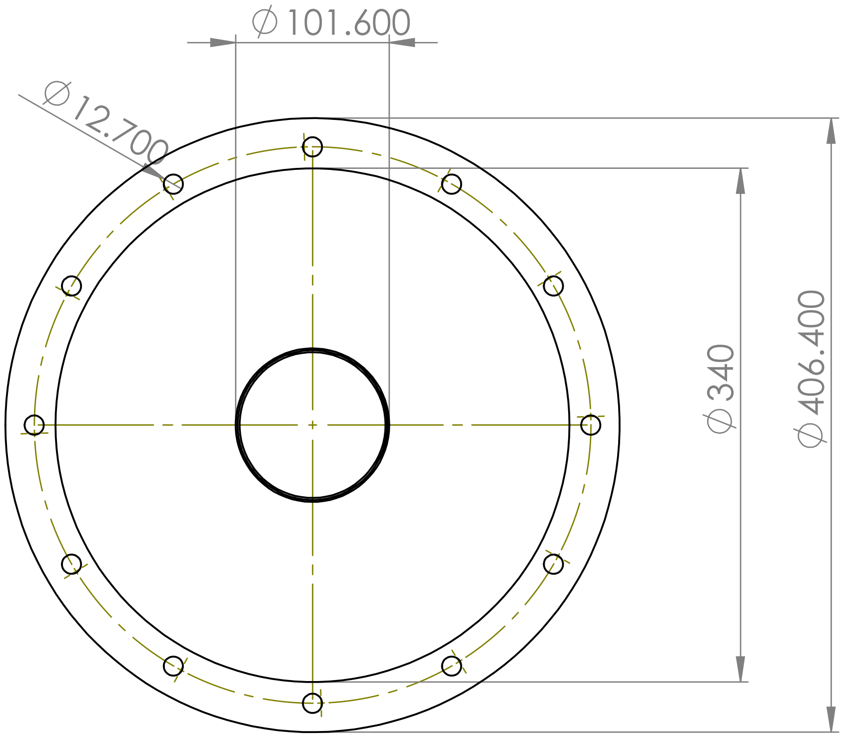
SHEET 6 OF 10



Reducer Isometric View



Reducer Side View



Reducer Front View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: 4" Inlet Reducer

MATERIAL: 304L SS

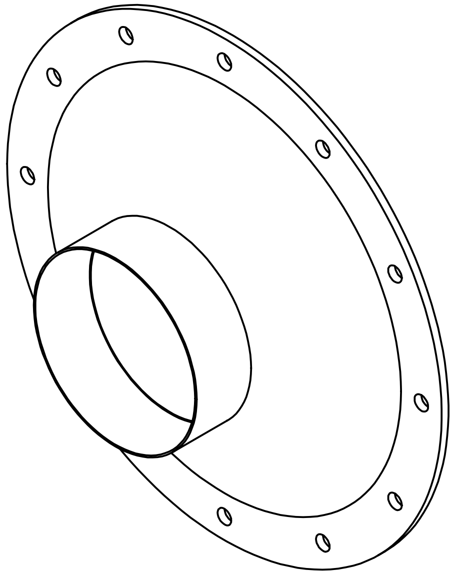
DWG NO. S&T Heat Exchanger Assy

A4

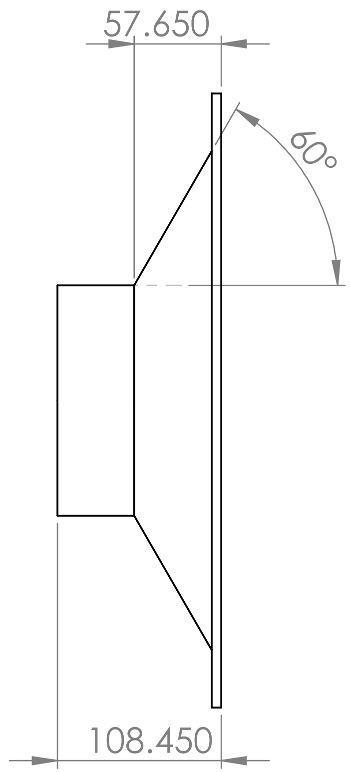
WEIGHT:

SCALE:1:20

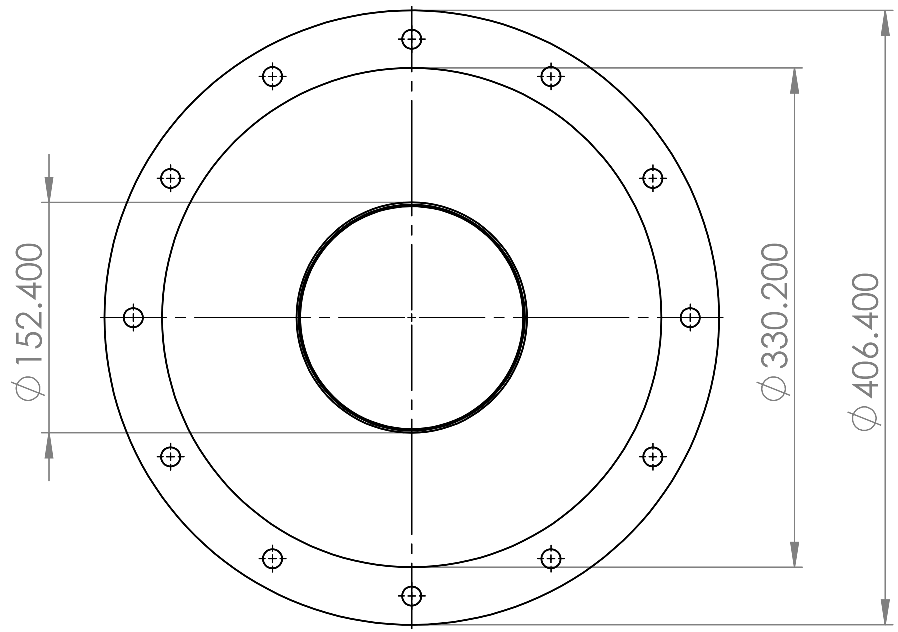
SHEET 7 OF 10



Outlet Reducer
Isometric View



Outlet Reducer
Side View



Outlet Reducer
Front View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE			
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A						

TITLE:

6" Outlet Reducer

MATERIAL: 304L SS

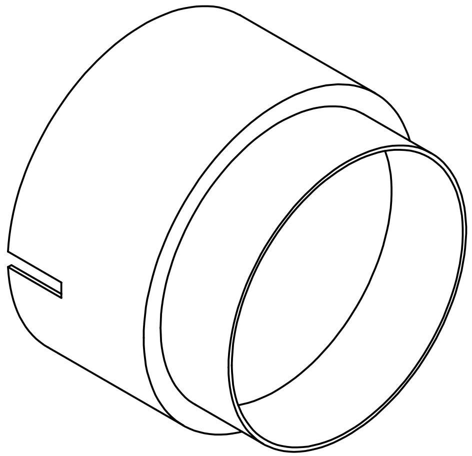
WEIGHT:

DWG NO. S&T Heat Exchanger Assy

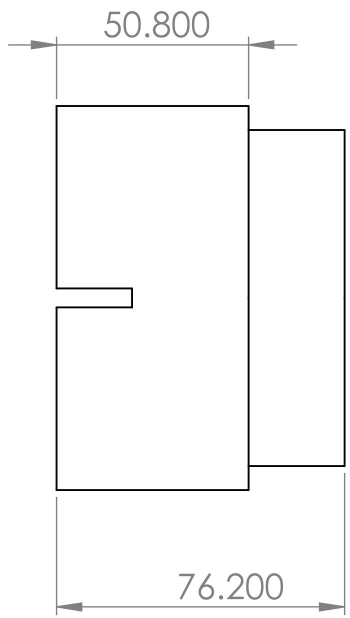
A4

SCALE:1:20

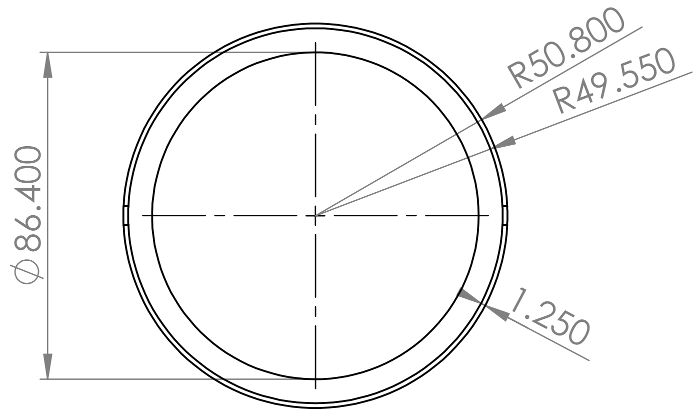
SHEET 8 OF 10



Exhaust Connector Isometric View



Exhaust Connector Side View



Exhaust Connector Front View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: 4" Exhaust Connector Pipe

MATERIAL: 304L SS

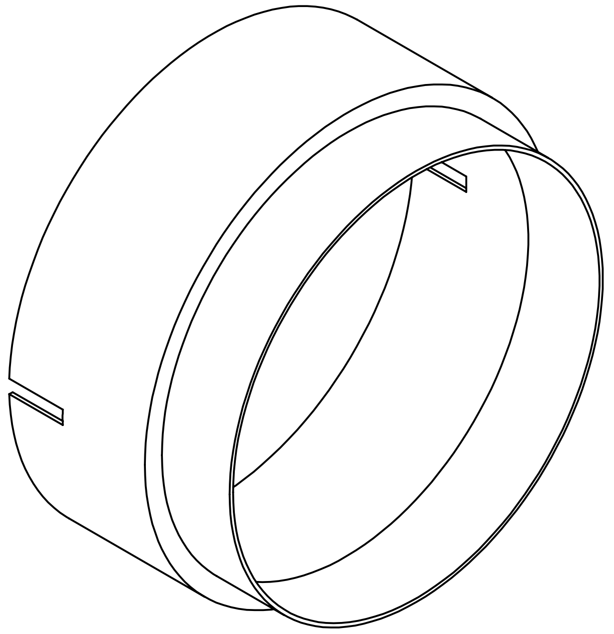
WEIGHT:

DWG NO. S&T Heat Exchanger Assy

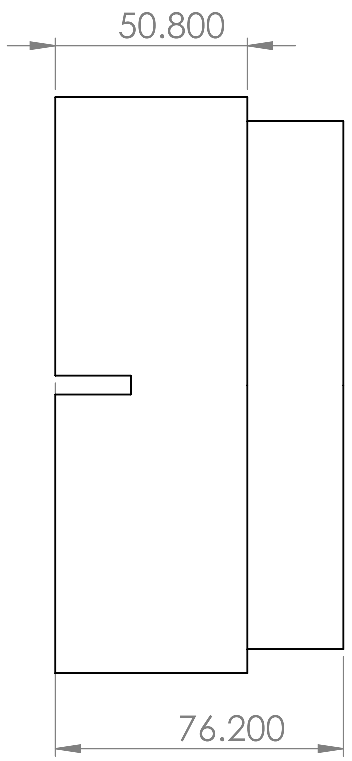
A4

SCALE:1:20

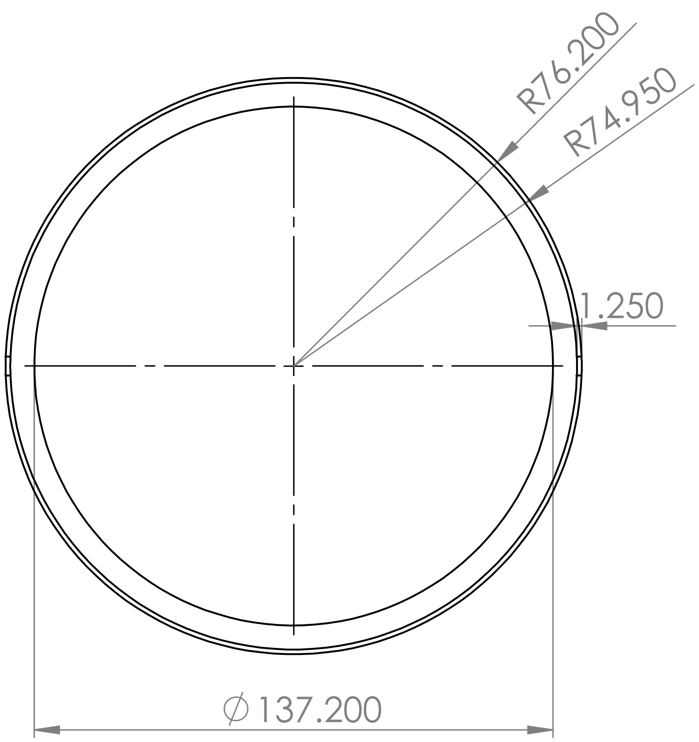
SHEET 9 OF 10



Exhaust Connector
Isometric View



Exhaust Connector Side View



Exhaust Connector Front View

UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN MM

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: 6'' Exhaust Outlet Connector

MATERIAL: 304L SS

DWG NO. S&T Heat Exchanger Assy

A4

WEIGHT:

SCALE:1:20

SHEET 10 OF 10

Appendix H: Total Backpressure Imposed by System

Total Back Pressure Imposed by System

Peak Torque			
Variable	**Symbol**	**Value**	**Unit**
Exhaust Gas Flow		401	*l/s*
Exhaust Gas Temperature	Ti	482	*°C*
Pipe Diameter	D	101.6	*mm*
Density	ρ	0.47	*kg/m^3*
Mass Flow Rate	ṁ	0.1875	*(kg/s)*
Cross Sectional Area	A	0.0081	*m^2*
Fluid Velocity	V	49.4615	*m/s*
Dynamic Viscosity	μ	3.50E-05	*kg/m s*
Reynolds Number	Re	6.72E+04	
Material Roughness	k	0.00003	*m*
Friction Factor	ƒ	0.020757995	

Section	System Components	Equivalent Length of Components	Number Components	Equivalent Length	Section Pressure Loss
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exchanger					
	Straight Pipe	1	762	762	0.133589492
	Butterfly Valve	2032	1	2032	0.356238644
	Heat Exchanger	n/a	1	n/a	0.470758704
	90° Elbow	3048	1	3048	0.534357966
Section Summary					**1.494944806**
Total					**25.87594481**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.133589492
	Butterfly Valve	2032	1	2032	0.356238644
	90° Elbow	3048	1	3048	0.534357966
Section Summary					**1.024186102**
Total					**25.4051861**

Maximum Power			
Variable	**Symbol**	**Value**	**Unit**
Exhaust Gas Flow		550	*l/s*
Exhaust Gas Temperature	Ti	489	*°C*
Pipe Diameter	D	101.6	*mm*
Density	ρ	0.46	*kg/m^3*
Mass Flow Rate	ṁ	0.2549	*(kg/s)*
Cross Sectional Area	A	0.0081	*m^2*
Fluid Velocity	V	67.8399	*m/s*
Dynamic Viscosity	μ	3.52E-05	*kg/m s*
Reynolds Number	Re	9.09E+04	
Material Roughness	k	0.00003	*m*
Friction Factor	ƒ	0.019736073	

Section	**System Components**	**Equivalent Length of Components**	**Number Components**	**Equivalent Length**	**Section Pressure Loss**
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exchanger					
	Straight Pipe	1	762	762	0.236834526
	Butterfly Valve	2032	1	2032	0.631558737
	Heat Exchanger	n/a	1	n/a	0.909146356
	90° Elbow	3048	1	3048	0.947338106
Section Summary					**2.724877725**
Total					**27.10587773**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.236834526
	Butterfly Valve	2032	1	2032	0.631558737
	90° Elbow	3048	1	3048	0.947338106
Section Summary					**1.815731369**
Total					**26.19673137**

Governed Speed			
Variable	Symbol	Value	Unit
Exhaust Gas Flow		542	*l/s*
Exhaust Gas Temperature	Ti	519	*°C*
Pipe Diameter	D	101.6	*mm*
Density	ρ	0.45	*kg/m^3*
Mass Flow Rate	ṁ	0.2417	*(kg/s)*
Cross Sectional Area	A	0.0081	*m^2*
Fluid Velocity	V	66.8532	*m/s*
Dynamic Viscosity	µ	3.60E-05	*kg/m s*
Reynolds Number	Re	8.41E+04	
Material Roughness	k	0.00003	*m*
Friction Factor	ƒ	0.019984934	

Section	System Components	Equivalent Length of Components	Number Components	Equivalent Length	Section Pressure Loss
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exchanger					
	Straight Pipe	1	762	762	0.224032275
	Butterfly Valve	2032	1	2032	0.597419401
	Heat Exchanger	n/a	1	n/a	0.86458676
	90° Elbow	3048	1	3048	0.896129101
Section Summary					**2.582167537**
Total					**26.96316754**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.224032275
	Butterfly Valve	2032	1	2032	0.597419401
	90° Elbow	3048	1	3048	0.896129101
Section Summary					**1.717580778**
Total					**26.09858078**

High Idle			
Variable	**Symbol**	**Value**	**Unit**
Exhaust Gas Flow		337.19	*l/s*
Exhaust Gas Temperature	Ti	440	*°C*
Pipe Diameter	D	101.6	*mm*
Density	ρ	0.50	*kg/m^3*
Mass Flow Rate	ṁ	0.1671	*(kg/s)*
Cross Sectional Area	A	0.0081	*m^2*
Fluid Velocity	V	41.5908	*m/s*
Dynamic Viscosity	µ	3.37E-05	*kg/m s*
Reynolds Number	Re	6.21E+04	
Material Roughness	k	0.00003	*m*
Friction Factor	ƒ	0.021049659	

Section	System Components	Equivalent Length of Components	Number Components	Equivalent Length	Section Pressure Loss
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exchanger					
	Straight Pipe	1	762	762	0.10149376
	Butterfly Valve	2032	1	2032	0.270650026
	Heat Exchanger	n/a	1	n/a	0.353281136
	90° Elbow	3048	1	3048	0.405975039
Section Summary					**1.131399961**
Total					**25.51239996**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.10149376
	Butterfly Valve	2032	1	2032	0.270650026
	90° Elbow	3048	1	3048	0.405975039
Section Summary					**0.778118825**
Total					**25.15911882**

Low Idle			
Variable	**Symbol**	**Value**	**Unit**
Exhaust Gas Flow		294.61	*l/s*
Exhaust Gas Temperature	Ti	400	*°C*
Pipe Diameter	D	101.6	*mm*
Density	ρ	0.53	*kg/m^3*
Mass Flow Rate	ṁ	0.1547	*(kg/s)*
Cross Sectional Area	A	0.0081	*m^2*
Fluid Velocity	V	36.3388	*m/s*
Dynamic Viscosity	µ	3.25E-05	*kg/m s*
Reynolds Number	Re	5.97E+04	
Material Roughness	k	0.00003	*m*
Friction Factor	ƒ	0.021201211	

Section	System Components	Equivalent Length of Components	Number Components	Equivalent Length	Section Pressure Loss
		mm		*mm*	*kpa*
Engine to SCR					
0-1	n/a	n/a	n/a		24.381
Valve #1 Open, Valve #2 Closed: Flow through Heat Exchanger					
	Straight Pipe	1	762	762	0.08270783
	Butterfly Valve	2032	1	2032	0.220554213
	Heat Exchanger	n/a	1	n/a	0.277495356
	90° Elbow	3048	1	3048	0.330831319
Section Summary					**0.911588719**
Total					**25.29258872**
Valve #1 Closed, Valve #2 Open: Bypassing Heat Exchanger					
	Straight Pipe	1	762	762	0.08270783
	Butterfly Valve	2032	1	2032	0.220554213
	90° Elbow	3048	1	3048	0.330831319
Section Summary					**0.634093362**
Total					**25.01509336**