Analysis of Heat Transfer in a Thermoacoustic Stove using Computational Fluid Dynamics

A Thesis

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By

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Abstract

Thermoacoustic devices have the potential to provide electricity from waste heat to more efficiently use energy resources and to provide new access to electricity for millions of persons around the world. The international SCORE (Stove for Cooking and Refrigeration) research team is studying, designing, testing, and disseminating a biomass cook stove that generates electricity using thermoacoustics. The device captures heat from a small cook stove, uses the thermoacoustic effect to transfer the energy to acoustical sound waves that are captured by a linear alternator and turned into electricity. This study highlights research to characterize and improve the efficiency of heat captured in the SCORE stove.

The SCORE stove prototype, named Demo2, had a measured loss of 45% in the process of capturing heat from a biomass fire to create an acoustic wave (Riley and Saha, 2010). This research used commercially available Computational Fluid Dynamics software to characterize the physical phenomena occurring inside the Demo2 unit. The simulation model was comprised of all components used in a thermoacoustic device including heat exchangers and regenerator. The thermoacoustic effect itself was not simulated, however. The simulation first derived the steady state temperature and flow fields, given boundary conditions extrapolated from observed experimental data. Secondly, an acoustic wave was induced over the steady state temperature solution to observe the impact on heat transfer. Finally, a simulation was run to calculate pressure transmission loss due to geometry.

Simulations predicted heat capture and transfer from the biomass fire’s exhaust gases to the working air inside the unit. The amount of heat captured was low and therefore it is recommended that design of the hot heat exchanger should be altered to boost heat transfer. Results indicate that during stove operation absent of acoustics, radiation is the dominant mode of transferring heat. Surfaces closest in space and parallel to receiving surfaces had the highest heat flux. Simulations modeling acoustics showed convection during all portions of the sound wave to be
greater the mode of heat transfer. It is recommended that heat exchanger geometry should be altered to expand the hottest sections of temperature distribution over the hot heat exchanger to improve both radiation at startup and convection during acoustic operation. Conduction in the air should be neglected at all times. Transmission pressure loss simulations for the acoustic wave due to geometry exceed 25%.
This is dedicated to all individuals, regardless of context, who are taking chances to try and make tomorrow a better place...
Acknowledgments

The author would like to acknowledge Dr. Ann Christy for advising him throughout the duration of this study. Her patience, optimism, and willingness to tackle new concepts directly led to the creation of this thesis. Her guidance on the journey is greatly appreciated and the author is forever grateful. Acknowledgement also extends to Paul Riley and Chitta Saha of the SCORE Project at the University of Nottingham for giving the author the opportunity to contribute to the project. Recognition is deserved for their willingness to accept the author onto the team and for being so accommodating. Special thanks to Dr. Scott A. Shearer and Dr. Lingying Zhao for serving on the author’s committee. Finally, the author would like to recognize the support of his loving family and especially his girlfriend for openly becoming an unofficial research collaborator. Her contributions to discussions and offerings of daily encouragement, specifically during the many challenging portions of this study, made this achievement possible.
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Chapter 1: Introduction

The human race has developed numerous machines to save and improve the quality of life. One caveat of the mechanization, and more recently the digitalization, of our world is that the modern way of life relies on the availability of energy. Fossil fuels are a finite resource, and 78.4 percent of the energy consumed in the United States in 2010 came from fossil fuels (Fichman, 2011), therefore research is needed to decrease that percentage and/or lower the overall consumption. Reducing the use of fossil fuels means finding new techniques to harness energy from previously under utilized sources. One of the common underutilized sources of energy in modern society is waste heat. Waste heat is all too often a necessary byproduct of many industrial and residential processes. Modern society has not been able to or cared to utilize this rich source of energy. The major problem with waste heat is that few energy transformers are designed to use temperature difference as the input. Fortunately, the re-emerging field of Thermoacoustics functions off this very input. In addition to using temperature difference as the input, when devices are designed to utilize the thermo-acoustic effect, the output can be electricity. Besides liquid and gaseous fuels, electricity has become the energy form of choice for most modern devices. Since devices using the thermoacoustic effect can transform a temperature difference such as waste heat into electricity, the field has great potential to reduce fossil fuel use and
improve efficiency in a host of applications. Given the potential for thermo-acoustic devices to be an important asset in the future energy discussion, the relatively small body of knowledge about this field must be extended through research to improve the design and expand the applications of thermoacoustic devices. This thesis describes research on an off-grid/developing world application of thermoacoustics in a stove that generates electricity during the cooking process.

1.1 Thermo-acoustic Phenomenon

Before describing the working principles of thermoacoustic devices, understanding the thermoacoustic effect will allow for greater appreciation of the operation of such devices. Thermoacoustics refers to the broad fields of either time-averaged heat/temperature-gradients driven by fluid oscillations or fluid oscillations driven by time-averaged heat/temperature gradients (Rott, 1980). How exactly are fluid oscillations (acoustics/sound waves) created from a temperature difference and vice versa? The conversion occurs during the interaction of a working fluid such as air or helium with-in a small, solid geometry. Broadly speaking, the interaction exploits fluid properties and an out of phase thermal relationship between the fluid and the geometry to allow the moving air molecules to do work.

As stated above, the thermoacoustic effect results from the interaction between a fluid and the restricted cross sectional area of a small geometry. The following explanation assumes the fluid is exposed to a temperature difference resulting in acoustic waves, although the reverse process is also true. In low end applications the working fluid is air (preferably dry), while more high-end applications use helium or a mixture of gases with high Prandtl numbers (Belcher et al., 1999). A high Prandtl
Figure 1.1: The movement of a gas particle between two parallel plates in a standing wave generated by a thermoacoustic engine. (Swift and Garrett, 2003)

number indicates that the fluid’s viscosity is high relative to its ability to transfer heat within the fluid. The fluid is then contained inside a geometry where the height and/or width are small enough to be within both the viscous and thermal boundary layers of the fluid. A diagram of a fluid (gas) molecule in a standing wave field between two flat plates is shown in Fig 1.1.

The solid black angled line in the bottom left graph of Fig. 1.1 is the temperature gradient between both flat plates with respect to the depth position. The ellipse is the path the gas molecule traces as it oscillates back and forth in the acoustic wave. Since the molecule is in the thermal boundary layer, the molecule is neither the temperature of the plate nor the free stream temperature. The molecule is always lagging behind the temperature of the plate, meaning that the temperature of the plates and the gas molecule are out of phase with each other. By being out of phase and the fluid being ideal gases, the volume of the molecule will also be oscillating out
of phase. As shown on the right in Fig. 1.1, the pressure-volume graph is also an ellipse; in this case the area within the ellipse in a pressure-volume graph shows the amount of work performed by the molecule. The work done, in this case of a heat engine or prime mover, is the amplification of the sound wave. In the reverse case of a heat pump, the work performed by the molecules would be heating or cooling the plates at the end of the path traced by each molecule. Swift (2003) provides a more complete description of the thermoacoustic effect and its use in varying types of engines.

The history of thermoacoustics is long, sporadic, and diverse, and is only recently receiving more attention as people look to new ways to use energy. The observation of the effect dates to the late 19th century with math describing the basic physics not being published until a series of papers by Nicolaus Rott in 1969, 1975, and 1980 (Rott, 1969). The recent focus on thermoacoustics has been sparked by the creation of various types of devices using the effect in the last two decades (Swift, 2004). With an understanding of how the thermoacoustic effect works, the effect can be exploited in devices to provide a service.

1.2 Thermo-acoustic Devices

The thermoacoustic effect can do work to amplify sound waves (as a prime mover) or to create a greater temperature difference (as a heat pump). Given the application of thermoacoustics presented in this thesis, this section will concentrate solely on the use of the thermoacoustic effect for designing prime movers.

Figure 1.2 shows how the concept from in Fig. 1.1 can be duplicated over the entire surface area of a ‘stack’ of parallel plates. The temperature difference is applied across
the plates or stack allowing for the thermoacoustic effect to do work. As the effect reaches steady state, the inside of the container will have a strong standing wave oscillating inside. The specifications of the sound wave are dictated by traditional acoustic theories which state that sound generated within a closed tube-like structure produces standing waves. Standing waves oscillate at a given frequency with the minimum and maximum of the amplitude remaining constant at a given position. This also produces the common nodes and anti-nodes present in basic wave theory. The acoustic wave can now be captured with either a piston to do mechanical work or a linear alternator (speaker) to produce electricity via a magnet and wire coil configuration.

The standing wave generating device schematic in Fig. 1.2 illustrates operation of the first generation of thermoacoustic devices (Hofler et al., 1988) with the addition of a cook stove application. After a decade of application and research, the efficiency of

Figure 1.2: Schematic of Initial SCORE Stove(Riley, 2007)
such devices was found to be limited to approximately 20% due to intrinsic inefficiencies of heat transfer during the thermoacoustic effect in the stack. In an attempt to boost efficiency, designers attempted to apply traveling wave technology. A schematic of a traveling wave device is shown in Fig. 1.3(a). The wave travels around the loop to be boosted or regenerated with more acoustical power as it passes through the regenerator. The regenerator is geometrically identical to a stack in a standing wave device but is renamed in a traveling wave device. Although efficiencies did improve, the waves suffered from major viscous losses in the regenerator (Swift, 1992). The major breakthrough in thermoacoustic design was the coupling of a traveling wave regenerator with the standing wave design allowing the traveling loop to amplify the standing wave component (Backhaus and Swift, 2000). The advanced traveling wave design is shown in Fig. 1.3(b) with the standing wave occurring in the resonance tube and the traveling wave occurring in the feedback loop. An expansion of knowledge about and new ways to handle the complex, nonlinear nature of thermoacoustic devices is necessary to improve the overall efficiency of thermoacoustic devices.

1.3 SCORE Project

The Stove for Cooking, Refrigeration and Electricity (SCORE) project is a $4 Million USD, five year international collaboration research project funded by the Engineering and Physical Science Council of the UK government that aims to improve the quality of life for those who do not have access to electricity (Riley and Saha, 2010). The target audiences are communities in developing nations and off-grid homesteaders. The project applies thermoacoustic technology to an appropriate design context focused on reliability and cost, in addition to electrical production.
The technology has the potential to also provide refrigeration, but the project leaders have chosen at the present time to solely focus on electricity production. Since the stove is producing electricity and is therefore a prime mover application, the stove needs a temperature difference to drive the system. The source of heat is envisioned to be a small biomass fire, such as one might cook over in a third world household or a wilderness setting. For an electrical benchmark, as little as 20 watts has been shown to be beneficial to such households but SCORE has chosen a target electrical
production of 100 watts (EPSRC, 2010). Market analysis suggests that at the upper-cost target of $120USD with 20 Watts of electricity as many as 60 million people could afford the stove. At the lower-cost target of $40USD and 100 Watts, it would be affordable to over one billion people (Riley and Saha, 2010). If the stove can be produced at the low cost target while providing enough usable electricity, adoption of the SCORE stove will be vastly greater than that of other small, non-electricity producing stoves, which despite dissemination programs have failed to be adopted by many developing country households (Barnes et al., 1994).

The initial concept for the SCORE project was shown in Fig. 1.2. After further calculations and testing, the leaders decided upon converting the device to a traveling wave design. A schematic of the first complete prototype is shown in Fig.1.4. The unit captures heat from a fire below the cooking surface; this heat was normally energy lost to the ambient environment or out the chimney. The first prototype is meant to be used in applications where cooking is performed with propane or other cooking fuels. The stove captures heat at the hot heat exchanger (HHX). A cold heat exchanger maintains a temperature close to ambient conditions. The exchangers are put on opposing sides of the regenerator which, in this design, was composed of layers of stainless steel mesh. This combination creates the thermoacoustic effect that produces a traveling sound wave in the tubes. Two thermoacoustic engines are used to amplify each other to create more acoustic power. The acoustic power is captured by a linear alternator (loudspeaker) and is converted to electricity.

After testing and validation in the design of the first prototype, the SCORE team worked towards reducing cost. The second prototype, referred to as Demo2, is similar to the first prototype in function, but uses less material and is more modular.
The stove, as pictured in fig 1.5, is designed to be built with a core unit labeled as “TAE System”. The TAE (ThermoAcoustic Engine) system holds the hot heat exchanger (HHX), ambient heat exchanger, the regenerator, and linear alternator. The TAE System is to be manufactured, sold, and disseminated with the complete plans for installers and distributors to build the remaining parts from locally available materials. The TAE Systems is to be used in conjunction with an improved biomass cook stove in order to improve overall efficiency and decrease the amount of biomass necessary to meet daily cooking needs.

The focus of the project has been designing the SCORE stove to be manufactured for around $40USD at high production rates. Even though the SCORE group has set the current record for electrical production from a biomass fueled thermoacoustic engine at 22.7 watts, averaging 8 watts over a 60 minute period on a 12 volt load, the performance numbers are far below the target 100 watts (Riley, 2010). With
the stove design near its economic benchmark, the focus has changed to boosting electrical production.

1.4 Scope of Research

The Demo2 stove has succeeded at the "proof of concept" stage, therefore the focus has shifted to improving efficiency and boosting electrical production. Using performance data from a working prototype, Fig. 1.6 presents a heat flow diagram of the first prototype. As noted, less than half of all energy coming from the propane flame is captured by the hot heat exchanger and only a quarter of that turns into acoustic power. A complete heat flow diagram has not yet been created, but initial experimental data shows similar or even less efficient results for Demo2.

If the SCORE project is to have the potential to reach a billion people, the transfer of energy from the flame to acoustic power needs to drastically improve. The focus of this thesis was to study the physics of heat transfer and fluid flow related to the
Figure 1.6: Summary of Heat Flow in First SCORE Stove Prototype. (Chen et al., 2011)

operation of the SCORE stove Demo2. The goal was to analyze the current methods of heat transfer using classical and numerical methods to gain an understanding and then suggest design changes in order to improve efficiency of the stove all while using low cost materials and passive methods. For examples, this precludes the use of expensive metals and forced convection systems.

The research highlighted in this thesis evaluated and compared the magnitude and effects of radiation and convection on the efficiency of transferring heat captured by the hot heat exchanger (HHX) to the outer screen mesh of the regenerator. Convection was thought by the author to not be accurately estimated as a mode of heat transfer due to the complex combination of the geometry and acoustics. Furthermore, the research calculated and characterized the transmission pressure losses through the complex geometry of the SCORE Demo2 stove, also believed to be a major loss of efficiency.
Chapter 2: Outlining the Problem

2.1 Introduction

Given the need for better efficiency in the SCORE Demo2 stove design, the SCORE leadership decided that three main areas were in need of study and improvement. The first was the linear alternator that turns sound into electricity. The SCORE team at Nottingham University has designed and built a new linear alternator that has the potential to improve efficiency and robustness. The second area was looking at losses in acoustical power through bends and fittings. Ongoing research is now attempting to characterize the phenomenon with the intent to reduce those losses. The final area, and the focus of this research, was to understand the physics of how heat is transferred in acoustical environments. Preliminary testing of earlier prototypes had shown around an 80% loss of energy in the transfer of heat from a flame source to the production of acoustical power (Chen et al., 2011). Furthermore, testing had shown that temperatures inside the stove were not reaching anywhere near design temperatures. For these reasons, research was conducted to discover exactly how and to what magnitude heat was being transferred in an acoustical environment inside the SCORE Demo2 device. This chapter will outline the specific geometry and physics that initially drove design and what additional physics were studied in this
study. Information outlined in this chapter was used in a CFD simulation software environment to characterize and quantify heat transfer in conjunction with acoustical phenomenon.

2.2 Geometry

The SCORE demo2 stove utilizes two ThermoAcoustic Engine (TAE) units. The units are identical and are placed side-by-side. Since the units are identical, CFD research was conducted on one unit only. Figure 2.1 shows an overview of the components of a TAE unit.

![Diagram of TAE unit](image)

Figure 2.1: Highlight of SCORE Demo2 ThermoAcousticEngine (TAE) unit. Modified from (Chen et al., 2011)

The TAE unit is composed of a hot heat exchanger (HHX), a cold heat exchanger (CHX), regenerator mesh, and a thermal buffer tube. A fire is lit on the underside of
the device with radiation and convective gases transferring heat up to the hot heat exchanger. The hot heat exchanger is a convoluted sheet of stainless steel as shown in the rear view of Figure 2.2. The hot heat exchanger captures and transfers heat to the regenerator mesh. The mesh is a stack of stainless steel mesh screens creating a porous cross sectional area that produces the thermoacoustic effect. Because the thermoacoustic effect in this application depends on a temperature difference, a cold heat exchanger is placed on the other side of the mesh to maintain ambient temperatures on that side. The cold heat exchanger is a section of a typical car radiator as shown inside the circle in the front view of Figure 2.2. The difference between heat captured from the fire and the cold heat exchanger will drive the thermoacoustic effect.

![Figure 2.2: Front and rear view of SCORE Demo2 ThermoAcousticEngine (TAE) unit.](image)

Figure 2.2: Front and rear view of SCORE Demo2 ThermoAcousticEngine (TAE) unit.
The acoustics created by the thermoacoustic effect at the regenerator mesh are sent out of the casing through the thermal buffer tube into the feedback piping. The traveling sound wave eventually returns from the feedback pipes through the cold heat exchanger where it is amplified continuously by the thermoacoustic effect in the regenerator mesh. The geometry is designed to maximize heat transfer from the flame source to the mesh while keeping the acoustic path sealed. The design does not use moving parts or exotic materials in an attempt to reduce cost and ultimately achieve the social impact described in the SCORE mission.

2.3 Initial Physics used in Design

2.3.1 Initial Design

The engineering discipline of heat exchanger design is well established and has developed general procedures for increasing heat transfer in varying applications. Given that the hot heat exchanger is capturing heat from natural convection flows originating at the fire, the obvious choice of physical system component is a fin or heat sink design. Initial testing by the SCORE team utilized readily available fins used in electrical cooling applications. Unfortunately, the cost of such units was deemed to be too high to meet the low cost requirement of the SCORE project. Other ideas such as casting fins separately or as part of the TAE unit casing also proved to be cost prohibitive.

In order to design a cheaper heat exchanger, knowledge of material costs and basic engineering equations were combined to produce a unit that was easy to produce while performing as efficiently as possible. From a material selection and handling point of view, the cheapest possible selection is flat sheet metal. Sheet metal would be easy to
find and replace in the field, but suffered one engineering flaw. Thermal convection
was assumed to be the greatest heat transfer mechanism, and thus the effective surface
area of a flat plate would be too low to allow for a large enough transfer of energy.
This concept is highlighted in the basic convective heat equation shown in Equation
2.1.

\[ q_{\text{conv}} = h_{\text{eff}} \times \text{Area}_{\text{eff}} \times (T_{\text{hot}} - T_{\text{cold}}) \] (2.1)

Where,

\( h_{\text{eff}} \)= Effective Heat Transfer Coefficient

\( \text{Area}_{\text{eff}} \)= Effective Area Transferring heat

\( T \)= Temperature

The equation gave priority to design features for the new hot heat exchanger. \( T_{\text{hot}} \)
and \( T_{\text{cold}} \) are important design criteria but are fixed by the size and source of the
flame for \( T_{\text{hot}} \) and by the ambient environment for \( T_{\text{cold}} \). The effective heat transfer
coefficient \( h_{\text{eff}} \) is hard to determine, let alone increase without the introduction of
expensive new materials or forced convection to create an increase in turbulence. The
equation then shows that the only easy way to increase heat transfer is to increase
the effective area. Combining the idea of flat sheet metal being the cheapest available
material with the idea of obtaining the largest area possible, the solution was to fold
a flat sheet metal piece to increase the area as much as possible. The resulting hot
heat exchanger is a flat piece of sheet metal bent into an accordion-like shape shown
in Figure 2.3.
2.3.2 Limitations and Issues

With the new design of the hot heat exchanger, testing has shown that temperatures were not reaching anywhere near the expected temperatures. Initial design equations assumed an operating temperature of 470°C. To the date of this writing, the temperature achieved by the Demo2 prototype have peaked around 315°C. SCORE designers believe that although the actual area of the plate has increased, limited knowledge of acoustic fluid dynamics has led to an effective heat transfer area that is smaller than the geometric surface area. The author was assigned the task of characterizing the fluid dynamics of the acoustical waves in the geometry of the SCORE Demo2 prototype.
In addition to concerns over effective surface area brought by SCORE designers, the author believes the inability to compute a realistic effective heat transfer coefficient for the complex geometry in an acoustical environment is also a potential contributor to the drastically inaccurate design calculations. In cases involving elevated temperatures as high as 500°C or more, the contribution of radiative heat transfer can be over-calculated as a proportion of overall heat flux because surface temperatures would be difficult to estimate. The potential for over or under estimating radiation during the design phase is high as suggested by Equation 2.2.

\[
\dot{q}_{1\rightarrow 2} = \varepsilon \star \sigma \star A \star F_{\text{view factor}1\rightarrow 2} \star (T_1^4 - T_2^4) \tag{2.2}
\]

Where,

\[\varepsilon = \text{Emissivity of Surface}\]

\[\sigma = \text{Stefan Boltzmann constant} = 5.670373108 \ \text{W/m}^2\text{K}^4\]

\[A = \text{Effective Surface Area}\]

\[F = \text{View Factor}\]

\[T = \text{Temperature}\]

Equation 2.2 shows that the rate of heat transfer due to radiation is a function of the temperature difference to the 4th power. Given the high temperatures seen in flame applications, estimating the contribution by radiation was difficult because calculating surface temperatures was complicated by the lack of knowledge about acoustically driven convection. The overlay of acoustics may raise or lower surface temperature which in turn affect the contribution of radiation. Furthermore, given that radiation transfers heat to all surfaces, the transfer of heat via radiation depends heavily on location and surrounding conditions that had not previously been considered. Radiation calculations were often assumed to be uniform as a function of space.
and time. This means that surface temperatures were unrealistically estimated to be consistent across large surface sections. The use of viewfacters was idealized and may also have led to overestimated heat transfer. Characterizing an effective heat transfer coefficient as a function of space and temperature would allow for more accurate designs.

Finally, in addition to heat transfer, the main purpose of the TAE is to create powerful acoustic waves. The current design of the hot heat exchanger suggests that very little concern was given to potential pressure losses due to turbulence inside the device’s geometry. Analysis of heat transfer inside the SCORE Demo2 stove must include characterization of the fluid dynamics to calculate pressure losses as the acoustic wave travels through the TAE. Pressure loss and heat transfer are inversely related; it is important to quantify the magnitude of how geometric designs, optimized to improve heat transfer, can affect pressure loss in the acoustic wave.

2.4 Operation and Related Physics

The operation of the SCORE stove from the point of view of heat transfer analysis was split into two parts: startup and acoustics. The startup phase involves the time from the lighting of the flame until the onset of acoustical waves. The second part is the physics of heat transfer in the presence of acoustical waves. The study was split in two since they act independently of each other and the scales of each phenomena are drastically different.

2.4.1 Startup

Startup is the time from the lighting of the flame to the onset of acoustical pressure oscillations. The onset of acoustics is a function of temperature difference across
the regenerator. The acoustical power potential is also a function of temperature difference across the regenerator as shown in Figure 2.4. For this reason, the cook stove was initially studied as a purely natural convection heat transfer problem. Inputs in the TAE were defined from experimental data.

Figure 2.4: Temperature Difference Across Regenerator and Related Acoustic Power. (98 kPa and 73 Hz) (Chen et al., 2011)
2.4.2 Acoustic

Once the fluid field was developed during the startup phase, acoustics were introduced into the simulation. Given that the design sound wave frequency of Demo2 is 72Hz, the period for the wave is 0.0138 seconds. With the short time period for acoustics, it was impractical to run long simulation lengths of time due to the short time steps needed to capture all components of the acoustical sine wave. Fortunately, experimental results as shown in Figure 2.5, indicate that the hot heat exchanger temperature remains relatively steady allowing for any conclusions obtained in a short simulation period to be applied over the majority of stove operation time. Since

![Figure 2.5: Time Dependent Results of Demo2 Stove Operation. (Riley, 2012)](image)

natural convection is a buoyancy driven phenomena, the introduction of acoustical
pressure oscillations is expected to wipe out any buoyancy forces thus eliminating any natural convection. Radiation will continue to contribute after the introduction of acoustics and is be a function of surface temperatures which can also be extrapolated over time.
2.5 Research Outcomes and Objectives

Characterizing and making design changes for the hot heat exchanger is difficult given the issues related to estimating realistic convective and radiative heat transfer in the complex geometry and inputs of the SCORE stove. In order to characterize the physics in the Demo2 prototype, commercial computational fluid dynamics (CFD) software was utilized to calculate specific physics and to handle the complex interactions between phenomena. The goal of the series of simulations was to quantify variables needed to both explain the current operation of the SCORE Demo2 stove and also set assumptions for modification and prototyping in the future. The objectives were to:

- Determine relative contributions of convection, conduction, and radiation in the SCORE stove during stove startup.
- Compare the predicted performance of using Stainless Steel and Aluminum for the Hot Heat exchanger.
- Determine relative contributions of convection, conduction, and radiation in the SCORE stove during acoustic operating conditions.
- Visualize the air flow pattern and conditions inside the SCORE stove during all operating conditions.
- Quantify the acoustic losses due to the geometry of the SCORE stove.

In order to accomplish the stated objectives, the following set of tasks were performed:

- Create CFD model of the SCORE stove.
- Steady state simulation of stove startup driven by radiation and natural convection.
• Transient simulation of effect of acoustics on steady state solution.

• Transient simulation of acoustic pressure losses due solely to TAE geometry.

The outcomes of the research were a quantification of 1) heat transfer parameters, 2) relative convective and radiation contributions, and 3) pressure losses due to system geometry. With this information, the SCORE team will be able to better understand the trade-offs of design and use the information to make improvements on stove prototypes in the future.
Chapter 3: Methods and Procedures

3.1 Introduction

The problem of characterizing and quantifying heat transfer in the SCORE Demo2 prototype stove was addressed using Computation Fluid Dynamics (CFD) modeling techniques. The code employed during the research was the commercially available software named Star CCM+ (CD-adapco, 2012). Star CCM+ is a complete CFD package that includes both meshing codes and equation solvers. The software is a finite volume solver that uses the algebraic multigrid solver for linear equations. This chapter explains how the CFD model was set up and run to analyze heat transfer within the stove.

3.2 Geometry

Although the physical dimensions of the SCORE Demo2 stove are about the size of a third of a meter per ThermoAcoustic Engine unit, the magnitude of scale for the radius of the heat exchanger convolutions are only 4 mm. Since the purpose of this study was to characterize the heat transfer between the convoluted HHX and the wire regenerator mesh, creating a simulation with that level of refinement for the entire stove was not feasible without access to a super computing cluster. To reduce the
size of the simulation, a vertical cross sectional slice of the geometry was used. The thickness of the slice was 23.5 mm which is slightly larger than a single convolution of the HHX. Using the slice, the model assumed that the slice was taken from the center of the prototype resulting in all edge effects from boundaries not modeled in the depth (Z) direction to be assumed negligible. A CAD model with labeled side views of the simulation geometry is shown in Figures 3.1. Each color in Figures 3.1(a) and 3.1(b) represents a different geometric region as labeled in Figure 3.1(b). Each region, except for the convoluted HHX, was modeled as either a porous or fluid region, meaning the CAD model is a model of the air volume inside the TAE unit and not of the physical unit itself.

The model resembles a side view of the stove fully installed which was important to enable correlation of numerical simulation results to observed experimental results. The fully installed stove includes ducting which channels exhaust gases from the burning biomass vertically up onto the underside of the convoluted HHX and then
out horizontally to the TAE to heat the cooking hob surfaces. A schematic of the Demo2 stove with associated ductwork is shown in Figure 3.2. The installed stove includes a chamber where the biomass is burned; exhaust gases exit the chamber by means of natural convection. During experimental operation, temperature readings were taken at many locations within the stove (Riley, 2012). Data were collected as a function of time including a point located directly above the fire, as shown in Figure 3.2. The locations indicated in Figure 3.2 are specifically important as data measured at these points were used as inputs into the simulation. The point temperature reading 1 in Figure 3.2, which was located upstream from the main unit, allowed for a more realistic simulation as the complex physics inside the unit were able to be modeled by the CFD code instead of manually entering estimated conditions. Manually entering
crucial boundary conditions usually results in conditions that are too simple or of too low resolution to match reality. The use of upstream values corrects this problem and reduces necessary experiential data to a few easy readings. The result is a more robust and complete simulation.

3.3 Model Regions

Star CCM+ breaks up different portions of any geometry into separate volumes that are called regions. Each region can have different physics and/or meshing models applied on each volume. The model developed for the SCORE Demo2 stove included nine separate regions that were treated differently in order to accurately simulate what occurs in the stove. The following sections will outline exactly how each region was handled.

3.3.1 Hot Side of Convoluted HHX - ‘HHX Hot’

The first region is the volume representing the air inside the duct work and on the hot side of the convoluted hot heat exchanger. The volume includes the empty space from inside the inlet and outlet ducts and inside the convoluted heat exchanger as shown in Figure 3.3. Since one concave convolution was used on the opposite, cold side of the convoluted heat exchanger, two convex convolution volumes worth of air were chosen for the hot side. This allowed for simulated heat to transfer from the hot exhaust gases through the solid heat exchanger and to the cold side from all directions creating a more accurate simulation. The length of the heat transfer area from the hot side into the solid hot heat exchanger is smaller than the full length of the solid heat exchanger due to the fact that the air gaps on the hot side when fully installed are filled with insulating material as shown in Figure 3.3(a). The volume in
the model only included the air that actually touches the solid heat exchanger which is labeled as ‘Output [In-Place-2]’ in Figure 3.3(b).

![Photo of Underside Duct Work](image1)

![Highlighted HHX Hot Region](image2)

Figure 3.3: Photo and CFD Volume of Modeled HHX Hot Region

The boundary conditions of the hot side includes an inlet, an outlet, an output interface, and wall conditions. The crucial boundary is the inlet which is labeled as "HHX Hot: Hot In" in Figure 3.3(b). The inlet was modeled as a constant velocity inlet with constant temperature. The natural convection startup portion of the simulation was run as a steady state case since the temperature distribution in the device only mattered at the temperatures where acoustics was tested. During the acoustic or operating portion of the simulation where the simulation was run as unsteady/transient/time marching, the velocity and temperature inputs into the hot region were assumed to be constant as well. This assumption was deemed appropriate because the time scale for the acoustical portion of the simulation was low multiples of 0.0138 seconds (the period of a 72 Hz acoustic wave). In such a short
time scale, the conditions of exhaust gases input into the TAE unit are nearly constant as the buoyantly driven conditions of exhaust gases have time scales of several seconds allowing for the constant assumption to be valid for short simulation times.

The crucial parameter into the region was velocity. No velocity values were recorded during experiments, meaning assumptions were needed to estimate velocity values at the inlet.

To estimate the velocity at the inlet, simulations were run on the lower portion of the duct work. As shown Figure 3.2, temperatures were recorded at a lower or upstream position - labeled as "Temperature Reading 2". The temperatures were used to run an idealized natural convection simulation where the temperatures at the bottom were assumed to be a constant wall boundary. The constant wall boundary assumption is not entirely accurate as the fluid would have momentum from below, but given the low velocities generated by natural convection, momentum input was treated as negligible. The CFD code recorded the peak velocity of the buoyancy "blooms" generated from the simulation at the upper location. Given the temperatures recorded at the upper (temperature 1) location were in the 600°C-1000°C range, a velocity of 0.43 m/s as measured in the simulation was assumed to exist at this related temperature range. The CFD velocity inlet for the stove simulation used the 0.43 m/s as a constant input value.

Picking an exact temperature for the inlet required some preliminary simulations. Although experimental readings existed, the chaotic nature of burning biomass results in extremely nonlinear observed experimental results. Efforts to combat the chaotic manner of the results by burning a more consistent fuel such as propane were attempted, but locations of data collection were different than on a fully installed
stove. For this reason, the simulation used experimental data to suggest a range of temperatures and then simulations were run to verify attainment of that range. The original design parameter used as the benchmark for initial design was that the solid, convoluted heat exchanger would be at a temperature of 470°C (Riley, 2012). With the surface of the convoluted heat exchanger not seeing uniform exhaust gas conditions, temperature varied across the surface. For this reason, simulations were run in order to achieve a 2-D pattern of temperatures on the solid heat exchanger that were in an appropriate range of temperatures with the maximum hovering around 470°C. Using experimental data and relating the slopes of exhaust temperatures to heat exchanger temperatures, initial simulations assumed a constant temperature of 850°C of the exhaust gases at the inlet. Upon full simulation trial and error, a final value of 880°C was chosen since it produced temperatures on the heat exchanger surface that peaked near 500°C while averaging at 450°C. Although the choice of 880°C is somewhat arbitrary, the chaotic temperatures in the stove vary over a wide range suggesting that the simulated values were representative of actual operating conditions.

The other remaining unique surfaces were the exhaust outlet and the solid interface with the HHX. The final outlet for the exhaust gases was treated as a pressure outlet whose temperature was set to an ambient temperature of 20°C. Pressure under ambient conditions were calculated with the CFD code treating the output as a wall during one simulation iteration allowing for the pressure field to be derived. Using the code to calculate space dependent pressure was necessary since the inclusion of gravity makes pressure dependent on elevation. The interface between the air volume and the solid, convoluted heat exchanger was handled by the CFD code as a direct
contact interface. All other surfaces were treated as walls with adiabatic thermal and no-slip wall conditions.

### 3.3.2 Convoluted Hot Heat Exchanger and HHX Cold Region

The focus of the study was centered around the use of a convoluted, solid heat exchanger. The heat exchanger is made of stainless steel with a thickness of 0.9 mm and an inside bend radius of 3.95 mm. Stainless steel thermal properties are listed in Table 3.1. For the purpose of comparison, the model’s material properties were numerically changed to those of aluminum for additional simulations. Aluminum is known to be much a better heat conductor than stainless steel. Thermal properties used for aluminum are also listed in Table 3.1. The fluid to solid contact type interface was used on both sides to interact with the fluid flow on both sides of the metal sheet. The interface on the acoustic or cold side of the sheet runs the entire length of the convolution as opposed to the under-side where insulating material blocks air flow from the outer edges of the metal sheet restricting the area of interaction to the region highlighted in purple in Figure 3.4(b).

<table>
<thead>
<tr>
<th>Property</th>
<th>Stainless Steel</th>
<th>Aluminum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>8055.0 kg/m³</td>
<td>2702.0 kg/m³</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>480.0 J/kg K</td>
<td>903.0 J/kg K</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>15.1 W/m K</td>
<td>237.0 W/m K</td>
</tr>
</tbody>
</table>

Table 3.1: Metal Thermal Properties (Incropera et al., 2007)
The air volume region on the cold side of the solid heat exchanger is shown in Figure 3.5. The volume fills one convolution and has direct interfaces (highlighted in purple) with the thermal buffer tube and the regenerator mesh.
3.3.3 Regenerator Region

The regenerator region is modeled after the regenerator mesh that allows the thermoacoustic effect to occur and the acoustic portion of the stove to operate. The regenerator mesh is a 10 mm thick stack of layered sheets of stainless steel screen. To model this region accurately, a porous module region type was used in Star CCM+. Star CCM+ employs a source term in the momentum equation to simulate the effects of flow through a porous region. Prior work experimentally calculated the relationship between velocity and pressure that allows for the inertia and viscous effects to be quantified (Khoo et al., 2012). The values were divided by the 10mm thickness to put each term in a per length value as expected by Star CCM+. Those values are

Figure 3.6: Photo and CFD Model of Regenerator Region
listed in Table 3.2. All thermal properties of the porous region were identical to those outlined for stainless steel in Table 3.1.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Porosity</td>
<td>0.792</td>
</tr>
<tr>
<td>Porous Inertial Resistance</td>
<td>743.0 kg/m(^4)</td>
</tr>
<tr>
<td>Viscous Thermal Resistance</td>
<td>1740.0 kg/m(^3) (-s)</td>
</tr>
</tbody>
</table>

Table 3.2: Regenerator Flow Properties (Khoo et al., 2012)

### 3.3.4 Radiator Regions

The regenerator is only able to produce the thermoacoustic effect if a temperature difference is created in the direction of the flow through the regenerator mesh. To create a cold side of the mesh in the as-built stove, a modified car radiator was installed on top of the regenerator and cooled with water circulated by an external pump. Experimental data had shown that the temperature on the cold side of the regenerator is not constant during operation. For this reason, the radiator needed to be modeled as opposed to simply enforcing a constant temperature boundary. To model the radiator, the volume was broken up into two porous regions: one region for the air and another for the water. An additional region was necessary to define the temperature and mass flow rates of the incoming water that fed the porous water region. The highlighted regions are shown in Figure 3.7. The air core inputs were dictated by the conditions of the flow field from other regions, as the radiator region has a direct interface to the regenerator region. The region must have all properties defined related to the porosity, viscous, and inertia dissipation, and solid
The values for these parameters were assumed to be constant and are listed in Table 3.3. The solid metal used in the radiator simulation was brass.

<table>
<thead>
<tr>
<th>Property</th>
<th>Air</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Porosity</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Porous Inertial Resistance</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X Direction</td>
<td>10000.0 kg/m$^4$</td>
<td>1.0E8 kg/m$^4$</td>
</tr>
<tr>
<td>Y Direction</td>
<td>10000.0 kg/m$^4$</td>
<td>200000.0 kg/m$^4$</td>
</tr>
<tr>
<td>Z Direction</td>
<td>90.0 kg/m$^4$</td>
<td>1.0E8 kg/m$^4$</td>
</tr>
<tr>
<td>Viscous Thermal Resistance</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X Direction</td>
<td>100000.0 kg/m$^3$ – s</td>
<td>1000000.0 kg/m$^3$ – s</td>
</tr>
<tr>
<td>Y Direction</td>
<td>100000.0 kg/m$^3$ – s</td>
<td>55000.0 kg/m$^3$ – s</td>
</tr>
<tr>
<td>Z Direction</td>
<td>450.0 kg/m$^3$ – s</td>
<td>1000000.0 kg/m$^3$ – s</td>
</tr>
</tbody>
</table>

Table 3.3: Porosity Values for Radiator

The only constant parameters used for the solid thermal calculations was thermal conductivity at a value of 109 W/m – K (Incropera et al., 2007).
A crucial criterion necessary for the two regions to act like a radiator is the heat exchanger interface. The model chosen was a dual stream simulator with actual heat transfer. This means the energy and flow equations are used to calculate the temperature difference between the regions and an overall user-specified heat transfer coefficient (U) was used to calculate energy transfer. Since the radiator is a basic water cooling air radiator, the U value of 120W/m²K was used (CheResources, 2012). A value of 120 was on the higher end for liquid to air heat transfer, but is considered conservative for this simulation to prevent the cold side of the regenerator from rising in temperature. Mass flow rate of the water was experimentally determined to be 0.8kg/s in the functioning SCORE stove. The temperature of incoming water was 20°C with a velocity of 0.129m/s via \( V = \frac{\dot{m}}{\rho A} \) with A being the cross sectional area of the water input slice of the simulation. The water core was modeled as having a pressure outlet that was constant at 20°C and 0.0 Pa gauge pressure. All other surfaces were modeled as adiabatic walls.

### 3.3.5 Thermal Buffer Tube and Acoustic Input Region

The thermal buffer and acoustic input regions simulate the feedback loop tubes that allow the traveling wave to resonate. The acoustic input tube is referred to as the ‘Lin Alt’ region in Figure 3.8. This is where the linear alternator would be located on one of the two units. The simulation assumes this modeled unit is the one with no linear alternator installed. The shape of the region is the same whether or not the linear alternator is present. The LinAlt region during the steady state startup simulation had an input of the direct interface with the air core region of the radiator and a pressure outlet on top. The other region highlighted in Figure
3.8 is the thermal buffer tube. As modeled, the geometry is a simple tube with no heat transfer or cooling of any sort. The values for the pressure outlet in each region were derived using the same method that was employed for the hot exhaust gases outlet. A single iteration was run under ambient conditions to allow a gravity driven pressure distribution to form, then boundary conditions were recorded and applied as constant input values. The pressure outlet conditions were only applied to the LinAlt region during the steady state, startup simulation. All other surfaces were deemed to be adiabatic walls. The only condition missing from the simulation was the inlet condition used to simulate acoustical waves during the steady state operation of the stove. Acoustics were introduced through the pressure outlet on top of the LinAlt region. The outlet was driven with an acoustical pressure input and related temperatures. The equation for pressure at the boundary is shown in Equation 3.1.

\[ P = P_o + A \sin(2 \pi f t) \]  

(3.1)
Where,

\[ P_o = \text{Mean Pressure} \]
\[ A = \text{Sound Wave Amplitude} \]
\[ f = \text{Acoustic Frequency} \]
\[ t = \text{Time} \]

Since the most likely application for the stove is going to be running at ambient pressure, \( P_o \) was set to an assumed value of 101,325 Pa. Based on experimental results, the peak amplitude and therefore \( A \) for simulated sound waves in the stove was assumed to be 3200 Pa. The frequency \( f \) for the experimental data used for this simulation was 72 Hz. Related temperatures from the values for the pressure, assuming an ideal gas, were derived assuming the volume was a portion of the long 100mm diameter feedback tube.

### 3.4 Physics and Meshing

Once all the geometry and boundary conditions had been detailed, the physics and related meshing were simulated. For the physics, regions were grouped based upon volumes that had the same physics. Meshing regions were grouped based upon the necessary level of refinement necessary to capture all required phenomena.

#### 3.4.1 Air Regions Physics

The physics applied to the air regions are as follows:

- Ideal Gas (compressible)
- Coupled Flow
The air needed to be modeled as compressible to simulate both density driven natural convection as well as the pressure oscillations during the acoustic wave generation. Coupled flow and energy were used to guarantee a more accurate solution given the compressible nature of the air. Gravity totaled $-9.8 \text{ m/s}^2$ at a $45^\circ$ angle to the HHX surface. Given the high temperatures associated with the stove, radiation was crucial to simulate operation. The surface-to-surface radiation was chosen as the radiation model since air was assumed not to participate in radiation. Furthermore, the surface-to-surface model after calculation of the viewfactors between surfaces required less memory to run than if the air was included. Radiation was not used on the exhaust gas region (HHX Hot) of the HHX to reduce computational time and computer memory. The K-epsilon turbulence model was used with the default values for air as the working fluid. Atmospheric conditions were assumed to be 101,325 Pa with 20°C temperature and zero initial velocity.

### 3.4.2 Solid Region Physics

The following CFD physics models were used in the solid, convoluted heat exchanger:

- Solid Material
- Coupled Energy
• Constant Density (Coupled Flow)
• Material - UNSS30200 Stainless Steel or Pure Aluminum (simulation specific)

Coupled energy and flow models were included to increase accuracy at the cost of computational effort. The temperatures seen by the solid were not hot enough to cause changes in material properties so all material properties were assumed constant.

3.4.3 Water Region Physics

The following CFD physics models were used in the water regions of the simulation:

• Coupled Energy
• Constant Density (Coupled Flow)
• Laminar Flow
• Working Fluid - Water

As with the solid region, coupled energy and flow modules were chosen for accuracy. Fluid flow in the water core was well within the laminar range and maintained a near constant temperature making changes in fluid properties negligible.

3.4.4 Meshing

Meshing for all regions, except the solid heat exchanger, was accomplished using a polyhedral mesh. Phenomena in the heat exchanger regions were the most important so they were meshed with a base size of 3.0 mm and included two prism layers to calculate any wall parameters. Base size in Star CCM+ is the global value that all other parameters are scaled upon. In this case, the maximum mesh size was 3.0 mm with refinement being controlled by the default meshing factors for fluid regions. The
remaining fluid and porous regions were meshed with a base size of 8.0 mm. Surface refinement to match different mesh sizes was not necessary in Star CCM+ as the code has built in features to handle mesh difference (CD-adapco, 2012). The solid heat exchanger was meshed with a mesh module designed to reduce the number of cells in thin regions. With the thickness of the solid heat exchanger wall being 0.9 mm regardless of the base size or other parameters, the mesh would be limited to a very high level number of cells if using standard polyhedral shapes. In order to reduce the number of cells, a prismatic or square type volume mesh referred to as a thin mesher was used instead. With a combination of a fine polyhedral, coarse polyhedral, and thin mesh, the overall volume mesh contained 546,486 cells as presented in Figure 3.9. The total cell count is not necessarily large, but the inclusion of radiation, gravity, coupled flow and energy, and compressible flow makes the simulation computationally heavy for the number of cells. For complete breakdown of the meshes, please refer to Appendix A.

![Image](image.jpg)

Figure 3.9: Regions with Mesh Overlay
3.5 Solvers and Convergence Criteria

The final task was to assign solver criteria to handle the matrix of equations created by the physics models and boundary conditions. The simulation was split into two sequential parts. The first portion was performed as a steady state case without the use of any acoustical boundary conditions. The second simulation used the solution of the first to then run a time-dependent implicit solution scheme for several acoustical oscillations with the pressure outlet on top of the LinAlt feedback tube region. One important variable used in all numerical schemes was the Courant number. The Courant number is the ratio of the time it takes for a flow condition to travel divided by a specified length. It is defined in Equation 3.2. The number dictated what resolution in relation to time and space was needed in order to either accurately describe the phenomena or to prevent nonlinear problems from causing divergence, most often early in the simulation for steady state simulations or at each time step for transient simulations.

\[ V = \frac{v \Delta t}{\Delta x} \quad (3.2) \]

Where,
\( v \)= Local Velocity
\( t \)= Time
\( x \)= Position

3.5.1 Steady State

The steady state simulation was run using a Courant number of 75. A value of 75 is much larger than normal, but given the need for the hot inlet temperature of 880°C
to reach all regions of the model, the temperature needed to move quickly into the mesh in order to shorten the number of iterations to convergence. Furthermore, with the input having such a small impact on everything except temperature, nonlinear phenomena were expected to be minimal. An ‘F’ cycle for the multi-grid solver was also used to achieve solution quicker by making the mesh more coarse and then interpolating the solution back to a fine mesh throughout the march to convergence. Convergence in the steady state case was considered to have been achieved when all solver residual values dropped below a certain threshold, meaning the changes from one iteration to the next were insignificant. The threshold was chosen to be when residuals for momentum, continuity, and turbulence dropped below a normalized value of $10 \times 10^{-6}$, combined with the maximum temperature on the underside of the solid heat exchanger not changing more than 0.5°C for a total of 10 iterations.

3.5.2 Unsteady Implicit

The implicit unsteady simulation was performed with a time step of $8.6806 \times 10^{-4}$ seconds. This time step allowed 16 discrete points to be generated for each period of the 72Hz acoustical wave. The solver was run as 2nd order accurate in time with a Courant number of 13. A Courant ramp input was used to drop the Courant number to 2.0 and then linearly rise to 13 over the first four iterations to control for nonlinearities potentially causing the solution to diverge. The solver used a traditional ‘V’ cycle solving scheme where the solver is run from a fine mesh to coarse and back to fine. This only differs from the ‘F’ cycle by returning to the finest mesh every time as opposed to every other in the ‘F’ cycle. The ‘V’ cycle is slower but allows for the solution to capture the detail at each time step. With taking 20 discrete steps
to compile the phenomena of the sound wave, non-normalized residuals were deemed convergent when they both dropped below $10 \times 10^{-6}$ and completed a minimum of 25 iterations. These convergent criteria meant that the residuals were absolutely low and air flow solutions were able to accurately march from one time step to the next.
Chapter 4: Results and Observations

Simulations outlined in Chapter 3 were performed, and the results were illustrated using the post-processing features of Star CCM+. In order to characterize heat transfer in the SCORE Demo2 stove prototype, values such as flow temperature and velocity were captured and displayed graphically. With the focus of this study being upon the convoluted heat exchanger, most data analyses involved documenting spatial values for temperature and heat fluxes. Results were generated for both the actual as-built stainless steel material as well as a simulated prediction of the effect of substituting aluminum for the convoluted heat exchanger material. Furthermore, all results were presented for the preliminary steady state, natural convection conditions and then for the overlain acoustic, time dependent conditions. In addition to the heat transfer study, results of the numerical simulation for transmission losses through the stove geometry were also generated.

4.1 Steady State

4.1.1 Flow Temperature and Velocity Profiles

The streamlines of the airflow and accompanying air and heat exchanger temperatures for both stainless steel and aluminum are shown in Figure 4.1. The colors indicate temperature and show the contact location of the rising hot air impinging
on the heat exchanger’s convolutions, moving along the sides, and exiting out the exhaust duct. The results indicate that air loses most of its heat near the location of initial contact with the heat exchanger. Air that receives the most contact with the metal loses the most heat by the time it exits the exhaust duct. Streamlines located near the top of the convolution best demonstrate this phenomenon. Although the physics modeled in the simulation were expected to be turbulent, the assumption of uniform and normal to the boundary velocity inlet air flow resulted in almost purely laminar flow. This laminar behavior was assumed true for the small section modeled, but overall air flow should be highly turbulent upon impinging on the convolution. Given the low velocities, the effect on heat transfer was assumed to be minimal, but further simulation is required to quantify and confirm this assumption.

![Figure 4.1: Air Temperature and Streamlines at the Hot Side of the Convoluted Heat Exchanger](image)

(a) Stainless Steel  
(b) Aluminum

Figure 4.1: Air Temperature and Streamlines at the Hot Side of the Convoluted Heat Exchanger

The air flow on the opposite, cold side of the heat exchanger is shown in Figure 4.2. The flow on this side is entirely driven by buoyancy forces. The resultant natural
convection means that as air temperature rises, density decreases causing the less dense air to rise in the direction opposite from gravity. Because the heat exchanger is sloped at a 45 degree angle from horizontal, air flows up along the convolutions until the temperature difference between an air molecule and its neighbors is great enough to force separation. For this reason, the greater the initial air temperature the easier for the flow to separate from the heat exchanger’s metal plate surface. Figure 4.2(a) shows that there is a higher temperature gradient with stainless steel which resulted in air leaving the metal surface along flow lines distributed across the full length of the convolution. The aluminum case in Figure 4.2(b) shows a much weaker temperature gradient resulting in the flow almost entirely remaining on the trough of the convolution until collecting at the top of the heat exchanger. In both cases, air moved upwards, but the stainless steel case is more likely to heat the regenerator mesh more evenly since air flowed into it over a larger surface area. The aluminum by contrast will cause the top of the regenerator to heat first. The effect of this on the startup of the thermoacoustic effect depends on the onset temperature difference necessary to start thermoacoustic waves for a given design. The lower the temperature difference necessary to initialize the thermoacoustic effect, the less it will matter whether stainless steel or aluminum is used. The onset temperature difference of 160°C seen in the SCORE Demo2 stove (Chen et al., 2011) is easily attainable in both cases, suggesting that material selection will have little effect on startup times. Applications needing a higher temperature difference to drive thermoacoustics will benefit from the use of stainless steel since the convective flows created by using stainless will cause the regenerator to reach the onset temperature difference over a larger surface area more quickly than aluminum.
Figure 4.2: Air Temperature and Streamlines at the Cold Side of the Convoluted Heat Exchanger

4.1.2 Exchanger Temperature Profiles

The difference in the flow field temperatures seen in Figure 4.2 is a direct effect of the temperature profile at the heat exchanger surface. Figure 4.3 shows the profile for both the stainless steel and aluminum simulations. In both cases, the highest temperature occurred at the location where the hot inlet air first contacts the plate.
The temperature then spread out from that location in the direction of the air flow. The simulations show that the air encountering the greatest obstruction will have more time to transfer heat and will also have the momentum to decrease fluid boundary layers thereby increasing heat transfer at the surface. To use this observation to improve heat transfer, future design alterations to the hot heat exchanger should attempt to include geometries that slow and/or deflect the incoming air. Such geometries could be created by inserting material (baffles, steel wool, etc.) into the air path or by decreasing the cross sectional area.

The difference in temperature profile between stainless steel and aluminum is the result of lower thermal conductivity of stainless steel trapping more heat closer to the initial point of contact. This created a higher temperature difference in the air flow on the opposite side, resulting in the more evenly dispersed natural convection flow of Figure 4.2(a). The low thermal conductivity of the stainless steel also results in heat not easily conducting to the edges of the heat exchanger plate. The prevention of heat flow to the edges was intentionally included in the Demo2 design by the SCORE team to prevent leakage of heat which could thermally short circuit the regenerator mesh without the use of expensive insulating materials. Figure 4.3(a) shows that conduction to the edge is low but is still a concern with edge temperatures ranging from 190°C to 300°C depending upon the location. For this reason, the design could potentially be improved by including a cheap and easy-to-install way to insulate the heat exchanger from the stove’s frame.

The temperature profile for the aluminum material followed a very similar pattern, with the highest temperature occurring at the point of initial initial contact.
and gradually cooling in the direction of the air flow. The patterns for both metals appear very similar in Figure 4.3, but the temperature values are quite different. As visualized in Figure 4.4 by setting the temperature scale to the same range, aluminum’s entire heat exchanger surface temperature is relatively even since all points fall in the range of 360°C to 414°C. The more uniform temperature distribution also means the edges on the aluminum heat exchanger had a higher temperature than the edges of stainless steel. The use of aluminum would require greater insulation than stainless steel. Installing insulation adds cost and complexity that the SCORE team is seeking to avoid. Based on these simulation results, neither natural convection air flows nor temperature distributions provide sufficient evidence to justify switching to aluminum.
Following analysis of temperature profiles on the underside of the heat exchanger as explored in sections 4.1.1 and 4.1.2, the focus shifted to the cold, acoustic side of the heat exchanger. The temperature profile is identical to that outlined in Figure 4.3 on the opposite side, given the thin 0.9 mm thickness of the sheet metal. With temperatures already known, the effect of greater thermal conductivity on heat transfer on the acoustic heat exchanger surface was quantified using parameters such as heat fluxes, heat transfer coefficients, and Nusselt numbers. When describing these parameters in the Star CCM+ software, the sign convention is positive for out of the region, except for the case of radiation where positive is into the region. The sign convention is stated in detail in each figure caption.

The total boundary heat flux in Figure 4.5 is shown from the perspective of the air region between the convoluted heat exchanger and the regenerator mesh. Because the
mass represented is the air, negative values represent heat flux coming into the fluid
with positive flux being back into the heat exchanger. Looking at Figure 4.5(a) for
the stainless steel case, the blue represents the greatest flux emitted into the air with
the small amounts of red denoting heat flux emitted back into the heat exchanger.
This sign and color convention also applies to the aluminum results in Figure 4.5(b).

The total heat flux gradient at the contact interface between the cold air and
the heat exchanger in Figure 4.5 shows similar patterns with blue areas indicating
large amounts of heat being removed in areas that do not correlate easily with the
observed temperature distribution. The greatest flux would be expected to occur at
locations of greatest temperature, not in the dark blue spots illustrated in Figure 4.5.
The physics missing from the picture is the effect of air flow. Figure 4.6 overlays
velocity vectors of the air flow with the color of those vectors indicating temperature.
Additional arrows in red were inserted to show the general motion of the air. The red

![Figure 4.5: Total Boundary Heat Flux](image)

(a) Stainless Steel  
(b) Aluminum

Figure 4.5: Total Boundary Heat Flux
arrows show that air is circulating in the regenerator mesh and radiator which leads to a much larger air flow impinging on the metal at the locations of the dark blue circles in Figure 4.5. The air flow cools the surface and thus was the source of the locally high values for heat flux. The aluminum simulation showed that circulation of air flow also occurred. Greater circulation occurred at the bottom of the heat exchanger surface as the values jump to 9550\(W/m^2\) just below the junction with the output thermal buffer tube. The greater air circulation along the heat exchanger surface, up into the thermal buffer tube, and back down is the result of higher temperatures near the edge of the heat exchanger surface. The increased air circulation and related heat fluxes are an undesired side effect of the use of aluminum, and for the natural convection portion of operation, increased heat losses. The areas with positive values for heat transfer (dark red in Figure 4.5) show areas where the air is actually heating the surface. This occurs in locations where the temperature of the air, after being heated at locations further down the heat exchanger surface, was higher than the temperature of the metal. This occurs mostly on the sides of the trough which are weakly heated by the exhaust gases underneath. Although the areal extent of the dark red zone is far greater in Figure 4.5(b) than Figure 4.5(a), the total quantity of heat flux is far greater in the stainless steel scenario since the edge temperatures were lower.

The heat flux illustrated in Figure 4.5 was the total heat flux at the boundary between air and metal. Given temperatures rising above 500°C, the contribution of radiation was thought to be a significant contributor to overall heat transfer. The contribution of air flow led to the surface pattern shown in 4.5, but, as shown in
Figure 4.6: Stainless Total Heat Flux With Velocity Overlay

Figure 4.7, the contribution of radiation was also significant depending upon location. The radiation pattern is very similar to the total heat flux’s spatial distribution suggesting that the main driver of heat transfer during natural convection was, in fact, radiation. Radiation was greatest on the top since the heat exchanger surface is very close to the regenerator mesh surface above it. This means nearly all radiation emitted by the heat exchanger on its top was absorbed by the surface directly above it. Radiation from locations in the trough was much lower since those locations were able to transmit only small amounts of radiation to cool surfaces. The ability to transfer heat to cool surfaces was a function of a surface’s ability to ‘see’ cool surfaces, defined as viewfactors in the surface-to-surface radiation model. This was true even at the location where the temperature was hottest, i.e., in the temperature ranges tested during the simulation, viewfactors were significantly more important.
than temperature during startup. To decrease startup time, designs should include methods to raise the temperature on surfaces closest to the mesh. Any other local variations besides viewfactors shown in Figure 4.9(a) are the effect of spatial temperature differences which easily correlate to the temperature distributions in Figure 4.4.

The visualization of total and radiative heat transfer is effective at enhancing understanding of the modes of heat transfer occurring in the stove, but do not easily provide design parameters to aid current and future design calculations. Figure 4.8 shows the distribution of heat transfer coefficients that could be used in convective equations. Overall, the values are very low with locations at or below $4 \text{W/m}^2 - \text{K}$. Locations with larger heat transfer coefficients are areas that had the greatest local air flow. In terms of natural convection only, designs attempting to shorten startup time should try to localize re-circulation patterns to foster better heat transfer inside the heat exchanger frame and to prevent heat from escaping out the feedback tubes. The

Figure 4.7: Boundary Radiation Heat Flux
very slow turbulent air circulation visualized in Figure 4.6 can double or triple the heat transfer coefficients shown in Figure 4.8. The remaining parameter that has not been considered is the effect of conduction on heat flux. With radiation being the main method prior to acoustic startup, the contribution of convection is shown to be low. With convection already being low, the best way to quantify the effect of conduction was to compare it to convection. The Nusselt number is a dimensionless parameter that is the ratio of convective heat transfer to conductive heat transfer. The ratio means that Nusselt numbers larger than 1 indicate a mostly convective contribution. Nusselt numbers less than 1 indicates a mostly conduction dominant heat transfer. Convection and conduction are equal when the Nusselt number is 1. As shown in Figure 4.9, values for the Nusselt numbers are much larger than one in nearly all locations. The conclusion is that conduction is vary small in comparison to convection, which in itself is small, compared to radiation. This means that conduction can
always be neglected. The fact that thermal conductivity of air is so low supports the simulation’s conclusion, even at the high temperatures seen in this study.

![Image of Nusselt Number simulation for Stainless Steel and Aluminum](image)

(a) Stainless Steel  (b) Aluminum

Figure 4.9: Boundary Nusselt Numbers

4.1.4 Calculated Energy Capture Efficiency

As noted in section 4.1.1, the uniform velocity inlet to the simulated system is assumed to be truly uniform only for the very center of the ducted exhaust gases. In reality, the incoming gases will have vortex-influenced momentum that will cause the gases to tumble along the underside of the heat exchanger causing an increase in heat transfer. The fact that this aspect of the input was not included in the simulation means that any attempt to quantify the amount of energy captured from the incoming air would be numerically inaccurate. Using the simulation to compare the effect of another variable directly related to energy would be a reasonable, however. Therefore, the different heat exchanger materials tested in this simulation was compared for their
ability to capture heat given identical flow and geometric conditions. Power values at
the gas inlet and exhaust outlet were tabulated in Table 4.1. Although aluminum has
a much higher heat transfer coefficient, it was only able to capture 3.3% more Watts
than stainless steel. The low performance gain, combined with the risk of extra heat
losses and fears over the robustness of aluminum in the presence of exhaust gases,
suggests that the costs incurred by switching to aluminum would not be justified.

<table>
<thead>
<tr>
<th>Material</th>
<th>Input</th>
<th>Output</th>
<th>Percent Captured</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stainless Steel</td>
<td>0.656</td>
<td>0.505</td>
<td>23.0%</td>
</tr>
<tr>
<td>Aluminum</td>
<td>0.656</td>
<td>0.483</td>
<td>26.3%</td>
</tr>
</tbody>
</table>

Table 4.1: Simulated Energy Into Heat Exchanger
4.2 Acoustics

During operation of the SCORE Demo2 stove, heat transfer will occur in the presence of acoustical waves. The acoustic conditions used in this simulation were 72Hz with an amplitude of 3200Pa. The frequency chosen was specified in the construction of the Demo2 prototype, but the peak pressure amplitude is a function of the thermoacoustic effect in the stove. Both the 72 Hz and the 3200 Pa peak pressure values were based on experimental data provided by the SCORE team (Riley, 2012). This section outlines the effect of overlaying an acoustic signal over the heat transfer simulation detailed in section 4.1. The sine wave acoustic input is time dependent, so most parameters were quantified at four distinct time steps. The time step data displayed in the following figures were captured during the second or third period of oscillation, with sixteen steps equaling one complete oscillation. With one full period being divided into sixteen steps, intervals of four time steps were chosen to show the conditions at a) mean, b) maximum, c) mean, and d) minimum input pressures. Figure 4.10 provides a graphical representation of where each time step was located on the input sine wave.

4.2.1 Flow Temperature and Pressure Profiles

When analyzing acoustics, it is important to remember that the wave travels through the device and not the air particles themselves. In acoustic applications, each air particle moves a short distance dependent upon acoustic and thermal conditions. In order to better comprehend how far particles will move, Figure 4.11 presents the relationship between peak particle displacement and air temperature. As the air temperature increases at a constant pressure, an air molecule will oscillate back and
forth over a larger distance. This is important to understand how air at a higher temperature is able to move relatively large distances in the simulation compared to cooler air. At 72 Hz the wavelength is so long that under ambient conditions only linear or planar oscillations will occur, however the addition of temperature creates non-linear profiles.

Pressure is the most important parameter in acoustics. Figure 4.12 shows the air pressure in the stove through the centerline cross section at each of the four selected time time steps. The pressure at zero amplitude input in Figure 4.12(a) shows a relatively even pressure field. The pressure is then increased to a maximum of 3200 Pa. Figure 4.12(b) shows a much higher pressure than the input 3200 Pa since pressure will build as the air flow encounters resistance in the radiator and regenerator mesh.
Pressure also builds near the top of the heat exchanger as the air is forced to change direction as it travels down the heat exchanger and out the thermal buffer tube. The pressure profile appears linear with clearly distinguished, uniform pressure regions.

The temperature profile of the air field inside the stove is shown in Figure 4.13. Since the simulation time range used was the second oscillation, a strong thermal profile already exists at the zero amplitude time step in Figure 4.13(a). The air at that elevated temperature was able to travel up to 50 mm as indicated in Figure 4.11, which is farther than the thickness of the space between the heat exchanger and regenerator. This causes hot air to easily enter the regenerator mesh before being cooled by the radiator. The time at the elevated temperature is short lived as the
pulse of air from the radiator cools the field drastically at the subsequent time step in Figure 4.13(b). Halfway through the sine wave in Figure 4.13(c), the pressure field remains relatively unchanged since the time between time steps is too short to facilitate any natural convection. In Figure 4.13(d), the air is moving back to its original location pulling heat from the heat exchanger back up to the regenerator.

The oscillatory motion of the air means that the air near the hot side of the regenerator will oscillate at a mean temperature in between the maximum and minimum temperatures shown in Figure 4.13. The extent to which this oscillating air temperature affects the temperature of the regenerator mesh is dependent on the thermal material properties and thickness of the wire used in the mesh. Given the low thermal conductivity of stainless steel used in the as-built stove, it is unlikely that heat can be able to conduct fast enough to mirror the full range of temperature oscillation. Whether the mesh temperature oscillates at a fraction of the amplitude or remains at the mean conditions was not studied in this simulation. The author suspects the temperature stays at the mean, but further research is necessary to calculate the time constant for conduction in the mesh.

4.2.2 Total and Radiative Heat Fluxes

The important parameters during the acoustic simulation are ones that will change over the same time scale as the excitation source, a 72 Hz sound wave. The period for a 72 Hz sound wave is 0.0138 sec. With such a short time period and acoustic conditions being introduced over the steady state thermal conditions, changes in heat exchanger surface temperatures were assumed negligible. For this reason, heat fluxes
were used as the indicator of the effect the acoustic wave had on interactions between the solid heat exchanger material and adjacent air flow and air temperature.

The boundary heat flux as shown in Figure 4.14 follows an acoustic oscillation pattern similar to the prior Figures. The starting timestep was shifted four timesteps to begin at the third oscillation’s maximum peak instead of the mean as labeled in Figure 4.10. The flux conditions parallel those shown in the temperature figures. The greatest heat flux occurred when the coolest air was impinging upon the heat exchanger surface in Figure 4.14(a). For the remaining time steps, a rather consistent heat flux occurred because the temperature of the air at the surface was relatively close to the metal’s surface temperature. The result indicated that the greatest heat input occurred during the positive portion of the sine wave as air was pushed down from the radiator and into the trough of the heat exchanger. The distance traveled will be on the order of 15 mm, meaning that air in the regenerator does not travel all the way down to the bottom of the heat exchanger trough but will come into contact with the heat exchanger near the top of its convoluted surface. This can be seen in Figure 4.14(a) as the top surface has a heat flux value that is orders of magnitude higher than at the bottom, with the exception of the hottest point. The effect of drawing more heat flux from the plate could be increased if the distance from the top to the bottom of the heat exchanger was decreased. This would allow more air to touch the surface of the heat exchanger as well as bring the hottest temperatures closer to the regenerator mesh. A potential concern, however, is whether the radiator would have the capacity to cool the increase in hot air as well as whether the modified cross sectional area would continue to be large enough to not suppress the pressure wave.
The values for the heat transfer coefficients under acoustic conditions are shown in Figure 4.15. The time steps are similar to those in the pressure and temperature figures where the first time step is at the mean or zero amplitude in the second oscillation of Figure 4.10. The heat transfer coefficients mirror the phenomena represented for the heat flux. Negative values indicate areas where the heat exchanger plate is heating the air, and positive values indicate where the air is heating the plate. The extremes shown near the bottom third on the top of the heat exchanger are due to the effects of higher velocities as air accelerates out from underneath the cross bar separating the regenerator from the thermal buffer tube. The observations about distance into the heat exchanger also apply to results shown in Figure 4.15(b).

To quantify the relative contributions of different modes of heat transfer, the total and radiative heat fluxes were graphed at each point along the midline direction (labeled as ‘length’ in Figure 4.14(a)) of the heat exchanger length for the time steps shown for the third oscillation in Fig 4.10. Graphing this way generated many data points for each location along the length. Although the graphs in Figure 4.16 appear complex, by including all data points the graph was able to show the full range of heat fluxes at each location along the heat exchanger. For example, Figure 4.16(a) at 70mm shows that the heat flux ranged from 40,000 to 130,000 $W/m^2 - K$. Graphing all the data points also showed the relative contribution of radiation to the total heat flux. Most of the locations in all time steps for a given position showed the total heat flux to be several orders of magnitude greater than the contribution of radiation. That was in sharp contrast to the conditions that existed in the natural convection case where radiation was the main driver of heat transfer in the absence of acoustic inputs.
The total heat flux was greater than radiation, but determining the relative contribution of conduction versus convection required additional analysis. The Nusselt numbers in Figure 4.17, were extremely high, suggesting that the contribution of conduction to total heat transfer should be assumed have been negligible.
4.3 Pressure Loss

Upon analysis of results from the pressure oscillations over time, there was concern that the geometry of the SCORE Demo2 prototype stove was causing transmission losses as the acoustic waves passed through the system. Given that the acoustic sound wave would eventually be converted to electricity in a linear alternator, losses in acoustics due to geometry should be minimized. To quantify the transmission loss, a simulation was done using ambient temperatures. Furthermore, the viscous losses included in the porous regions of the regenerator mesh and radiator were removed so that any calculated losses would be solely geometry-induced. Gravity was also removed from the simulation. The CFD model mesh of the feedback input and output tubes was extended to ensure that reflective waves would be avoided for one acoustic input pulse. The results of the simulation are shown in Figure 4.18.

The surface average pressure was measured just after the inlet and just beyond the circulating air flow in the outlet tube, then graphed in Figure 4.19. The peak of the pulse values are tabulated in Table 4.2. The difference between peak input and outlet pressures indicated a peak transmission loss of 807 Pa which translates to a loss of 26.12%. No Fourier transforms were considered necessary as the 72 Hz signal was the only input and the only frequency of interest.

<table>
<thead>
<tr>
<th>Max Input (Pa)</th>
<th>Max Output (Pa)</th>
<th>Pressure Loss (Pa)</th>
<th>Percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>3089.71</td>
<td>2281.98</td>
<td>807.09</td>
<td>26.12</td>
</tr>
</tbody>
</table>

Table 4.2: Pressure Input and Output of Demo2 Geometry
A sound wave at a frequency of 72 Hz has such a long wavelength that any resonant effects that might cancel out the signal would not occur. Furthermore, the geometry does not contain any cavities that could produce Helmholtz resonance. Helmholtz resonance is resonance in a nearly closed cavity with one opening, and when used as a silencer is effective at damping long wavelengths (Selamet and Ji, 2000). With the possibility of losses due to general acoustic silencer theory discounted, the source for the pressure loss appears to be from a swift change in cross sectional area and airflow direction causing turbulence. Attempting any massive redesigns of the SCORE stove to smooth out the air flow would necessitate a reworking of the entire heat transfer theory developed in this study. To avoid this, a simple fix for the device would be to ensure that the opening from the heat exchanger case to the thermal tube is as large as the acoustic feedback pipe to reduce the viscous losses created by air flow being forced through such a small opening. Other attempts such as increasing the height of the convolutions in the heat exchanger would decrease loss but have a negative effect on the overall rate of heat transfer.
Figure 4.12: Flow Field Pressures at Discrete Acoustic Time Steps

(a) Zero Amplitude  
(b) Positive Peak Amplitude  
(c) Zero Amplitude  
(d) Negative Peak Amplitude
Figure 4.13: Flow Field Temperatures at Discrete Time Steps
Figure 4.14: Spatial Boundary Heat Fluxes at Discrete Acoustic Time Steps
Figure 4.15: Heat Transfer Coefficients at Discrete Acoustic Time Steps
Figure 4.16: Total and Radiative Boundary Heat Flux at Discrete Acoustic Time Steps Along the Length of the Heat Exchanger.
Figure 4.17: Nusselt Numbers at Discrete Acoustic Time Steps
Figure 4.18: Pressure Wave in Geometry at Discrete Acoustic Time Steps and Wire-frame Schematic
Figure 4.19: Pressure Input and Output of Demo2 Geometry.
Chapter 5: Conclusions and Future Work

The CFD simulation of the components of the SCORE Demo2 prototype stove gave insight into and numerical values for the heat transfer phenomenon associated with the stove’s thermoacoustic operation. A summary of conclusions is outlined in the first section of this chapter. The remaining portions discuss potential areas of interest for future research. Work must be done, first to improve and validate the simulation model, and then to make design changes to the Demo2 stove based on the results of the simulation. This discussion of future work in this chapter is split into two sections. The first section provides recommendations directly applicable to the SCORE project that focus on research directions for the Demo2 prototype. The second section is written from a larger body of knowledge point of view on how the physics of thermo-acoustics could be studied differently and how to improve CFD simulations in the future.

5.1 Summary of Conclusions

The following is a summary of conclusions from this study, separated into the steady state, transient acoustic, and transient acoustic pressure loss simulations.
5.1.1 Steady State

- The location of the highest temperature on the heat exchanger was where exhaust gases impinged on the heat exchanger.
- Radiation was the predominant mode of heat transfer. Surfaces closest to the regenerator mesh transferred the most heat flux.
- Conduction through the air was negligible.
- Stainless steel, as the heat exchanger material, provided more even heating of the regenerator mesh compared to aluminum.
- The use of aluminum spread out the heat exchanger’s temperature profile and caused unnecessary increases in convection. Furthermore, the simulated gain in captured energy was a mere 3.3%. Therefore the use of aluminum was not recommended.
- The ability to transfer more heat was a function of temperature distribution on the heat exchanger.
- The geometry of the current design does not capture enough heat from the exhaust gases.

5.1.2 Transient Acoustic

- Convection was the dominant mode of heat transfer in the presence of acoustical waves.
- The acoustic oscillations pulled heat up from the heat exchanger to the regenerator mesh during the negative portion of the sound wave.
• The temperature of the air around the regenerator mesh oscillated between the maximum observed at the heat exchanger and the minimum observed at the radiator surface.

• More heat on the exhaust gas side must reach the top of the heat exchanger convolutions to increase heat transfer.

• The time scale for conduction in the regenerator mesh needs further investigation to see if there is an effect of the oscillating temperature due to acoustics.

5.1.3 Transient Acoustic Pressure Loss

• Pressure loss through the geometry alone was 26%.

• High turbulence occurred near the attachment of the thermal buffer tube and HHX case.

• Losses occur due to large changes in cross sectional area.

5.2 SCORE Project

The first thing the project could do to utilize the results of this simulation is to validate that the results are consistent with the physical operation of the stove. Experimental data were used to estimate inputs and qualitatively check simulations, but more values are needed for a setup identical to the one modeled in this study. Specifically, operating the stove in the fully installed state while using gas fuels would be the experimental validation setup. Operating in fully installed state includes all the masonry and ducting that could affect the conditions at the inlet duct to the Demo2. Gas fuels should also be used for validation since the burning of biomass creates conditions are are heavily chaotic. The experiment should use a hot wire anemometer.
to measure time dependent air velocity at the inlet duct to ensure the modeled inlet velocity of 0.43 m/s is within the proper range. Velocity data may also give insight into whether the laminar flow assumption is correct. Furthermore, thermocouple locations need to be more carefully chosen to allow for the best correlation between collected data and simulation results. The temperature distributions on the heat exchanger and in the air vary greatly with location in space, so conditions set in the simulation must exactly match the positions and values in the experiment.

If the data collected are similar to the inputs and results of this study, then a few steps should be done to improve heat transfer in the Demo2 stove design. First, loose steel wool or horizontal fins should be inserted on the exhaust gas side of the convoluted heat exchanger to capture more heat. Including either of these pieces will create more surface area thus increasing the time that hot exhaust gases are in contact with the heat exchanger. Future design iterations should attempt to decrease the number and height of convolutions to move the higher temperatures closer to the regenerator mesh. The obvious trade off is potential reduction in cross sectional area for the acoustics to pass, so maintaining similar cross sectional area is suggested. Simulations can be run for future designs to give insight into changes in physics caused by the design changes. Additional simulations can easily be run for changes in boundary conditions, material properties, or the physics to be modeled. The most obvious choice is running the model with different temperature inputs to see how that affects heat transfer in the rest of the stove. Changes in geometry will cause running new simulations to be more involved since, outside of running the simulations, recreating the volume mesh is the most labor intensive step of CFD modeling. Future runs should only be attempted after validation of inputs.
If a current or future member of the SCORE team would like to look into improving or re-creating the model in this study, there are a few areas that would be of greatest importance. One area of interest that was not modeled in detail for this study was the regenerator mesh. Meshes are modeled as porous regions during typical CFD applications, however such porous elements do not have the level of refinement necessary to make precise predictions about mesh parameters. As was done in this simulation, most porous regions in CFD codes are only concerned about their effect on the air flow. Focus is placed on how the introduction of a porous region affects the velocity, pressure, and dissipation of the fluid flow. The handling of thermal parameters in this study was as a one-way coupling. This type of coupling means that the air will heat or cool the mesh, but the reverse will not happen. In thermoacoustic applications, the thermal interaction between the air flow and and the porous region is two-way coupled. Any further study into the mesh and its temperature profile needs careful attention to better understand exactly how the energy equations are being handled in the porous region representing the regenerator mesh.

Another concept that needs to be studied is how to better handle the vastly different time scales of heat transfer and acoustics. The acoustic simulation done in this study shows the impact that the oscillating field had on heat transfer, but only over a short period of time. The simulation was run for test cases for up to nine oscillations, but the thermal properties of the solid regions remained constant over such a short period. Having the ability to validate heat fluxes or heat transfer coefficients in an acoustic environment as a function of temperature would allow for the acoustic simulation to be run with large time steps or as a steady state model. Running in large time steps would allow for the effects of acoustics on surface
temperatures to feed back to inputs and radiative outputs. This would also allow for the chaotic inputs that exist when the stove is heated with biomass combustion to be run, creating more nearly realistic simulations for the heat transfer portion of the model.

If a high level of simulation for the Demo2 prototype stove is desired, it may be of interest to include the acoustic linear alternator in the model. The inclusion of the linear alternator directly is not suggested or necessary if the purpose of the simulation is to focus on heat transfer, but it does become important if the goal is to predict electrical output of the stove. The effects of the alternator should be experimentally captured and then the data should be inserted into a heat transfer simulation in the definition of the boundary conditions. The coupling of how much the linear alternator dissipates the acoustic flow can be modeled by defining the incoming acoustic wave as a function of modified outgoing acoustic properties. The relationship between acoustic pressure loss or dissipation at the linear alternator and the impinging acoustic wave would need to be experimentally determined.

5.3 Simulations and Related Physics

Given that the fields of acoustics and heat transfer were entirely separate disciplines until the creation of thermoacoustic devices, several areas of the interaction between these disciplines need further study especially when applied to a simulation environment. Although many areas need attention, a few would directly improve the use of thermoacoustics in a low tech, fundamental physics environment.
The first area where further fundamental research is necessary is to validate traditional fluid flow and heat transfer principles under different acoustic scenarios. Questions remain about the characterization of the effect of acoustics on fluid and thermal boundary layers as functions of other parameters. Equations that can be derived to describe parameters need to be tested and tabulated to provide easy-to-use correlations that currently exist only for laminar and turbulent mean flow regimes. Specific areas related to the SCORE project include determining how parameters change based on both high temperature and low frequency acoustics. A solid literature review combined with an attempt at improving the communication between traditionally separate acoustic, electrical, and mechanical engineering disciplines would allow for future designers to more easily navigate the complex and interdisciplinary world of thermoacoustics.

The main area of recommended future research related to CFD simulation of thermoacoustic devices is the creation and validation of techniques to model the thermoacoustic effect in varying types of porous regions. Prior attempts at modeling the thermoacoustic effect were performed using simplified geometries such as flat plates or circular rings (Yu et al., 2010; Saat and Jaworski, 2010). Having the ability to approximate the acoustic contribution in the regenerator mesh as a function of temperature difference would provide a better acoustic simulation. Simulating the actual thermoacoustic effect would be challenging, so an alternative approach may be to quantify the contribution of the thermoacoustic effect so it could be included as a source term in the momentum and energy equations. The creation and validation of such a method would make it universally simple to include the thermoacoustic effect in a wide array of CFD codes.
The last area of great interest for advanced research is the handling of the porous regions in simulations. Porous regions are often modeled as source terms in momentum and energy equations, and their ability to handle convection and radiation heat transfer is limited. Some codes have developed methods to handle such effects, but relating effects to a physical mesh would be important. The most crucial aspect of thermoacoustic devices is the temperature difference across the mesh, so developing techniques to better model and visualize conditions in the porous region would be a huge step forward.

There are other miscellaneous topics related to simulation that merit further research. Validation of turbulence models is necessary to ensure that predicted dissipation values are accurate. Furthermore, the optimization of mesh types and sizes needs attention to reduce the simulation size while maintaining accuracy. Another issue is the time scale difference between acoustics and heat transfer. Methods to accurately use time averaged values and still produce accurate physics need to be developed. Finally, the use of some Lagrangian or particle tracking methods should be perfected in order to better visualize the exact path individual air particles take in a thermoacoustic device. Acoustics causes air particles to move back and forth while the acoustic wave travels throughout the entire looped domain. Using simple scalar fields or velocity vectors to visualize air movement can give the wrong impression that the air itself is actually flowing when, in reality, the particles are only oscillating slightly. Having enhanced visualization tools would allow better communication of these concepts to people with limited acoustic backgrounds, and facilitate collaboration between acoustic, electrical, and mechanical engineering engineers working on innovative thermoacoustic designs.
Appendix A: Data on Meshing

A.1 Mesh Diagnostics

--- Overall Statistics:

-> ENTITY COUNT:

   # Cells: 546486
   # Faces: 2513604
   # Verts: 2502790

-> MESH VALIDITY:

   Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

   Minimum Face Validity: 1.000000e+00
   Maximum Face Validity: 1.000000e+00

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1.00 \leq \text{Face Validity} & 546486 & 100.000\%

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Maximum Volume Change: 1.000000e+00

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--- Computing statistics in Region: HHX
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-> ENTITY COUNT:

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# Verts: 656164

-> EXTENTS:
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y:  [-1.0926e-04,  4.0280e-02] m
z:  [ 5.0190e-04,  2.2198e-02] m

-> MESH VALIDITY:

Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00
Maximum Face Validity: 1.000000e+00

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--- Computing statistics in Region: LinAlt

--- ENTITY COUNT:

- # Cells: 1971
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- # Verts: 11036

--- EXTENTS:

- x: [1.3050e-01, 2.8050e-01] m
- y: [8.3000e-02, 1.6300e-01] m
- z: [-4.0000e-04, 2.3100e-02] m

--- MESH VALIDITY:

Mesh is topologically valid and has no negative volume cells.

--- FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00
Maximum Face Validity: 1.000000e+00

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Minimum Volume Change: 7.312426e-03
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--- Computing statistics in Region: Air Core

-> ENTITY COUNT:

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  # Verts: 5902

-> EXTENTS:

  x: [ 9.6000e-02, 2.9000e-01] m
  y: [ 5.3000e-02, 8.3000e-02] m
z: [-4.0000e-04, 2.3100e-02] m

-> MESH VALIDITY:

Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00

Maximum Face Validity: 1.000000e+00

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-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 1.236141e-02

Maximum Volume Change: 1.000000e+00

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--- Computing statistics in Region: Water Core

-> ENTITY COUNT:
   # Cells: 653
   # Faces: 3450
   # Verts: 3551

-> EXTENTS:
   x: [ 9.6000e-02, 2.9000e-01] m
   y: [ 5.3000e-02, 8.3000e-02] m
   z: [-4.0000e-04, 2.3100e-02] m

-> MESH VALIDITY:
   Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:
Minimum Face Validity: 1.000000e+00
Maximum Face Validity: 1.000000e+00

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-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 1.236141e-02
Maximum Volume Change: 1.000000e+00

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--- Computing statistics in Region: Water Inlet
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-> ENTITY COUNT:

  # Cells: 147
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-> EXTENTS:

  x: [8.4000e-02, 9.6000e-02] m
  y: [5.3000e-02, 8.3000e-02] m
  z: [-4.0000e-04, 2.3100e-02] m

-> MESH VALIDITY:
Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00

Maximum Face Validity: 1.000000e+00

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-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 4.697016e-03

Maximum Volume Change: 1.000000e+00

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--- Computing statistics in Region: Regen

-> ENTITY COUNT:

  # Cells: 5805
  # Faces: 25158
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-> EXTENTS:

  x: [ 9.6000e-02, 2.9000e-01] m
  y: [ 4.3000e-02, 5.3000e-02] m
  z: [-4.0000e-04, 2.3100e-02] m

-> MESH VALIDITY:

  Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00

Maximum Face Validity: 1.000000e+00

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Maximum Volume Change: 1.000000e+00

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<tr>
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<td>1e-02 &lt;=</td>
<td>1e-01</td>
<td></td>
</tr>
<tr>
<td>1e-01 &lt;=</td>
<td>1e+00</td>
<td></td>
</tr>
</tbody>
</table>

--- Computing statistics in Region: TBT

--> ENTITY COUNT:

  # Cells: 1034
  # Faces: 5825
  # Verts: 6817

--> EXTENTS:

  x: [-9.0000e-03, 6.6000e-02] m
  y: [ 4.3000e-02, 1.9800e-01] m
  z: [-4.0000e-04, 2.3100e-02] m

--> MESH VALIDITY:

  Mesh is topologically valid and has no negative volume cells.

--> FACE VALIDITY STATISTICS:
Minimum Face Validity: 1.000000e+00

Maximum Face Validity: 1.000000e+00

<table>
<thead>
<tr>
<th>Face Validity &lt; 0.50</th>
<th>Count</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.50 &lt;= Face Validity &lt; 0.60</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.60 &lt;= Face Validity &lt; 0.70</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.70 &lt;= Face Validity &lt; 0.80</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.80 &lt;= Face Validity &lt; 0.90</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.90 &lt;= Face Validity &lt; 0.95</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.95 &lt;= Face Validity &lt; 1.00</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1.00 &lt;= Face Validity</td>
<td>1034</td>
<td>100.000%</td>
</tr>
</tbody>
</table>

-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 5.011542e-03

Maximum Volume Change: 1.000000e+00

<table>
<thead>
<tr>
<th>Volume Change &lt; 0e+00</th>
<th>Count</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>0e+00 &lt;= Volume Change &lt; 1e-06</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-06 &lt;= Volume Change &lt; 1e-05</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-05 &lt;= Volume Change &lt; 1e-04</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-04 &lt;= Volume Change &lt; 1e-03</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-03 &lt;= Volume Change &lt; 1e-02</td>
<td>3</td>
<td>0.290%</td>
</tr>
<tr>
<td>1e-02 &lt;= Volume Change &lt; 1e-01</td>
<td>47</td>
<td>4.545%</td>
</tr>
<tr>
<td>1e-01 &lt;= Volume Change &lt;= 1e+00</td>
<td>984</td>
<td>95.164%</td>
</tr>
</tbody>
</table>

--- Computing statistics in Region: HHX_Hot

---
-> ENTITY COUNT:

  # Cells: 123231
  # Faces: 619084
  # Verts: 588001

-> EXTENTS:

  x: [4.1094e-02, 2.7391e-01] m
  y: [-4.8255e-02, 3.9410e-02] m
  z: [5.0000e-04, 2.2201e-02] m

-> MESH VALIDITY:

  Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00

Maximum Face Validity: 1.000000e+00

<table>
<thead>
<tr>
<th>Face Validity</th>
<th>Count</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 0.50</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.50 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 0.60</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.60 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 0.70</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.70 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 0.80</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.80 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 0.90</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.90 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 0.95</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.95 &lt;=</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>&lt;= 1.00</td>
<td>123231</td>
<td>100.000%</td>
</tr>
</tbody>
</table>

-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 2.481194e-03

Maximum Volume Change: 1.000000e+00
Volume Change < 0e+00 0 0.000%
0e+00 <= Volume Change < 1e-06 0 0.000%
1e-06 <= Volume Change < 1e-05 0 0.000%
1e-05 <= Volume Change < 1e-04 0 0.000%
1e-04 <= Volume Change < 1e-03 0 0.000%
1e-03 <= Volume Change < 1e-02 376 0.305%
1e-02 <= Volume Change < 1e-01 6041 4.902%
1e-01 <= Volume Change <= 1e+00 116814 94.793%

--- Computing statistics in Region: HHX_Cold

-> ENTITY COUNT:

  # Cells: 281354
  # Faces: 1379049
  # Verts: 1206482

-> EXTENTS:

  x: [ 1.4000e-02,  2.9000e-01] m
  y: [ 9.0838e-04,  4.3000e-02] m
  z: [-4.0000e-04,  2.3100e-02] m

-> MESH VALIDITY:

  Mesh is topologically valid and has no negative volume cells.

-> FACE VALIDITY STATISTICS:

Minimum Face Validity: 1.000000e+00
Maximum Face Validity: 1.000000e+00

98
<table>
<thead>
<tr>
<th>Face Validity Range</th>
<th>Count</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Face Validity &lt; 0.50</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.50 &lt;= Face Validity &lt; 0.60</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.60 &lt;= Face Validity &lt; 0.70</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.70 &lt;= Face Validity &lt; 0.80</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.80 &lt;= Face Validity &lt; 0.90</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.90 &lt;= Face Validity &lt; 0.95</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0.95 &lt;= Face Validity &lt; 1.00</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1.00 &lt;= Face Validity</td>
<td>281354</td>
<td>100.000%</td>
</tr>
</tbody>
</table>

-> VOLUME CHANGE STATISTICS:

Minimum Volume Change: 1.360043e-03

Maximum Volume Change: 1.000000e+00

<table>
<thead>
<tr>
<th>Volume Change Range</th>
<th>Count</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume Change &lt; 0e+00</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>0e+00 &lt;= Volume Change &lt; 1e-06</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-06 &lt;= Volume Change &lt; 1e-05</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-05 &lt;= Volume Change &lt; 1e-04</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-04 &lt;= Volume Change &lt; 1e-03</td>
<td>0</td>
<td>0.000%</td>
</tr>
<tr>
<td>1e-03 &lt;= Volume Change &lt; 1e-02</td>
<td>115</td>
<td>0.041%</td>
</tr>
<tr>
<td>1e-02 &lt;= Volume Change &lt; 1e-01</td>
<td>7213</td>
<td>2.564%</td>
</tr>
<tr>
<td>1e-01 &lt;= Volume Change &lt;= 1e+00</td>
<td>274026</td>
<td>97.395%</td>
</tr>
</tbody>
</table>
Appendix B: Particle Displacement in Simulated Acoustic Conditions

%Acoustic Displacement Calcs
%03/21/2012 revised 5/1/2012
freq=72; %Hz
omega=2*pi*freq;
invomega=1./omega;

Dia=.100; %m tube dia
Area=pi*(Dia/2)^2; %Area of Tube

T=[20 100 200 300 400 450 730] %Temps
rho=[1.177 .9413 .7060 .580 .50 .47 .353] %Density of Dry Air at given Temps kg/m^3
Z=rho.*343 %Impedance
P=1600; %Pascals

I=P.^2./Z %Sound Intensity
Pac=I.*Area %Acoustic Power W
partdisp=invomega.*(sqrt(Pac./(Z.*Area))).*10^3 %mm
figure(2)

plot(T, partdisp, '-o', 'LineWidth', 2)

title('Particle Displacement as a Function of Temperature')

xlabel('Temp (C)')

ylabel('Particle Displacement (mm)')
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