Design and Modeling of an Exhaust Gas Waste Heat Autoclave

by

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Abstract

In order to provide proper sterilization and cleaning of medical equipment for field hospitals and third-world countries while also decreasing the reliance on electricity of traditional sterilization methods, a new steam sterilizer/autoclave system was designed and modeled. This system uses waste engine heat from the exhaust system of a diesel generator set to boil water and produce the pressurized steam conditions necessary for effective medical sterilization.

Currently, the design utilizes a 0.59 meter, concentric tube cross-flow heat exchanger and high-temperature heat transfer fluid to draw thermal energy from the exhaust pipe and deposit it into the autoclave pressure vessel to create steam. The system is designed to run a 35-minute sterilization cycle, requiring 15 minutes to produce saturated steam at 2 atmospheres within a 50-liter autoclave, and 20 minutes to sterilize medical instruments in the steam environment. Furthermore, the system uses basic, off-the-shelf fluid transfer materials to provide a robust, effective system that can be easily maintained in the field without need for specialized parts or technicians.

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1 Introduction

In many third world countries and remote parts of the world, biological and medical sanitation can be difficult to maintain due to limited funds for purification equipment. Furthermore, many tasks that depend on sterile instruments, such as surgery and biological testing, suffer without proper means of sterilizing tools. Inconsistent sterilization practices can lead to the spread of disease and infection, further exacerbating the problem of ill-equipped medical facilities. One way to sterilize tools for either surgery or cooking is by flushing them with steam. However, a steady heat source is required, meaning extra constraints on the already scarce resources in such locations.

Typically, facilities such as these are powered by large diesel generator sets which produce electricity; these sets are limited in their electrical output, thus constraining the daily operations of the facility. However, as is the case with most internal combustion engines, most of the energy output by the system comes in the form of thermal energy. Currently, this energy is released into the atmosphere via the generator set's exhaust system and wasted. This project involved designing a system to utilize the waste heat expelled by the generator set to boil water and create pressurized steam for medical sterilization.

The exhaust gas waste heat autoclave is broken into three major system components: the autoclave pressure vessel, the exhaust gas waste heat exchanger, and the working fluid transfer piping between the two. Since the operational characteristics of each component rely on the specific design of the other two components, all three components were designed simultaneously, individually optimized, and then combined and optimized as a full working system. The system itself was designed around real-world materials and products that would be available in the areas in which the system was most likely to be deployed, such as field hospitals in third-world countries and forward deployed military areas. By designing the system based on regionally available materials, system downtime can be greatly reduced. The system requires no specialized parts or technicians to maintain it, thus repairs and maintenance can be done locally to save on cost and time. Furthermore, the waste heat autoclave system is designed to utilize the waste heat created by the generator sets without decreasing their performance or efficiency.

2 Initial Design

The exhaust gas waste heat exchanger is essentially a closed loop heat transfer system similar to that found in a household refrigerator (Fig. 1).

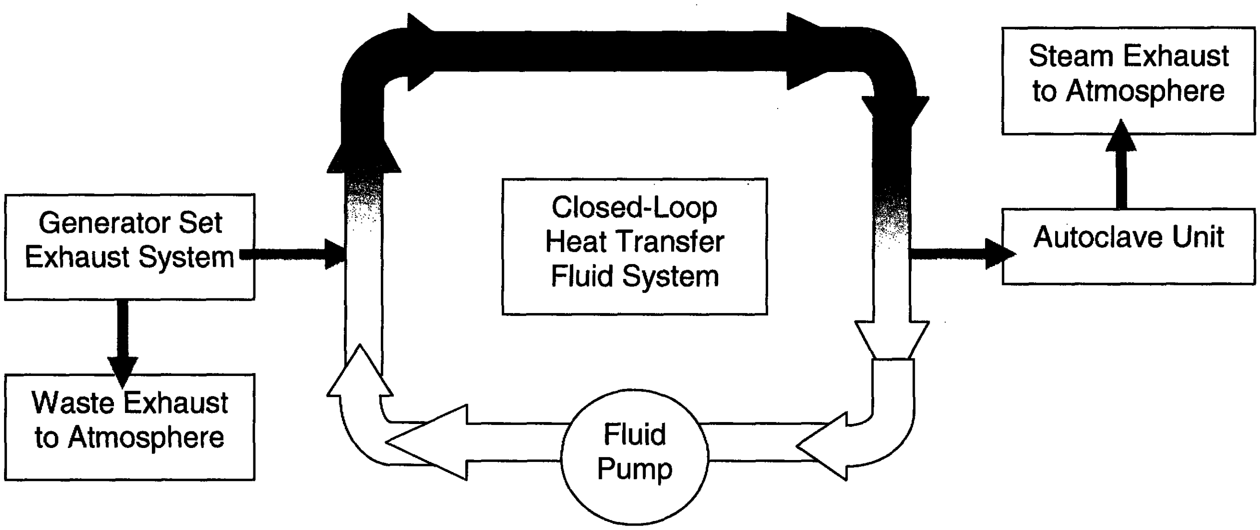

Fig. 1: Thermal energy is transferred into the heat transfer fluid from the hot exhaust stream. The hot heat transfer fluid (gray) is pumped to the autoclave, where it releases its stored thermal energy into the water within the autoclave. The cool heat transfer fluid (white) is then pumped back to the exhaust stream to be reheated.

Thermal energy is extracted from the exhaust gas of a diesel generator set via natural conduction with a heat transfer fluid within a heat exchanger. The heat transfer fluid is then pumped from the generator set to the autoclave via an electric pump and piping. Once in the autoclave, the thermal energy is then transferred from the heat transfer oil to the liquid water within the autoclave via natural conduction within the water. The heat transfer tubing is placed directly into the water reservoir to achieve the most effective heat transfer. The thermal energy released by the heat transfer fluid boils the water, creating pressurized saturated steam, ideal sterilizing conditions for medical equipment. Once the sterilization cycle is complete, the pressurized steam is released into the atmosphere via a butterfly valve on the autoclave unit. The medical equipment is then allowed to cool and removed for use.

3 Developing a Basic Model

To accurately determine the required heat transfer rate necessary to create ideal sterilizing conditions within the autoclave, it was first important to develop a thermal model to describe how each section of the system was operating. Once the system was fully defined in terms of its fundamental properties, it would be possible to use these properties to design and model the separate components of the waste heat autoclave.

3.1 Pure Substance Model

Initially, it was decided to model the waste heat autoclave as a pure substance system. This is a reasonable assumption because the autoclave/heat transfer system, like pure substance systems, is a simple, purely thermodynamic system which does not experience significant effects from outside forces, such as gravity, magnetic fields, or shear stresses [1]. Another important characteristic of the pure substance model is that the pressure within the system is considered uniform at equilibrium. This assumption allows us to consider the pressure within the autoclave at operating temperature to be uniformly distributed throughout the vessel, rather than localized; this becomes a very relevant distinction when designing such vessels. Finally, the pure substance model best allows us to define our system because it is possible to define a stable thermodynamic state of the system based on any two independent system properties. These characteristics of the pure substance model allowed for a simple autoclave model and allow for accurate analysis of the specific states within the system.

4 Exhaust Gas Waste Heat Exchanger

Once an adequate thermodynamic model was chosen, the model was tailored to the specific autoclave system by utilizing the laws of thermodynamics and fluid mechanics. The most critical element of the waste heat autoclave is the exhaust gas waste heat exchanger. Precise design is necessary here to both extract enough thermal energy to produce steam at the necessary pressure and temperature required for sterilization, and operate efficiently within the spatial confines of a generator set exhaust system. The thermal energy required to produce steam at a pressure of 2 atmospheres can be found by analyzing the initial and final states of the water being boiled (Eq. 1).

$$\Delta Q_{cycle} = m_{water} \cdot \left(u_{final} - u_{initial}\right) \qquad \text{(Eq. 1)}$$

Here, the required energy input from the heat exchanger ΔQ_{cycle}, is equal to the difference in energies between the initial and final states of the autoclave. The initial state of the autoclave was that of the autoclave at equilibrium before being turned on, defined as pure liquid water at 1 atmosphere of pressure at 25 degrees Celsius. The final state of the autoclave was that of the autoclave at equilibrium during operating conditions, defined as saturated steam at 2 atmospheres of pressure at 121 degrees Celsius. This final state was chosen based on the temperature and pressure requirements of commercial autoclaves [2]. Once the initial and final system states were chosen, the specific energies for each state were found using steam tables.

After determining the specific energies of each state, it was necessary to determine the amount of water required to produce saturated steam at the given final state. This was done using the volume of the autoclave as well as the specific of the steam in the final state (Eq. 2).

$$m_{water} = \frac{V_{autoclave}}{v_{final}} \qquad \text{(Eq. 2)}$$

Combining Equation 1 and Equation 2, the resulting thermal energy required to operate the autoclave is given in (Eq. 3).

$$\Delta Q_{cycle} = \frac{V_{autoclave}}{v_{final}} \cdot \left(u_{final} - u_{initial}\right) \qquad \text{(Eq. 3)}$$

However, to ensure that the system can create saturated steam at 2 atmospheres within the desired operating time, it is more logical to analyze the exhaust gas waste heat exchanger in terms of a heat transfer rate. This can be accomplished by simply dividing the required thermal energy by the desired cycle time t (Eq. 4).

$$\dot{Q}_{cycle} = \frac{\Delta Q_{cycle}}{t} = \frac{V_{autoclave}}{v_{final} \cdot t} \cdot \left(u_{final} - u_{initial}\right) \qquad \text{(Eq. 4)}$$

Once the required heat transfer rate has been determined, it is necessary to determine the changes in temperature of both the exhaust and the heat transfer fluid based on the operation of the heat exchanger. This is done for two reasons; first, the required temperature change of each fluid will determine the overall dimensions of the heat exchanger. Smaller diameter piping may not transfer heat from the exhaust stream to the heat transfer fluid quickly enough, resulting in inadequate sterilization conditions within the autoclave. The second reason it is necessary to determine the temperature changes within both streams is because it is this temperature range that determines the type of working fluid to be used. Using a fluid above or below its operating temperature range could damage components of the autoclave system and eventually lead to system failure.

Before analyzing the exhaust gas waste heat exchanger further, it is necessary to discuss the assumptions made in terms of modeling. Here, the major assumption for the heat exchanger is that there are no losses experienced during the heat transfer between the exhaust stream and the heat transfer fluid. While it is impossible to perfectly insulate a system and heat loss does occur, here it is assumed to be negligible based on the physical characteristics of the system. Losses to the atmosphere are minimized by using insulation on the heat exchanger. Furthermore, the tight spacing within the generator set exhaust area does not allow for good convection or a large temperature difference between the working fluid and the surrounding air. Thus, a no-loss assumption is appropriate for this design. Using the first law of thermodynamics, the change in energy of the heat transfer fluid is given by (Eq. 5).

$$Q_{exhaust} = m_{exhaust} \cdot c_p \cdot \left(T_{in} - T_{out}\right)_{exhaust} \qquad \text{(Eq. 5)}$$

Here, $\dot{m}_{exhaust}$ represents the exhaust mass flow rate, c_p represents the specific heat of the exhaust, and T_{in} and T_{out} represent the entrance and exit temperatures of the exhaust flow, respectively. It is important to note how powerful the no-loss assumption can be in terms of designing. Here, based on the energy conservation principle, and under the assumption that the system experiences no losses, all the energy lost by the exhaust stream must be transferred to the heat transfer fluid.

However, the mass of the exhaust is not a quantity that can easily be changed to optimize the system. It is more advantageous to analyze the system in terms of a mass flow rate, which can be controlled based on the operation of the generator set; for a diesel engine under moderate load, the exhaust mass flow rate is roughly equal to 0.002 kg/s [3]. To change the exhaust mass into a mass flow rate, Eq. 5 is again divided by time to express the heat transferred between the exhaust and the heat transfer fluid as a rate (Eq. 6).

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,exhaust} \cdot (T_{in} - T_{out})_{exhaust} \qquad \text{(Eq. 6)}$$

Based on the energy conservation principle, to produce saturated steam at 2 atmospheres, heat transfer rate from the heat transfer fluid to the water must be equal to the heat transfer from the generator set exhaust stream to the heat transfer fluid. Equation 6 can now be set equal to the required heat transfer rate found for the system cycle time found in Equation 4 (Eq. 7).

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,exhaust} \cdot (T_{in} - T_{out})_{exhaust} = \dot{Q}_{cycle} \qquad \text{(Eq. 7)}$$

Rearranging Equation 7, the change in exhaust temperature required to produce the prescribed sterilization state can now be solved for (Eq. 8). Note that the temperature change has been replaced by $\Delta T_{exhaust}$ for simplicity.

$$\Delta T_{exhaust} = \frac{\dot{Q}_{cycle}}{\dot{m}_{exhaust} \cdot c_{p,exhaust}} \qquad \text{(Eq. 8)}$$

Using a similar equation, the resulting change in heat transfer fluid temperature can also be calculated (Eq. 9).

$$\Delta T_{oil} = \frac{\dot{Q}_{cycle}}{\dot{m}_{oil} \cdot c_{oil}} \qquad \text{(Eq. 9)}$$

Solving Eq. 8 and Eq. 9 yields a temperature increase of 22°C in the heat transfer fluid and a corresponding temperature drop of 5°C in the exhaust gas stream.

After finding the temperature changes for each fluid, a few assumptions are needed to allow for the inlet fluid temperature and exhaust exit temperatures to be found. After the heat transfer fluid leaves the exhaust gas waste heat exchanger, it is pumped to the autoclave to boil water. After leaving the autoclave, it is pumped back to the exhaust gas waste heat exchanger to get reheated. As a method of simplifying the analysis of the heat exchanger, it is assumed that all the thermal energy collected by the heat transfer fluid as it passes through the exhaust gas waste heat exchanger is transferred into the water in the autoclave for boiling. While this assumption is not correct due to the losses incurred during the pumping of the fluid, it is still acceptable as a starting point from which the actual heat exchanger entry temperature can be calculated. Under this assumption, the final water temperature must be equal to the temperature of the heat transfer fluid, again assuming that there are no significant system losses. Thus, for saturated steam at 2 atm, the heat transfer fluid will equal the water saturation temperature at 2 atm, or 121° C. For internal combustion engines, such as those in work trucks or generation sets on which the waste heat autoclave could be used, the average exiting exhaust temperature can get up to 600° C. As this is the temperature before any cooling effects, it can be used as the entry temperature for the exhaust gas stream.

4.1 Heat Exchanger Physical Dimensions

Once the necessary thermodynamic characteristics for the exhaust gas waste heat exchanger have been found, it was necessary to determine what type of heat exchanger setup should be used. For the exhaust pipe waste heat exchanger, a concentric tube heat exchanger design was chosen because of the relative simplicity involved in its installation and design and the heat transfer characteristics of this design. In terms of simplicity, a parallel tube heat exchanger that was wound around the exhaust pipe would have been simpler to install because it would not require extra welding or heat transfer fluid contacting the exhaust pipe. However, the effective heat transfer area between the exhaust pipe and the heat exchanger is much too small for the required temperature increase of the fluid. By using a concentric tube design, the effective heat transfer area is drastically increased, allowing for a shorter required length for the heat exchanger itself. Essentially, a larger cylindrical shell can be slid over and welded to the exhaust pipe, negating the need for multiple passes or more complicated designs.

After deciding upon a concentric tube heat exchanger platform, it was necessary to determine whether a counter-flow or parallel-flow setup would be more effective for this application. To relate the thermodynamic characteristics of the heat exchanger to the physical dimensions of the actual system, the log mean temperature difference (LMTD) equation is used (Eq. 10).

$$\dot{Q}_{cycle} = U \cdot A \cdot \Delta T_{LM} \quad \text{(Eq. 10)}$$

Here, ΔT_{LM} refers to the log mean temperature difference for a given heat exchanger setup, U represents total heat transfer coefficient for the system, and A represents the overall area of the heat exchanger. To determine whether a counter-flow or parallel-flow design should be used, the log mean temperature differences for a counter-flow heat exchanger (Eq. 11) and a parallel flow heat exchanger (Eq. 12) are compared.

$$\Delta T_{LM,CF} = \frac{\left(T_{exhaust,in} - T_{oil,out}\right) - \left(T_{exhaust,out} - T_{oil,in}\right)}{\ln\left(\frac{\left(T_{exhaust,in} - T_{oil,out}\right)}{\left(T_{exhaust,out} - T_{oil,in}\right)}\right)} \quad \text{(Eq. 11)}$$

$$\Delta T_{LM,PF} = \frac{\left(T_{exhaust,in} - T_{oil,in}\right) - \left(T_{exhaust,out} - T_{oil,out}\right)}{\ln\left(\frac{\left(T_{exhaust,in} - T_{oil,in}\right)}{\left(T_{exhaust,out} - T_{oil,out}\right)}\right)} \quad \text{(Eq. 12)}$$

Analyzing Eq. 11 and Eq. 12, for the same inlet and exit temperatures, the log mean temperature for the counter-flow heat exchanger is greater than that of the parallel-flow heat exchanger. From Eq. 10, it can be see that for a higher log mean temperature difference, a smaller effective surface area is required to achieve a prescribed heat transfer rate. Thus a counter-flow, concentric tube heat exchanger is ideal to accommodate for the tight space constraints associated with the exhaust system of either a truck engine or a generator set

Once a heat exchanger setup was determined, it was now possible to calculate the physical dimensions of the system. Rearranging Eq. 10, the LMTD equation can now be used to solve for the effective area of the heat exchanger (Eq. 13).

$$A = \frac{\dot{Q}_{cycle}}{U \cdot \Delta T_{LM}} \quad \text{(Eq. 13)}$$

Here, the effective area A is a function of the overall heat transfer rate q, the log mean temperature difference ΔT_{LM}, and the overall heat transfer coefficient for the heat exchanger U. To solve for U, it is first necessary to solve for the individual heat transfer coefficients for each section of the heat exchanger. Starting from inside the heat exchanger, the thermal conductivity of the exhaust flow can be calculated by analyzing the thermodynamics of the stream. First, a Reynolds number is calculated for the exhaust stream to determine if the exhaust flow is laminar or turbulent (Eq. 14).

$$\mathrm{Re} = \frac{4 \cdot \dot{m}}{\pi \cdot D_h \cdot \mu} \qquad \text{(Eq. 14)}$$

Based on the exhaust pipe diameter for commercial trucks, the exhaust pipe for the waste heat exchanger is modeled as being a 2-in diameter circular pipe of negligible thickness (D_h=0.0508 m). Here, D_h represents the hydraulic diameter of the pipe and μ represents the viscosity of the exhaust flow. Again using an exhaust mass flow rate $\dot{m}$ of 0.002 kg/s and modeling the exhaust gas as air, the Reynolds number is solved for using Eq. 15. This yields an exhaust gas flow Reynolds number of 12,065, indicating that the flow is turbulent. However, this turbulence is to be expected due to the high mass flow rate and low viscosity of air.

Next, the heat transfer coefficient for the exhaust stream can be calculated using the Nusselt number Nu and the thermal conductivity of air k (Eq. 15).

$$h = \frac{Nu \cdot k}{D_h} \qquad \text{(Eq. 15)}$$

Here, the Nusselt number used corresponds to the correlation for turbulent flow in a smooth pipe, given by (Eq. 16):

$$Nu = \frac{\frac{f}{8} \cdot (\mathrm{Re} - 1000) \cdot \mathrm{Pr}}{1 + 12.7 \cdot \sqrt{\frac{f}{8}} \cdot \left(\mathrm{Pr}^{\frac{2}{3}} - 1 \right)} \qquad \text{(Eq. 16)}$$

The pipe friction factor f and the Prandlt number Pr can be calculated using Eq. 17 and Eq. 18, respectively.

$$f = (0.79 \cdot \ln(\mathrm{Re}) - 1.64)^{-2} \qquad \text{(Eq. 17)}$$

$$\mathrm{Pr} = \frac{\mu \cdot c_p}{k} \qquad \text{(Eq. 18)}$$

Here, the Prandlt number is a function of properties of the fluid, namely the viscosity μ, the specific heat capacity c, and the thermal conductivity k. The heat transfer coefficient for the exhaust gas stream is calculated by inserting Eq. 16 and Eq. 17 into Eq. 15.

The next heat transfer coefficient to be calculated is that of the wall of the exhaust pipe. However, due to the relatively small thickness of the tubing compared to the overall size of the heat exchanger, the wall heat transfer coefficient can be neglected.

Assuming no heat loss to the atmosphere, the final heat transfer coefficient for the exhaust gas waste heat exchanger is that of the heat transfer fluid flowing through the outer pipe. The heat transfer coefficient here is calculated in the same manner as that of the exhaust gas heat transfer coefficient. Initially, the Reynolds number is recalculated using the mass flow rate and thermodynamic properties of the fluid. However, since the heat transfer fluid is flowing through an effective annulus, the characteristic diameter changes from the diameter of the pipe to the difference in diameters of the inner and outer pipes (Eq. 19).

$$D_h = D_{out} - D_{in} \qquad \text{(Eq. 19)}$$

As can be seen from Eq. 14, a smaller hydraulic diameter leads to a higher Reynolds number, and as a result, a higher heat transfer coefficient. To maximize the heat transfer between the exhaust pipe and the fluid, and to accommodate the tight space constraints, an outer pipe diameter of 3 inches was chosen. Solving Eq. 14 using the thermodynamic properties of fluid and the new hydraulic diameter it was determined that the heat transfer fluid flow was laminar. To increase heat transfer, baffles were added along the inner walls of the heat exchanger to increase the Reynolds number.

As before, the heat transfer coefficient can be solved for using Eq. 15, but replacing k with the thermal conductivity of the heat transfer fluid, and using the annulus hydraulic diameter. To recalculate the Nusselt number for the heat transfer fluid flow, the correlation for turbulent flow through an annulus is used (Eq. 20).

$$Nu = \frac{\frac{f}{8}\cdot(\mathrm{Re}-1000)\cdot \mathrm{Pr}}{1+12.7\cdot\sqrt{\frac{f}{8}}\cdot\left(\mathrm{Pr}^{\frac{2}{3}}-1\right)}\cdot\left[0.86\cdot\left(\frac{D_i}{D_o}\right)^{-0.16}\right] \qquad \text{(Eq. 20)}$$

Solving Eq. 20 and substituting into Eq. 15 yields a heat transfer coefficient for the heat transfer fluid annulus.

To solve for U, the heat transfer coefficients from each section of the heat exchanger are summed. The value of U is then inserted into Eq. 13 and the overall effective heat transfer area was calculated for $q = \dot{Q}_{cycle}$. For an exhaust pipe of 2-in outer diameter, this translates into a required heat exchanger length of 0.59 m.

This analysis represents a basic design for the exhaust gas waste heat exchanger. However, there are further design steps which can be taken to minimize the required length of the heat exchanger. One such design step, the addition of baffles within the heat transfer fluid annulus, was utilized in the final design of the waste heat autoclave. Baffles are essentially walls placed in the fluid stream to create turbulence. By increasing the turbulence of the heat transfer fluid, the heat transfer coefficient of the fluid increases, yielding a higher heat transfer rate and smaller required effective area. Baffles could also be placed in the exhaust pipe of the generation set to increase turbulence; however, the baffles would also increase the back pressure experienced within the exhaust pipe, leading to deteriorated engine performance. Baffles within the heat transfer fluid would have this same effect, increasing the required pump head necessary to pump the fluid. The higher pump head would require more electricity to run, thus defeating the purpose of the waste heat autoclave.

Increasing the mass flow rate of the heat transfer fluid would also increase the fluid turbulence and increase the heat transfer coefficient. These effects would be doubled due to the increase in the heat transfer coefficient within the autoclave as well. However, as with the baffles, any constraints or physical changes to the fluid would require a change in the pump operation and thus a change in the required electricity.

5 Autoclave Heat Exchanger

The autoclave heat exchanger is responsible for using the thermal energy from the heat transfer fluid to boil liquid water into pressurized steam. As the heat transfer fluid tubing passes through the autoclave, thermal energy from the heat transfer fluid is used to heat the water. However, once the water reaches its boiling temperature, the heat transferred into the water will be used to initiate a phase change, turning the water into two-phase steam. As this process continues, the steam generated is trapped within the sealed autoclave. The steam produced is unable to exit the sealed volume of the autoclave, thus causing an increase in the internal pressure of the autoclave. This increase in pressure also results in an increase in the boiling temperature of the water, thus raising the internal temperature of the autoclave. This process continues until all the liquid water has been changed into steam and the system is in a saturated vapor state. This final state was chosen because it is the optimal environment for sterilization of most common bacteria. The steam remains sealed in the autoclave for the full cycle time, after which it is vented to the atmosphere, allowing the sterilized instruments to cool.

In Section 4.1, Eq. 4 was used to calculate the total required heat transfer rate to boil water and set the sterilization conditions. However, to effectively extract the thermal energy from the heat transfer fluid, it is necessary to design a heat exchanger to work within the autoclave. To maximize the effective heat transfer area of the heat exchanger while conforming to the size constraints of the autoclave, a multi-pass tube heat exchanger was chosen. This design takes up minimal space along the bottom inside the autoclave, while also providing a large heat transfer area for effective boiling.

To calculate the total necessary tube length required to release all the thermal energy from the heat transfer fluid into the water within the autoclave, Eq. 11 is solved for the thermodynamics of the system. First, to determine the Nusselt number for the heat exchanger, the system is modeled as natural convection over a horizontal cylinder. This is an appropriate model under the condition that the fluid flow through the pipe is assumed to have no thermal transients in the radial direction. However, based on the mass flow rate of the fluid, as well as the high thermal conductivity of the copper piping and heat transfer coefficient of the heat transfer fluid, this is a valid assumption. For this model, the Nusselt number and be calculated based on the appropriate correlation (Eq. 23).

$$Nu = 0.36 + \frac{0.518 \cdot Ra^{\frac{1}{4}}}{\left[1 + \left(\frac{0.559}{\Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \qquad \text{(Eq. 23)}$$

It is interesting to note here that for the proposed model of the autoclave heat exchanger, the heat transfer rate is not dependent on any fluid dynamics properties, such as the Reynolds number or the mass flow rate. The heat transfer rate is based mainly on the Rayleigh number Ra and the Prandlt number Pr, both of which are dependent only on the thermodynamics of the system; this is due to the assumption made earlier in regards to thermal transients within the pipe flow. Here, the Rayleigh number can be defined as (Eq. 24):

$$Ra = \frac{g \cdot \beta \cdot \Delta T \cdot D^3}{\alpha \cdot v} \qquad \text{(Eq. 24)}$$

where g is the gravitational constant, β is the coefficient of thermal expansion of the water in the autoclave, ΔT is the temperature difference between the bulk temperature of the water and the log mean temperature difference of the heat transfer fluid flow, D is the outer diameter of the heat exchanger tubing, and α and v are the thermal diffusivity and kinematic viscosity of water, respectively. Inserting Equation 24 and 23 into Equation 15 and solving for the overall heat transfer coefficient of the autoclave heat exchanger, it is now possible to find the required heat exchanger surface area required to transfer all of the thermal energy from the heat transfer fluid to the water. Using Newton's Law of Cooling, the effective area of the autoclave heat exchanger can be given by (Eq. 25).

$$A = \frac{\dot{Q}_{cycle}}{h \cdot (\Delta T)} \qquad \text{(Eq. 25)}$$

Under the assumption that all of the thermal energy carried in the fluid is transferred to the water, the final temperature of the heat transfer fluid can be assumed to be equal to the final temperature of the saturated steam within the autoclave. Although this is an assumption, it is safe to make because it actually over-predicts the required heat exchanger length necessary to heat and pressurize the water.

After finding the required length for the exhaust gas waste heat exchanger based on the necessary thermodynamic properties of the autoclave, it was necessary to analyze the durability and safety of the heat exchanger. The major safety concern with a high heat transfer rate such as that required for the autoclave is the chance of material failure and meltdown. Such an incident is most likely to occur if the system experiences a heat flux higher than the critical heat flux for the system. At this point, all fluid around the heat source (in the case of the autoclave, the exhaust pipe) is instantly vaporized and the system experiences film boiling. However, since there is no longer fluid in contact with the heat source, the heat transfer coefficient drops drastically from that of the liquid to that of the vapor. For such a relatively low heat transfer coefficient, the heat source cannot conduct heat away quickly enough, resulting in a sharp increase in the surface temperature of the heat source. Here, the temperature can quickly rise above the melting temperature of the heat source material itself, leading to mechanical failure of the system.

To ensure the safety and longevity of the system, a design safety factor limiting the heat transfer rate of the autoclave heat exchanger to 80% of the critical heat flux must be maintained. This safety factor was chosen to maximize the heat transfer from the exhaust flow to the heat transfer fluid, while maintaining a significant heat transfer buffer to ensure that the system does not approach the critical heat flux (Eq. 26).

$$q_{critical} = K \cdot h_{fg} \cdot \left[\sigma \cdot g \cdot \rho_g^{\ 2} \cdot \left(\rho_f - \rho_g\right)\right]^{\frac{1}{4}} \qquad \text{(Eq. 26)}$$

Here, h_{fg} represents the latent heat of vaporization of water, σ represents the surface tension of water, and ρ_f and ρ_g represent the densities of water in liquid and vapor states, respectively. To solve for the heat exchanger geometric constant K, a critical heat flux correlation must be used. However, to determine which correlation is correct for this system, it is necessary to first determine the characteristic radius of the heat exchanger (Eq. 27):

$$R^* = \frac{R}{L_c} \qquad \text{(Eq. 27)}$$

where L_c represents the characteristic length of the heat exchanger (Eq. 28).

$$L_c = \left[\frac{\sigma}{g \cdot \left(\rho_f - \rho_g\right)}\right]^{\frac{1}{2}} \qquad \text{(Eq. 28)}$$

Combining Equations 27 and 28 yields a characteristic radius of 16.1. For this value, the correlation for a small horizontal cylinder can be used (Eq. 29).

$$K = \frac{0.123}{\left(R^*\right)^{\frac{1}{4}}} \qquad \text{(Eq. 29)}$$

After solving for K, the critical heat flux is then determined using Equation 26. However, from a design standpoint, determining the minimum allowable heat exchanger length for the autoclave system is much more useful than the critical heat flux. The minimum heat exchanger length based on the critical heat flux and the 80% safety factor is determined using (Eq. 30).

$$L_{critical} = \frac{\dot{Q}_{cycle}}{0.8 \cdot q_{critical} \cdot \pi \cdot 2R} \quad \text{(Eq. 30)}$$

Solving Eq. 30 yields a minimum autoclave heat exchanger length of 0.87 m, roughly on par with the calculated heat exchanger length of 0.91 m. By modeling the system such that the maximum heat transfer rate sustained during operation is less than the critical heat transfer rate, the system retains durability and effective use while minimizing the risk of failure.

6 Autoclave Pressure Vessel

To reach the effective sterilization conditions, a pressure vessel is necessary to both contain the pressurized steam and prevent thermal leaking into the atmosphere. However, it is also necessary that the pressure vessel be cheap enough and relatively common enough so that maintenance, cost, and component availability do not outweigh the usefulness of the autoclave.

Initially, it was decided that industrial pressure vessels would be used. These are often welded steel vessels capable of high internal temperatures and pressures. However, such vessels are often expensive due to the materials and engineering involved in the construction. Furthermore, these vessels usually have large internal volumes and are very heavy and obtrusive. In order for the exhaust gas waste heat exchanger to be a viable alternative to the electric autoclaves currently employed in field hospitals, the pressure vessel used in the design had to be consistent with the size and weight of current autoclaves.

The final design idea came from home document safes. Traditional home lock boxes are made to be fireproof and waterproof to protect valuable documents during home fires. These safes are made with thick insulated walls to prevent the heat from fires from damaging computer disks and sensitive paper documents. However, the reverse of this situation is perfect in designing the autoclave; the thick insulated walls help prevent parasitic heat leak from the autoclave into the surrounding atmosphere, and the interior coating of the safe is able to stand up to the high temperatures of pressurized steam while also preventing condensation. Furthermore, fireproof document safes are relatively cheap in cost when compared to larger industrial autoclaves and pressure vessels.

Further conversations with safe manufacturers as well as materials research showed that a traditional fireproof document safe would be able to serve as an adequate autoclave pressure vessel after some alteration. The major problem with using the fireproof safe is that although they are designed to completely isolate the contents within the safe from the environment in the case of a fire or flood, the vessels themselves are not completely sealed. To make the safes usable as sealed pressure vessels, it is necessary to apply gaskets to the doorframes of the safe to create a completely sealed environment that can be pressurized.

6.1 Autoclave Sealing Gaskets

Ensuring a proper seal during autoclave operation is necessary to create proper sterilization conditions within the autoclave. For the design of the autoclave system, it was necessary for the seals used to be able to withstand pressure differences of 1 to 2 atmospheres and operating temperatures from 100 to 200 degrees Celsius in order for the system to operate reliably. Furthermore, the seals must have a very low coefficient of thermal expansion; essentially, for the gaskets to work correctly, they cannot deform due to fluctuations in temperature. Ideally, some expansion would occur as the system temperature increased to create a tighter seal at higher temperature/pressure, but too much expansion could cause the gaskets to fail or for the seal to become weak.

After conducting research into a material for the gaskets, expanded-polytetrafluoroethylene (PTFE) was chosen based on the operating requirements and thermodynamic modeling of the autoclave (Appendix A). Expanded-PTFE is a chemically-inert sealant material typically used in gaskets for flanges. It has an operating temperature range between -240 °C and 315 °C and can maintain a seal up to 3000 psi [3]. In terms of accessibility, PTFE gaskets are an industry standard and are widely used. Gaskets made of PTFE are also water and steam resistant, non-toxic and FDA compliant, making them an ideal choice for use in sealing the autoclave.

7 Heat Transfer Fluid Pump/Piping System

The fluid transfer system is the next critical component in the design of the exhaust gas waste heat autoclave. This component is responsible for transferring the hot fluid from the heat exchanger to the autoclave. This is done through the two primary components of the system: the heat transfer fluid pump and the fluid transfer piping. While both elements do not require actual designing, it is necessary to perform analysis on the overall system to determine what specifications each component is required to have. In this section, both the fluid pump and the piping are analyzed based on the mass flow rate and required entry temperature for the autoclave heat exchanger.

7.1 Heat Transfer Fluid Pump

To design a heat fluid transfer system to move the heat transfer fluid most effectively and efficiently, it was necessary to determine the correct type of pump to use for this given application. In terms of fluid transfer, there are many different pump layouts available on today's market. However, for this application, two specific pump types were analyzed: gear pumps and centrifugal pumps. Both pump types were chosen based on the characteristics of the flow and pressure head created by the pumps, as well as their ease of installation in a system such as the waste heat autoclave.

7.1.1 Gear Pumps

Gear pumps transfer fluid by using two or more internal gears to create a vacuum pressure that drives the fluid forward. As the gear teeth make contact, the fluid load is pushed forward and more fluid is drawn into the gears (Fig. 1).

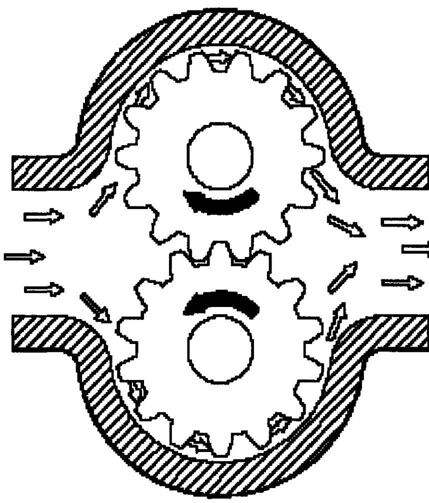

Fig. 1: In a gear pump, two tightly interlock gears create a suction head by transferring fluid via the space between gear teeth. The tight tolerance of the gears also prevents fluid backflow within the pump.

Here, tight tolerances between the driving and free-wheeling gears within the pump serve to prevent fluid from traveling backwards through the system. Due to the constant rotation of the gearing as well as the precise tolerances between the internal gears, gear pumps are capable of delivering relatively continuous, non-pulsating flow at high pressures and flow rates [4]. This characteristic is very important in a heat transfer fluid application such as this autoclave system because the thermodynamics of the system rely on the mass flow rate of the system.

Gear pumps are also advantageous because many designs have "dry running" capabilities, meaning that if the fluid supply is cut off, the pump can continue to operate undamaged. This is an important safety mechanism in terms of leaks in the system. Given a leak in the heat transfer fluid piping, a dry-running gear pump would not experience catastrophic failure, contributing to the overall robustness of this autoclave system.

7.1.2 Centrifugal Pumps

Centrifugal pumps operate by using rotating vanes and stationary stators to accelerate and pressurize the fluid through the pump. As the fluid contacts the rotating vanes, it is imparted with the kinetic energy of the vanes, increasing the fluid velocity. The stationary stators force the fluid to discharge, changing velocity into pressure (Fig. 2).

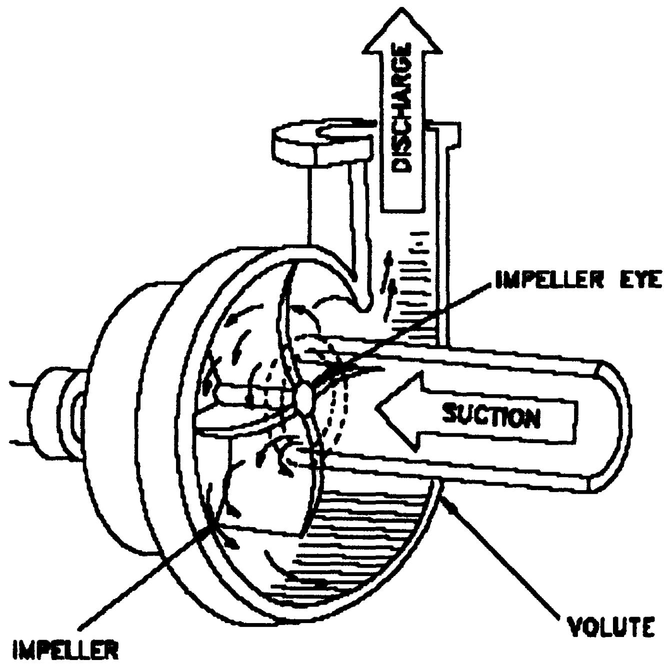

Fig. 2: In a centrifugal pump, a rotating impeller creates a vacuum to draw in fluid and increases the energy within the working fluid. The fast-moving fluid is channeled toward the discharge by the stationary walls of the pump, thus increasing the pressure of the fluid.

This conversion is accomplished by maintaining a tight tolerance between the stators and the vanes. Pressure is further increased due to the gradual increase in the width of the vane blades. The blades gradually widen from their center to the edges, thus decreasing the rotational velocity and increasing the discharge pressure. This also allows the centrifugal pumps, like the gear pumps, to produce very continuous, non-pulsating flow throughout the piping [5]. As with the gear pump, this is important in maintaining the thermodynamic conditions for which the system is designed.

While both pumps are more than capable of working in this system, a centrifugal-type pump was chosen for the autoclave system due to its ability to provide a constant velocity flow and because of its ability to handle fluids of varying viscosities. Gear pumps, while able to handle fluids of varying viscosities, can experience diminished performance based on buildup or interference between the working fluid (the heat transfer fluid) and the internal mechanics of the pump. The heat transfer fluid has a viscosity that is temperature dependent, meaning that the fluid is much more viscous during a cold system startup than during optimal operating temperatures. The ability to efficiently pump the fluid throughout the autoclave system despite its viscosity makes the centrifugal pump the logical choice for this application.

After choosing a pump configuration for the fluid transfer system, it is necessary to specify the flow characteristics of the pump, specifically the required pressure head of the pump to ensure a proper flow rate through the autoclave setup. The required pressure head was calculated using head loss equation for viscous pipe flow (Eq. 31).

$$\Delta P = \rho_{oil} \cdot \frac{v_{average}^{2}}{2}\left(f \cdot \frac{L}{D} + \Sigma K\right) \qquad \text{(Eq. 31)}$$

Here, the pressure drop experienced throughout the piping is a function of the heat transfer fluid properties, such as density ρ_{oil} and average flow velocity $v_{average}$, as well as properties of the piping, such as the friction factor for the pipe wall *f*, the total length of the piping *L*, and the hydraulic diameter of the piping *D*. Additional losses due to pipe bends and valves within the system are defined within the system by the sum of their respective loss coefficients *K*; however, these losses can be assumed negligible due to their minimal effect on the overall pressure drop experienced across the system. From Eq. 31, a pressure head of 3 psi is required to pump the heat transfer fluid though the piping system.

Losses due to fouling and its effects on the pump operation and pipe flow characteristics were also deemed negligible, since the heat transfer fluid component of the autoclave setup is a closed-loop system. If special care is taken to thoroughly clean and adequately seal the pump and fluid system during installation, losses and degraded system performance based on fouling should be nonexistent.

7.2 Heat Transfer Working Fluid

To efficiently transfer thermal energy from the exhaust gas stream to the autoclave, it is necessary to choose a working fluid with the right characteristics to best operate in this system. The most important characteristic is fluid operating temperature. The heat transfer fluid is required to operate at very high temperatures to effectively transfer enough thermal energy to create sterile conditions within the autoclave. As it was seen in Section 5, the heat transfer fluid must be able to operate between approximately 25°C and 140°C without breaking down or boiling to maintain a good heat transfer rate.

Another important characteristic of heat transfer fluid is its heat transfer coefficient. To remove heat from such a small heat exchanger such as that designed for the exhaust tube, a high heat transfer coefficient is necessary. A high heat transfer coefficient also assures that the heat collected from the exhaust gas is quickly transferred to the water in the autoclave to produce effective boiling.

The mechanical properties of the fluid are equally important in determining the proper working fluid to use as they heavily affect the flow characteristics through the autoclave piping system. As the working fluid heats up, it becomes less viscous within the piping. This results in a continuous decrease in the required pump pressure head as the system heats up. In terms of efficiency, a working fluid with a lower viscosity at both system startup and under operating conditions is preferred.

Based on these operating conditions, it was chosen to model the system using Duratherm S heat transfer fluid due to its high thermal conductivity and specific heat capacity, as well as its low viscosity and large operational temperature range (Appendix B). Duratherm S is a silicone based heat transfer fluid with an operational temperature range of nearly 400°C [6]. This is ideal for the waste heat autoclave due to the large temperature range the system experiences between startup and operation. Duratherm S is also non-toxic and oxidation-resistant, meaning that it will not foul the piping if exposed to air. This is a critical characteristic from a maintenance and robustness standpoint, allowing the system to be taken apart for maintenance without risking damaging or fouling the fluid transfer piping or pump.

7.3 Heat Transfer Piping

The piping system is a critical component of the entire autoclave setup not only because it is responsible for transferring the heat transfer fluid from the exhaust gas waste heat exchanger to the autoclave, but because it represents the section of the system subject to the most thermal energy lost. As soon as the fluid leaves the high-temperature exhaust pipe, it immediately starts conducting thermal energy through the pipe walls and releasing it into the atmosphere. To prevent as much of this parasitic heat loss as possible, adequate insulation must be used.

Initially, traditional polyethylene foam insulation was chosen based on its good insulation qualities as well as its general availability as a very common insulation material. However, polyethylene is soft, open-cell foam, making it very weak in terms of durability. For settings such as field hospitals in which the waste heat autoclave would be used, a much tougher, more durable insulation material is required to withstand the abuses of weather and daily use. Furthermore, it is necessary for the foam insulation to maintain the good insulation qualities of regular polyethylene insulation.

Based on the required system parameters, Micro-Lok fiberglass resin insulation was chosen for insulating the fluid transfer piping. Micro-Lok is made from glass fibers bonded in a thermosetting resin, resulting in a semi-rigid, highly durable insulation. Furthermore, the fiberglass-resin construction and vapor retarded jacketing makes the insulation is ideal for both concealed and exposed piping with operating temperatures up to 454 °C. To fully insulate the fluid transfer piping to eliminate any thermal energy loss, 2-inch thick insulation tubes were chosen for all exposed piping.

For the heat exchanger and fluid transfer elements of the autoclave design, it was necessary to use piping with a high thermal conductivity and relative hardness to allow for efficient heat transfer and durability. Here, the most common choice is copper; copper is commonly used in most plumbing and heating systems due to its high thermal conductivity. This makes it a perfect choice for the heat exchanger elements of the autoclave system. However, for the fluid transfer pipes that run between the autoclave and the generator set, piping with a low thermal conductivity and relatively high strength is a better choice to help limit thermal losses and maintain a durable system. Although copper is not the best choice to use, the system uses copper fluid transfer pipes because it is much easier to construct the closed-loop system out of one material. Furthermore, the thick fiberglass insulation effectively prevents heat loss while also serving as a protective strengthening layer for the pipe system. Here, 0.375-in type L hard copper piping was used due to its higher durability and strength. The pipe diameter was chosen based on the characteristics of the autoclave heat exchanger.

8 Discussion

Overall, the exhaust gas waste heat autoclave is designed to provide proper sterilization for remote locations both in need of medical equipment and with scarce electricity resources. To effectively accomplish this functional requirement, and for the exhaust gas waste heat exchanger to be a marketable, usable product, certain design criteria must be met. First, as this system is intended to replace traditional electronic autoclaves in forward deployed military clinics and field hospitals, it is necessary that the waste heat autoclave be able to operate independent of electricity. Aside from the electricity required to power the fluid pump, the waste heat autoclave is entirely powered by thermal energy from the waste heat source. By using a minimal amount of electricity, the waste heat autoclave allows more electricity to be saved for other critical processes.

The waste heat autoclave must also be cost-effective to serve as a viable alternative to electronic autoclaves. Essentially, this means that the waste heat autoclave must be no more expensive than an equivalently-operating counterpart both in initial cost and maintenance costs. The waste heat autoclave is made from "off-the-shelf" components that can be easily installed without professional help or intensive maintenance (Table 1). The waste heat autoclave also costs roughly ten times less than a standard electric autoclave while maintaining nearly all the same performance characteristics (Table 2).

Autoclave Price Comparison			
Waste Heat Autoclave		Electric Autoclave	
Component	Price	Component	Price
Sentry DA5781 Fireproof Safe [7]	$531.12	Systec D-65 Front-Load Autoclave; 380V	$19,875.00
ASTM B75 Copper Tubing (32 m) [8]	$434.56		
Grainger 1P795 Centrifugal Pump [9]	$185.00		
Duratherm S Heat Transfer Fluid (5gal) [10]	$175.00		
Micro-Lok Fiberglass Insulation (32 m)	$211.97		
Total Cost	**$1,537.65**	**Total Cost**	**$19,875.00**

Table 1: The exhaust gas waste heat exchanger is made from standard industrial components for a cost over ten times lower than a traditional electronic autoclave.

Autoclave Performance Comparison		
Waste Heat Autoclave	System Characteristic	Systec D-65 Autoclave [11]
115 V	Electrical Requirement	380 V
93.4 kg	Autoclave Unit Weight	88.5 kg
60H x 47W x 49D (cm)	Exterior Dimensions	63H x 75W x 77D (cm)
35-minute cycle, 10 minute cooldown	Sterilization Cycle Time	30-minute cycle, 10 minute cooldown
None	Temperature Control	Dedicated microprocessor control/thermal sensors
50 L	Sterilization Volume	56 L
$1,537.65 (est.)	Price	$19,875.00

Table 2: The exhaust gas waste heat autoclave is designed to maintain similar performance characteristics to a traditional electronic autoclave.

Although the waste heat autoclave does have performance characteristics similar to a traditional electric autoclave, it is important to also analyze the performance limits of the waste heat autoclave. The most notable feature lacking from the waste heat autoclave is the temperature controls. The traditional electric autoclave features specific software and sensors to accurately measure and control the internal temperature and pressure of the autoclave. Controls such as these are necessary for ensuring proper operating conditions during sterilization. However, the waste heat autoclave has been designed such that its operating temperature and pressure are those necessary for most sterilization processes. By eliminating costly electronics, the system consumes less electricity while still being operationally effective.

The waste heat autoclave also relies on a constant heat source; the autoclave can only be operated while the diesel generator set is running. However, many field hospitals run on commercially available electricity, meaning that the generator set is only in operation when power goes down. This makes the waste heat autoclave useful only for very remote hospitals where diesel generators are the primary source of electricity. While this may lead to the waste heat autoclave providing little profit by only appealing to a niche market, the waste heat system provides reliable sterilization for markets where electricity is a scarce commodity.

As it can be seen from the calculations and modeling done for each component of the waste heat autoclave, each element of the system was specifically over-designed to allow for the losses and assumptions made during the design process. While this would not prove to be cost effective in terms of designing and manufacturing for profit, by over-designing the autoclave, the system is made more robust and less prone to failure. Over-designing the system also takes into account externalities and random situations that could impact the system during field use to further protect the system during operation

9 References

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Appendix A – MATLAB Model

```
% Exhaust Gas Waste Heat Autoclave Design - MATLAB Model

% Heat oil properties
c_oil = 1772; % specific heat capacity of oil
m_oil = .003; % mass flow rate of heat transfer oil
k_oil = 0.120; % thermal conductivity of heat transfer oil
rho_oil = 935; % density of heat transfer oil
mu_oil = 0.03 % viscosity of oil

% Exhaust properties
m_ex = 0.02; % mass flow rate of exhaust
c_ex = 1128.1; % specific heat capacity of exhaust (@ 660 C) (J/kgK)
k_ex = 0.06501; % thermal conductivity of exhaust (@ 660 C) (W/mK)
mu_ex = 4.1549*10^-5; % viscosity of exhaust (@ 660 C) (kg/m sec)

% Water properties
k_w = 0.3129; % thermal conductivity of water (W/mK)
mu_w = 5.073*10^(-4); % viscosity of water (kg/m sec)
c_w = 4196; % specific heat capacity of water (J/kgK)
B_w = 5.322*10^(-4); % expansion coefficient of water (1/K)
a_w = 1.5*10^(-7); % thermal diffusivity of water (m^2/s)

% System Physical Characteristics
r_heat = 1.5*2.54/100; % radius of counterflow heat exchanger shell
r_oil = 0.1875*2.54/100; % radius of oil piping
r_ex = 1*2.54/100; % radius of exhaust pipe
d_heat = 2*r_heat; % diameter of heat exchanger pipe
d_oil = 2*r_oil; % diameter of heat transfer oil piping
d_ex = 2*r_ex; % diameter of exhaust pipe
k_p = 400; % thermal conductivity of oil piping (copper, W/mK)

% System Thermodynamic Characteristics
uf1 = 83.906*10^3; % specific energy of liquid at initial startup (P = 101.3kPa, T =
20 C)
ug2 = 2529.1*10^3; % specific energy of vapor at final state (saturated vapor @ P =
202.6kPa)
vg2 = 0.8857; % specific volume of vapor at final state
hfg = 2201.5; % heat of vaporization of water @ saturation @ 202.6kPa

s = 0.05497; % surface tension of water @ saturation @ 202.6kPa
V = 50*0.001; % volume of autoclave (m^3)
t = 1200; % desired cycle run-time (s)
m_w = V/vg2; % total mass of water required for autoclave cycle
rho_g = 1/vg2; % density of vapor (kg/m^3)
rho_f = 1000; % density of liquid (kg/m^3)
g = 9.81; % gravity (m/s^3)

% AUTOCLAVE HEAT EXCHANGER
% Find total heat flux required for autoclave steam cycle
Q_boil = (m_w * (ug2 - uf1)) / t;

% Determine minimum allowable heat transfer area for heat exchanger and boiler
L_crit = (s/(g*(rho_f-rho_g)))^(0.5); % characteristic length of heat exchanger
R_crit = r_heat/L_crit;
K = 0.123/(R_crit)^(0.25); % critical heat flux constant (based on geometry)
q_crit = K*hfg*(s*g*rho_g^2*(rho_f-rho_g))^(0.25); % critical heat flux
L_min = Q_boil/(0.8*q_crit*pi*d_heat)
```

```
% EXHAUST HEAT EXCHANGER
% Thermal fluid characteristics for heat exchanger
T_exin = 650+273; % initial exhaust temperature (K)
T_oilin = 393; % initial oil temperature (K)
T_oilout = Q_boil/(m_oil*c_oil) + T_oilin; % exiting oil temperature (K)
T_exout = T_exin - Q_boil/(m_ex*c_ex); % exiting exhaust temperature (K)

% Determine length for counterflow heat exchanger based on heat transfer coefficients
Re_ex = 4*m_ex/(pi*mu_ex*d_ex);
Re_heat = 4*m_oil/(pi*mu_oil*(d_heat-d_ex))+1000;
f_ex = (0.790*log(Re_ex)-1.64)^(-2);
Pr_ex = mu_ex*c_ex/k_ex;
Nu_ex = ((f_ex/8)*(Re_ex-1000)*Pr_ex)/(1+12.7*(f_ex/8)^(0.5)*((Pr_ex^2/3)-1));
h_ex = Nu_ex*k_ex/(d_ex);
f_heat = (0.790*log(Re_heat)-1.64)^(-2);
Pr_oil = mu_oil*c_oil/k_oil;
Nu_heat = (((f_heat/8)*(Re_heat-1000)*Pr_oil)/(1+12.7*(f_heat/8)^(0.5)*((Pr_oil^2/3)-
1)))*(0.86*(d_ex/d_heat)^(-0.16));
h_heat = Nu_heat*k_oil/(d_heat-d_oil);
U = h_heat + h_ex;

% LMTD analysis for exhaust pipe waste heat exchanger
dT_ex = T_exin-T_exout;
dT_oil = T_oilout-T_oilin;
TLM_ex = (dT_ex-dT_oil)/log(dT_ex/dT_oil);
L_heat = Q_boil/((pi*d_ex)*U*TLM_ex)

% AUTOCLAVE HEAT EXCHANGER
% LMTD analysis for autoclave heat exchanger
T_w = 120; % ambient steam temperature in autoclave
T_o = 120; % exit temperature of heat transfer oil from autoclave
T_in = T_oilout; % entry temperature of heat transfer oil into autoclave
TLM_oil = (T_in - T_o)/log(T_in/T_o);
dT_w = TLM_oil - T_w;

% Determine length for finned shell-and-tube heat exchanger based on
% thermal resistances
Re_oil = 4*m_oil/(pi*mu_oil*d_oil);
f_oil = (0.790*log(Re_oil)-1.64)^(-2);
Pr_w = mu_w*c_w/k_w;
Nu_oil = ((f_oil/8)*(Re_oil-1000)*Pr_oil)/(1+12.7*(f_oil/8)^(0.5)*((Pr_oil^2/3)-1));
Ra_w = (g*B_w*dT_w*d_oil^3)/(a_w*nu_w);
Nu_w = 0.36 + (0.518*Ra_w^(1/4))/(1+(0.559/Pr_w)^(9/16))^(4/9);
h_w = Nu_w*k_w/d_oil;
A_f = 0; % area of heat transfer fins
N = 100; % number of fins
A_h = N*A_f;
L_boil = Q_boil/((pi*d_oil)*h_w*dT_w)-A_h/(pi*d_oil);
```

Appendix B – Duratherm S Properties

DURATHERM
EXTENDED LIFE FLUIDS

P.O. Box 563, Lewiston, NY 14092 • P 800 446 4910 • F 905 984 6684 • info@heat-transfer-fluid.com

Property Vs. Temperature
Duratherm S - Metric Units
Temperature (Celsius)
Minimum: -50 Maximum 343

Temperature (Celsius)	Density (kg/m3)	Kinematic Viscosity (Certistoke)	Thermal Conductivity (W/m.K)	Heat Capacity (kJ/kg.K)	Vapour Pressure (kPa)
-50	999.85	334.25	0.146	1.551	0.00
-40	997.36	287.80	0.144	1.561	0.00
-34	994.86	241.34	0.143	1.570	0.00
-29	992.36	194.89	0.142	1.580	0.00
-23	989.87	136.74	0.141	1.590	0.00
-18	987.37	125.06	0.140	1.599	0.00
-15	984.87	113.37	0.139	1.607	0.00
-12	982.38	106.40	0.138	1.614	0.00
-7	979.88	99.42	0.138	1.621	0.00
-1	977.38	92.45	0.137	1.629	0.00
0	974.89	85.47	0.136	1.636	0.00
4	972.39	78.50	0.135	1.643	0.00
10	969.90	71.67	0.134	1.651	0.00
16	967.40	64.83	0.133	1.658	0.00
21	964.90	58.00	0.132	1.665	0.00
27	962.41	51.17	0.132	1.673	0.00
32	959.91	44.33	0.131	1.680	0.00
38	957.41	37.50	0.130	1.687	0.00
43	954.92	35.72	0.129	1.696	0.00
49	952.42	33.93	0.127	1.706	0.00
54	949.92	32.15	0.126	1.715	0.00
60	947.43	30.37	0.125	1.724	0.00
66	944.93	28.58	0.124	1.733	0.00
71	942.43	26.80	0.123	1.742	0.00
77	939.94	25.02	0.122	1.751	0.00
82	937.44	23.23	0.121	1.761	0.00
88	934.95	21.45	0.119	1.770	0.00
93	932.45	19.67	0.118	1.779	0.00
99	929.95	17.88	0.117	1.788	0.00
104	924.96	16.70	0.115	1.807	0.00
110	922.46	15.30	0.114	1.816	0.00
116	919.97	14.90	0.112	1.825	0.00
121	917.47	14.49	0.111	1.834	0.21
127	914.97	14.09	0.110	1.843	0.25
132	912.48	13.69	0.109	1.852	0.29
138	909.98	13.29	0.108	1.862	0.33
143	907.48	12.89	0.107	1.871	0.37
149	904.99	12.49	0.106	1.880	0.41
154	902.49	12.09	0.104	1.890	0.50
160	900.00	11.68	0.103	1.899	0.58
166	897.50	11.28	0.102	1.909	0.66
171	895.00	10.88	0.101	1.918	0.74
177	892.51	10.48	0.100	1.928	0.83
182	890.01	10.08	0.099	1.938	0.91
188	887.51	9.68	0.098	1.947	0.99
193	885.02	9.28	0.097	1.957	1.08
199	882.52	8.88	0.095	1.967	1.16
204	880.02	8.47	0.094	1.976	1.24
210	877.53	8.07	0.093	1.986	1.32
216	875.03	7.67	0.092	1.995	1.41
221	872.54	7.27	0.091	2.005	1.49
227	870.04	6.87	0.090	2.015	1.57
232	867.54	6.47	0.089	2.024	1.65
238	865.05	6.07	0.088	2.034	1.74
243	862.55	5.66	0.086	2.044	1.82
249	860.05	5.26	0.085	2.053	1.90
254	857.56	4.86	0.084	2.063	1.99
260	855.06	4.46	0.083	2.072	2.07
266	852.56	4.35	0.082	2.082	2.56
271	850.07	4.24	0.081	2.092	3.05
277	847.57	4.13	0.080	2.101	3.54
282	845.07	4.02	0.079	2.111	4.03
288	842.58	3.92	0.078	2.121	4.52
293	840.08	3.81	0.077	2.130	5.01
299	837.59	3.70	0.076	2.140	5.50
304	835.09	3.59	0.075	2.150	5.98
310	832.59	3.48	0.074	2.159	6.47
316	830.10	3.37	0.073	2.169	6.96
321	827.60	3.26	0.072	2.178	13.77
327	825.10	3.15	0.071	2.188	20.58
332	822.61	3.04	0.070	2.198	27.39
338	819.61	2.93	0.069	2.207	34.20
343	817.62	2.83	0.067	2.217	41.01