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Design and Optimization of a 1 kW Stirling Engine

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Design and Optimization of a 1 kW Stirling Engine

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Dissertation Submitted to the
Benjamin M. Statler College of Engineering and Mineral Resources
At West Virginia University
in partial fulfillment of the requirements for the degree of
Doctorate of Philosophy in Mechanical Engineering

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Keywords: Stirling Engine, Additive Manufacturing, Flexure Bearings, Regenerator, Heater
Head, Radiation Shields

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Abstract

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Garrett T. Rinker

Stirling engines are often overlooked as a means of energy conversion, though its versatility offers many advantages. Since it is an external combustion engine, it can be supplied with heat through the combustion of a variety of fuels. Its maintenance- and degradation-free aspects, as well as its excellent turn down ratio, make it a suitable candidate for applications such as remote power generation, combined heat and power (CHP), cryogenics, solar power, and even space exploration. The continual advancement of additive manufacturing (AM) offers opportunity to potentially further improve the already efficient Stirling cycle. This dissertation presents the design and optimization of a 1 kW Stirling engine with the heater head and regenerator developed through AM.

This work was divided into four specific tasks: heater head geometry selection for reduced conduction losses, analysis of a foil regenerator made through AM, the impact of flexure bearing geometry and clamping method on stress and fatigue life, and how the configuration and quantity of radiation shields in the displacer affect radiation heat transfer losses. The results of these tasks were discussed, and design recommendations for these four components were provided.

A head heater head was designed and developed with AM that allowed for a previously unattainable geometry with a fully integrated pressure vessel and heat exchanger that could significantly reduce detrimental dead volumes. In addition, tapering of the heater head wall was shown to significantly reduce conduction losses; considered taper options had losses, which were 23.8 to 37.7% less than that of a constant thickness heater head. A regenerator was designed and developed with both foil thickness and gaps between foils set as 300 μm . A foil regenerator developed through AM has the advantages of having less friction losses than traditional random fiber regenerators, and the manufacturing process is less expensive than micro-machining. If 100 μm foils could be printed with AM (current technology achieves a minimum thickness of approximately 200 μm), the Sage results show a 3.6 percent increase in cycle efficiency is possible compared to a random fiber regenerator of the same dimensions. FEA showed the clamping method of flexure bearings had a significant effect on stress and fatigue life; the radially clamped condition resulted in maximum stress values 26.3%-28.2% less than those obtained with the outer ring clamped. A spiral flexure was designed and manufactured with Sandvik 7C27Mo2, which can provide theoretically infinite life. The flexure design was validated with laboratory testing. FEA showed that for a given number of radiation shields, there is an optimal placement along the length of the displacer that yields the lowest radiation losses. The results also show that uniformly spacing the radiation shields out along the length of the displacer has the best performance. Based on the dimensions and boundary conditions considered, five radiation shields provide optimal resistance to heat transfer. The insertion of radiation shields can reduce the temperature and total heat flux in the displacer body by 40 % and 55 %, respectively.

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II. Table of Contents

I. Acknowledgements	iii
II. Table of Contents	iv
III. List of Figures	vi
IV. List of Tables	ix
V. Nomenclature	x
Chapter 1: Introduction	1
1.1 System Overview	1
1.2 Description of Tasks.....	3
1.2.1 Heater Head Geometry for Reduced Conduction Losses	3
1.2.2 Design of an Additively Manufactured Regenerator.....	3
1.2.3 Effect of Clamping Method on Flexure Bearing Stress and Fatigue Life	5
1.2.4 Effect of Radiation Shield Placement on Heat Loss.....	5
Chapter 2: Literature Review	7
2.1 Stirling Engine History	7
2.2 General Stirling Engine Design	8
2.3 Stirling Engine Applications	11
2.4 Heater Head	14
2.5 Regenerators	15
2.6 Flexure Bearings.....	18
2.7 Radiation Shields.....	25
2.8 Additive Manufacturing	26
2.9 Sage Modeling.....	28
Chapter 3: Heater Head Design for Reduced Conduction Losses	29
3.1 Effect of Shape	29
3.2 Effect of Length of Minimum Wall Thickness	36
3.3 Heater Head Dimensions and Manufactured Product	38
3.4 Conclusions	40
Chapter 4: Regenerator	41
4.1 Methodology	41
4.2 Results	42
4.3 Regenerator Dimensions and Manufactured Product.....	46
4.4 Conclusions	50
Chapter 5: Flexure Bearings	51
5.1 Methodology	51
5.2 Results	56

5.3 Experimental Validation	63
5.4 Conclusions	65
Chapter 6: Radiation Shields	67
6.1 Methodology	67
6.2 Results and Discussion.....	70
6.2.1 Effect of Shield Location.....	70
6.2.2 Effect of Number of Shields Employed	77
6.2.3 Effect of Emissivity	80
6.3 Conclusions	81
Chapter 7: FEA of Stirling Engine Components	83
7.1 Displacer.....	83
7.2 Heat Rejecter Assembly	84
7.3 Heater Head with Bolt Pretension.....	87
Conclusions and Suggested Future Work	89
References	90

III. List of Figures

Figure 1: Stirling Engine Schematic	2
Figure 2: Displacer Assembly Schematic	2
Figure 3: Ideal Stirling Cycle and its Processes [22]	9
Figure 4: Comparison of Ideal, Simulated, and Experimental Stirling Cycles [23]	9
Figure 5: Alpha, Beta, and Gamma Stirling Engine Schematics [24]	10
Figure 6: Spiral flexure without (left) and with (right) clamping	20
Figure 7: Schematic of typical flexure bearing set up	20
Figure 8: Flexure Design Variables	23
Figure 9: Shape Factor Influence on Flexure Geometry	23
Figure 10: Sage Model	28
Figure 11: Representation of Heater Head Geometries	30
Figure 12: Heater Head Temperature Boundary Conditions	31
Figure 13: Heater Head Static Structural Boundary Conditions	31
Figure 14: Heater Head Top Stress Contour for Constant Wall Thickness	32
Figure 15: Heater Head Bottom Stress Contour for Constant Wall Thickness	32
Figure 16: Heater Head Top Stress Contour for Convex Above Midpoint	33
Figure 17: Heater Head Bottom Stress Contour for Convex Above Midpoint	33
Figure 18: Constant Thickness Geometry Temperature Profile	34
Figure 19: Convex to Above Midpoint Temperature Profile	34
Figure 20: Temperature Profile through the Bottom Segment of the Heater Head Based on Geometry	35
Figure 21: Dimensions Considered for Heater Head Analysis	36
Figure 22: Factor of Safety vs. Conduction Losses for Various C/L Ratios	37
Figure 23: Stress Contour for C/L = 15%	37
Figure 24: Stress Contour for C/L = 95%	38
Figure 25: Heater Head Dimensions for Additive Manufacturing	38
Figure 26: Heater Head Dimensions after Finish Machining	39
Figure 27: Manufactured Heater Head	40
Figure 28: Model of Regenerator Top Segment	42
Figure 29: Regenerator Temperature Distribution	43
Figure 30: Regenerator Stress Distribution	44
Figure 31: Regenerator Axial Deformation	45
Figure 32: Regenerator Radial Deformation	46
Figure 33: Regenerator Top Section Dimensions	47
Figure 34: Regenerator Middle Section Dimensions	48
Figure 35: Regenerator Bottom Section Dimensions	49
Figure 36: Additively Manufactured Regenerator Top Segment	50
Figure 37: Sandvik 7C27Mo2 S-N Curve [111]	52

Figure 38: Flexure Bearing Mesh	53
Figure 39: Boundary Conditions for both Rim (Left) and Radially (Right) Clamped Flexures...	54
Figure 40: All Flexure Geometries Utilized in the Sensitivity Analysis	55
Figure 41: Stress contour for Sandvik 7C27Mo2 Flexures Clamped at the Outer Rim (Left) and Radially (Right)	56
Figure 42: Stress Contour for the Baseline Flexure Geometry ($f = 0.04$ and Kerf Width = 1.2 mm)	59
Figure 43: Flexure Stress Contours for Various Kerf Widths and Shape Factor of 0.04	60
Figure 44: Flexure Stress Contours for Various Shape Factors and Kerf Width of 1.2 mm	61
Figure 45: Maximum Stress and Axial Spring Rate for Varying Kerf Width and Constant Shape Factor	62
Figure 46: Maximum Stress and Axial Spring Rate for Varying Shape Factor and Constant Kerf Width.....	63
Figure 47: Flexure Developed through Waterjet Cutting (Left), Flexure Mask (Center), Flexure after Shot-Peening (Right)	63
Figure 48: Flexure Bearing Dimensions	64
Figure 49: Flexure Test Rig	65
Figure 50: Boundary Conditions and Displacer Assembly Mesh for the Simulations	68
Figure 51: Radiation Shield Dimensions	69
Figure 52: Example Displacer Cross-Sectional Temperature Profile.....	70
Figure 53: Two Shields at 20 mm, 50mm, 60 mm, and 80 mm from the Displacer Body.....	72
Figure 54: Displacer Wall Temperature Profile Based on the Location of Two Shields	72
Figure 55: Three Shields at 20 mm, 50 mm, and 80 mm from the Displacer Body.....	73
Figure 56: Displacer Wall Temperature Profile Based on the Location of Three Shields	73
Figure 57: 80 mm to Top Shield, 15 mm between Sets.....	75
Figure 58: 60 mm to Top Shield, 15 mm between Sets.....	75
Figure 59: Displacer Wall Temperature Profile Based on Number of Shields (60 mm to Top Radiation Shield)	76
Figure 60: Displacer Wall Temperature Profile Based on Number of Shields (80 mm to Top Radiation Shield)	76
Figure 61: 1-6 Shields, 15 mm Spacing between Shields.....	78
Figure 62: 15 mm Spacing between Shields Variations	78
Figure 63: Displacer Wall Temperature Profile Based on Number of Radiation Shields (15 mm between Shields)	79
Figure 64: Displacer Wall Temperature Profile Based on Number of Radiation Shields Variations	79
Figure 65: Displacer Wall Temperature Profile Based on Number of Shields (0.8 Emissivity)..	81
Figure 66: Displacer Thermal and Structural Boundary Conditions	83
Figure 67: Displacer Temperature and Stress Profiles	84
Figure 68: Heat Rejecter Boundary Conditions.....	85

Figure 69: Heat Rejecter Stress Contour	85
Figure 70: Heat Rejecter Shell Stress Contour	86
Figure 71: Heat Rejecter Flow Distributer Stress Contour.....	86
Figure 72: Heat Rejecter Inner Surface Stress Contour.....	87
Figure 73: Heater Head with Bolt Pretention Boundary Conditions	88
Figure 74: Stress Contour from the Bolt Pretension FEA	88

IV. List of Tables

Table 1: Pros and Cons of the Alpha, Beta, and Gamma Schematics [24], [25]	11
Table 2: Spring Rate Comparison of Flexures Based on Number of Arms [8].....	24
Table 3: Conduction Losses in the Axial Direction of the Heater Head Wall Based on Geometry	35
Table 4 – FEA Mesh Independence Study	41
Table 5 – Thermal Properties of Inconel 718 [109], [110]	43
Table 6: FEA Results for the Regenerator	46
Table 7: Material Properties Utilized for Flexure FEA	52
Table 8: Flexure Geometry Considered in the Sensitivity Analysis.....	54
Table 9: Maximum Stress Comparison.....	57
Table 10: Axial Spring Rates	57
Table 11: Modal Analysis Results	58
Table 12: Constant Shape Factor Results	59
Table 13: Constant Kerf Width Results	62
Table 14: Thermal Properties of Inconel 718	67
Table 15: Maximum Displacer Body Temperature and Radiation Heat Loss for the Two Shield Cases at 80 mm, 50 mm, and 20 mm.....	72
Table 16: Maximum Displacer Body Temperature and Radiation Heat Loss for the Three Shield Cases at 80 mm, 50 mm, and 20 mm.....	74
Table 17: Maximum Displacer Body Temperature and Radiation Heat Loss for the 60 mm to Top Radiation Shield Cases	76
Table 18: Maximum Displacer Body Temperature and Radiation Heat Loss for the 80 mm to Top Radiation Shield Cases	77
Table 19: Maximum Displacer Body Temperature and Radiation Heat Loss for the 15 mm Spacing between Shields Cases	80
Table 20: Maximum Displacer Body Temperature and Radiation Heat Loss for the 15 mm Spacing between Shields Case Variations	80
Table 21: Maximum Displacer Body Temperature and Radiation Heat Loss for the 0.8 Emissivity Simulations	81

V. Nomenclature

APU.....	Auxiliary Power Unit
CAD	Computer-Aided Design
CFD.....	Computational Fluid Dynamics
CHP.....	Combined Heat and Power
COP.....	Coefficient of Performance
CQ.....	Radiation Heat Loss
DB	Cylinder Diameter
DEVC.....	Diesel Engine-Vapor Compression
DMLS	Direct Metal Laser Sintering
E	Young's Modulus
EDM.....	Electrical Discharge Machining
f	Shape Factor
FA	Area Factor
FEA.....	Finite Element Analysis
FM.....	Emissivity Factor
FN	Radiation Shield Factor
F _R	Resulting Force
ICE	Internal Combustion Engine
k.....	Conduction Coefficient
k _a	Surface Condition Modification Factor
k _{axial}	Axial Spring Rate
k _b	Size Modification Factor
k _c	Load Modification Factor
k _d	Temperature Modification Factor
k _e	Reliability Factor
k _{radial}	Radial Spring Rate
LMP	Larson-Miller Parameter
N.....	Number of Cycles

P	Pressure
PM.....	Particulate Matter
R.....	Polar Distance from the Center of the Flexure
Re	Reynolds Number
R _i	Active Inner Radius
R _o	Active Outer Radius
S _e	Fatigue Limit
S _e '	Specimen Fatigue Limit
S _{max}	Maximum Stress
S _{ut}	Ultimate Stress
t	Time
t _r	Rupture Life
T	Temperature
T _H	High Temperature
T _L	Low Temperature
V.....	Volume
v.....	Specific Volume
WHR	Waste Heat Recovery

Greek Letters

σ	Stefan-Boltzmann Constant
ε	Emissivity
ρ	Density
δ	Displacement
θ	Intermediate Arc Angle
θ_0	Total Included Arc Angle
ν	Poisson's Ratio
ω	Frequency

Chapter 1: Introduction

In the field of energy conversion, the versatile Stirling engine is often overlooked. Since the Stirling engine is an external combustion engine, a variety of materials can be used as fuel sources. Additionally, their maintenance- and degradation-free operation makes Stirling convertors ideal for use in remote power generation, solar power generation, combined heat and power applications, cryocoolers, and even for space applications [1]–[3]. Stirling engines will benefit greatly from the increasingly advanced technology in the field of additive manufacturing. Not only would the use of additive manufacturing to produce a robust and reliable foil regenerator, it can also be employed to manufacture the entire Stirling heater head. This would allow for a previously unattainable complex geometry of a fully integrated pressure vessel and heat exchanger design that can potentially eliminate detrimental dead volumes. This work outlines the design and optimization of a 1 kW Stirling engine with several components developed through additive manufacturing.

1.1 System Overview

The designed Stirling engine and displacer assembly are shown in Figure 1 and Figure 2. FEA, performed with ANSYS Workbench, was conducted to ensure every component would be able to withstand applied temperature and/or force boundary conditions. Most analyses involved importing a temperature profile from a steady state thermal analysis into a static structural analysis, which could be easily performed directly in the Workbench layout. Optimization of the Stirling engine was done with Sage modeling.

Figure 1: Stirling Engine Schematic

Figure 2: Displacer Assembly Schematic

1.2 Description of Tasks

There were four primary design tasks in this work, which are specific areas in component design intended to add to the body of knowledge concerning Stirling engines.

1.2.1 Heater Head Geometry for Reduced Conduction Losses

The heater head is a critical component of a Stirling convertor, which must be designed to ensure it will not fail due to creep over its expected design life. Creep is a significant limiting factor in the design life of high temperature components. Conduction losses along the wall of the heater head should be kept to a minimum for improved thermal performance. This can be accomplished by minimizing the heater head wall thickness and utilizing an appropriate taper design [4]. Efficient heater head designs are often the result of a compromise between thin walls for heat transfer characteristics and thick walls for improved resistance to creep and durability [5]. Creep is a significant factor that must be considered in the high temperature region of the heater head. Creep is even more significant for thin walls such as those of the heater head, since stress rupture of super alloys occurs slower with thick solid bars under the same temperature and pressure [6]. The majority of nickel based super alloys have enhanced creep strength due to fine and uniformly dispersed γ' - particles precipitating during ageing treatment following a solution annealing [7]. In this work, an analysis was performed to investigate the effects of varying wall taper profiles on stresses.

1.2.2 Design of an Additively Manufactured Regenerator

The regenerator is a crucial component of a Stirling engine which allows for the engine to achieve high efficiencies [8]. The regenerator acts as a means of internal energy storage where a fraction of the waste thermal energy is stored in the solid matrix and retrieved by the working fluid later in the cycle. This results in a reduction of the required input energy from the fuel source

leading to an increase in the overall system efficiency. A variety of regenerator designs exist, such as wire mesh, stacked screens, foil, micro-channel, and metal foam. Most Stirling engines incorporate the wire mesh regenerator design due to its excellent heat transfer characteristics between the wires of the regenerator and the working fluid, and it has low axial conduction in the flow direction [9]. Foil regenerators are very efficient because the working fluid flows through straight channels, which usually exhibit lower pressure drop values compared to wire mesh regenerators. Foil regenerators also have the advantage of well-defined flow characteristics whereas wire mesh regenerators may display preferential flow due to the random nature of the design, which leads to performance variation [10]. The use of foil regenerators is limited as it is nearly impossible to keep a uniform foil spacing throughout the regenerator due to foil deformation. Foil deformation can occur due to thermal expansion of the foils during the repeated heating and cooling cycles, or the regenerator can be damaged by the insertion process during assembly. The fragility of foil regenerators is a direct result of the way foil regenerators are manufactured. Therefore, Stirling engine manufacturers use a less efficient but more reliable regenerator to ensure that the desired operation is achieved over the lifetime of the unit.

Despite their desirable flow characteristics, foil regenerators have not been utilized in Stirling engine in the past due to their low reliability, as they were difficult to manufacture robustly. Recent advancements in additive manufacturing would allow for the production of foil regenerators with foils that are several hundred microns thick. This manufacturing method would be vastly less expensive than previously used micro-machining techniques. The drawback of previously manufactured foil regenerators was their inability to maintain foil spacing over prolonged thermal cycling as the foils deform due to thermal expansion. The use of additive manufacturing allows for the production of a novel foil regenerator where the foils are intricately

connected for added rigidity. A FEA of the designed regenerator with 300 μm foils was conducted to examine both the thermally induced stresses in the regenerator as well as the thermal deformation. This work presents the design and fabrication of a robust foil regenerator developed through additive manufacturing.

1.2.3 Effect of Clamping Method on Flexure Bearing Stress and Fatigue Life

In general, Stirling engines are divided into two groups: kinematic or free-piston engines. In kinematic engines, manual linkages are used to connect the various moving parts within the engine like in a conventional internal combustion engine. Free-piston engines on the other hand use the pressure variation of the working gas to induce motion in the reciprocating components and work is extracted via a linear alternator. The lack of manual linkages coupled with the use of clearance seals in a free-piston engine increases the reliability and lifespan of the engine as there is no wear on the parts. The critical component to the long, maintenance-free life of these engines is the novel implementation of flexure bearings in both the Stirling engine displacer assembly and within the linear alternator. Characteristics of flexure bearings, such as clamping method, shape factor, and number of arms were observed in terms of their impact on longevity. Also, a sensitivity analysis was performed to observe the impact of kerf width and shape factor on stress and spring rate.

1.2.4 Effect of Radiation Shield Placement on Heat Loss

This document goes over a journal article published by the author [11], though more details are included here. Inserting radiation shields between surfaces with a significant temperature differential is a common method for reducing heat transfer to the cooler surface due to radiation. The amount of heat transfer can be further reduced by evacuating the space between the surfaces to result in negligible conduction and convection effects [12]. Several applications of radiation

shields are in cryogenics [13], refrigeration [14], building design [15], and Stirling engines [16]. Radiation shields, for most purposes, are typically made out of copper and aluminum for low-temperature applications [17]. Inconel 718 is a popular material to make displacers out of due to its ability to withstand high temperatures and its resistance to creep and corrosion [67] [87]. This work presents an analysis that studies the effects of quantity of shields and their arrangement in the displacer on the temperature profiles and heat loss from the hot to cold side of the displacer.

Chapter 2: Literature Review

2.1 Stirling Engine History

The Stirling engine was invented by Reverend Robert Stirling in 1816 [20]. The Stirling engine did not initially achieve commercial success. From 1818 to 1845, only four Stirling engines were produced, mainly due to inadequate materials of the time. The components would fail shortly after the engine was in operation, causing complete engine failure [21].

As described by Bauer [21], the Stirling engine had three distinct phases of success, which were from the 1860's to 1880's, the 1930's to 1950's, and the 1970's to 1980's. Improved Bessemer steel in the 1860's allowed for reliable Stirling engine performance. Though the power output was limited to approximately 3.5 kW, these engines were safer and less expensive than the popular steam engines of the time. However, Stirling engines still had some disadvantages, such as sealing problems, energy losses, and high power to weight ratios (as great as three tons per kW). The Otto engine began to dominate the power generation market with its introduction in the 1870's, as it was more efficient than the Stirling engines and steam engines of the time, and it had a better power to weight ratio.

In the 1930's the Stirling engine made a comeback because of the Dutch company Philips Gloeilampenfabriken. The company wanted to design a small engine to generate electricity to power radios in remote areas, and they turned to Stirling engines as a solution. Though the engine proved to be satisfactory, the progress of transistor technology halted the use of Stirling engines for this purpose [21].

In the 1970's public concern over the environmental impacts of internal combustion engines, as well as an oil crisis in 1973, caused the U.S. government to pass emissions and fuel

consumption standards. The government also subsidized development of more efficient engines to meet these goals, which drew interest back to Stirling engines. An automotive Stirling engine was introduced in 1989 by Mechanical Technology Incorporated, which was based on the design made by Philips in the 1940's. This engine produced far lower emissions than the standards set as a maximum value, and it consumed approximately one-third less fuel than the gasoline engines. However, the diesel engines of the time only consumed slightly more fuel than the Stirling engines, and the emissions standards easily be met with some modifications to the existing internal combustion engines. Therefore, manufacturers were not willing to dismiss the well-established technology of internal combustion engines to take the financial risk of developing Stirling engines for automotive purposes [21].

2.2 General Stirling Engine Design

Figure 3 is a P-v diagram of the ideal Stirling cycle along with the processes involved in it. Isothermal expansion occurs from State 1 to State 2, where heat is transferred to the working fluid from the high temperature reservoir (usually exhaust gases). Work is done by the system during this process. Both pistons move to the right (according to Figure 3) from State 2 to State 3, while maintaining constant volume. During this process, heat is transferred from the working fluid to the regenerator, and the temperature decreases from T_H to T_L . Isothermal compression occurs from State 3 to State 4, and heat is transferred to the low temperature reservoir to maintain the temperature at T_L . Finally, both cylinders move to the left in a constant volume process from State 4 to State 1, where the working fluid obtains heat that was stored in the regenerator during process 2-3, raising the temperature back to T_H [22].

As stated by Walker [20], practical Stirling engines do not operate on the ideal Stirling cycle for a variety of reasons; It is impossible to have isothermal compression and expansion

processes, the working fluid is never completely contained in the expansion or compression space, dead spaces decrease the efficiency, the regenerative process is not perfect, and frictional effects cannot be neglected. Figure 4 (from the work performed by Tavakolpour-Saleh et al. [23]) is an example showing how simulated and experimental Stirling cycles deviate from the ideal case. The figure shows the actual case will neither be as efficient nor have as great a power output as the ideal scenario, due to the fact that the net work output (the area contained within the curve of the cycle) is smaller.

Figure 3: Ideal Stirling Cycle and its Processes [22]

Figure 4: Comparison of Ideal, Simulated, and Experimental Stirling Cycles [23]

The three basic Stirling engine configurations are referred to as the alpha, beta, and gamma arrangements, which are depicted in Figure 5. The alpha configuration has two cylinders, both with their own piston, with a regenerator between them. The beta configuration consists of a piston and displacer located in the same cylinder. The gamma configuration consists of a displacer and power piston in different cylinders. Table 1 lists some of the pros and cons of each of the configurations.

Figure 5: Alpha, Beta, and Gamma Stirling Engine Schematics [24]

Table 1: Pros and Cons of the Alpha, Beta, and Gamma Schematics [24], [25]

Engine Configuration	Advantages	Disadvantages	Typical Drive Arrangements
Alpha	Conceptually the simplest.	Both pistons need to have seals to contain the working space.	V-arrangement and yoke drive (Ross linkage). Can be compounded into a multiple cylinder configuration.
Beta	High compression, efficiency, and power can be obtained due to lower dead volume.	Mechanical disadvantage of the drive rod from the displacer extends through the piston.	Rhombic drive.
Gamma	Mechanically more efficient than the other arrangements (simple crank mechanism).	Higher dead volumes, specifically the connecting pipe that connects both the compression space and the lower part of the expansion space.	Standard crank drive.

2.3 Stirling Engine Applications

Stirling engines have found success in a variety of applications. One such application is with combined heat and power (CHP) systems. CHP systems simultaneously generate electricity and heat for other applications (such as heating water). Cardozo et al. [26] integrated a Stirling engine with a wood pellet burner to produce a CHP system. The burner supplied 20 kW of heat, and the Stirling engine could produce 1 kW of electrical power. The heat not utilized for the Stirling cycle was utilized to produce hot water. The overall CHP system efficiency was 72%. Thiers et al. [27] experimentally investigated the performance of a wood pellet Stirling engine micro-CHP system intended to provide electricity and heat for a residential building. Testing of the unit showed it could provide approximately 1.38 kW of electricity 5.4 kW of heat. Lipp [28]

presented the results of field testing of four Stirling engine micro-CHP units in residential buildings located close to Berlin, Germany. The units were shown to operate at a steady state efficiency above 90%.

The field of cryogenics employs Stirling technology. Caughley et al. [29] presented a free-piston Stirling cryocooler with utilizing metal diaphragms for extended life. The prototype could remove 29 W of heat at 77 K, and could reach a no-load temperature of 56 K. Wang et al. [30] conducted a study on a free-piston Stirling cryocooler that operated at approximately 30 K. The lowest achieved by the cryocooler was 27.6 K. The cryocooler could remove 78 W of heat at 40 K, and obtained a relative Carnot efficiency of 14.8 %. Riabzev et al. [31] utilized a low vibration Stirling cryogenic refrigerator to cool a high definition microscopy unit. The refrigerator removed 5 W at 40 K.

Stirling engines have been considered for waste heat recovery (WHR) systems in automobiles. Andriollo and Tortella [32] presented a theoretical model of two control strategies a Stirling free piston engine that serves as a WHR system for a diesel engine. The Stirling engine is coupled to a linear electric generator, which will be utilized as an auxiliary power unit (APU). The mean power output of the strategy with the greater power output is 511.5 W, and the mass of the engine is approximately 10 kg. Alfarawi et al. [33] conducted a thermal analysis of a beta configuration Stirling engine WHR that would be utilized to power an alternator. The alternator would be decoupled from the internal combustion engine (ICE) for this configuration. Decoupling the alternator from the ICE can provide the benefits of increased fuel efficiency and less CO₂ emissions. The engine model was ideal and adiabatic. The authors conducted a CFD analysis to optimize the heater and cooler sections of the Stirling engine. The results of their analysis showed the Stirling engine could provide 1.5-2 kW of electrical power at a thermal efficiency of 40%. The

overall mass of the engine was 11-14 kg. Cullen and McGovern [34] presented a study on the feasibility of using a Stirling engine as an exhaust gas WHR for an Otto Cycle ICE. Their model predicted a Stirling engine with 193 cc of total displacement would provide 14.23 kW of brake power output at 3000 rpm. The engine operated at about 32% brake thermal efficiency considering an 812 K high temperature reservoir that provided 46 kW of heat. The shaft power of the combined ICE and Stirling system was increased by 30% compared to the ICE alone. A study conducted by Yu et al. [35] agree with the results gathered by Cullen and McGovern [34] in that a model of a Stirling engine utilized as a WHR for an Otto cycle gasoline engine could increase the overall power output of the combined system by 32.5% when operated in the range of 2000-3500 rpm. Flannery et al. [36] developed and tested a hybrid Stirling engine-adsorption chiller for HDD4.1962-4 engines. The APU was composed of a 1 kW linear, propane fired (propane was utilized during the test because the Stirling engine was intended for domestic CHP applications), free-piston Stirling engine and a 2 kW zeolite-water adsorption chiller. The adsorption chiller had an average coefficient of performance (COP) of 0.42 ± 0.06 and 2.3 ± 0.1 kW of cooling capacity at the baseline testing conditions. The system indicates a payback period of 4.6 years, and is comparable to diesel engine-vapor compression (DEVVC) technology in terms of cost, with the Stirling adsorption system offering a potential of approximately \$600 cost saving over the lifetime of the truck. The authors indicate these types of Stirling engine systems can achieve approximately 29% and 97% reduction in NO_x and particulate matter (PM) emissions, respectively, compared to DEVVC APU with a diesel particulate filter. However, CO emissions may be approximately 39% greater.

Barreto and Canhoto [37] go over the modelling and simulation of a Stirling engine that obtains heat from a parabolic dish solar receiver. In direct illumination, the engine had an

efficiency of 23.8%, which led to a global efficiency of 10.4%. Kadri and Abdallah [38] evaluated the performance of a solar dish Stirling system for power generation in off-grid rural settings in Tunisia. Ferreira et al. [39] presented designs of a solar dish cogeneration system, where several options had thermal efficiencies between 66.3% and 76.1%.

Stirling engines are utilized for space applications. Dang [40] present a variety of cooling devices for such systems, most of which meant for optics subsystems. Cooling the optic sensors helps to minimize background noise and enhances sensitivity. Qiu et al. [41] presented the design of a Stirling convertor intended for NASA missions. The convertor was calculated to have a power density greater than 44 W/kg, and the efficiency was estimated to be greater than 60% of the Carnot efficiency. Brandhorst and Chapman [18] presented the design of a 5 kW Stirling convertor intended to be utilized for NASA's Vision of Exploration of the moon.

2.4 Heater Head

Many Stirling engine heater head durability and creep analyses exist in the literature ([42]–[50]). In order to maximize the efficiency of a Stirling engine, the operating temperature at the heat acceptor must be high and losses must be minimized. Operating at higher pressures increases the power density of the engine. When the engine is operated at higher pressures and temperatures, however, its life and reliability are reduced, cost is increased, and superior material properties are required [51].

Zhang and Shores [52] state that the formation and maintenance of a dense oxide scale is a key aspect of the resistance of metals and alloys to corrosion and oxidation at elevated temperatures. The resistance to oxidation and corrosion is a primary criterion when selecting an alloy to utilize for a Stirling engine heater head [53]. Excessive scale spalling will cause the heater

head to fail due to decreased wall thickness, which leads to high pressure working fluid rupturing the tubes or permeating through the walls [53].

Stephens and Barrett [53] reported the results of an experimental investigation in which 16 alloys were subjected to a Stirling engine simulator test rig for 3500 hours at 820°C under cyclic conditions. The alloys were evaluated based on their oxidation and corrosion resistance after exposure to combustion gases. Five alloys were considered to have excellent oxidation and corrosion resistance, which were alloys CG-27, Incoloy 800, HS-188, N-155, and Inconel 718. These alloys had a positive change in specific weight (the samples gained weight).

The most commonly used formulations to correlate creep life to applied stress and temperature include Larson-Miller, Orr-Shelby-Dorn, Manson-Haferd, and Manson-Succop [54]. The Larson-Miller parameter (LMP) is calculated by utilizing Equation 1.

$$\text{LMP} = T(\log t + C) \quad \text{Equation 1}$$

In Equation 1, T is the temperature in absolute units, t is the time to rupture in hours, and C is a constant, which typically has a value of 20 [55].

2.5 Regenerators

Because of its importance to the overall efficiency of a Stirling engine, extensive research has been conducted on characterizing the flow dynamics and heat transfer of various regenerator types. Regenerators are designed to have high thermal mass and they are used to temporarily store thermal energy [22]. Without a regenerator, Stirling engines would require five times the amount of heat supply needed to generate the same power as those would do with regenerators. The heat transferred to the gas in the regenerator during one cycle is about four times the amount that passes through the heater for the cycle [24].

Alfarawi et al [9], [56] reported correlations for both the friction loss and Nusselt number of a novel micro-channel regenerator. While the flow losses within the regenerator were greatly reduced, the high thermal conduction losses had a negative impact on their performance. However, the authors suggested that segmentation of the regenerator could potentially minimize the effect of conduction loss improving their thermal performance of the regenerator. Costa et al. [57] used experimental results to validate their numerical model of a stacked woven screen regenerator and evaluated both the pressure drop and flow losses. After their model was validated, Costa et al. [58] employed it to characterize the performance of a wound woven regenerator while also incorporating the effects of heat transfer. Using the correlations they developed, Costa et al. [59] then developed a non-equilibrium porous media model to investigate the performance of a real Stirling engine without typically employed simplifications. Their results indicated that a stacked woven regenerator outperforms a wound woven regenerator. Li et al. [60] numerically evaluated the performance of a compact porous sheet regenerator by developing a porous media model. The regenerator showed significantly less flow loss while exhibiting comparable heat transfer to traditional regenerators. Xiao et al. [61] simulated and conducted an experimental study on the characteristics of steady and oscillating flows through a wire screen regenerator and concluded that while steady flow correlations can be used to predict oscillating flows in a certain Reynolds (Re) number range the flow in the regenerator is greatly affected by the compressibility of the gas. Nielsen et al. [62] demonstrated the impact of flow maldistribution on the performance of microchannel parallel plate heat exchangers through a numerical model and cyclic steady-state regenerator experiments. Hsu and Biwa [63] measured the resistance of stacked-screen regenerators in oscillatory flows and compared the results to empirical equations. Nam and Jeong [64] investigated the use of a parallel wire regenerator for use in cryocoolers. While their results

showed the friction factor was 20-30% lower than a mesh screen regenerator, the effectiveness of the parallel wire regenerator was lower due to high conduction losses.

The impact of replacing a random fiber regenerator with an involute or foil regenerator was demonstrated by Ibrahim et al. [65]–[68] where an increase in efficiency of 4% was shown by simply replacing the existing regenerator with a new micro-fabricated regenerator without further optimization of the engine. Their regenerator was produced using electrical discharge machining (EDM) and LiGA X-ray lithography. The regenerator consisted of a stack of forty-two 0.25 mm thick disks with a 19 mm diameter. Each disk had flow channels that were 85 μm wide separated by 15 μm thick “foils.” Ibrahim et al. [65]–[68] tested the designed involute-foil regenerator in an oscillating-flow test rig that was previously used to test various regenerator types [69]–[72]. The experimental results were extremely promising as the involute-foil regenerator demonstrated a figure of merit double that seen for a 90% random fiber regenerator. The figure of merit is the ratio of heat transfer to flow losses. A detailed design analysis of the regenerator was conducted during the development of the regenerator. Qiu and Augenblick [10] performed a thermal and structural analysis of the designed involute-foil regenerator to examine its structural rigidity and showed that the regenerator had a high axial stiffness and that there was minimal deformation of the foils as they were intricately interconnected, therefore a uniform foil spacing would be maintained during operation. The regenerator design was further evaluated by Tew et al [73]. Using a slightly simplified model of the regenerator, they evaluated both the heat transfer and flow losses of the regenerator under both unidirectional and oscillating flow conditions. Additionally, they investigated the effect that jetting has on the performance of the proposed involute foil regenerator. Despite the regenerator’s promising performance, the manufacturing process was time-consuming and expensive thus the regenerator was not put into mass production. However, recent

advancements in additive manufacturing would significantly reduce the associated manufacturing costs, making a foil-type regenerator a viable mass-market option.

2.6 Flexure Bearings

Flexure bearings have been used in Stirling engines for over 30 years. The flexures can be manufactured in a variety of ways to form one or more support legs between the clamped inner and outer rings of the flexure as shown in Figure 6. The first patented flexure was by Wolf et al. [74] in 1938, which were included in a vibration detector unit to observe Earth's vibrations. Their flexure had straight arms rather than the spiral arms that are primarily included in designs today. The first use of a flexure bearing in a Stirling convertor was in a cryocooler at the University of Oxford in 1981 [75]. In 1992, Wong et al. [76] developed a novel linear flexure that displayed 70% greater radial stiffness and 30% less stress than an optimized three-arm spiral design at a set maximum displacement.

Flexure bearings provide a simple, yet robust, technique to provide the necessary supporting of the moving components within a free-piston engine [77]. Furthermore, they offer the benefits of 1) eliminating the rubbing in the seals and the need for lubrication, 2) eliminating the possibility of gas-bearing port plugging failures, 3) they demonstrate good predictability, repeatability, reliability, and stability, and 4) minimized performance and operating frequency variations due to variations in operating conditions mainly temperature. Flexure bearings are essentially planar springs that provide stability for the moving components. The flexures are desired to allow for axial motion while providing high radial stiffness. The radial stiffness ensures that the piston does not contact the stationary cylinder while it is in motion, while the axial stiffness assists in ensuring the desired dynamic resonance characteristics within the Stirling engine. During the design process, axial stiffness, radial stiffness, subassembly rocking modes, and flexure

arm modes must be considered [77]. The necessary spring rate of the flexure is determined through a linear dynamic analysis, whereas the necessary radial spring rate is determined based on the requirements to handle side loads and rocking modes. Also, the peak predicted stress must be well below the fatigue limit [40]. Additionally, since flexures are passive in nature they are always functioning unlike gas bearings that are activated via check valves and the pressure of the working fluid within the cycle. Thus, gas bearings are inoperable during engine startup and low power operation, which leads to unnecessary wear that decreases the life span of the engine. For a free-piston Stirling engine, the flexure bearing spring rate is temperature independent, which greatly reduces the impact of environmental conditions on the systems performance.

Typically, when a flexure bearing is used in a linear alternator, the outer ring is clamped to the stationary stator assembly while the inner ring is clamped to the mover rod, thus these are called moving inner diameter flexure bearings. When the flexures are used in the displacer assembly, the inner ring is clamped to the stationary displacer rod while the outside of the flexure is connected to the moving displacer piston, so being called moving outer diameter flexure bearings. Two or more flexures can be stacked with an axial separation to provide a support-couple wheelbase that prevents the moving members from rocking. This setup is shown in Figure 7. Additionally, since the designed flexures have wide arms they provide an extremely high resistance to radial motion while allowing for the flexure to spring axially with a relative degree of freedom. The flexure and moving member form a mass-spring dynamic system where the system frequency is mainly dependent on the total spring rate of the flexure stack and moving mass. The high radial spring rate also enables the moving member to function with a tight clearance seal of only 12 to 25 microns.

Figure 6: Spiral flexure without (left) and with (right) clamping

Figure 7: Schematic of typical flexure bearing set up

The flexures can be designed to have an almost infinite life by ensuring that the maximum stress within the flexure is below the allowable endurance stress limit. Furthermore, the allowable stress used during the design process is obtained by subtracting three times the standard deviation from the vendor's listed mean endurance limit for the selected flexure material. This leads to a 99% confidence level in the endurance of the part. Amoedo et al. [78] performed fatigue analysis to ensure a theoretically infinite endurance life by multiplying the endurance limit of a rotating beam specimen with a number of stress-raising factors, an expression similar to that presented by Budynas and Nisbett [79] as shown in Equation 2:

$$S_e = k_a k_b k_c k_d k_e S'_e \quad \text{Equation 2}$$

Amoedo et al. [78] utilized a maximum fatigue stress value of 366 MPa to obtain an infinite bearing fatigue life for materials with an ultimate tensile strength above 1380 MPa. Each arm of a flexure is exposed to alternating stresses at a frequency equal to the operating frequency of the Stirling engine. For a certain axial displacement, the magnitude and location of the flexure's maximum stress are dependent upon the spiral profile, diameter, and thickness of the flexure. The maximum stress should be much less than the endurance limit of the material to ensure the flexure has virtually infinite life [80].

The axial stiffness of the flexure bearing is usually significantly less than the stiffness of the gas spring above the piston to minimize the moving mass, which influences the level of vibrations in the system [80]. Flexures must be made out of materials that have superior thermal and mechanical properties, such as a high ratio of fatigue stress to Young's modulus. Beryllium copper and austenitic stainless steel are two of the most common materials used for flexures [40], [81]. The operating frequency of the Stirling convertor is heavily dependent upon the overall spring rate of the flexure bearings. As flexures provide nearly frictionless operation without requiring any lubrication, they can be utilized along with clearance seals, they are inexpensive, and they can be manufactured easily with a dye in a punch press [78]. A well-developed and proven flexure design can easily be incorporated into a wide range of Stirling converters that operate at various power levels as the flexures are designed independent of the absolute power of the engine. For instance, a flexure that was designed to operate within an engine at 60 Hz with an amplitude of 12 mm can be used over a range of power levels from 1 kW to in excess of 7 kW without any modification. When this aspect is combined with the flexure's insensitivity to the mean operating pressure, the power range for a design can be further expanded. The axial stiffness of the flexure

stack is directly related to the dominant spring rate within the system that also determined the resonant frequency of both the pistons and the displacer. The utilization of flexure bearings within a free-piston Stirling engine increases the robustness, reliability, and lifespan of the engine. These aspects allow for a reduction in the associated maintenance costs of the engine's operation, making them ideal for use in remote power generation applications.

Prior to the advent of high-speed computational FEA software, flexures were analyzed as straight or curved cantilevered beams. Dimensional analysis can be used to derive a set of equations and parameters that along with testing can lend insight into the fundamental behavior of flexures. While closed form solutions can readily be found for straight beams, it is difficult for curved beams. With advances in FEA software and overall computing power, FEA has become the primary tool by which the stresses and spring rate for various flexure arm shapes are analyzed and optimized. The numerical predictions of spring rates for spiral flexures are typically within 10% of the tested values. This is because not only do they experience bending but also a torsional load as well that leads to higher stresses at the edges of the fixed end of the flexure arm. Spiral flexures can be mathematically modeled with Equation 3:

$$R = R_i + (R_o - R_i)[\theta/\theta_0 + f \sin(2\pi\theta/\theta_0)] \quad \textbf{Equation 3}$$

where R is the polar distance from the center of the flexure, θ/θ_0 is the relative sweep angle ratio, R_i is the active inner radius, R_o is the active outer radius, and f is the arm shape factor. The variables R_i , R_o , θ , and θ_0 are depicted in Figure 8, and the impact of the shape factor on flexure geometry is illustrated in Figure 9. Both the inner and outer radii amplitudes are calculated through optimization techniques for typical Stirling engine designs. A more evenly distributed stress contour over the arms of the flexure can be achieved by varying arm width with a non-zero shape factor.

Figure 8: Flexure Design Variables

Figure 9: Shape Factor Influence on Flexure Geometry

The kerf width, or the gap between the arms of the flexure, can be adjusted during the optimization process depending on the resulting stress fields and desired operational parameters. Increasing the kerf width usually results in lower spring rates, less mass, and higher modal frequencies. The ideal kerf width yields reduced system mass and vibration.

A multi-arm spiral flexure is a multiple degree-of-freedom system that has many modes. Therefore, a modal analysis must be utilized to ensure none of the modal frequencies are close to

the operating frequency or multiples of the operating frequency. The first mode of modal analysis is the natural frequency of the flexure, describing the situation when it is supporting no additional mass. The natural frequency must be greater than the operating frequency to allow each flexure to reciprocate additional mass. The greater the natural frequency of a single flexure, the fewer total number of flexures required to achieve the designed operating frequency. The 2nd and 3rd modes of modal analysis are the flexure rocking frequency about the in-plane axis. The 4th and 5th modes of modal analysis represent what is referred to as “surge” frequency and are considered to be the most important modes. If the surge frequency is near a flexure harmonic, the flexure arms can contact each other, resulting in quick failure of the component.

A spiral flexure can have any number of arms, but three is the most commonly utilized in designs. The number of arms to be used for a flexure is highly dependent upon the design requirements. Including more arms in a flexure design usually results in higher axial spring rates and lower radial stiffness. Qiu et al. [77] made a spring rate comparison between flexures having the same physical size, amplitude, and thickness, but the number of arms was varied. The results of this comparison are shown in Table 1. Generally, for a flexure with a diameter of two inches or less, the maximum axial spring rate occurs when four or five arms are included in the design. For flexures with a diameter of six inches or greater, the maximum axial spring rate occurs with approximately eight arms [77].

Table 2: Spring Rate Comparison of Flexures Based on Number of Arms [8]

	3 arms	4 arms	6 arms	8 arms
k_{axial} (N/m)	25643	31138	37000	43960
k_{radial} (N/m)	838964	758970	651146	583931

2.7 Radiation Shields

Formulations exist in the literature for heat transfer characteristics of radiation shields for common geometries. Jabbari and Saedodin [82] and Torabi et al. [83] performed calculations for concentric spheres and concentric hemispheres, respectively. Barforoush and Saedodin [84] presented experimental and numerical analyses of the effect of using multiple concentric cylinders as radiation shields. Siegel [85] presents radiation heat transfer equations for multiple parallel plate shields, as well as for concentric cylinders and spheres. Martini [86] provides an equation for calculating the heat transfer due to radiation in a cylindrical displacer:

$$CQ = FA * FM * FN * (\pi/4) * DB^2 * \sigma * (TH^4 - TL^4) \quad \textbf{Equation 4}$$

where CQ is the radiation heat loss, FA is the area factor, FM is the emissivity factor, FN is the radiation shield factor, DB is the cylinder diameter, σ is the Stefan-Boltzmann constant, and TH and TL are the hot and cold surface temperatures, respectively. Though this is a good approximation for the radiation heat loss, the equation does not consider the spacing of the radiation shields within the displacer and it assumes the displacer is a perfect cylinder.

As demonstrated by Chia and Kiang [87], the heat transfer due to radiation is reduced by a certain factor based on how many shields are employed, though there is a point at which increasing the number of shields results in a negligible change in the factor. Multiple radiation shields are usually employed in a displacer, with the quantity based on acceptable heat transfer characteristics and geometric considerations.

Maiorova et al. [88] conducted an optimization study of a thermal protection system involving radiation shields for reusable space vehicles, in which up to ten shields were considered made of two different materials (nickel and aluminum oxide). The position of the shields was also considered. It was determined that utilizing one or two shields led to an optimized design.

Kasthuriangan et al. [89] conducted an experimental study on the performances of single and two-stage pulse cryocoolers with and without radiation shields. The vacuum level was also varied. It was observed that both the single and two-stage pulse cryocoolers were able to achieve lower temperatures with radiation shields installed. Sun et al. [90] performed an analysis involving adjustment parameters for thermal control methods for an orbital transfer vehicle involving radiation shields. Seven adjustable parameters were considered, which were associated with the thermal and radiative properties of the shields.

The emissivity of a material may vary greatly depending on its temperature and surface finish [86]. Keller et al. [91] reported the emissivity of Inconel 718 “as-received” (sheets) from the manufacturer to be in the range of 0.21-0.28 in the temperature interval from 760-1275 K, whereas Tanda and Misale [92] reported “as-received” (EDM machined) emissivity values of approximately 0.36-0.46 in the temperature interval from about 375-580 K. Kieruj et al [93] observed the emissivity of Inconel 718 to be approximately 0.16 in the range of 200-600°C, and 0.28 at 800°C. A variety of materials can be applied to a surface to reduce its emissivity, such as coatings, films, foils, and paints [94]. Coatings with silver nanoparticles have been investigated [95].

2.8 Additive Manufacturing

Additive manufacturing is becoming increasingly utilized for the manufacturing of components that would be difficult to fabricate through conventional processes, with applications in a variety of fields. One such field benefitting from this technology is the biomedical field. Maji et al. [96] presented the development of a patient-specific femoral prosthesis made through rapid prototyping. Demir and Previtali [97] investigated the production of cardiovascular stents through selective laser melting. Costa et al. [98] developed a scaffold material for bone regeneration, which

was produced through additive manufacturing. Berretta et al. [99] developed and studied the characteristics of cranial implants made through laser sintering. The field of engineering is another beneficiary of advancements in additive manufacturing. The potential benefits of additive manufacturing on the aerodynamics, structures, and materials utilized for unmanned aerial vehicles (UAVs) were explored by Goh et al. [100]. Scheithauer et al. [101] presented the pros and cons of developing ceramic heat exchangers of complicated geometries through lithography-based ceramic manufacturing (LCM). Robinson et al. [102] developed a small, water-cooled heat sink through additive manufacturing that includes micro-jet arrays and micro-channels. Kirsch and Thole [103] compared the pressure loss and heat transfer performance of pin fin arrays made through laser powder bed fusion to those made through conventional processes.

Additive manufacturing has some challenges, including anisotropic material properties, manufacturing-induced porosity, small build volumes, and cost. The anisotropic properties and porosity aspects are due to the layer-by-layer printing technique [100]. However, some additive manufacturing processes can produce specimens that rival or surpass those made through conventional methods in terms of mechanical properties. Trosch et al. [104] compared the properties of Inconel 718 components made by Selective Laser Melting (SLM), also known as rapid prototyping, to those made through conventional methods. It was shown that SLM could produce parts with better mechanical properties than parts made through casting and forging at room temperature. One major limitation to current additive manufacturing techniques is that the thickness of the generated components is limited by the particle size of the powder metal used and well as heat dissipation in metals [105]. This limitation on size was considered in the design of the foils of the regenerator presented in this paper. Based on the feedback of several additive manufacturing vendors, a minimum foil thickness of 300 microns is achievable.

2.9 Sage Modeling

The initial design of a Stirling engine starts with a thermodynamic modeling tool. In this study, the Sage produced by Gedeon Associates [106] was used for this purpose. Using the Sage key design parameters such as the diameters, lengths, fin spacing, temperature, pressures, etc. of the system and components can be determined as well as an estimated system efficiency. Sage software will be utilized to model the Stirling cycle with various foil regenerator configurations. The impact of replacing a random fiber regenerator with an additively manufactured regenerator on the performance of the Stirling converter will be observed. The original Sage model is shown in Figure 10.

Figure 10: Sage Model

Chapter 3: Heater Head Design for Reduced Conduction Losses

3.1 Effect of Shape

Inconel 718 was selected as the alloy to be utilized for these simulations due to its corrosion resistance at elevated temperatures. Inconel 718 has excellent creep rupture strength at temperatures up to 700°C, and is considered to be superior to other aerospace materials in terms of its corrosion resistance and weldability [19]. Brinkman et al. [107] provides an equation for the rupture life of Inconel 718 based on stress and temperature:

$$\log t_r = 162.319 - 193.662 \log \sigma + 88.117 (\log \sigma)^2 - 12.807 (\log \sigma)^3 - 0.01052 T \log \sigma \quad \text{Equation 5}$$

Where t_r is the rupture life (hrs), σ is the stress (MPa), and T is the temperature (K). Equation 5 will be utilized to estimate the life of the heater head.

The heater head considered in this work is composed of two sections. Five different geometries were tested for the bottom section, whereas the top section was kept the same in all the assemblies. The first geometry is a constant thickness wall from the top to the bottom. The second type is a single linear taper with a maximum thickness at the top to a minimum thickness at the bottom. The third type is a double taper, which starts at the maximum thickness at the top, goes to the minimum thickness at middle, and remains at the minimum thickness until the bottom. The fourth type is a concave curve to the midpoint, and the fifth type is a convex curve to a height greater than the midpoint. A representation of the geometries of the bottom halves of the heater heads tested is given in Figure 11.

Figure 11: Representation of Heater Head Geometries

A steady-state thermal and a static structural analysis was performed simultaneously for each of the five geometries in ANSYS Workbench. The thermal and structural boundary conditions are shown in Figure 12 and Figure 13, respectively. As shown in Figure 12, a temperature of 80°C was applied to the entirety of the outer shell of the low temperature heat exchanger and the bottom ledge of the top section of the heater head that extends into the low temperature heat exchanger region. A temperature of 700°C was applied to the outer surfaces of the top half of the heater head. As shown in Figure 13, a pressure of 3.5 MPa was applied to all the inner surfaces of the heater head, a frictionless support was applied to all surfaces on the XY plane to represent symmetry, and the bottom surface of the outer shell of the low temperature heat exchanger was fixed. The two sections of the heater head were modeled as Inconel 718, and the shell of the low temperature heat exchanger was modeled as the default stainless steel material in ANSYS Workbench. The stress contours for the top and bottom halves of the heater head assembly for two of the cases are shown in Figure 14 through Figure 17. The stress contours for all five cases were similar, with the top and bottom portions of the heater head having maximum stress values of approximately 53 MPa and 220 MPa, respectively. Utilizing Equation 2 for the top, higher temperature section of the heater head, a temperature of 973.15 K and pressure of 53 MPa results in an expected life of $1.17\text{E}7$ hours.

Figure 12: Heater Head Temperature Boundary Conditions

Figure 13: Heater Head Static Structural Boundary Conditions

Figure 14: Heater Head Top Stress Contour for Constant Wall Thickness

Figure 15: Heater Head Bottom Stress Contour for Constant Wall Thickness

Figure 16: Heater Head Top Stress Contour for Convex Above Midpoint

Figure 17: Heater Head Bottom Stress Contour for Convex Above Midpoint

The temperature profile of both sections of the heater head for the constant thickness profile and the convex to above the midpoint simulations are given in Figure 18 and Figure 19, respectively.

Figure 18: Constant Thickness Geometry Temperature Profile

Figure 19: Convex to Above Midpoint Temperature Profile

Figure 20 compares the temperature profiles of the bottom sections of the heater heads for all five simulations. The conduction losses along the wall of the heater head due to resulting temperature profile for each case are presented in Table 1. The table shows the geometry with a constant wall thickness has the greatest conduction losses at 113.4 W. Utilizing any form of a taper results in a significant reduction in conduction losses, as the geometry with the second greatest conduction loss value, the single taper, shows a 23.8% reduction. The geometry with a convex

curve to above the midpoint of the bottom half of the heater head has a 37.7% reduction in conduction losses compared to the constant thickness geometry.

Figure 20: Temperature Profile through the Bottom Segment of the Heater Head Based on Geometry

Table 3: Conduction Losses in the Axial Direction of the Heater Head Wall Based on Geometry

Geometry	Conduction Losses (W)
Constant Thickness	113.4
Single Taper	86.4
Double Taper	73.1
Convex to Midpoint	75.1
Convex to Above Midpoint	70.7

The effect of heater head geometry on conduction losses along the wall was observed in this work. Five cases were considered, which included a constant thickness geometry and four variations of tapering to a minimum thickness. Effective tapers result from keeping the thickness of the heater head to a minimum to reduce conduction losses, and increasing the thickness at the hot end to improve durability and creep resistance. The results of the simulations performed for this work show keeping the thickness of the heater head constant at the maximum value lead to the greatest conduction losses of all the cases considered. Tapering significantly reduced the

conduction losses; the remaining four cases had losses 23.8 to 37.7% less than that of the constant thickness heater head. The estimated life of all the heater head assemblies is approximately 1.17E7 hours.

3.2 Effect of Length of Minimum Wall Thickness

An analysis was performed to observe the factor of safety and conduction losses based on the dimensions of the lower section of the heater head. The dimensions A, B, and L were constant, and C varied so that $0.05 \leq C/L \leq 0.95$ (these dimensions are shown in Figure 21).

Figure 21: Dimensions Considered for Heater Head Analysis

Figure 22 plots the factor of safety and the conduction losses through the wall of the heater head based on C/L ratios. The plot shows that factor of safety increases and the conduction losses decrease as the C/L ratio increases, both of which are desired results. However, based on Equation 5, C/L must be 0.85 or less in order to provide a lifetime of at least 30 years. If C/L is 0.95, the expected lifetime is only approximately 5 years.

Figure 23 and Figure 24 are stress contours for C/L values of 15% and 95%, respectively, which are depictions of the minimum and maximum limits of the values tested. The plots show

that the stress is more concentrated near the bottom of the heater head for the 15% case, whereas the stress is somewhat more distributed throughout the length of the wall for the 95% case.

Figure 22: Factor of Safety vs. Conduction Losses for Various C/L Ratios

Figure 23: Stress Contour for C/L = 15%

Figure 24: Stress Contour for C/L = 95%

3.3 Heater Head Dimensions and Manufactured Product

Figure 25 and Figure 26 are drawings with dimensions for the final version of the heater head produced by additive manufacturing and after finish machining, respectively. Direct metal laser sintering (DMLS) was utilized to develop the heater head, and then finish machining was performed to drill holes for bolts, apply tapering, smooth surfaces, improve tolerances, etc. The finished product can be seen in Figure 27.

Figure 26: Heater Head Dimensions after Finish Machining

Figure 27: Manufactured Heater Head

3.4 Conclusions

A head heater head was designed and developed with AM that allowed for a previously unattainable geometry with a fully integrated pressure vessel and heat exchanger that could significantly reduce detrimental dead volumes. Based on the results of the work done for Task 1, it is recommended that the heater head wall have some form of tapering; considered taper options had losses that were 23.8 to 37.7% less than that of a constant thickness heater head. Also, the wall should be as thin as possible, though adequate thickness should be allowed near the high temperature regions to provide more resistance toward creep, thus extending the lifetime of the heater head.

Chapter 4: Regenerator

4.1 Methodology

To investigate the robustness of the proposed additively manufactured regenerator design, a finite element analysis (FEA) was conducted. Based on the feedback of the additive manufacturing vendors consulted, it was proposed that the regenerator be made of several sections to ensure the foils remain straight and do not deform during the manufacturing process as the tolerance of the printing process decreases with the height of the foils. The foils have a thickness of 300 μm with a foil spacing of 300 μm . A CAD model of the top section of the regenerator is shown in Figure 28. The top portion of the regenerator was curved to match the profile of the heater head to minimize detrimental dead volumes in the knuckle region of the heater head between the heat acceptor and regenerator. A comprehensive report on the effect of dead volumes in a Stirling engine was given by Puech and Tishkova [108]. The model of the regenerator was imported into ANSYS Workbench where a $\frac{1}{4}$ symmetric model was developed. A $\frac{1}{4}$ model was used to reduce the total computational size of the model and decrease the overall run time. A mesh independence study was conducted where the mesh size was decreased from 1 mm to 0.5 mm. Table 4 shows the maximum stress at various mesh sizes. Ultimately, a mesh size of 0.5 mm was chosen, as the maximum stress in the regenerator remained unchanged.

Table 4 – FEA Mesh Independence Study

Element Size	Number of Nodes	Maximum Stress
1 mm	1,331,462	126.6 MPa
0.9 mm	1,582,256	129.1 MPa
0.75 mm	2,182,665	129.3 MPa
0.5 mm	4,587,690	128.1 MPa

Figure 28: Model of Regenerator Top Segment

4.2 Results

First, a steady state thermal simulation was conducted to determine the temperature distribution within the regenerator. This temperature distribution was then used as the input to a static structural simulation to examine the thermally induced stresses present in the regenerator. For the thermal simulations, the bottom surface of the regenerator was set to 80 °C while the top surface of the foils was set to 810 °C. The regenerator is made from Inconel 718 and the properties used in the simulations are listed in Table 5. Both the thermal conductivity and thermal expansion coefficient were functions of temperature. The resulting temperature distribution is shown in Figure 29. As expected, a primarily linear distribution forms over the length of the regenerator. In the top section of the regenerator, the temperature contours are curved slightly because the outer diameter of the regenerator is hotter than the inner diameter. This occurs because the top surface of the regenerator foils was set to 810 °C and the outer foils are shorter leading to the increased temperature at the same axial location.

In addition to the temperature distribution within the regenerator, the axial conduction losses were also examined. The main drawback previously seen in literature with an axially continuous type regenerator were the significantly higher axial conduction losses seen compared

to a random fiber regenerator, which has no clear axial conduction path. Although the regenerator is likely to be produced in multiple sections to increase the tolerance of the additive manufacturing process, it was modeled as a continuous solid body. A total of 198.5 W of axial conduction losses were predicted over the 110 mm length of the designed regenerator. As previous work in the literature has stated ([9], [64]), segmentation of the regenerator will significantly reduce the total conduction loss. For example, Nam and Jeong [64] saw a 1% decrease in the ineffectiveness of their parallel wire regenerator when seven-segmentations were used over none during experimental testing.

Table 5 – Thermal Properties of Inconel 718 [109], [110]

Temperature (°C)	Thermal Expansion Coefficient (1/°C)	Thermal Conductivity (W/m°C)	Specific Heat (J/kg°C)
21	1.229×10^{-5}	11.7	430
538	1.449×10^{-5}	19.0	560
871	1.728×10^{-5}	23.9	645

Figure 29: Regenerator Temperature Distribution

The resulting temperature profile was used as the input to a static structural analysis. For the structural analysis, the bottom edge of the outer ring of the regenerator was fixed. Unlike in

previous engine designs where the regenerator was sandwiched between the hot and cold heat exchangers, this design uses an outer ring at the base of the regenerator to fix the regenerator in place. The outer ring is clamped between the bottom of the heater head and the rejection system. In addition, frictionless supports were applied to the symmetry planes to simulate the symmetric boundary conditions. The resulting stress distribution in the regenerator is shown in Figure 30. The maximum stress occurs in the outer most foil near the regenerator ribs. A similar stress distribution is seen in the other foils. A maximum stress of 128.1 MPa is predicted. This is well below the yield stress of Inconel 718 at 760 °C, which is roughly 740 MPa. This yields a margin of safety of 4.9. An additional area of stress concentration is seen where the outer ring is attached to the regenerator ribs. A maximum stress of 94.3 MPa is seen in this area. The temperature at the locations is roughly 86 °C and the yield stress of Inconel 718 is 1185 MPa at 70 °C leading to a factor of safety of 11.6 in this region. It is unlikely the regenerator will fail due only to thermally induced stresses.

Figure 30: Regenerator Stress Distribution

The axial deformation of the regenerator is shown in Figure 31. The total length of the regenerator increases by 0.74 mm because of thermal expansion. The radial deformation of the regenerator and a detailed view of the deformation of the top section of the regenerator foils are

shown in Figure 32. A maximum radial displacement of 0.58 mm occurs at the top of the outer most regenerator foils. While this deformation may seem alarming and that the foil spacing will not be maintained, the innermost channel radially deforms 0.38 mm. and the deformation from foil to foil is only 0.01 mm. The maximum deformation seen along a single foil is under 0.07 mm and occurs in the outer most foil. At the bottom of the regenerator, the foils deform 0.01mm. Therefore, despite the radial deformation that occurs in the hot portion of the foils, the radial spacing between the foils remains mostly uniform across the width of the regenerator.

Using the one-dimensional modeling tool Sage, the effect of replacing a 90% random fiber regenerator with a foil regenerator was examined. At present, additive manufacturing cannot produce foils thinner than 200 μm . If the regenerator in an existing engine is replaced with a 200 μm foil regenerator the efficiency of the Stirling cycle decreases because the surface area of the regenerator decreased. For the engine examined, replacing the random fiber regenerator with a 200 or 300 μm foil regenerator resulted in a 5.3 and 7.3 percent decrease in the cycle efficiency. If 100 μm foils were achievable, a 3.6 percent increase in efficiency is possible. While this thickness is currently unattainable, the results indicate the improvement using a foil over random fiber regenerator can still be achieved on the performance of a Stirling engine.

Figure 31: Regenerator Axial Deformation

Figure 32: Regenerator Radial Deformation

Table 6 is a summary of the results obtained from FEA for the regenerator. The maximum stress is well below the yield stress of Inconel 718 at 760 °C, which is approximately 740 MPa. As explained before, the gaps between the foils will be maintained, so the axial and radial deformation values are acceptable.

Table 6: FEA Results for the Regenerator

Maximum Stress (MPa)	Axial Deformation (mm)	Radial Deformation (mm)
128.1	0.74	0.58

4.3 Regenerator Dimensions and Manufactured Product

Figure 33 through Figure 35 show the three different types of regenerator components to be manufactured. The regenerator was divided into different sections to improve tolerance and reduce conduction losses. Three middle components will be utilized to assemble the regenerator.

Figure 33: Regenerator Top Section Dimensions

Figure 35: Regenerator Bottom Section Dimensions

Figure 36: Additively Manufactured Regenerator Top Segment

4.4 Conclusions

The potential improvement to the efficiency of a Stirling engine by using a foil type regenerator was investigated. The resulting stresses were well below the yield stress of Inconel at both high (810 °C) and low (80 °C) temperatures leading to margins of safety over four. A total axial deformation of 0.74 mm occurs while the regenerator radially deforms 0.58 mm. Although the radial deformation seems concerning, the maximum deformation along a single foil is under 0.07 mm and the deformation from foil to foil is only 0.01 mm. Therefore, the spacing between the foils is maintained and the performance of the regenerator should not change after numerous thermal cycles. Lastly, the flow losses in the regenerator were examined using a CFD model. Currently, the model uses constant properties to gain preliminary information about the flow dynamics of the regenerator. The predicted pressure drop from the CFD model agreed well with the predictions of the Sage model. An experimental test rig is being designed to further examine the behavior of the regenerator and validate the numerical model. Using the predicted pressure drop, the friction factor for the regenerator over a range of Re numbers was calculated. The friction losses in the proposed foil regenerator were significantly lower than those seen in both a random fiber and woven screen regenerator, particularly at higher Re numbers. Based on the investigation of the designed regenerator, additively manufactured foil regenerators can be used to significantly improve the efficiency of the next generation of Stirling engines.

Chapter 5: Flexure Bearings

5.1 Methodology

The five materials considered in this work are the default stainless steel, copper alloy, and titanium alloy from ANSYS Workbench, Sandvik 7C27Mo2 [111], and Beryllium Copper. The default stainless steel and titanium alloy has properties similar to that of Stainless Steel 18-8 and Ti-6Al-4V, respectively. Though the grade of copper alloy is not specified in ANSYS Workbench, copper alloy in general is a metal with a copper content between 50-94% [112]. Sandvik 7C27Mo2 is a hardened and tempered martensitic chromium steel developed by the Sandvik Group, a company based in Stockholm, Sweden. Figure 37 is a S-N curve for the material up to a maximum thickness of 1 mm. Property values for Beryllium Copper were obtained from Cardarelli [113] and Budynas and Nisbett [79]. Table 7 lists the material properties relevant to the performed FEA simulations. When performing fatigue analysis for an actual mechanical component (rather than a material specimen), S_e' is reduced to S_e , which is less than $0.5S_{ut}$ [79]. For steels, Budynas and Nisbett [79] recommend utilizing S_e' values of $0.5S_{ut}$ and 700 MPa for S_{ut} values less than 1400 MPa and greater than 1400 MPa, respectively. This agrees with the S-N curve shown in Figure 37, and this criterion will be used for the default stainless steel from ANSYS Workbench. The value of S_e' for copper alloy, titanium alloy and Beryllium Copper were obtained based on data from the literature [114]–[117]. It should be noted that Sandvik 7C27Mo2 has the greatest S_e' value of the five materials considered, which will lead to greater life cycles.

Figure 37: Sandvik 7C27Mo2 S-N Curve [111]

Table 7: Material Properties Utilized for Flexure FEA

Material	ρ (kg/m ³)	ν	E (GPa)	S_{ut} (MPa)	S_e' (MPa)
Stainless Steel	7750	0.31	193	586	293
Sandvik 7C27Mo2	7700	0.31	210	1800	700
Copper Alloy	8300	0.34	110	430	180
Beryllium Copper	8219	0.285	124	1300	276
Titanium Alloy	4620	0.36	96	1070	380

A sweep mesh, composed of solid-shell elements, with 10 elements through the thickness of the flexure was developed in ANSYS Workbench 18.1. A 0.4 mm element face mesh was applied to all surfaces on the x-y plane of the model. The resulting mesh is shown in Figure 38. For the static structural simulations, two cases were examined; one where just the outer rim is fixed (rim clamped) and one where the outer rim with a portion of the flexure arms is held fixed (radially clamped). The inner ring was displaced by 5.47 mm in both cases. The setup of the boundary conditions is shown in Figure 39. The results considered from the static structural simulations were

the equivalent (von-Mises) stress plots of the whole flexure and a force reaction probe, applied to the displacement boundary condition. The force probe was utilized to obtain values for axial spring rates (k_{axial}) by dividing the resulting force from the probe (F_R) by the displacement (δ), as shown in Equation 3.

$$k_{axial} = F_R / \delta \quad (3)$$

Separate modal analyses were performed to obtain the natural frequency, as well as the second through fifth modal frequencies, for the flexures. Just the fixed supports shown in Figure 39 were utilized for the modal analyses, not the displacement conditions.

Figure 38: Flexure Bearing Mesh

Figure 39: Boundary Conditions for both Rim (Left) and Radially (Right) Clamped Flexures

A sensitivity analysis was conducted to observe the effects of modifying kerf width and shape factor on stress and axial spring rate. Since the radially clamped boundary condition displayed reduced stress concentrations and greater spring rates than clamping at the outer ring, this clamping method was utilized in the analysis. Aspects of the flexure geometry utilized are listed in Table 8. During the analysis, all variables were kept constant except for either the kerf width or the shape factor. Figure 40 shows the nine different configurations considered in the analysis.

Table 8: Flexure Geometry Considered in the Sensitivity Analysis

Material	Sandvik 7C27Mo2
Number of Arms	Six
θ_0	325° (5.672 radians)
R_i	5 mm
R_o	30 mm
Thickness	1 mm
Baseline Kerf Width	1.2 mm
Baseline f	0.04

Figure 40: All Flexure Geometries Utilized in the Sensitivity Analysis

5.2 Results

Figure 41 shows the equivalent (von-Mises) stress results of a Sandvik 7C27Mo2 flexure with both applied clamping methods. The same results were generated for the other four materials. The maximum stress point for the ring clamped flexures was located on the inside corner where the arms meet the ring, and the maximum stress for the radially clamped flexures occurred where the arms attached to the fixed support. For all materials, the radially clamped condition resulted in maximum stress values 26.3%-28.2% less than those obtained with the outer ring clamped, and the stress distribution is more even along the arms.

Figure 41: Stress contour for Sandvik 7C27Mo2 Flexures Clamped at the Outer Rim (Left) and Radially (Right)

Based on the maximum stress values from the FEA results and the fatigue limits of the materials, it was determined that a radially clamped Sandvik 7C27Mo2 or titanium alloy flexure

could provide infinite cycles for the specified conditions since $S_{\max}$ is less than S_e for those conditions.

Table 9: Maximum Stress Comparison

Material	$S_{\max}$ (MPa), Ring Clamped	$S_{\max}$ (MPa), Radially Clamped	S_e (MPa)
Stainless Steel	879.7	631.5	293
Sandvik 7C27Mo2	957.2	687.1	700
Copper Alloy	497.3	358.9	180
Beryllium Copper	569.3	419.5	276
Titanium Alloy	431.7	314.5	380

Table 10 shows the calculated axial spring rates of the flexures based on the results from the force probe and the given displacement of 5.47 mm. The radially clamped flexures have greater spring rates due to the fixed support, which is applied on a small portion of each arm.

Table 10: Axial Spring Rates

Material	Ring Clamped		Radially Clamped	
	F_R (N)	k_{axial} (N/m)	F_R (N)	k_{axial} (N/m)
Stainless Steel	60.0	11.0E3	62.5	11.4E3
Sandvik 7C27Mo2	65.3	11.9E3	68.0	12.4E3
Copper Alloy	33.6	6.14E3	35.0	6.40E3
Beryllium Copper	39.1	7.15E3	40.8	7.46E3
Titanium Alloy	29.0	5.30E3	30.1	5.50E3

Table 11 lists the first five modal frequencies of the flexures obtained through modal analysis. The modal frequencies for the radially clamped flexures are greater than those clamped at the ring. These results work in favor of the radially clamped flexures because greater natural frequencies lead to a fewer total number of flexures required to obtain the desired operating frequency.

Table 11: Modal Analysis Results

Material	Mode Number	Frequency (Hz), Outer Ring Clamped	Frequency (Hz), Radially Clamped
Stainless Steel	1	213.1	219.3
Stainless Steel	2	389.5	398.6
Stainless Steel	3	393.6	402.7
Stainless Steel	4	514.0	525.4
Stainless Steel	5	514.3	525.8
Sandvik 7C27Mo2	1	223.0	229.5
Sandvik 7C27Mo2	2	407.6	417.1
Sandvik 7C27Mo2	3	411.9	421.5
Sandvik 7C27Mo2	4	537.9	549.8
Sandvik 7C27Mo2	5	538.2	550.2
Copper Alloy	1	154.3	158.7
Copper Alloy	2	283.2	289.8
Copper Alloy	3	286.3	292.9
Copper Alloy	4	373.6	381.9
Copper Alloy	5	373.9	387.4
Beryllium Copper	1	166.9	171.9
Beryllium Copper	2	303.9	311.1
Beryllium Copper	3	307.2	314.3
Beryllium Copper	4	401.3	410.2
Beryllium Copper	5	401.5	410.5
Titanium Alloy	1	192.3	197.7
Titanium Alloy	2	354.0	362.2
Titanium Alloy	3	357.8	365.9
Titanium Alloy	4	466.8	477.1
Titanium Alloy	5	467.1	477.4

Figure 42 is the stress contour for the baseline flexure geometry of the sensitivity analysis. The maximum stress and spring rate for this geometry were used for comparison purposes to describe the significance of changing kerf width or shape factor.

Figure 42: Stress Contour for the Baseline Flexure Geometry ($f = 0.04$ and Kerf Width = 1.2 mm)

Figure 43 shows the stress contours of the cases where the shape factor was kept constant at 0.04 and the kerf width varied between 0.6 and 1.8 mm. The resulting maximum stress and spring rate values for these simulations are listed in Table 12. Both the stress and spring rate decrease with increasing kerf width.

Table 12: Constant Shape Factor Results

Shape Factor	Kerf Width (mm)	Maximum Stress (MPa)	Spring Rate (kN/m)
0.04	0.6	859	18.3
0.04	1.0	786	15.6
0.04	1.2	747	14.2
0.04	1.4	711	12.9
0.04	1.8	633	10.4

Figure 43: Flexure Stress Contours for Various Kerf Widths and Shape Factor of 0.04

Figure 44 shows the stress contours for the flexure geometries with a constant kerf width of 1.2 mm, and the shape factor varied between 0 and 0.08. The stress and spring rates for these cases are listed in Table 13. The results show the stress and spring rate decrease with increasing shape factor. However, the stress increases when the shape factor goes from 0.06 to 0.08. It should be noted that the maximum stress in this situation changes location from where the arm meets the

clamped region to a thin portion of the arm, as can be seen in Figure 44. This shows further increasing the shape factor will negatively affect the stress concentration.

Figure 44: Flexure Stress Contours for Various Shape Factors and Kerf Width of 1.2 mm

Table 13: Constant Kerf Width Results

Shape Factor	Kerf Width (mm)	Maximum Stress (MPa)	Spring Rate (kN/m)
0	1.2	814	14.2
0.02	1.2	778	14.4
0.04	1.2	747	14.2
0.06	1.2	616	13.6
0.08	1.2	655	12.4

Figure 45 and Figure 46 are the plots depicting how the results of the sensitivity analysis compare to those of the baseline geometry. The plots show the kerf width and shape factor varied from the baseline value by $\pm 50\%$ and $\pm 100\%$, respectively. Within the ranges considered, for the kerf width variation cases, the stress and spring rates changed by as much as 26.3% and 43.5%, respectively, from the maximum values obtained. On the other hand, for the shape factor variation cases, the stress and spring rates changed by 19.5% and 12.7%, respectively, from the maximum values obtained. Therefore, for the dimensions considered, variations in kerf width had a more significant impact on stress and axial spring rates.

Figure 45: Maximum Stress and Axial Spring Rate for Varying Kerf Width and Constant Shape Factor

Figure 46: Maximum Stress and Axial Spring Rate for Varying Shape Factor and Constant Kerf Width

5.3 Experimental Validation

The FEA results were verified by fabricating Sandvik 7C27Mo2 flexures through waterjet cutting and testing them in a flexure test rig. Each flexure had six arms, a R_o of 32 mm, and a thickness of 1 mm. The flexures were post-processed with a shot-peening system to improve the regions where they would be clamped. A flexure mask was developed to ensure only the clamped regions would be affected by the shot-peening process. Figure 47 shows the flexures before and after shot-peening. Figure 48 shows the dimensions of the produced flexure.

Figure 47: Flexure Developed through Waterjet Cutting (Left), Flexure Mask (Center), Flexure after Shot-Peening (Right)

Figure 48: Flexure Bearing Dimensions

Figure 49 shows the test rig utilized to validate that the flexures would be able to achieve an infinite number of cycles. A stack of ten flexures was subjected to an axial displacement of 5.4 mm at 45 Hz for more than 10^7 cycles. Based on Figure 37, if the flexure has passed 10^7 cycles without failure, it is reasonable to conclude the component will be able to provide an infinite number of cycles. This is a proven method for testing flexures; Qiu et al. [77] utilized the same approach to design flexures for a 10 W Stirling converter that operated at 65% over-stroke without fail for more than 80,000 hours.

Figure 49: Flexure Test Rig

5.4 Conclusions

In this work, the impact of the flexure clamping method on stress, fatigue life, axial spring rates, and modal frequencies was investigated with FEA, conducted with ANSYS Workbench 18.1. Two options were considered, which were flexures clamped at the outer ring and flexures clamped radially along a small portion of each arm. Five materials were utilized, which were the default stainless steel, copper alloy, and titanium alloy from ANSYS Workbench, Sandvik 7C27Mo2, and Beryllium Copper. It was determined that the clamping method had a significant effect on stress and fatigue life; the radially clamped condition resulted in maximum stress values 26.3%-28.2% less than those obtained with the outer ring clamped. The radially clamped flexures

had greater modal frequencies, which is favored because greater natural frequencies lead to a fewer total number of flexures required to obtain the desired operating frequency. The radially clamped Sandvik 7C27Mo2 and titanium alloy flexure designs will provide infinite cycles. This was validated with laboratory testing of Sandvik 7C27Mo2 flexures. An additional sensitivity analysis was performed to observe the effect of changing either kerf width or shape factor while keeping all other variables constant. It was determined, for the dimensions and degree of variations utilized, that kerf width had a greater impact on maximum stress and axial spring rate. Based on the results, it is recommended that flexure designs incorporate a clamping mechanism that involves a small portion of each arm for increased spring rate and lower stress values during engine operation. It is also recommended to utilize lower kerf widths and shape factors.

Chapter 6: Radiation Shields

6.1 Methodology

As stated by Kobus and Tison [118], the two main types of radiation shield models are lumped method and numerical analysis. Due to the relative complexity of the geometry involved, a FEA approach with ANSYS Workbench was utilized to determine the steady-state temperature profile throughout the displacer assembly. This approach was similar to the work done by Nam et al. [119], in which 3D FEA models were developed to perform the thermal analysis associated with a heat shield. The designed displacer has a radius of 34.35 mm with an overall height of 174.66 mm. . The radiation shields had a thickness of 0.1 mm. The material properties of Inconel 718 were utilized for all components of the assembly. Both the specific heat and thermal conductivity varied with temperature, Table 14, and a constant density of 8220 kg/m³ was used [120]. The effect of distance between the high-temperature surface and the radiation shields as well as the spacing of the shields was investigated. Additionally, the effect of emissivity on the radiation losses was examined.

Table 14: Thermal Properties of Inconel 718

Temperature (°C)	Specific Heat (J/kg°C)	Thermal Conductivity (W/m°C)
21	430	11.4
538	560	19.6
871	645	24.9

Figure 50 depicts how the boundary conditions were applied to the displacer assembly for the steady-state thermal analyses. A temperature of 750 °C and 80 °C was applied to the top and bottom surfaces of the displacer assembly, respectively. All the internal surfaces were subjected to a surface-to-surface radiation condition. The emissivity values of the high-temperature inner surfaces of the displacer were set to 0.7, and all the other surfaces were set to 0.3. An additional

set of simulations were performed in which the emissivity of all the internal faces was set to 0.8 in order to observe the importance of surface finish on the radiation heat transfer. The radiation heat transfer is modeled using the surface-to-surface method. All bodies are treated as gray-diffuse surfaces. Therefore, the emissivity and absorptivity are equal, independent of wavelength and direction. Symmetry regions were applied to the faces on the XY and YZ planes to employ a 1/4 model. According to analytical and experimental studies presented by Wood et al.[120], the convective heat transfer within the displacer is well suppressed by the implementation of the shields. Therefore, the possible convection losses due to the motion of gas contained inside the displacer were not considered in this study. Figure 50 shows the type of mesh that was utilized for all the simulations conducted in the study. The maximum element size was set to 1.5 mm and resulted in approximately 102,514 nodes. In addition to the temperature profiles, the heat flux into the displacer body was examined. The dimensions of the radiation shields to be utilized are shown in Figure 51.

Figure 50: Boundary Conditions and Displacer Assembly Mesh for the Simulations

6.2 Results and Discussion

The effects of employing radiation shields on the temperature distribution with a displacer assembly were investigated using FEA. The effectiveness of the radiation shields can be determined by examining the temperature profile through a vertical cross-section of the entire displacer assembly as well as the heat loss through the displacer body. An example temperature profile is shown in Figure 52 for a configuration that has four radiation shields.

Figure 52: Example Displacer Cross-Sectional Temperature Profile

6.2.1 Effect of Shield Location

Numerous configurations of the radiation shields were considered to gain a greater understanding of the effect position has on their ability to reduce radiation effects between the hot displacer cap and cold displacer body. First, the effect of distance between the hot surface and the radiation shields was investigated. A set of two shields that were 7.5 mm apart were placed 20, 50, 60, and 80 mm from the top of the displacer body, Figure 53. The temperature profile along the

displacer wall for these configurations is given in Figure 54. Two inflection points are seen in the four curves. The first occurs at roughly 42% of the total displacer height, which corresponds to the junction between the displacer body and the displacer cap. The temperature profile in the displacer body is similar for the four heights considered. As the height of the radiation shields is increased, both maximum temperature and thermal gradient in the displacer body decreases. However, there appears to exist an optimal distance between the displacer body and radiation shields as further increasing the distance from 60 mm to 80 mm results in an increase in the maximum temperature of the displacer body as well as the total heat flux. Table 15 lists the maximum displacer body temperature as well as total heat flux into the displacer body.

While the temperature distribution in the displacer body is relatively the same for the three cases considered, it is vastly different in the displacer cap section where the radiation shields are located. A second inflection point is seen in the temperature profile at the location of the radiation shields. For the 20 mm case, this occurs at roughly 54.4% of the total height. A rapid increase in temperature is seen between the top of the displacer and the radiation shield for the 20 mm case. The rate of temperature increase is significantly lower between the radiation shield and the top of the displacer cap. The rate at which the temperature between the displacer body and the radiation shields increases decreases as the shields are moved further away.

Figure 53: Two Shields at 20 mm, 50mm, 60 mm, and 80 mm from the Displacer Body

Figure 54: Displacer Wall Temperature Profile Based on the Location of Two Shields

Table 15: Maximum Displacer Body Temperature and Radiation Heat Loss for the Two Shield Cases at 80 mm, 50 mm, and 20 mm

Location (mm)	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
80	379.0	213.1	52.6
60	362.7	181.6	49.0
50	367.4	204.9	49.8
20	409.4	216.9	57.4

When the number of radiation shields was increased to three, both the temperature and heat flux decreased for all cases. The schematics of the three radiation shield cases for heights of 20, 50, and 80 mm are shown in Figure 55. Two inflection points were again seen in the temperature profile curves, the first at the top of the displacer body and the second at the location of the radiation shields. Again, the results indicate that there is an optimal distance for the location of the radiation shields. This location is approximately located at the center of the displace cap. Increasing the number of radiation shields from two to three resulted in a 5% decrease in the maximum displacer body temperature and a 7% decrease in the total heat flux for the 50 mm case.

Figure 55: Three Shields at 20 mm, 50 mm, and 80 mm from the Displacer Body

Figure 56: Displacer Wall Temperature Profile Based on the Location of Three Shields

Table 16: Maximum Displacer Body Temperature and Radiation Heat Loss for the Three Shield Cases at 80 mm, 50 mm, and 20 mm

Location (mm)	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
80	356.6	201.3	47.9
50	350.5	195.2	46.2
20	407.1	211.1	56.2

As the results indicated that increasing the number of shields reduces the radiation to the displacer body, a series of simulations with two, four, and six shields at a maximum height of 60 and 80 mm from the displacer body were conducted. The schematics are shown in Figure 57 and Figure 58. For these simulations, the two radiation shields are still 7.5 mm apart and there is a gap of 15 mm between the radiation shield sets. A variation on the four-shield design with a 30 mm gap was also considered. The temperature profiles are shown in Figure 59 and Figure 60. For both heights, the six-shield and four-shield variation 2 cases have nearly identical temperature profiles. For the 80 mm height, employing six radiation shields results in the lowest displacer body temperature and heat flux. However, in the 60 mm cases, the temperature increases from four shields to six shields by 2%. The maximum and average temperature along with the heat flux for both the 60 and 80 mm cases are listed in Table 17 and Table 18.

Figure 57: 80 mm to Top Shield, 15 mm between Sets

Figure 58: 60 mm to Top Shield, 15 mm between Sets

Figure 59: Displacer Wall Temperature Profile Based on Number of Shields (60 mm to Top Radiation Shield)

Figure 60: Displacer Wall Temperature Profile Based on Number of Shields (80 mm to Top Radiation Shield)

Table 17: Maximum Displacer Body Temperature and Radiation Heat Loss for the 60 mm to Top Radiation Shield Cases

# of Shields	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
2	363.7	203.6	49.0
4 Var 1	324.2	181.6	40.9
4 Var 2	346.0	185.3	44.2
6	330.1	176.9	41.0

Table 18: Maximum Displacer Body Temperature and Radiation Heat Loss for the 80 mm to Top Radiation Shield Cases

# of Shields	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
2	379.0	213.1	52.6
4 Var 1	327.4	186.0	41.9
4 Var 2	323.8	181.0	40.7
6	303.8	163.5	36.8

6.2.2 Effect of Number of Shields Employed

It appears that using more shields spread over the length of the displacer cap will lead to the lowest heat flux in the displacer body. Therefore, simulations were conducted by increasing the number of radiation shields from one to six with a uniform spacing of 15 mm starting at a distance of 80 mm from the top of the displacer body, Figure 61. Variations of the three- and four-shield designs were considered as seen in Figure 62. A base case with no radiation shields was also examined. The temperature profiles for the zero to six shield uniform spacing cases are shown in Figure 63 and the temperature for the three- and four-shield variations are given in Figure 64. For the zero-shield case, the rate of temperature increase is relatively constant over the length of the displacer cap and the top half of the displacer body (>20% of the total height). Simply using a single radiation shield reduces the temperature seen in the displacer body by 100 °C leading to a 28% decrease in the total heat flux. As the number of shields increases, the temperature of the displacer body decreases and reaches a minimum when five radiation shields are used. However, the decrease in the displacer body temperature is significantly smaller when more than four shields are used. For the five-shield case, the temperature has been decreased by 208 °C and the total heat flux has decreased by 55%. Increasing the number of shields from five to six actually increases the maximum temperature in the displacer body, but the average temperature decreases. The total heat flux only increases by 0.2kW/m². In all cases considered the variations to the three- and four-

shield designs yield a higher displacer body temperature than the uniform spacing configurations. Thus, the radiation shields perform better when they are uniformly spaced versus having large gaps between shields. Additionally, there is an optimal number of radiation shields after which point further increasing the number of shields has a smaller impact on the temperature distribution and can lead to an increase in the displacer body temperature.

Figure 61: 1-6 Shields, 15 mm Spacing between Shields

Figure 62: 15 mm Spacing between Shields Variations

Figure 63: Displacer Wall Temperature Profile Based on Number of Radiation Shields (15 mm between Shields)

Figure 64: Displacer Wall Temperature Profile Based on Number of Radiation Shields Variations

Table 19: Maximum Displacer Body Temperature and Radiation Heat Loss for the 15 mm Spacing between Shields Cases

# of Shields	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
0	518.7	288.4	84.2
1	418.2	233.4	61.0
2	368.4	207.3	50.3
3	336.7	190.6	43.7
4	316.7	179.2	39.5
5	310.8	173.6	38.0
6	314.5	171.0	38.2

Table 20: Maximum Displacer Body Temperature and Radiation Heat Loss for the 15 mm Spacing between Shields Case Variations

# of Shields	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
3	336.7	190.6	43.7
3 Var 1	362.8	193.3	47.6
3 Var 2	373.5	197.0	49.6
4	316.7	179.2	39.5
4 Var 1	344.1	183.9	43.8

6.2.3 Effect of Emissivity

The final set of simulations conducted were conducted with a uniform internal emissivity of 0.8 to examine the impact that surface finish has on the radiation heat transfer. For the zero radiation shield cases, the temperature of the displacer body is 9.3 degrees lower than for the previous 0.7/0.3 emissivity cases, however, the average temperature is 3.4 degrees lower and the total heat flux is only 1.3 kW/m² lower. As with the previous investigation, as the number of shields increases both the temperature and the total heat flux seen in the displacer body decreases. After four shields are employed, the decrease in temperature seen by increasing the number of shields vastly decreases. Increasing the number of shields from four to five reduces the temperature by 14.2 degrees and further increasing the number of shields to six only results in a temperature by

0.6 degrees. The high emissivity case for five shields results in a total heat flux of 45.6 kW/m², which is 20% higher than seen in the low emissivity case. Therefore, great care should be paid to the surface finish in order to minimize the effects of radiation in the displacer.

Figure 65: Displacer Wall Temperature Profile Based on Number of Shields (0.8 Emissivity)

Table 21: Maximum Displacer Body Temperature and Radiation Heat Loss for the 0.8 Emissivity Simulations

# of Shields	Max Temperature (°C)	Average Temperature (°C)	Total Heat Flux (kW/m ²)
0	509.4	285.0	82.9
1	460.0	256.2	70.7
2	414.6	232.6	60.8
3	382.6	215.6	53.8
4	359.1	203.0	48.7
5	344.9	194.4	45.6
6	344.3	189.9	44.8

6.3 Conclusions

The implementation of radiation shields in the displacer of a Stirling engine is an effective method for reducing heat transfer losses. The effects of radiation shield spacing and surface

emissivity (related to the surface finish and temperature of the material) on the heat transfer losses to the cool side of a displacer were examined through FEA analyses. The temperature profiles of the displacer and total heat flux to the cool side of the displacer for each case were gathered from the simulations for comparison. The results show that surface finish is a critical aspect of reducing radiation losses since the higher emissivity cases resulted in greater displacer body temperatures and radiation losses compared to the lower emissivity cases with the same number of shields. For a given number of shields, there is an optimal placement along the length of the displacer that yields the lowest radiation losses. The results also show that uniformly spacing the radiation shields out along the length of the displacer has the best performance. Furthermore, placing a greater gap between the shields yield greater radiation losses, as observed between the variation cases with the same number of shields. The implantation of radiation shields can reduce the temperature and total heat flux in the displacer body by 40 % and 55 % respectively. While the use of a single radiation shield will reduce radiation losses to the cold displacer body, care should be given to the overall number and relative placements of the radiation shields to ensure the best performance. Based on the results, it is suggested that five radiation shields be utilized for optimal resistance to radiation losses. Four shields could be utilized in the interest of cost savings and weight reduction. The shields should be placed centrally in the displacer if possible.

Chapter 7: FEA of Stirling Engine Components

7.1 Displacer

FEA was also performed for a displacer model. The thermal and structural boundary conditions for the analysis are shown in Figure 66. The cap of the displacer was simulated as Inconel 625, whereas the body was stainless steel. The resulting temperature and stress contours are shown in Figure 67. The maximum stress (68.9 MPa) in the assembly occurs near the interface of the displacer cap and body. Stainless steel 304 has a yield stress of approximately 215 MPa at room temperature. With the assumption that the yield stress at 80°C is similar to the value at room temperature, the factor of safety of the displacer body is approximately 3.1. The yield stress of Inconel 625 at 21°C and 810°C is approximately 490 MPa and 410 MPa, respectively. Even utilizing the conservative case of using the yield stress value at 810°C, the factor of safety is at least 5.9 for all locations of the displacer cap. Based on these results, the displacer assembly will not fail due to temperature and pressure.

Figure 66: Displacer Thermal and Structural Boundary Conditions

Figure 67: Displacer Temperature and Stress Profiles

7.2 Heat Rejecter Assembly

The heat rejecter (or the low temperature heat exchanger) is responsible for transferring heat to the low temperature reservoir and maintaining stable temperatures within the Stirling engine. Figure 68 shows the boundary conditions utilized in the FEA analysis of the heat rejecter. The top surface, where the rejecter is bolted to the heater head, is constrained by a displacement condition, which allows expansion in the x- and z-directions, but prevents motion in the y-direction. A pressure of 2.5 MPa was applied to the inner surfaces exposed to the working fluid, and frictionless supports were applied to apply symmetry. Figure 69 is a stress contour of all the components of the heat rejecter, and Figure 70 through Figure 72 show the three individual components with their stress contours. The outer shell is made of stainless steel, and the other components are made of copper alloy. Based on the maximum von Mises stresses, none of the components will fail due to the pressure.

Figure 68: Heat Rejecter Boundary Conditions

Figure 69: Heat Rejecter Stress Contour

B: Static Structural
 Equivalent Stress 2
 Type: Equivalent (von-Mises) Stress
 Unit: Pa
 Time: 1
 5/29/2018 2:25 PM

3.8196e7 Max
 3.3966e7
 2.9736e7
 2.5506e7
 2.1276e7
 1.7046e7
 1.2815e7
 8.5853e6
 4.3551e6
1.2494e5 Min

Figure 70: Heat Rejecter Shell Stress Contour

B: Static Structural
 Equivalent Stress 4
 Type: Equivalent (von-Mises) Stress
 Unit: Pa
 Time: 1
 5/29/2018 2:26 PM

2.2823e7 Max
 2.0511e7
 1.82e7
 1.5888e7
 1.3576e7
 1.1264e7
 8.9525e6
 6.6407e6
 4.3289e6
2.0171e6 Min

Figure 71: Heat Rejecter Flow Distributer Stress Contour

Figure 72: Heat Rejecter Inner Surface Stress Contour

7.3 Heater Head with Bolt Pretension

A FEA analysis of the heater head was conducted that included bolt pretension. The boundary conditions are shown in Figure 73. The model was fixed at a single point on the bottom surface, and a displacement condition preventing y-direction motion was applied. A frictionless support represented a symmetry condition. A pressure of 4.015 MPa was applied to the inner surfaces of the heater head. Bolt pretension values of 4000 N were applied to all locations where bolts would be applied. Figure 74 shows the resulting stress contour of the lower portion of the heater head. The maximum stress in the component (located where a bolt pretension condition was applied) was approximately 731 MPa, which is well below the yield stress of Inconel 625 for those conditions.

Figure 73: Heater Head with Bolt Pretention Boundary Conditions

Figure 74: Stress Contour from the Bolt Pretension FEA

Conclusions and Suggested Future Work

Research has been conducted toward developing a 1 kW Stirling engine with several components made through additive manufacturing. FEA has been performed for the critical components, including the regenerator with dimensions that push the limits of additive manufacturing. Four tasks were completed that were meant to introduce innovative design concepts for Stirling engines. The following conclusions were drawn from the results:

- A head heater head was designed and developed with AM that allowed for a previously unattainable geometry with a fully integrated pressure vessel and heat exchanger that could significantly reduce detrimental dead volumes.
- Tapering of the heater head wall was shown to significantly reduce conduction losses; considered taper options had losses, which were 23.8 to 37.7% less than that of a constant thickness heater head.
- A regenerator was designed and developed with both foil thickness and gaps between foils set as 300 μm . A foil regenerator developed through AM has the advantages of having less friction losses than traditional random mesh regenerators, and the manufacturing process is less expensive than micro-machining.
- If 100 μm foils could be printed with AM (current technology achieves a minimum thickness of approximately 200 μm), the Sage results show that a 3.6 percent increase in cycle efficiency is possible compared to a random fiber regenerator of the same dimensions.
- FEA showed the clamping method of flexure bearings had a significant effect on stress and fatigue life; the radially clamped condition resulted in maximum stress values 26.3%-28.2% less than those obtained with the outer ring clamped.

- A Sandvik 7C27Mo2 that could provide theoretically infinite life was designed and manufactured through waterjet cutting. The design was validated with laboratory testing.
- FEA showed that for a given number of radiation shields, there is an optimal placement along the length of the displacer that yields the lowest radiation losses. The results also show that uniformly spacing the radiation shields out along the length of the displacer has the best performance. Based on the dimensions and boundary conditions considered, five radiation shields provide optimal resistance to heat transfer.
- The insertion of radiation shields can reduce the temperature and total heat flux in the displacer body by 40 % and 55 %, respectively.

Though some key components have been manufactured (such as the heater head, regenerator, and flexure bearings) and verified by component testing, the fabrication of entire Stirling engine yet to be completed and tested to validate the engine design.

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