THE DESIGN, CONSTRUCTION, AND EXPERIMENTAL EVALUATION OF A COMPACT THERMOACOUSTIC-STIRLING ENGINE GENERATOR FOR USE IN A MICRO-CHP APPLIANCE.

A Dissertation in
Acoustics
by
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Abstract

Micro combined heat and power or micro-CHP is the simultaneous generation of useful heat and electricity on a residential scale. The heat and electricity are produced at the point of use, avoiding the distribution losses associated with a centralized power plant. These appliances combine a conventional gas-fired condensing boiler with an electric power module capable of generating electricity from the heat of combustion. Currently, the leading power modules for micro-CHP appliances are free-piston Stirling engines (FPSEs) which can generate 1050 watts of electricity at a thermal-to-electric efficiency of 26%.[1] These external combustion engines have been under development for the last 25 years, with FPSE micro-CHP appliances only recently being introduced to the commercial market. Publications by developers assert unlimited service life and high efficiency, with low noise and emissions; but despite these claims, the actual reliability and cost of manufacturing has prevented their successful mass-market adoption. A Thermoacoustic-Stirling Engine Generator or TaSEG is one possible alternative to FPSE’s. A TaSEG uses a thermoacoustic engine, or acoustic heat engine, which can efficiently convert high temperature heat into acoustic power while maintaining a simple design with fewer moving parts than traditional FPSE’s. This simpler engine is coupled to an electrodynamic alternator capable of converting acoustic power into electricity. This thesis outlines the design, construction, and experimental evaluation of a TaSEG which is appropriate for integration with a gas burner inside of a residential micro-CHP appliance. The design methodology is discussed, focusing on how changes in the geometry affected the predicted performance. Details of its construction are given and the performance of the TaSEG is then outlined. The TaSEG can deliver 132 watts of electrical output power to an electric load with an overall measured thermal-to-electric (first law) efficiency of $\eta_{T-E}=8.32\%$, corresponding to 14% of Carnot $\eta_c$. The volumetric power density of this TaSEG is 8.9 kW/m³. While the demonstrated overall efficiency is modest (for reasons that are largely understood), this TaSEG has moved the technology away from laboratory prototypes toward a commercially viable power module having a design configuration suitable for implementation in a micro-CHP appliance. Based on the TaSEG’s measured experimental performance results, recommendations for future work that might improve the overall efficiency of the TaSEG are also presented.
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\( \dot{W} \) Time-averaged Work Flow, p. 4
\( T_C \) Thermal Source/Sink, Cold Temperature, p. 4
\( T_H \) Thermal Source/Sink, Hot Temperature, p. 4
\( \dot{Q}_C \) Time-Averaged Cold Heat Flow, p. 4
\( \dot{Q}_H \) Time-Averaged Hot Heat Flow, p. 4
\( \eta \) Efficiency p. 5
\( \eta_c \) Carnot’s Efficiency, p. 5
\( Z \) Acoustical Impedance, p. 8
\( p_1 \) Acoustic Pressure, p. 8
\( U_1 \) Acoustic Volume FLOW Rate, p. 8
\( \rho \) Density of the Working Fluid, p. 8
\( c \) Speed of Sound, p. 8
\( \lambda \) Acoustic Wavelength, p. 10
\( \delta_\kappa \) Thermal Penetration Depth, p. 10
\( \delta_\nu \) Viscous Penetration Depth, p. 10
\( k \) Thermal Conductivity, p. 10
\( \kappa \) Thermal Diffusivity, p. 10
\( \omega \) Angular Frequency, p. 10

\( c_p \) Specific Heat at Constant Pressure, p. 10

\( \mu \) Average Coefficient of viscosity, p. 10

\( \nu \) Kinematic Viscosity, p. 11

\( \xi_1 \) Gas Displacement Amplitude, p. 11

\( u_1 \) Gas Velocity Amplitude, p. 11

\( f \) Frequency, p. 16

\( R_{\text{alt}} \) Alternator Resistance, p. 17

\( X_{\text{alt}} \) Alternator Reactance, p. 17

\( T_m \) Mean Temperature, p. 18

\( \rho_m \) Local Mean Density of the Working Fluid, p. 18

\( U \) Volumetric Velocity, p. 18

\( \tau \) Ratio of Absolute Temperatures at the Regenerator Ends, p. 19

\( U_{1,h} \) Volumetric Velocity at the Hot End of the Regenerator, p. 19

\( U_{1,c} \) Volumetric Velocity at the Ambient End of the Regenerator, p. 19

\( p_{1,c} \) Pressure at Cold Side of Regenerator, p. 19

\( U_{1,fb} \) Volumetric velocity in the Feedback Intertance, p. 19

\( R \) Regenerator Resistance, p. 19

\( L \) Feedback Inertance, p. 19

\( C \) Feedback Compliance, p. 19

\( L_r \) Length of the Regenerator, p. 19

\( r_h \) Hydraulic Radius of the Regenerator, p. 19

\( A_r \) Cross-sectional Area of the Regenerator, p. 19

\( L_l \) Length of the Inertance, p. 19
$A_1$ Cross-sectional Area of the Inertance, p. 19

$V_c$ Gas Volume of the Compliance, p. 19

$\gamma$ Ratio of Isobaric to Isochoric Specific Heats of the Working Fluid, p. 19

$p_{\text{in}}$ Mean Pressure of Working Fluid, p. 19

$\phi_{pU}$ Phase Angle Between Acoustic Pressure and Volumetric Flow, p. 21

$p_1$ First-order Acoustic Pressure, p. 21

$\dot{E}_2$ Time-averaged Acoustic Power, p. 21

$\dot{E}_c$ Acoustic Power Flowing into the Cold End of the Regenerator, p. 21

$\dot{E}_{fb}$ Time-averaged Power Fed Through Intertance, p. 21

$p_{1,h}$ Pressure at Hot Side of Regenerator, p. 21

$\dot{E}_h$ Time-averaged Acoustic Power Flowing Out of the Hot Side of the Regenerator, p. 21

$\dot{E}_{\text{alt}}$ Time-averaged Acoustic Power Flowing to Alternators, p. 21

$\dot{Q}_{\text{h}}$ Heat Input to Hot Heat Exchanger, p. 22

$\eta_{Th-El}$ Thermal-to-Electric Conversion Efficiency, p. 30

$V_c$ Front Compression Space Volume, p. 53

$V_b$ Back Volume per Side, p. 53

$S_b$ Surface Area in the Back Volume, p. 53

$L$ Clearance Seal Length, p. 53

$\delta$ Clearance Seal Gap (Width), p. 53

$p_C$ Dynamic Pressure Amplitude in the Compression Space, p. 54

$p_B$ Dynamic Pressure Amplitude in the Back Volume, p. 54

$P_{\text{m}}$ Mean Pressure, p. 54

$A_P$ Alternator Piston Area, p. 54
\( K_M \) Alternator Mechanical Stiffness, p. 54
\( t_r \) Pressure Vessel Wall Thickness, p. 60
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Dedication

I would like to dedicate this work to my wife, Kristina, and children, Ella, Mya, and Piper, who have unconditionally supported my desire to become “Dr. Daddy,” while constantly providing motivation, love and support. I love you!
ATCHOO!

UH OH.

I'M LEAKING BRAIN LUBRICANT.

I LET MY MIND WANDER AND IT DIDN'T COME BACK.

I FIGURED YOU'D LOST YOUR MIND YEARS AGO.

The Days are Just Packed: A Calvin and Hobbes Collection
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Introduction

Since 1816, when Reverend Dr. Robert Stirling received a patent for his novel heat engine, numerous efforts have been undertaken to utilize the Stirling engine’s essentially reversible thermodynamic cycle and produce a commercially viable product. Over 100 years ago the English physicist John Strutt (Lord Rayleigh) established Rayleigh’s criteria, which states that the heating and cooling of a gas can create acoustic power. While Rayleigh’s qualitative understanding was correct, it would take another half-century before Nikolaus Rott, *et al.* would establish an accurate theoretical foundation for the field of thermoacoustics. Even with the theoretical understanding, thermoacoustic engines were not developed until much later, with the first patent for a standing-wave[2] acoustic engine granted in 1958. Standing-wave engines are inherently irreversible and can never match the efficiency of a Stirling cycle, even in theory; but in 1979, Peter Ceperley realized that gas in a porous medium experiences essentially a Stirling cycle when in the presence of a *traveling* acoustic wave. This realization gradually led to the development of traveling-wave acoustic engines, which use the same basic cycle as FPSE’s but use acoustic instead of mechanical feedback to achieve the cycle phasing, eliminating one major moving part. These engines make use of the heat flow from a high temperature source to a low temperature sink to generate acoustic power. In 1999, researchers at Los Alamos National Lab constructed the first large-scale thermoacoustic-Stirling heat engine, which converted heat into acoustic power at an efficiency of 41%. While this high thermal-to-acoustic conversion is impressive, acoustic power itself is not as generally useful as shaft power or electricity.
Coupling an acoustic engine with an alternator to produce electricity is itself a considerable challenge.

Thermoacoustic-Stirling Engine Generators (TaSEG) combine a thermoacoustic (Stirling) engine and a linear alternator to produce a single, small-scale device capable of efficiently converting heat into electricity. Researching small scale TaSEG technology could result in the development of devices suitable for residential micro-combined heat and power, or μCHP appliances. However, to fully realize their potential, more in-depth research is required.

Existing μCHP appliances currently on the market have high initial investment costs, long payback periods and/or require routine maintenance. Currently, the leading power modules for μCHP appliances are free-piston Stirling engines (FPSE’s). These external combustion engines have been under development for the last 25 years, with two FPSE based μCHP appliances recently being introduced to the commercial market. Publications by developers suggest unlimited service life and high efficiency with low noise and emissions. Despite these claims, the actual reliability and cost of manufacturing has prevented their successful mass-market adoption.

A TaSEG is fundamentally very much like a FPSE, sharing the key idea that the gas in the regenerator experiences thermal expansion when the pressure is high and thermal contraction when the pressure is low, thus the working fluid in the regenerator does work every cycle, pumping power into the working fluid. They also share a similar arrangement of regenerator and heat exchangers. Figure 1.1 conceptually compares a FPSE (right) to a TaSEG (left). Careful examination of the figure shows both the similarities and differences of these engines.

The most striking difference is that all of the costly internal displacer mechanisms found in the FPSE (displacer dome, seal, support, bearings and spring) are not present in the TaSEG. Instead, the displacer components are replaced with a tuned acoustic network consisting of a thermal buffer tube with porous flow straighteners at each end, an annular space providing the required compliance (gas spring), an acoustic resistance, and inertance. This displacer replacement is possible because the Stirling “thermal core” does not know or care if the Stirling cycle

\[1\] Being heated at high pressure and cooled at low pressure is common to virtually all types of engines.
phasing is being created by a mechanical displacer or a tuned acoustic network. Comparing FPSEs to thermoacoustic engines suggests that the reduced number of complex parts in the TaSEG provides a simpler mechanical design, a reduction in production component costs and an improvement in operational reliability. Thus, thermoacoustic technology addresses the shortcomings of FPSE technology while it still preserves the fundamentally high efficiency of the Stirling cycle.

Small scale thermoacoustic engine technology is a highly innovative and truly unique technology. To the best of the authors knowledge there has been little research on small scale TaSEGs with a restriction placed on both size and cost. The proposed research deals directly with a new, novel technology compared to the state of the art. The main technical objectives of this work are the develop a smaller, lighter, cheaper thermoacoustic-Stirling engine generator appropriate for use as a \( \mu \)CHP power module. This TaSEG should have a coaxial thermal core, a realistic hot heat exchanger, compact acoustic feedback, static Gedeon streaming suppression, minimal vibration, be able to generate 600 to 1000 watts of electricity and have a thermal-to-electrical conversion efficiency of at least 15%. All features that make it suitable for a \( \mu \)CHP appliance.
1.1 History and Governing Principles

Elementary thermodynamics courses often begin with the two basic types of heat engines shown in Figure 1.2. In a heat pump, heat is pumped from a low temperature source to a high temperature sink by means of an energy-consuming engine. In prime movers, heat flows from a high temperature source to a low temperature sink through an engine. This heat flow results in net work being generated by the engine.[3]

![Figure 1.2.](image)

**Figure 1.2.** The two types of heat engines: heat pumps often referred to as “refrigerators” and prime movers often referred to as “engines.”

The maximum efficiency of these heat engines is restricted by the first and second laws of thermodynamics. Assuming steady-state engine operation and defining $\dot{W}$ as the time-averaged work flow, $T_C$ and $T_H$ as the cold and hot temperatures of the thermal sources/sinks and $\dot{Q}_C$ and $\dot{Q}_H$ as the time-averaged cold and hot heat flows, it is possible to explore the governing laws of basic engine operation.

Applying the law of conservation of energy to these heat engines provides a statement of the first law of thermodynamics, which states

$$\dot{Q}_H - \dot{Q}_C - \dot{W} = 0.$$  \hspace{1cm} (1.1)

The second law of thermodynamics governs the entropy generation of the engine, depending on its intended operation, and restricts the generated entropy to be zero or positive. During steady state operation of the engine, the net entropy increase in the reservoirs is of greatest interest:
\[
\frac{\dot{Q}_C}{T_C} - \frac{\dot{Q}_H}{T_H} > 0 \text{ Prime Mover,}
\]
(1.2)
\[
\frac{\dot{Q}_H}{T_H} - \frac{\dot{Q}_C}{T_C} \geq 0 \text{ Heat Pump.}
\]
(1.3)

Focusing solely on prime movers, the desired output divided by the required input is known as the “first law efficiency”, \( \eta = \frac{\dot{W}}{\dot{Q}_H} \). This is referred to as the efficiency of the engine. Combining Eqs. (1.1) and (1.2) and eliminating \( \dot{Q}_C \), the efficiency then becomes

\[
\eta = \frac{\dot{W}}{\dot{Q}_H} \leq \frac{T_H - T_C}{T_H}.
\]
(1.4)

By rearranging the temperature ratio on the right hand side of Eq. (1.4) it is possible to define the maximum efficiency that a prime mover can achieve:

\[
\eta_c = 1 - \frac{T_C}{T_H}.
\]
(1.5)

This efficiency, more commonly known as Carnot’s efficiency \( \eta_c \), is an expression for the thermal efficiency of an engine undergoing a process in which the engine and its surroundings are restored to their respective initial state. A process of this type is known as a reversible process. While no macroscopic process is perfectly reversible, Carnot’s efficiency represents the limiting case; the ideal efficiency that a heat engine can achieve. Furthermore, from an engineering perspective, real engine designs must also consider safety, reliability, convenience, and cost, which often compromise efficiency.

Nearly 200 years ago Reverend Robert Stirling unknowingly\(^2\) made use of the first and second laws of thermodynamics in his pursuit to invent a heat engine that provided a safer alternative to steam engines.[4, 5] The Stirling engine is a closed-cycle, regenerative heat engine that employs an “economiser,” more commonly known as a regenerator, and carries out a thermodynamic process known as the Stirling cycle. The primary effect of the regenerator in a Stirling engine is to considerably increase the thermal efficiency by internally recovering heat that

\(^2\)The first law of thermodynamics was established in 1850 by German scientist Rudolf Clausius and the second law of thermodynamics was formulated by Sadi Carnot in 1824, while Reverend Stirling patented his engine in 1816.
would otherwise be lost to the inherent irreversibility of the engine.

The Stirling cycle has long fascinated engineers because in the ideal it can reach Carnot’s efficiency, but with higher power density than an actual Carnot engine would have; and being a closed-cycle, a Stirling engine can theoretically run on any external heat source. Despite the intrinsically high efficiency and fuel flexibility of Stirling engines, the mechanical complexities of practical designs have limited their commercial success against the more cost-effective internal combustion engine.

Over a century ago, glass-blowers noticed that when blowing a hot bulb on the end of a cooler tubular glass stem, the stem tip would sometimes radiate sound. This phenomenon was experimentally studied by C. Sondhauss, a German physicist, who observed that the application of a heat source to the closed-bulb end would cause the air in the entire tube to oscillate and produce sound.[6, 7] He went on to quantitatively investigate the relationship between the pitch of the emitted sound and the dimensions of the apparatus under test.

In the *Theory of Sound* published in 1878, Lord Rayleigh (J. W. Strutt) not only confirmed the results obtained by Sondhauss, but expanded on them. Rayleigh explained that acoustic power could be generated by means of oscillatory thermal expansion and contraction, stating “if heat be given to the air at the moment of greatest condensation, or be taken from it at the moment of greatest rarefaction, the vibration is encouraged.”[8] Though Rayleigh’s qualitative understanding was correct, a quantitatively accurate theoretical understanding was not achieved until Nikolaus Rott *et al.* developed the mathematics which govern acoustic oscillations of a gas in a channel having an axial temperature gradient.[9, 10, 11, 12, 13, 14, 15] The conclusions of their work laid the theoretical foundation applicable to fundamental experiments involving Stirling based thermoacoustic heat pumps and prime movers.[16, 17]

Initially, thermoacoustic prime movers and heat pumps being developed were all of the “standing wave” type. This style of thermoacoustic engine is termed “standing-wave” because the time phasing between acoustic pressure and acoustic particle velocity oscillations is \( \approx 90^\circ \), or close to that of a standing acoustic wave. The work of the engine is actually produced in the boundary layer of a component know as a “stack” - a collection of closely spaced plates, an array of channels, or a bundle of pins. In a standing-wave engine work output is only possible due to
the phase lag incurred by imperfect thermal contact between the working fluid and the stack. While the imperfect thermal contact produces the required phase lag necessary for the engine to produce acoustic work, it also generates entropy, resulting in a less efficient thermodynamic cycle. Over the years, numerous attempts have been made to demonstrate the generation of acoustic power from heat energy by means of a standing wave thermoacoustic engines.[18] One such effort successfully demonstrated the coupling of a thermoacoustic standing wave engine to an electric generator (linear alternator).[19] In contrast to standing wave engines, traveling-wave (thermoacoustic-Stirling) engines rely on acoustic pressure oscillations being virtually in phase with acoustic particle velocity at the location of a regenerator. As in a Stirling engine, the regenerator in a traveling-wave acoustic engine is designed for intimate thermal contact with the working fluid, to very closely approximate isothermal heat transfer when absorbing or returning (“regenerating”) sensible heat during the cycle.[18, 20] A regenerator could in theory be made like a stack from a standing-wave engine, i.e. an array of parallel channels, though with smaller pore size. However, in a traveling-wave engine, there is no need to maintain laminar flow, as there is in the standing-wave engine (where all the important phasing happens in the undisturbed boundary layer). The phasing in a traveling wave engine is imposed on the bulk fluid by the tuning of the overall acoustic circuit (as is described in the following section), and the regenerator simply needs to store and release sensible heat. Therefore the working fluid can follow a tortuous path without penalty (this may even enhance the heat transfer), and regenerators are typically made of stacked screens or random-fiber mats, which are simpler to manufacture than the precise arrays of laminar channels demanded by standing-wave engines.

In the late 1970’s, Peter Ceperley realized that a traveling acoustic wave propagating through a regenerative heat exchanger (regenerator) located within an acoustic network with essentially toroidal topology containing Stirling heat exchanger components undergoes a thermodynamic cycle that is similar to the Stirling cycle.[21, 22, 23, 24] He suggested that by using acoustics to govern the motion of the gas and its pressure, it would be possible to eliminate the pistons found in traditional Stirling engines and produce a “traveling wave heat engine.” These engines would rely only on the working gas itself to establish the proper phasing
between pressure and velocity to produce work. As an acoustic wave travels up the temperature gradient through the regenerator, it carries the gas in the regenerator through a sequence of displacement toward higher temperature, depressurization, displacement toward lower temperature, and pressurization. Thus, the gas experiences thermal expansion while it is displaced toward a higher local temperature and thermal contraction while it is displaced toward a lower local temperature. This ensures that acoustic power is amplified as the wave travels towards the higher temperature, which is the source of the engine’s net work. At the same time, the working gas gains entropy from the regenerator during the depressurization and loses entropy to the regenerator during pressurization. Thus, the two displacement processes yield a net convection of entropy down the regenerator’s temperature gradient. This is the mechanism by which a thermoacoustic traveling-wave engine extracts heat from a high temperature heat source and rejects heat to an ambient temperature heat sink.[20]

Ceperly’s publications discuss the creation of a resonantly enhanced traveling wave field inside a looped resonator and established the fundamental foundation for traveling wave thermoacoustic-Stirling devices. However, his experimental engine was incapable of amplifying acoustic power because of the low complex ratio of pressure $p_1$ to volume flow rate $U_1$

$$Z = p_1/U_1,$$

also known as acoustical impedance[18], of the working gas. Yazaki et al. were the first to demonstrate such a thermoacoustic-Stirling prime mover (engine), but at a very low efficiency.[25] Both Ceperley and Yazaki et al. realized that the acoustical impedance at the regenerator would have to be increased to reduce the acoustic power being lost due to viscous dissipation in the regenerator, however neither anticipated the presence of acoustic streaming.[26, 27, 28, 29]

Swift et al. suggested placing an engine’s regenerator in a compact feedback loop near a pressure antinode in a standing wave resonator.[30] By ensuring that the feedback loop dimensions are small to minimize thermoviscous power losses and sized such that the specific acoustic impedance at the regenerator is much larger than $\rho c$ (the gas density times the speed of sound), their approach resulted in a geometry that creates a Stirling cycle, in-phase relationship between velocity and
pressure in the regenerator at a high enough impedance to enable useful efficiencies.

The toroidal geometry of traveling-wave engines does have a drawback in that steady flow can exist around the loop. This steady mass flux, commonly referred to Gedeon acoustic streaming in the literature, creates an undesirable heat leak that wastefully convects heat away from the hot heat exchanger without generating acoustic power. Left unaddressed, this type of acoustic streaming can significantly impact the achievable performance of a thermoacoustic-Stirling engine.

In the same manner as conventional Stirling engines, a thermoacoustic-Stirling engine can accept almost any form of heat, which can then be converted into acoustic power. Exhaust heat from internal combustion engines, by-product heat created from industrial processes, like paper production, or the heat produced by traditional premixed air-gas home heating systems are all examples of heat sources that can be accepted by thermoacoustic-Stirling engines. The output power from these types of engines can be used to drive various types of acoustic loads\cite{16, 31} including thermoacoustic chillers\cite{32, 33}, bi-directional turbines\cite{34} and linear alternators\cite{35}, the latter of which in turn generates electricity.

Thermoacoustic-Stirling chillers and engines, comprised of a regenerator situated at the location of traveling wave phasing within a compact, lumped element acoustic feedback loop, have been developed and tested\cite{16, 20, 30, 35}. The thermoacoustic-Stirling engine generator, or TaSEG, developed in this thesis, consists of two mirrored, compact, coaxial TaSEG’s, similar in form factor to the Stirling engine-generator set developed by Lane et al., that are each coupled to a linear alternator\cite{38, 39}. There are several design constraints placed on the TaSEG which stem from its intended application in a residential micro-combined heat and power, or \(\mu\)CHP, appliance\cite{40, 41}. Micro-CHP is the simultaneous production of useful heat and power (electricity) within a residential dwelling. In these types of appliances electricity and heat are produced at the location they are consumed, avoiding the thermal and electrical distribution losses associated with centralized production plants. This is an emerging, relatively young, home-heating appliance market that is slowly becoming established in both Europe and the United States\cite{42}. Internal combustion (IC) engines and fuel cells have been adapted to this market, but there is still plenty of opportunity for alternatives. Stirling engines are well-suited for \(\mu\)CHP, especially where the waste heat is carried away.
in hot water. Linear free-piston Stirling and thermoacoustic-Stirling engines are particularly attractive because they require no lubricants and can have a theoretically infinite product life without any scheduled maintenance. The TaSEG variant aims to further improve the basic Stirling reliability, by replacing the mechanical displacer with a piece of tubing. The Stirling engine displacer is traditionally one of the costliest and most failure-prone components in a Stirling engine, and a source of performance sensitivity.[81] Therefore, replacing a moving displacer with moving helium in a tube, as a thermoacoustic-Stirling engine does, will improve reliability and reduce cost.

1.2 A Note on Important Length Scales

There are several important length scales in thermoacoustic devices that should be mentioned. These include the wavelength of sound in the direction of wave propagation $\lambda$, the thermal $\delta_\kappa$ and viscous $\delta_\nu$ penetration depths perpendicular to the direction of gas motion, and the gas displacement amplitude $|\xi_1|$.

The importance of the acoustic wavelength defined by Eq. 1.11 to the lumped element approximation has already been discussed and should not be understated. Its worth noting that typically the size of thermoacoustic devices and their components (e.g. heat exchangers, regenerator, etc.) are much smaller than the acoustic wavelength of the operating frequency.

The thermal interaction of the pressure and velocity oscillations with the regenerator’s solid boundary result in a heat flux along the regenerator, thus it is important to define the distance over which heat can diffuse laterally to and from the solid during the time interval of the order of one period of oscillation divided by $\pi$. This distance is defined as the thermal penetration depth and is a function of properties of the working gas and the device’s frequency of operation,

$$\delta_\kappa = \sqrt{\frac{2k}{\omega \rho c_p}} = \sqrt{\frac{2\kappa}{\omega}},$$  \hspace{1cm} (1.7)

where $k$ and $\kappa$ are the thermal conductivity and diffusivity of the gas respectively, $\omega$ is the angular frequency of operation, $\rho$ is the density of the working gas and $c_p$ is the gas’s specific heat per unit mass at constant pressure. Heat exchange
components should be designed to have lateral dimensions comparable to this depth in order to ensure sufficient heat exchange can occur between the working gas and exchanger solid. Similarly, it is possible to define a depth that momentum can diffuse laterally during the same time interval. This distance is defined as the viscous penetration depth and is again a function of the properties of the working gas and the device’s frequency of operation,

\[ \delta_\nu = \sqrt{\frac{2\mu}{\omega \rho}} = \sqrt{\frac{2\nu}{\omega}}, \]

where \(\mu\) and \(\nu\) are the gas’s dynamic and kinematic viscosities, respectively. The working gas at distances much greater than these penetration depths feels no thermal or viscous contact with the nearest solid boundary. Working gas throughout any thermoacoustic device that is within these penetration depths will feel both thermal and viscous effects from the nearby boundaries. Interestingly enough, the square of the ratio of these penetration depths,

\[ \left( \frac{\delta_\nu}{\delta_\kappa} \right)^2 = \frac{\mu c_p}{\kappa} = \sigma \leq 1, \]

yields a ratio of the product of viscosity and heat capacity (momentum diffusivity) to thermal conductivity (thermal diffusivity), more commonly referred to as the Prandtl number. The Prandtl number provides a measure of the relative effectiveness of momentum and energy transport by diffusion in the velocity and thermal boundary layers, respectively.[58] For gases typically used in thermoacoustic devices the Prandtl number is approximately unity, thus the thermal and viscous penetration depths are comparable. Unfortunately this means that most thermoacoustic devices suffer from substantial viscous effects.[18]

The last important length scale that should be noted is the gas displacement amplitude \(\xi_1\), or the absolute zero-to-peak distance that a gas parcel travels during a period of oscillation, defined as the velocity amplitude of the gas divided by the angular frequency of operation,

\[ \xi_1 = \frac{|u_1|}{\omega} = \frac{|u_1|}{2\pi f}. \]

This is an important dimension as it aids in defining the useful lengths of the
heat exchangers in the direction of wave propagation (e.g., having a heat exchanger that is longer than twice this distance is typically of no benefit). It should be noted that the displacement amplitude is usually much larger than the aforementioned penetration depths and always shorter than the wavelength.\[18\]

1.3 Thermoacoustic-Stirling Engine Generator Overview

![Figure 1.3](image)

**Figure 1.3.** Simple coaxial thermoacoustic-Stirling engine generator, or TaSEG, comprises a gas-filled pressure vessel containing a regenerator, hot and ambient heat exchangers and a linear alternator. Thermal power is injected into the working fluid at the hot heat exchanger and waste thermal power is rejected at the ambient exchanger. As the acoustic wave propagates through the regenerator it is amplified by the temperature gradient imposed across it by the hot and ambient heat exchangers. This results in the acoustic power flowing out of the hot end of the regenerator being greater than that flowing into the cold end. However, to maintain the acoustic oscillation, a fraction of the acoustic power flowing out of the hot end of the regenerator must be fed back through the feedback inerterance and compliance to the cold end of the regenerator. This acoustic power flow is indicated by the blue path, where the grey arrow heads indicate flow direction. The remainder of the acoustic power that is not fed back is used to drive the power piston of the linear alternator, generating electricity.

A basic sketch of a thermoacoustic-Stirling engine generator, or TaSEG, is shown in Figure 1.3. Like traditional Stirling engines, the conversion of heat to
power occurs in the regenerator through which the working gas oscillates. The regenerator consists of numerous small, often random, channels which smoothly span a temperature gradient established by the placement of a hot and an ambient heat exchanger at its end. The “channels” or gas passages within the regenerator must be very small to ensure that the fluid contained within them is ideally at the same temperature as the local regenerator solid. This intimate thermal contact permits heat transfer with minimal entropy production, nearly doubling the achievable efficiency over stack based standing wave engines.[43]

Figure 1.4 shows a section view of the dual, mirrored TaSEG SOLIDWORKS model that provides a glimpse of its internal components, which will be discussed later. Careful examination of the model shows the TaSEG’s two, mirrored, acoustically coupled engines that are symmetric about the flow direction plate found in the middle of the assembly.[44, 45] Each engine consists of a lumped element feedback loop that mainly consists of an annular “compliance” volume, or gas-spring volume (compliance being the inverse of spring stiffness), an inertance (mass) of the gas in the inertance tube, and the primarily resistive impedance of the regenerator.
Figure 1.5. The thermoacoustic-Stirling engine-generator.

The constructed TaSEG, which is illustrated in Figure 1.5, consists of an approximately 8,260 cubic centimeter pressure vessel that fits inside a space envelope that is 0.96 m long by 0.32 m wide by 0.22 m high. The TaSEG is filled with 3.9 MPa helium gas and has a maximum allowable working pressure (MAWP) of 5.6 MPa.[46]

The white tube protruding from the side of the TaSEG in Figure 1.5 carries water into and out of the two ambient heat exchangers that are connected in series with each other. Several pressure sensors[47] measure the pressure at important locations and two displacement sensors[48] monitor the stroke of each engine’s electrodynamic, linear alternator. Numerous thermocouples are used at various locations to measure the local metal and water temperatures. These measurement instruments are used to investigate the heat flux into the engine, the acoustic power being generated by the engine, and the overall efficiency of the TaSEG. It should also be observed that this TaSEG has a realistic, internally finned hot heat exchanger designed to accept heat from a gas burner through the pressure vessel wall, as opposed to numerous laboratory engines that have electrically generated heat applied directly to the working fluid from inside the pressure vessel.[16, 36]
While direct internal heat application is an effective and efficient way to deliver the heat to the working fluid it is not practical for real-world applications. One of the explicit design criteria for the TaSEG was that it have a realistic hot heat exchanger able to accept thermal input from a combustion or radiant heat source. However, for safety reasons all experimental tests were performed with a custom made electrical resistance heater[49] designed to provide heat to the hot exchanger from outside the pressure vessel. The TaSEG uses this thermal input to generate acoustic power that is then converted into useful electricity through the two linear alternators coupled to the two engines. In the end, the electricity produced is measured then dissipated in an electric load.

1.4 Power Generation by Engine

All thermoacoustic heat engines are monofrequency devices and use the same basic method to amplify acoustic power. The acoustic wave is a single frequency wave consisting of a time-dependent sinusoidal pressure amplitude. During the low pressure portion of the cycle, the temperature of the working fluid drops due to thermal expansion. Likewise, during the high pressure portion of the cycle the temperature of the working fluid increases due to compression. As described by Lord Rayleigh[8], oscillatory thermal expansion and contraction of a gas can generate acoustic power. All that is required to amplify a sound wave is the addition of heat during the compression portion of the pressure cycle and removal of heat during the expansion portion of the pressure cycle. This basic principle is used in thermoacoustic heat engines to generate acoustic power from heat.

In devices like the TaSEG the working gas ideally oscillates with (or nearly with) traveling wave time phasing in a regenerator that has a steep temperature gradient in the direction of acoustic power flow. Recall that a regenerator is a porous material whose characteristic dimensions are small enough to ensure intimate thermal contact between its solid material and the gas it contains. In an ideal regenerator, the temperature of a gas parcel within the regenerator is identical to that of the adjacent solid portion of the regenerator. However, in reality there does exist a finite difference between the gas parcel temperature and the adjacent regenerator temperature, which generates entropy. As the gas in the regenerator moves
towards the hot heat exchanger it is heated, thus expanding while the pressure is high. As it moves towards the ambient exchanger it is cooled, thus contracting while the pressure is low. The oscillating thermal expansion and contraction of the gas in the regenerator has the time phasing with respect to oscillating pressure to meet the power producing criteria set forth by Rayleigh.

1.5 Basic Lumped Element Circuit Analysis

This section presents a general explanation of how a TaSEG should be sized to ensure that the phasing between gas motion and pressure oscillations is properly set so that the amplitude of the wave propagating through the regenerator’s axial temperature gradient is amplified.

To aid in the analysis, we first consider a lumped-element approximation.[16] Before proceeding, however, the validity of the lumped element approximation must be examined as this assumption is only valid if all the TaSEG’s components are much shorter than one quarter of the wavelength of sound at the intended operating frequency. Equation 1.11,

\[ \lambda = \frac{c}{f}, \]

states that the wavelength of sound is the ratio of the speed of sound in the working gas \( c \) to the frequency of the sound \( f \).[50] In the case of the TaSEG, the operating frequency is \( f = 122 \text{ Hz} \) and the working fluid is helium, which has a speed of sound of \( c = 965 \text{ m/s at } 25^\circ\text{C} \). Using these values in Eq. 1.11 yields \( \lambda = 7.58 \text{ m} \) and \( \lambda/4 = 1.90 \text{ m} \). Comparing \( \lambda/4 \) to the TaSEG’s overall height of 0.22 m, width of 0.32 m, and total length of the two engines combined of 0.96 m it is clear that all of the components that make up the TaSEG are far smaller than a quarter wavelength of its operating frequency. Therefore, it is possible to capture all the most important features of the engine using a simple lumped-element electric circuit model demonstrated by Backhaus and Swift.[20] This circuit analysis provides an outline of the power flows within the engine and shows what establishes the operating temperatures and pressures within the engine. Contained within the TaSEG is an acoustic feedback network first analyzed by Swift et al.[30] and patented by C.M.
Figure 1.6. A cutaway view of the TaSEG that shows its practical coaxial topology, appropriate for external heating. The yellow line and grey arrows show the flow of pV work in the TaSEG. Close examination shows that only a fraction of the pV work that circulates is drawn off to generate electricity.

de Blok[51]. In order to easily analyze the engine, its components are replaced by idealized electrical equivalent components.[52]

Figure 1.6 and Figure 1.7 show a cross section of the TaSEG and the electric circuit analog of the simplified lumped-element acoustic feedback loop. The feedback tube that is hidden from view in Figure 1.6, but labeled as $L$, acts like a lumped mass, dominated by inertia; while the volume adjacent to the cold heat exchanger, labeled with a $C$, acts as a gas spring (a compliant volume). Thus the two together act like the neck and the bulb of a Helmholtz resonator. Hence, the feedback tube is often referred to as an “inertance tube” while the gas-spring volume is called a “compliance.”[53] These elements can be seen on the left hand side of Figure 1.7. The alternators can be modeled as a simple resistance, $R_{alt}$, in parallel with a reactive impedance, $iX_{alt}$. These elements can be seen on the bottom of Figure 1.7. Ignoring the spatial temperature dependence of viscosity across the regenerator as well as the compliance of the gas within it, the regenerator and adjacent heat exchangers can be modeled as a volume velocity source and a single resistor
Figure 1.7. TaSEG electrical equivalent lumped element circuit. The TaSEG components are much shorter than one quarter of the wavelength of sound at its operating frequency. Therefore, they can be modeled as lumped elements. The subscripts in the equivalent circuit correspond to physical locations in the TaSEG.

in the simplified equivalent electrical circuit. This is shown in the upper right hand side of Figure 1.7. As the regenerator is made up of tightly packed stacked screen or random fiber elements, the flow in the regenerator is viscously-dominated, and the regenerator as a lumped element appears resistive\(^3\) rather than inertial or compliant.\cite{54} In the absence of acoustic streaming, the local temperature \(T_m\) in the regenerator has a linear profile from \(T_C\) to \(T_H\). Thus, \(\rho_m\) changes according to \(\rho_m \propto 1/T_m\). The subscript “m” signifies a mean, time invariant value of a variable, “1” signifies a variable that is first order in acoustic amplitude and “2” signifies a variable that is second order in acoustic amplitude. Ignoring the compressibility of the gas within the regenerator, conservation of the first-order mass flux dictates that \(\rho_m U_1\) remains constant throughout the regenerator. The decrease in \(\rho_m\) with

\(^3\)This is also, very importantly, a consequence of its location in the acoustic circuit. If the regenerator were placed in a dead-ended volume, it would not act like a resistor at all.
increasing local temperature means that the temperature gradient across the regenerator serves to increase $U_1$ by the ratio of the absolute temperatures at the regenerator ends, $\tau = T_H/T_C$. Ideally, the volumetric velocity at the hot end of the regenerator $U_{1,h}$ is equal to the product of the volumetric velocity at the ambient end of the regenerator $U_{1,c}$ and the ratio of absolute temperatures at the regenerator ends $\tau$. Another way to think of this is that $U_1$ must grow as $T_m$, thus it is justifiable to model the regenerator’s temperature gradient as a volume velocity source.[22]

The lumped element electrical circuit provides a simplified interpretation of the fundamental aspects of the TaSEG. By applying Kirchhoff’s circuit laws to the lumped element circuit, two loop equations can be derived.[55] Current in the circuit analogy is the equivalent of volume velocity in the acoustic domain; and Kirchoff’s junction rule states the algebraic sum of the currents (analogous to volume velocity in the acoustic domain) flowing into a junction is equal to the sum of the currents flowing out of the same node. Applying this rule to the lumped element circuit’s compliance node yields

$$-i\omega C p_{1,c} = U_{1,c} - U_{1,fb},$$

where $p_{1,c}$ is the pressure at the ambient side of the regenerator and $U_{1,fb}$ is the volumetric velocity in the feedback inertance.

In a similar way, making use of Kirchoff’s second rule, which states that the algebraic sum of the changes in electric potential (analogous to pressure in the acoustic domain) around any complete loop in a network is zero, yields

$$-i\omega L U_{1,fb} = U_{1,c} R.$$  

(1.13)

The $R$, $L$, and $C$ in Eqs. (1.12) and (1.13) are[18]

$$R = \frac{6\mu L_r}{A_r r_h^2}, \quad L = \frac{\rho_m L_i}{A_i}, \quad C = \frac{V_c}{\gamma \rho_m},$$  

(1.14)

where $\mu$ is the average coefficient of viscosity, $L_r$ is the regenerator length along the acoustic axis, $A_r$ is the cross-sectional area of the regenerator, $r_h$ is the hydraulic radius of the regenerator mesh (defined as the volume of the working fluid within
the regenerator divided by the wetted surface area of the regenerator), $\rho_m$ is the gas density, $L_i$ is the length of the inertance, $A_i$ is the cross-sectional area of the inertance, $V_c$ is the compliant gas volume, $\gamma$ is the ratio of isobaric to isochoric specific heats of the working fluid, and $p_m$ is the mean pressure.[18]

Combining Eqs. (1.12) and (1.13) and solving for the volume velocity entering the regenerator yields Eq. (1.15),

$$U_{1,c} = \frac{\omega^2 L C p_{l,c}}{R(1 + \frac{\omega L}{R})}.$$  (1.15)

If the magnitude of the feedback inertance impedance is small compared to the regenerator resistance $R$, Eq. (1.15) can be further simplified yielding

$$U_{1,c} \simeq \frac{\omega^2 L C p_{l,c}}{R},$$  (1.16)

which shows that the volume velocity and the pressure at the ambient end of the regenerator are significantly in-phase, corresponding to traveling wave phasing at the regenerator.[22] As Eq. 1.16 shows, the magnitude of $U_{1,c}$ is set by the magnitudes of all three impedances: $R$, $\omega L$, and $1/(C\omega)$. Rearranging Eq. 1.16 yields

$$Z_{1,c} = \frac{p_{l,c}}{U_{1,c}} \simeq \frac{R}{\omega^2 L C},$$  (1.17)

which is the acoustic impedance at the ambient end of the regenerator.

Without the lumped element circuit, a pure traveling wave would have an acoustic impedance of $Z=(\rho_m c)/A$ where $\rho_m$ is the mean density of the working fluid, $c$ is the speed of sound in the working fluid and $A$ is the cross-sectional area of the waveguide containing the wave. In order to reduce the viscous dissipation in the regenerator to an acceptable level, the physical dimensions of the inertance $L$, compliance $C$, and regenerator resistance $R$ should be selected so that \(\frac{|p_{l,c}|}{U_{1,c}}\), at the ambient side of the regenerator, is approximately 15 to 30 times $\frac{\rho_m c}{A}$.

As Backhaus and Swift pointed out, this lumped element circuit can also be used to examine the time-averaged power flows throughout the engine and find the regenerator’s power gain.[20] Most introductory acoustics textbooks define acoustic power as
\[
\dot{E}_2(x) = \frac{\omega}{2\pi} \int \text{Re}[p_1(x)e^{(\omega t)i}]\text{Re}[U_1(x)e^{(\omega t)i}]dt = \frac{1}{2}|p_1||U_1|\cos\phi_{pU}, \tag{1.18}
\]

where \(\phi_{pU}\) is the phase angle between \(p_1\) and \(U_1\), and \(\dot{E}_2\) is the time-averaged acoustic power flowing in the \(x\)-direction.\[18\] Making use of this definition and multiplying Eq. 1.16 by \(p_{1,c}\) gives the acoustic power flowing into the ambient end of the regenerator:

\[
\dot{E}_c = \frac{1}{2}p_{1,c}U_{1,c} \simeq \frac{\omega^2LC|p_{1,c}|^2}{2R}, \tag{1.19}
\]

where second order terms in \(p_1\) and \(U_1\) have been disregarded. Because neither inertance nor compliance generate or absorb time-averaged power in this approximation, the power fed back through the inertance must be equal to the power flowing into ambient side of the regenerator, \(\dot{E}_c = \dot{E}_{fb}\).[56] If the pressure drops caused by the parallel combination of the regenerator resistance and the feedback inertance are small compared to the total acoustic pressure, then \(p_{1,c} \simeq p_{1,h}\). Combining this assumption with the regenerator’s amplification of the volumetric flow implies that the time-averaged acoustic power flowing out of the hot side of the regenerator is given by

\[
\dot{E}_h \simeq \tau\dot{E}_c. \tag{1.20}
\]

Applying conservation of time-averaged power at the junction between the engine and the alternators and assuming a lossless feedback path, the amount of acoustic power \(\dot{E}_{alt}\) flowing down to the alternators is given by

\[
\dot{E}_{alt} \simeq (\tau - 1)\dot{E}_c. \tag{1.21}
\]

To determine \(\tau\) to establish how the engine parameters set the operating temperatures, further equivalent circuit analysis can be performed. Substituting Eq. (1.19) into Eq. (1.21) and knowing that \(\dot{E}_{alt} = \frac{|p_{1,c}|^2}{2R_{alt}}\), \(\tau\) then becomes

\[
\tau \simeq \frac{1 + R}{\omega^2LCR_{alt}}. \tag{1.22}
\]
In an ideal, insulated regenerator where thermal conduction is ignored, \( \dot{E}_h = \dot{Q}_h \), where \( \dot{Q}_h \) is the heat input to the TaSEG’s hot heat exchanger. Using this idealization combined with Eqs. (1.19) and (1.20) yields \( |p_{1,c}|^2 \) as a function of \( \dot{Q}_h \),

\[
|p_{1,c}|^2 \approx \frac{2R\dot{Q}_h}{\tau\omega^2LC}.
\]  

(1.23)

Eqs. (1.22) and (1.23) illustrate how the operating point of a fixed engine geometry is set by external controls. The input heat determines the acoustic pressure amplitude, while the acoustic load (electrodynamic alternators in the case of the TaSEG) presented to the engine dictates the required hot end temperature.

While the simplified model presents an intuitive depiction of how the power flows are distributed, there are several issues that should be noted. The predicted \( U_{1,c} \) and \( U_{1,h} \) are not highly accurate, as it is not obvious what effect the impedances of the actual components will have on these phasors. Research has shown that they can be very sensitive to non-ideal behavior of the various device components. Additionally, while the lumped element regenerator model presented above is adequate for a simplified fundamental analysis, a more detailed model should be used as the volumetric velocity gain occurs in a distributed manner and the large temperature difference between the hot and ambient ends of the regenerator causes critical parameters, such as thermal diffusivity \( \kappa \), dynamic viscosity \( \mu \), thermal penetration depth \( \delta_\kappa \) and viscous penetration depth \( \delta_\nu \) (defined below), to change within the regenerator as a function of axial position.[18] The interaction between the load presented by a linear electrodynamic alternator to the engine portion of a device also requires a more in-depth evaluation.[35, 57] Finally, “real world” design constraints placed on a particular device, in some cases prior to the actual design of the device, due to its intended application can force the design to stray from the ideally defined engine design guidelines.
1.6 Thesis Overview

Chapter 2 focuses on the iterative analytical modeling and machine design process used to establish the TaSEG design. This includes discussion of various design considerations and constraints placed on the TaSEG and a detailed analytical and SOLIDWORKS model of the TaSEG. In Chapter 3 the various engineering details and construction techniques associated with each component of the TaSEG are discussed. Chapter 3 ends with a description of the as-built, experimental TaSEG. Chapter 4 outlines the sensors and measurement instruments used to measure the TaSEG’s performance. Chapter 5 outlines the achieved experimental performance of the TaSEG and the various modifications made and challenges encountered along the way. Chapter 6 draws conclusions based on the outcome of the TaSEG experiments, makes recommendations for improving the TaSEG’s overall thermal-to-electric conversion efficiency, and explores applicable future work.
TaSEG Design Selection

2.1 Introduction

This chapter provides an overview of the TaSEG design goals and constraints, including the balance between the necessary attributes that make this it a viable “real world” product option and the technical goals that advance the fundamental understanding of thermoacoustic-Stirling engines. The main goals and design constraints placed on the TaSEG before mechanical design work had even started are also outlined. These goals and design constraints represent a careful balance between the expected performance of a TaSEG prototype defined in this chapter, its manufacturability, and its suitability for a $\mu$CHP appliance. Two preliminary TaSEG design options (a single 175 alternator and a dual 132 alternator) are presented and the advantages and disadvantages of each are discussed. The suitability of each option as it pertains to integration into a $\mu$CHP appliance is evaluated. Based on this preliminary design review, a decision is made as to which TaSEG design is developed into a functional prototype for experimental testing.

2.2 TaSEG Design Goals and Constraints

Ideally, the geometry of a TaSEG would be formulated using as a guide an expanded version of the rudimentary, lumped element circuit model discussed in the previous chapter. This expanded model would add lumped elements to the sim-
plified circuit shown in Figure 1.7 that more accurately models the geometries of the TaSEG’s internal components by accounting for various acoustic power loss mechanisms, etc.[16] This approach is beneficial in that it allows the tuning of $U_{1,c}$ and $U_{1,h}$, which determine whether the acoustic wave has the desired traveling wave phasing at the regenerator. A fine control of this phasing is used to maximize acoustic power flow with minimum viscous loss.[67] However, this design methodology not place any constraints on the device regarding its overall size and shape, nor does it address features that must be included in the design to make the TaSEG appropriate for commercialization and implementation in a $\mu$CHP household appliance, which is the main focus of this research and development work$^1$. By approaching the TaSEG design from a commercial product perspective, several main goals can be established to guide its development:

1. The working TaSEG prototype shall be able to achieve steady state operation.

2. The TaSEG design shall be cost effective from a production point of view and shall be appropriate for integration into a $\mu$CHP household appliance. It shall also be competitive with alternative $\mu$CHP power modules, including internal combustion and free-piston Stirling engines.

3. Input heat shall be externally applied to a realistic hot heat exchanger from outside the pressure vessel.

4. The TaSEG shall generate 600 to 1000 watts of electrical output with a thermal-to-electric efficiency of $\geq 15\%$.

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$^1$As specified by the project sponsor.
In addition to these four main goals, several additional constraints are imposed on the TaSEG design. Not all of these are hard constraints, but they all represent desirable features that support one of the main goals.

1. The engine portion of the TaSEG shall have a compact, coaxial design similar to that described by C. M. de Blok.[51]

2. The pressure vessel shall meet the EU pressure Equipment Directive (PED) 97/23/EC and/or the ASME boiler and pressure vessel code.[68, 71, 72]

3. The TaSEG shall operate with a hot end metal temperature no greater than 700°C.

4. The hot heat exchanger shall be a finned block type.

5. The ambient (reject) heat exchanger shall be a finned block type surrounded by a water jacket.

6. An elastomer barrier (e.g. latex diaphragm) and a collimator having a flow resistance “tuning knob” shall be implemented in the design to suppress Gedeon streaming and aid in optimizing cycle phasing. These components will then be used to design a static, compact “jet plate” fluid-diode element similar to those implemented by Backhaus and Swift[16] and Wilcox[93] with the end goal of optimizing the jet plate as part of the final TaSEG design.

7. The TaSEG length ≤ 400 mm, diameter ≤ 280 mm, and weight ≤ 48 kg.

8. TaSEG shall have a compact acoustic feedback that fits within the pressure vessel envelope.

9. The linear alternator power pistons shall be “shaped” to address potential piston drift that could occur during operation.[101]

### 2.3 Preliminary TaSEG Designs and Design Selection

The main focus of this thesis is to develop a TaSEG prototype with the necessary attributes required for a real µCHP product. Previously published work on ther-
Figure 2.1. Toroidal geometry developed by Backhaus and Swift [20] (left); Coaxial geometry used in TaSEG (right). The two engines depicted (Not to scale) have approximately the same regenerator volume, but the coaxial TaSEG design has a much greater power density, even if the large resonator volume in the Backhaus engine is neglected.

Thermoacoustic engines [20, 73, 75, 74] used electric heaters to apply heat directly to the working fluid, from inside the pressure vessel. This approach is an acceptable practice when studying the fundamental thermodynamics and acoustics of a thermoacoustic device, but it leaves unaddressed the design of a hot heat exchanger capable of exchanging heat from an external heat source. Internal heating can be applied to an engine in almost any configuration, however if the hot zone is to be heated by a gas burner, then that zone cannot be embedded inside the pressure vessel. Therefore, the requirement of accepting heat from an external heat source is the first design constraint placed on the hot heat exchanger design, which means placing a heating element within the pressure vessel is not a viable option. As it turns out, the ability to operate on an externally applied heat source is a dominant factor in the overall physical layout of the TaSEG prototype.

Figure 2.1 shows the well known Backhaus and Swift thermoacoustic-Stirling engine geometry (left) [16] and a coaxial TaSEG geometry (right). The Backhaus engine makes use of an internal electric heater, represented by $T_H$, while the TaSEG has a hot heat exchanger called “heat acceptor,” which allows the application of an
external heat source. The two engines depicted in Figure 2.1 have approximately the same regenerator volume, but the coaxial TaSEG design has a greater power density, even if the resonator volume (not shown in the figure) of the Backhaus engine is neglected. The lack of any volume constraint or a salient hot zone allows a freedom in sizing of components not shared by the more compact coaxial concept. Given the need for a hot heat exchanger having the ability to accept external heat, a coaxial TaSEG configuration is selected. This decision, coupled with the other aforementioned constraints, requires a radical departure from all previous work.

Two different alternator sizes are deemed appropriate to meet the desired 600 to 1000 watts of electrical output. When operating at 60 Hz, a single 175 Chart STAR alternator is capable of generating 750 watts of electricity; while two 132 Chart STAR alternators\(^4\) are capable of generating 500 watts of electricity together. These two alternator sizes are used to form the basis of two preliminary TaSEG design ideas.

---

\(^4\) See Appendix A for alternator operating specifications. The “132” and “175” alternator designations refer to the diameter of the stator iron used to produce the alternator.
bient heat exchanger, which all surround a thermal buffer tube. Below the thermal buffer tube there is a jet plate that is intended to suppress Gedeon streaming.[83] Further down is the compact inertance tube, which connects the compression space in front of the alternator’s power piston to the compliance space located below the ambient heat exchanger. Together these components form an approximate lumped element feedback loop that, if correctly tuned, creates acoustic traveling wave phasing in the regenerator. The single 175 design resembles its \( \mu \text{CHP} \) appliance Stirling-engine power module counterpart, and would function almost as a drop-in replacement, using the same gas-fired burner configuration and the same type of vibration balancer.[76, 77, 78] The idea behind the single 175 TaSEG design is the desire to produce a TaSEG having smaller physical dimensions, weighing less, and costing less to manufacture, while preserving similar performance to conventional free-piston Stirling engines appropriate for \( \mu \text{CHP} \) appliances.[79]

In order to start the TaSEG mechanical design a selection of a TaSEG configuration needs to be made. A DeltaEC (Design Environment for Low-amplitude ThermoAcoustic Energy Conversion) engine simulation produced by Scott Backhaus of Los Alamos National Laboratory as part of a previous project serves as the starting point for predicting the performance of a TaSEG having the configuration shown in Figure 2.2.[63, 65, 35] DeltaEC is a free computer program designed to predict the behavior of thermoacoustic devices. The program makes use of a sequence of segments, i.e. ducts, regenerators, heat exchangers, etc., which define the geometry of the device. The program then simultaneously numerically integrates the momentum, continuity and power equations through this one-dimensional geometry to predict a device’s performance. The validity of DeltaEC’s calculations is limited to low acoustic amplitude, as acoustic approximations are made to both the momentum and continuity equations.

With support from Phil Spoor\(^5\) of Chart, the Backhaus simulation is further refined into a simulation that serves to establish the space envelope and the optimized performance of a single 175 alternator TaSEG having limited size restrictions. The simulation suggests that a TaSEG of this sort would have an overall height of approximately 445 mm and a maximum diameter of 210 mm, and be capable of generating 950 watts of electrical output with a thermal-to-electric First

\(^5\)“or Dr. Phil.”
Law) efficiency of $\eta_{Th-El} = 26\%$ operating at 92 Hz.

Figure 2.3 shows the sketch of a second design concept similar to Figure 2.2, except the alternator size is reduced to Charts 132 STAR alternator size, and two alternators are required. The idea behind this design is to mirror a smaller, single TaSEG about a central axis to reduce the complexity and cost of the required pressure vessel. By making use of a mirrored, two engine design having a common hot zone, there is no need for a domed-cap pressure vessel, as it is replaced by a simple, single diameter tube (which is much easier to manufacture and much less expensive). Additionally, this TaSEG design makes use of a pair of balanced linear alternators, one for each engine, to convert acoustic power into electricity. The use of two alternators instead of one has the advantage of dynamic balance and vibration cancellation without requiring the separate mass balancer found in almost all traditional free-piston Stirling engine based $\mu$CHP appliances. However, this design requires two of sets of internal components (i.e. hot heat exchanger, cold heat exchanger, regenerator, thermal buffer tube, alternator, etc.) that may increase cost, even with the cheaper pressure vessel. This design also makes the integration into the complete $\mu$CHP appliance a bit more challenging, but not unrealistic. What is unknown with this design is whether the twin configuration is as efficient or cost-effective as the single 175 TaSEG design.

In order to investigate the validity of the dual 132 TaSEG design concept and compare it to the single 175 alternator TaSEG, a production cost analysis is performed on both designs. A list of the pros and cons of each TaSEG design
with respect to their integration into a μCHP appliance is generated and an initial DelatEC simulation of a 132 TaSEG are produced, again supported by Phil Spoor, in order to establish its performance within the given design constraints. The simulation suggests that a TaSEG of this sort would have an overall height of approximately 670 mm and a maximum diameter of 156 mm, and be capable of generating 910 watts of electrical output with a thermal-to-electric efficiency of $\eta_{Th-El} = 28\%$ operating at 110 Hz. The simulation results reveal another benefit to the twin 132 configuration in that the TaSEG naturally runs at a higher frequency (a 132 alternator has much less moving mass than a 175 alternator), which helps the TaSEG achieve the target output power with a lower acoustic amplitude.

<table>
<thead>
<tr>
<th>Table 2.1. Pros and Cons for a Single 175 Alternator TaSEG Design</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Pros</strong></td>
</tr>
<tr>
<td>Heat Cell compact</td>
</tr>
<tr>
<td>Development of heat cell not required</td>
</tr>
<tr>
<td>Stirling burner easily adjustable for TaSEG</td>
</tr>
<tr>
<td>Burner seal developed already</td>
</tr>
<tr>
<td>Retrofit to existing Stirling based μCHP appliance</td>
</tr>
<tr>
<td>Reduced appliance time to market</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>
Table 2.2. Pros and Cons for a Dual 132 Alternator TaSEG Design

<table>
<thead>
<tr>
<th>Pros</th>
<th>Cons</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cheaper, simpler pressure vessel</td>
<td>μCHP appliance development required</td>
</tr>
<tr>
<td>No mass balancer required</td>
<td>Increased appliance length to accommodate longer TaSEG</td>
</tr>
<tr>
<td>No carcinogenic refractory required in burner</td>
<td>Burner development required</td>
</tr>
<tr>
<td>Increased power density</td>
<td>Longer time to market required</td>
</tr>
<tr>
<td>Higher operation temperature possible with cylindrical hot zone</td>
<td>μCHP appliance reliability unknown</td>
</tr>
<tr>
<td>Simpler hot heat exchanger design with cylindrical hot end</td>
<td></td>
</tr>
<tr>
<td>Offers single engine option if desired</td>
<td></td>
</tr>
<tr>
<td>Reduced Start-Stop noise</td>
<td></td>
</tr>
<tr>
<td>Chart 132 alternator most commercialized</td>
<td></td>
</tr>
</tbody>
</table>

The estimated production cost of the single 175 TaSEG at quantities of 50,000 per year is estimated to be $1,054, while the estimated production cost of the dual 132 TaSEG design (also at quantities of 50,000 per year) is found to be $1,104.\(^6\) A separate independent production cost estimate was carried out by a global leader in the home heating appliance manufacturing industry[81]. They estimated a single 175 TaSEG at quantities of 50,000 per year would cost approximately $1,075 while a dual 132 TaSEG at the same production rate would cost approximately $1,086. These estimates are well aligned with the industry opinion that the production cost of the power module of a μCHP appliance that is commercially viable should be less than $1,000. The details of the cost estimate can be found in Appendix B.

Given the results of the DeltaEC TaSEG performance simulations and production cost estimates, no dramatic advantages of either TaSEG design are apparent. Therefore, the emphasis is place on the integration of each design into a μCHP appliance to help establish which TaSEG design to focus on. A single 175 TaSEG and a dual 132 TaSEG is produced to investigate each space envelope and possible integration within a μCHP appliance. Both the dual and single alternator mock-up

\(^6\)These production costs were calculated by the author and are based on time spent working in the μCHP appliance industry.
Figure 2.4. The dual 132 alternator TaSEG (left) has a height of 700 mm and a diameter of 165 mm, while the single, 175 alternator TaSEG (right) has a height of 450 mm and a diameter of 220 mm. These dimensions neglect the outside diameter of any flange joints that maybe required as part of the experimental prototype to allow assembly and disassembly. Such flange joints would be removed in a production-ready appliance.

Table 2.3. TaSEG Design Supporting Values

<table>
<thead>
<tr>
<th>Supporting Values</th>
<th>Design Advantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost</td>
<td>Both Equal</td>
</tr>
<tr>
<td>Safety</td>
<td>Both Equal</td>
</tr>
<tr>
<td>TaSEG $\mu$CHP Appliance Development Time</td>
<td>Single</td>
</tr>
<tr>
<td>TaSEG Power Density</td>
<td>Dual</td>
</tr>
<tr>
<td>TaSEG Lifetime</td>
<td>Dual</td>
</tr>
<tr>
<td>Noise Features</td>
<td>Dual</td>
</tr>
<tr>
<td>$\mu$CHP Appliance Layout</td>
<td>Single</td>
</tr>
<tr>
<td>Unique Selling Points</td>
<td>Dual</td>
</tr>
</tbody>
</table>

TaSEGs are shown in Figure 2.4. The scale of each is indicated with a standard dry-erase marker. The physical dimensions for both shells are derived from the preliminary DeltaEC TaSEG simulations. These TaSEGs are then combined with a commercially available heat cell[87] and housing representative of the size of a $\mu$CHP appliance to provide a general idea of the possible CHP appliance layouts, shown in Figure 2.5. Both appliances make use of a housing that has a width of 575 mm, a height of 840 mm and a depth of 450 mm, also shown in Figure 2.5.

Table 2.3 shows a list of supporting values established to further justify which
Figure 2.5. Mock-up TaSEGs combined with a commercially available heat cell. These provide a general idea of the possible μCHP appliance layouts. The appliance housing has a width of 575 mm, a height of 840 mm and a depth of 450 mm. One difference between the two TaSEG concepts, not addressed in this work, is that the dual 132 alternator TaSEG integrated in the vertical configuration shown, results in one of its two engine halves having a reversed thermal gradient (i.e. the ambient temperature zone will physically be located above the hot temperature zone). Previous work[93] suggests that the TaSEG may perform in any orientation without a significant impact on the performance. The planned experimental testing will be carried out with the TaSEG positioned horizontally on a bench top, therefore the convective stability[82] of operating a dual 132 alternator TaSEG in the configuration shown merits further work.

TaSEG design to focus on. Using Tables 2.1, 2.2, and 2.3, along with the TaSEG sketches and physical mock ups, production cost estimates, and preliminary simulation results for both designs, the dual 132 alternator, mirrored TaSEG is chosen for further development.
Chapter 3

Dual 132 TaSEG Mechanical Design

3.1 Introduction

This section is intended to show how the dual 132 TaSEG design evolved from the general specifications established outlined in Chapter 2 into a finished, working prototype, and how compromises along the way steadily eroded the predicted power output. Some of those compromises were unique to this prototype, and have nothing to do with the technology *per se*. Others are consequences of producing a design that is appropriate for a μCHP appliance, and still others are inherent to the TaSEG itself. Reviewing the TaSEG design and understanding which compromises belong to which category is essential in order to make the most of the knowledge gained in this work. There are several design parameters that clearly distinguish this engine from other previously designed laboratory acoustic-Stirling engines.\[16, 36, 59, 60, 61, 62\] The development of the numerical DeltaEC[63, 64] simulations used to iterate between the TaSEG’s predicted performance and its solid mechanical model are presented and discussed, along with the final TaSEG components and prototype design.
3.2 TaSEG Simulation, Performance Analysis, and Mechanical Design Iterations

The preliminary dual 132 TaSEG DeltaEC simulation established in the design selection process is used as a starting point for the mechanical design. This simulation has morphed from its original form into a baseline simulation that dimensionally reflects the solid model TaSEG design shown in Figure 3.1 and establishes the beginnings of a buildable piece of hardware rather than a theoretical simulation. While rigorous mechanical design is distinct from specifying critical dimensions based on thermoacoustic simulations, it often helps to refer to some initial mechanical design to ensure that the baseline simulations are kept within realistic limits. This crude solid model is the first sanity check that the component dimensions specified in the baseline simulation to obtain the desired results yield a design that is not impossible or unrealistic to construct and integrates with a $\mu$CHP appliance. A careful balance is required to preserve performance (thermal-to-electric efficiency, electrical output, etc.) while producing a design that is acceptable from a manufacturing and integration standpoint.

In addition to the DeltaEC model, a simulation of the 132 TaSEG is also produced in SAGE.[66] SAGE is another software package that is used to simulate and optimize Stirling cycle engines, coolers, pulse-tube cryocoolers and various other types of cooler. SAGE and DeltaEC are two of the most extensively used modeling tools within the thermoacoustics community. These codes are fundamentally different from each other, with SAGE using finite differencing and DeltaEC using integration of the wave equation to solve the simulation. Generally, there is very good agreement between the two codes, with the exceptions being conventional thermoacoustic stacks, where SAGE loses accuracy, and low temperatures, where DeltaEC does not include real gas properties. A difference between these codes from the user’s perspective comes down to the physical viewpoint and specific features. DeltaEC presents an acoustic picture of the system displaying first-order pressure amplitudes and volume flows, whereas SAGE presents physical components, mass flows, and pressures. Both codes model mechanical components such as transducers, masses, and springs. SAGE has a click-and-drag graphical user interface, which is advantageous in presenting a picture of the system while DeltaEC
Figure 3.1. Early solid model of a Dual 132 TaSEG. The end to end length is 678.6 mm and the largest flange diameter is 156.2 mm.

uses a DOS-based interface, which can be cumbersome to a casual user. A particularly powerful feature of SAGE is the optimizer, which does a multivariable search to optimize a user-selected parameter. Additionally, the SAGE software is able to model random fiber regenerators, while DeltaEC cannot. However, SAGE is not as convenient for making changes on the fly as DeltaEC, but it is generally known to get more accurate results for certain kinds of components. A comparison of the simulation results from both simulation packages for the 132 TaSEG is shown in Table 3.1.

In the Table 3.1 the RGR column denotes the type of regenerator material used in the simulation where screen refers to a woven wire stainless steel cloth screen. The HHX column is the temperature of the hot heat exchanger solid, the
Table 3.1. Dual 132 TaSEG Simulation Comparison

<table>
<thead>
<tr>
<th>Model</th>
<th>RGR Type</th>
<th>HHX</th>
<th>AHX</th>
<th>Freq</th>
<th>$x_1$</th>
<th>$\frac{P_1}{P_m}$</th>
<th>$Q_h$</th>
<th>$pV$</th>
<th>Power</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>DeltaEC</td>
<td>Screen</td>
<td>873</td>
<td>347</td>
<td>110</td>
<td>5</td>
<td>9.5</td>
<td>2.9</td>
<td>992</td>
<td>784</td>
<td>.27</td>
</tr>
<tr>
<td>SAGE</td>
<td>Screen</td>
<td>873</td>
<td>347</td>
<td>111</td>
<td>5</td>
<td>9.4</td>
<td>2.8</td>
<td>1060</td>
<td>936</td>
<td>.34</td>
</tr>
</tbody>
</table>

AHX column is the temperature of the ambient heat exchanger solid, Freq is the operating frequency, $x_1$ is the displacement amplitude of the TaSEG’s alternator, $\frac{P_1}{P_m}$ is the ratio of the pressure oscillation amplitude and the mean working pressure $Q_h$ is the heat added to the thermoacoustic gas in the hot heat exchanger, $pV$ is the acoustic power generated by the TaSEG, Power is the electrical power generated by the TaSEG, and $\eta$ is the predicted thermal-to-electric efficiency of the TaSEG. The DeltaEC simulation on which Figure 3.1 is based predicts a TaSEG of this design could theoretically produce 784 W (392 W for a single 132 TaSEG stage) of electricity at a thermal-to-electric efficiency of 27%. The corresponding SAGE simulation suggests a dual 132 TaSEG could produce 936 W (468 W for a single 132 TaSEG stage) of electricity at a thermal-to-electric efficiency of 34%. While these simulation results do not match perfectly, the predicted performance is close enough at this stage to move forward with the mechanical design. It is noted that while the estimated conversion efficiency meets the established performance goal, the predicted electrical output is a bit low.

The mechanical design process is an iterative process that involves using the DeltaEC simulation to establish the specifics of the TaSEG component shapes and dimensions, implementing the components into the mechanical model, modifying as needed to allow interfacing with other components, then seeing how these modifications based on the mechanical solid model affected the simulation predicted performance.

From iteration to iteration the development of the TaSEG mechanical design changes impacted the predicted simulation performance results. Beginning with the baseline simulation, “simulation one,” moving to simulation iteration two, all of the “guesses” and “targets”\(^1\) that affect the component dimensions were removed to investigate the TaSEG performance as a function of amplitude. Additionally, the back volume, or volume on the back side of the alternator piston, was also reduced.

\(^1\)See the DeltaEC users manual for more information on guesses and targets.
from a size of 3.0 liters to 1.5 liters. This reduction increased thermal relaxation loss in the back volume as it increases the pressure wave there and seal gap, but it makes the overall design smaller and hence lighter. This is important as the intention for a TaSEG based µCHP appliance is that it can be hung on a wall like most European conventional boilers. This iteration also saw the areal porosity (ratio of the gas flow cross-sectional area divided by the total heat exchanger cross-sectional area) of the hot heat exchanger change to increase the hot heat exchanger’s length while maintaining an acceptable hot section outside diameter. Additionally, the dimensions of the ambient heat exchanger and the “expansion space,” the small plenums between the heat exchangers and the regenerator that allow the flow there to diffuse, were modified for compactness within the solid model.

Iterating from simulation two to three saw only a minor change in the hot heat exchanger and corresponding expansion space dimension, again done for compactness in the solid model, but at the expense of performance on the basis of a production ready design. The TaSEG solid model after design iteration three is shown in Figure 3.2 below.

Figure 3.2. TaSEG solid mechanical model after design iteration three.

Once the dimensions of the heat exchangers in the simulation reflected those present in the solid model, the focus of the design shifted to examining the hot and ambient flow straighteners and the thermal buffer tube. Based on experience, the flow straightener lengths in the simulation seemed too short, and thus needed to be lengthened. The input dimensions of the flow straightener screens also needed to be changed to reflect readily available, off-the-shelf screen sizes. In doing so one has to
consider how any added flow resistance caused by changing the flow straighteners might alter the cycle phasing. Traditionally, cryocoolers\(^2\) make use of copper flow straighteners that have their own tubular copper housings and are placed over the ends of the thermal buffer tube. This is an effective solution that seals the joint between the buffer tube and the flow straighteners, but it costs more to produce and requires two additional parts. Therefore, the TaSEG’s thermal buffer tube is designed to allow flow straighteners to be press fit into the ends (the exact flow straighteners implemented in the TaSEG are discussed in more detail in Section 3.3.5). This ensures sealing along the seated edges and reduces the complexity, part count and production cost. Implementing these changes in the model provided a welcome boost in the predicted performance.

One of the most important, challenging tasks of the dual 132 TaSEG mechanical design was the strict requirement to integrate the necessary acoustic inertance and compliance feedback components within the TaSEG pressure vessel in a compact manner. Thermoacoustic engines based on the toroidal “Backhaus and Swift[20]” engine design have large, externally accessible feedback components that are part of the pressure vessel wall, while other concentric engine designs[35] place the feedback inertance component as an appendage to the pressure vessel. Neither of these options are desirable in a TaSEG suitable for a µCHP appliance. Therefore, numerous different feedback configurations were investigated in order to include these feedback components compactly within the TaSEG’s pressure vessel space envelope, while also maintaining their acoustic performance. Figure 3.3 shows several different design configurations that were investigated. These configurations were constructed and evaluated. Each was given a score for the manufacturability and transitions of each feedback configuration from 1 to 4. The manufacturability factors taken into consideration included the ease in which each assembly could be mass produced and how the assembly might be integrated into the TaSEG design, while the transition score was based on the minor losses that each might cause.[18, 90, 91]

\(^2\)Specifically Chart Inc.’s cryocoolers with which the author is familiar.
Figure 3.3. Acoustic Inertance-Compliance Feedback Concepts.

Based on these scores the “cochlea” acoustic feedback configuration shown in the center of Figure 3.3 was selected for the TaSEG. Figure 3.4 shows the refined cochlea acoustic feedback (described in more detail in Section 3.3.10) design that was implemented in the TaSEG solid model.

Once this design is selected, its geometry that resulted from the development of the solid model is input into the simulation and the performance simulation is run again. The minor loss effects associated with the transitions within this design are approximated based on the literature.[18, 90, 91] The effect of including the compact cochlea acoustic feedback is seen in the “simulation five” performance results plotted in Figure 3.5.
Figure 3.4. Compact “Cochlea” Acoustic Inertance-Compliance Feedback. The back side of the component interfaces with the ambient heat exchanger and forms the acoustic feedback compliance volume. A plate is welded over the front side covering the visible channel, thus forming the feedback inertance tube, as described in more detail in Section 3.3.10. The entrance to the inertance tube is then connected to the compression space in front of the TaSEG alternator piston. The overall outside diameter of the feedback is 185 mm.

The next and final phase of the TaSEG design that is examined involves the electro-acoustic alternator, which is responsible for converting the acoustic power delivered to the alternator’s power piston into electricity. Initially the simulation made use of standard 132 STAR alternator parameters provided by Chart Inc.\[38\] In parallel to the simulation and mechanical design work, a pair (Serial #132427 and #132433) of custom alternators were constructed. The simulation input parameters were experimentally determined\(^3\) and implemented in the simulation. In the same simulation iteration, the back volume in the solid model (the gas space behind the alternator power piston) and its associated surface area are increased to provide adequate room for the inclusion of a displacement sensor. This increase in back volume reduces the pressure wave amplitude, which in turn decreases the associated acoustic thermal and viscous losses associated with this space. This design iteration also saw the regenerator housing material type change from stainless steel to Inconel 625 due to pressure vessel concerns, which will be discussed in more detail in Section 3.2.1. The development of an anti-drift piston, discussed in more detail in Chapter 5.

\(^3\)Measurement of the alternator parameters is discussed further in Chapter 5.
detail in Section 3.2.1, is also a part of this design iteration. The measured alternator quantities, back volume modifications, changes to front and back volumes due to the inclusion of an anti-drift piston and the pressure vessel wall change are incorporated into “simulation six.” At this point, a realistic TaSEG solid mechanical model that is appropriate for a µCHP appliance and deemed manufacturable for experimental testing was achieved. Additionally, there existed a simulation that matched all the TaSEG component geometries and predicted its performance.

Simulation six and seven are exactly the same except in simulation seven the seal gap, or the clearance seal between the alternator power piston and the bore it reciprocates in, was increased to from 12.7 µm to 25.4 µm based on input from Chart Inc. pertaining to the seal gaps they normally achieve on 132 sized motors. This change had no effect on the solid model, but did adversely affect the predicted TaSEG performance. Figure 3.5 provides an overview of how the changes and compromises made in the TaSEG design evolved with each design and simulation iteration.

Once the DeltaEC simulation completely matches the component geometries represented in the TaSEG solid model, a corresponding simulation in SAGE is completed to compare the results. Table 3.2 shows the DeltaEC and SAGE simulation results based on the component geometries in the dual TaSEG solid model. It also shows the SAGE simulation results for a geometrically identical TaSEG that makes use of a random stainless steel fiber regenerator instead of a woven, stainless steel, stacked screen regenerator. Additional details on the regenerators used in the TaSEG are given in Section 3.3.7. The random fiber regenerator material is approximately four times cheaper than its stacked screen counterpart and should be considered the regenerator material of choice for a low cost µCHP TaSEG power module. Figure 3.6 below shows both a woven, wire-mesh, stainless steel regenerator screen and a sintered, stainless steel, random fiber regenerator element.

The table abbreviations remain the same as those found in Table 3.1, with the inclusion of the “fiber” regenerator type that refers to a sintered, stainless steel, random fiber regenerator.

As Table 3.2 indicates, the DeltaEC and SAGE simulation results differ in their estimated electrical output power and thermal-to-electric efficiency. This
Figure 3.5. Evolution of the predicted TaSEG electrical output (per side) and thermal-to-electric efficiency.

Table 3.2. Dual 132 TaSEG Simulation Results

<table>
<thead>
<tr>
<th>Model</th>
<th>RGR Type</th>
<th>HHX K</th>
<th>AHX K</th>
<th>Freq [Hz]</th>
<th>$x_1$ [mm]</th>
<th>$\eta_{EN}$ [%]</th>
<th>$Q_h$ [kW]</th>
<th>$pV$ [W]</th>
<th>Power [W]</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>DeltaEC</td>
<td>Screen</td>
<td>873</td>
<td>330</td>
<td>141</td>
<td>5</td>
<td>9.1</td>
<td>3.4</td>
<td>1028</td>
<td>676</td>
<td>.20</td>
</tr>
<tr>
<td>SAGE</td>
<td>Screen</td>
<td>873</td>
<td>330</td>
<td>139</td>
<td>5</td>
<td>9.1</td>
<td>3.6</td>
<td>730</td>
<td>502</td>
<td>.14</td>
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<tr>
<td>SAGE</td>
<td>Fiber</td>
<td>873</td>
<td>330</td>
<td>139</td>
<td>5</td>
<td>8.6</td>
<td>4.2</td>
<td>748</td>
<td>564</td>
<td>.13</td>
</tr>
</tbody>
</table>

difference is mainly caused by a lower thermal-to-acoustic conversion efficiency predicted by SAGE compared to DeltaEC; 21% compared to 30%. What the table does not show is that the helium temperature gradient across the regenerator in the SAGE simulation is lower than in the DeltaEC model. It is believed that this is because the conductive path in the hot and ambient heat exchangers are defined axially rather than radially.[86] To address this matter the point source in the heat exchangers in the SAGE simulation are replaced with a line source. The result
Figure 3.6. Woven wire mesh regenerator screen (left) and a sintered random fiber regenerator screen (right). Both screens are made of stainless steel and have the same inside and outside diameters. The random fiber screen is approximately one fourth the cost of the woven mesh screen.

is a slight rise in the thermal-to-acoustic conversion efficiency. Additionally, the heat source and sink temperatures in the SAGE model were adjusted so that that average helium temperature gradient across the regenerator is equal to that in the DeltaEC model. This results in an increase in the predicted electrical output in the SAGE model and leads to better agreement between the two simulations. Table 3.3 shows the DeltaEC and SAGE simulation results after the necessary modifications were made to the SAGE simulations.

<table>
<thead>
<tr>
<th>Model</th>
<th>RGR Type</th>
<th>HHX K</th>
<th>AHX K</th>
<th>Freq Hz</th>
<th>$x_1$ mm</th>
<th>$\frac{pV}{Pm}$ %</th>
<th>$Q_h$ kW</th>
<th>$pV$ W</th>
<th>Power W</th>
<th>$\eta$</th>
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</thead>
<tbody>
<tr>
<td>DeltaEC</td>
<td>Screen</td>
<td>873</td>
<td>330</td>
<td>141</td>
<td>5</td>
<td>9.1</td>
<td>3.4</td>
<td>1028</td>
<td>676</td>
<td>.20</td>
</tr>
<tr>
<td>SAGE</td>
<td>Screen</td>
<td>901</td>
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<td>140</td>
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<td>3.7</td>
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<td>901</td>
<td>317</td>
<td>137</td>
<td>5</td>
<td>8.5</td>
<td>4.5</td>
<td>870</td>
<td>676</td>
<td>.15</td>
</tr>
</tbody>
</table>

As Table 3.3 indicates, the predicted performance results from the two different simulations packages agree and give confidence in the TaSEG mechanical design.
3.2.1 TaSEG Design Challenges

Before the ordering of TaSEG components could commence, there were three issues that still needed to be addressed with respect to the solid model. The first was the need to include a method to suppress Gedeon streaming within the TaSEGs two engine segments.[18, 93] One of the drawbacks of the TaSEG design is the intrinsic closed-loop feedback flow path in its engine’s sections. This path is required for the TaSEG to function, but it also introduces the potential for a DC gas flow, illustrated in Figure 3.7.

![Figure 3.7](image)

**Figure 3.7.** Illustration of generally undesirable DC gas flow, or Gedeon streaming, within thermoacoustic engines. Gedeon streaming can exist in thermoacoustic systems that comprise of a closed-loop path, such as the required feedback path present in the TaSEG design. Left unaddressed in thermoacoustic engines, Gedeon streaming creates an unwanted heat leak by wastefully convecting heat away from the hot heat exchanger without producing any acoustic power. The time-averaged mass-flux density, which is superimposed on a much larger oscillating flow, is signified by the red arrows in the figure.

This DC gas flow arises when the “positive” and “negative” halves of the acoustic cycle do not average to zero. David Gedeon first discovered these DC gas flows while simulating double-inlet pulse-tube cryocoolers.[26] The DC pressure gradients that tend to be produced by in-phase density and velocity fluctuations in
resistive flow elements of the feedback loop are the underlying cause of the DC gas flow. Unbalanced DC pressure gradients within the feedback loop that do not completely cancel result in DC flow and additional thermal losses. The regenerator in a TaSEG is a resistive flow element in which the oscillations of density and velocity are largely in phase (by design); hence, it can be a driver of Gedeon streaming. Based on past experience and to keep the design process moving forward it was decided to incorporate a “collimator” and a latex membrane/barrier into the TaSEG design. The collimator occupies space, reducing “dead volume,” and serves as the mounting support for the latex diaphragm. Additionally, it acts as a flow resistance inserted into the feedback loop, which can be used to “tune” the cycle phasing at the expense of some lost work. The latex diaphragm is a physical barrier that oscillates with the acoustic wave and restricts DC flows from occurring, which prevents Gedeon streaming around the TaSEG engines closed-loop feedback paths. The collimator and latex diaphragm barrier are integrated into the TaSEG mechanical design in a manner that easily allows them to be replaced by a more reliable, compact, static “jet plate” fluid-diode component that suppresses Gedeon streaming.[16, 93] By simply unthreading the collimator and attached diaphragm from the ambient heat exchanger one can remove this streaming suppression assembly from the TaSGE’s feedback path and replace it with a static, asymmetric flow element (the jet plate).

The inclusion of these components allowed the TaSEG design to continue to move forward without delay, but the compactness of the TaSEG design resulted in an available space envelope for the latex diaphragm that is significantly limited. Assuming a simple bowl-shaped deflection of the latex membrane during operation, a ratio of 1 to 6.7 exists between its swept volume and that of the TaSEGs alternator at the intending operating conditions. Based on this, there is a risk that the diaphragm could fail during operation as the pressure amplitude and alternator piston displacement increase toward their intended operating condition. Nonetheless, this is the simplest, least time consuming option, hence it is the one that is selected. The decision is further justified as the low-amplitude TaSEG performance data will provide important information required for the design of the jet plate.

The second issue that needs to be addressed is to verify that the TaSEG pres-
sure vessel design is safe for operation at the operating conditions defined by the DeltaEC simulations. Given that the TaSEG’s pressure vessel could simultaneously be subjected to a high operating pressure of 4.5 MPa and an approximate 650°C to 40°C linear temperature gradient, which induces bending stresses on it, a full pressure vessel analysis of the TaSEGs vessel is completed per Section VIII, Div. 1 of the American Society of Mechanical Engineers (ASME) pressure vessel code.[71, 72] It should be noted that this code is well aligned with the EU pressure Equipment Directive (PED) 97/23/EC.[68, 69] A vessel that is deemed safe per the ASME code will meet the requirements of the EU directive.[70]

Several months were spent simulating, evaluating, and redesigning the thermally active portion of the pressure vessel using finite element analysis (FEA) in SOLIDWORKS simulation.[94, 95, 96] Care had to be taken in balancing the desired heat transfer requirements of the vessel with the vessel’s ability to adequately withstand the stresses it would be subjected to during experiments. Figure 3.8 shows a simulation result for the thermally active portion of the TaSEG’s pressure vessel, which is subjected to both hoop stress from the internal pressure and bending stress caused by the change in the linear temperature gradient across the ends of the TaSEGs regenerator.

Figure 3.8. Meshed thermally active portion of the TaSEG pressure vessel (left) and TaSEG pressure vessel stress results (right).

The FEA simulations suggested that the pressure vessel would not fail during testing, when subjected to the intended operating conditions defined by the
DeltaEC simulations. Despite this, the pressure vessel was evaluated\(^4\) by Dana Pezzimenti, a Professional Engineer[97] at RJR Engineering[98] to address concerns with respect to how the vessel design would hold up to the cyclic stress and fatigue it would experience during testing. His analysis[46] suggest that the cyclic stresses the TaSEG vessel would be subjected to due to the thermal and pressure loading during its operation would lead to vessel failure. As such, two additional TaSEG pressure vessel design options are produced and analyzed. These options include making the 347 stainless steel, thermally active vessel wall thicker or changing the thermally active center portion of the vessel wall material to Inconel 625 (UNS N06625), which has excellent mechanical properties at extremely high temperatures and is appropriate for components exposed to high mechanical stresses, but is a more expensive material.

Analysis of these redesign options suggests that both the thicker 347 stainless steel and the Inconel 625 pressure vessel meet the requirements of the ASME Section VIII, Div. 1. They are therefore deemed safe for the given operating conditions, as shown in Figure 3.9. Because cyclical thermal operation of the TaSEG is expected, a fatigue analysis is also carried out on both vessel options.

Based on the fatigue analysis, the thicker 347 stainless steel vessel is not recommended for cyclic use until a more stringent analysis including material testing is performed, or experience with similar equipment/loading over a sufficient time frame can be obtained. The vessel as analyzed is adequate for approximately 113 pressure and thermal cycles. The Inconel 625 vessel option is appropriate for cyclic use based on past Chart Inc.’s experience with similar vessel loading. The vessel design as analyzed is adequate for approximately 210 pressure and thermal cycles. Cycles here refers to taking the vessel from 0 to 4.5 MPa, then instantaneously increasing the vessel’s inner surface from 20°C to 650°C. This cycling profile is assumed to carry out the analysis of the vessel per Section VIII, Div. 1 of the ASME pressure vessel code and would not be typical of what the TaSEG would experience during typical operation of a \(\mu\)CHP appliance.

In both cases, the results are based on fatigue curves for the materials at 427°C or less. Due to the actual anticipated operating temperature of 650°C, the fatigue life is likely to be substantially less. The low fatigue life is due primarily to the high

\(^4\)The pressure vessel was evaluated for safety but not ASME stamped.
Figure 3.9. Drawing of the thermally active portion of the TaSEG pressure vessel showing the intended working pressure and zone of applied high temperature heat.
transient stress that occurs in the vessel wall should the vessel’s inner surface be instantly heated to 650°C. The analysis is carried out by applying this temperature to the inner surface of the vessel. This causes a very high peak stress due to the hot inner surface being constrained by the cooler outer portion of the vessel wall. Because a much less severe “heat up” condition is anticipated in experiments (i.e. the vessel wall is heated externally and slowly so that the metal temperature is brought to its maximum value over a some duration of time, say half an hour, rather than instantaneously), the peak stresses will be reduced and the fatigue life will be increased. Based on these results it was decided to move forward with the Inconel 625 pressure vessel design for the dual TaSEG.

Figure 3.10. Scaled SOLIDWORKS model of the dual 132 TaSEG. The boxes outline the TaSEG’s four flange joints. The red boxes outline the square flange joints that are clamped by four, \( \frac{3}{4}\)-10, Grade 8 steel bolts and Grade 8 steel nuts equally spaced on a 20.5 cm bolt circle. These flanged joints are both made out of aluminum. The blue boxes outline the circular flange joints that are clamped by eight, \( \frac{3}{8}\)-16, Grade 8 steel screws inserted into threaded holes equally spaced on a 14.8 cm bolt circle. One side of the circular flange joint is made of 316 stainless steel welded to the central Inconel hot tube, while the other is made of aluminum. The aluminum side of the circular flange joint has the threaded holes that have stainless steel helical inserts installed in them to provide stronger threads.

As an additional part of the pressure vessel analysis, the TaSEG’s four flange joints shown in Figure 3.10 are evaluated. This is done to check for potential flange joint separation, to calculate the fatigue loading safety factor of the bolts used in each joint, and to verify that there will be no thread shearing issues in the bolts, nuts and tapped holes used in these flange designs.[99] The Excel spreadsheets used to evaluate the TaSEG’s flange joints can be found in Appendix B. Analysis of these joints suggests that they are adequate for the loading that they will be
subjected to during experimental testing, provided the flange is operated at ambient temperature. The bolts that are to be used to bolt the flanges together must be made of a high strength material with an allowable strength of 29 ksi or higher.

The dual 132 TaSEG design consists of a two of Chart’s 132 alternators arranged in opposition for dynamic vibration balancing, running in clearance-seal piston bores, just like Chart’s typical pressure-wave generators. In the TaSEG as opposed to acoustic-Stirling cryocoolers, acoustic power generated by the TaSEG drives the alternator pistons, which in turn generates electrical power, rather than the reverse, but the construction and the dynamics are very much the same.\[100\] The biggest decisions concerned piston size and mass. These were determined mainly by the DeltaEC simulation, but an upper limit was placed on the moving mass. The normal moving mass of a 132 alternator is 722 grams, with a 4.83 cm diameter piston. Acoustic engines require more volume flow per unit power throughput than their cryocooler counterparts. Because the maximum stroke is mechanically fixed, this means the piston diameter must grow. A larger piston area, however, means a larger reactive force on the piston\(^5\). In the TaSEG design the piston diameter is increased to 9.2 cm to handle the additional required volume flow necessary for TaSEG operation, with the moving mass increased to 974 grams.

Another unfortunate consequence of a large piston area is that a large area makes the pistons more susceptible to drifting axially off-center. It is well known that free-piston devices, like the TaSEG, are susceptible to a shift in the equilibrium power piston position (i.e. moving it off-center), due to a nonzero time-averaged mass flux through the piston’s clearance seal, which results in the development of unbalanced mean pressure difference across the piston. Considerations such as cost and power density accounted for in the design of the TaSEG drove the design in a direction that increased the susceptibility of the alternator’s piston to drift. The TaSEG has a large moving-magnet mass, which is most effectively balanced by using the reactive component of the acoustic pressure acting on the power piston face as the restoring force (rather than mechanical springs). This moved the design of the power piston to a piston with a large frontal area, but hollow insides to keep it relatively lightweight. In general, as the diameter of a piston becomes larger it becomes more difficult to maintain tight tolerances on the

\(^5\)A stiffer “gas spring.”
Figure 3.11. A simplified, labeled schematic of a Chart pressure wave generator that has twin-opposed linear motors/alternators compressing a common front, compression volume. Several important dimensions pertaining to the alternator and anti-drift piston required in the DelatEC TaSEG simulations are visually defined. The reciprocating motion of these motors is ensured by the use of a suspension with relatively low axial stiffness, but high stiffness against rotation or transverse rocking motion. Thus the pistons pure axial motion is guaranteed by the suspension only, and the piston should have only incidental contact with the bore it rides within. The acoustic isolation of the front and back volumes is maintained by a tight, unlubricated clearance seal that has a length of $L$ and a radial gap of $\delta$.[86]

clearance seal gap. These factors increased the time-averaged mass flux through the TaSEG piston seal, resulting in a piston offset (suggested by simulation). This offset compromises the TaSEGs performance by reducing the alternator’s useful stroke, which lead us to include an anti-drift piston in the alternator sub-assembly design.

Figure 3.11 shows a simplified schematic of a Chart pressure wave generator, similar to the TaSEG’s alternators, in which several important alternator and piston dimensions as shown. Some of these dimensions are mentioned above and are inputs required in the DeltaEC simulation including the piston area $A_p$, the front compression space (volume) $V_c$, the back volume $V_b$ per side, the back volume surface area $S_b$ per side that includes the surface of the alternator, the radial clearance seal gap $\delta$\textsuperscript{6} and the seal length $L$.

\textsuperscript{6} The seal gap will be discussed in more detail in Chapter 5.
Spoor[101] suggests an approximate formula to predict the drift of solid pistons due to this second-order mass flux through the piston seal:

\[
x_0 = \frac{2\gamma + 1}{4\gamma P_m} (p_C^2 - p_B^2) \frac{A_p}{K_M},
\]

where \( p_C \) is the dynamic pressure amplitude in the compression space, \( p_B \) is the dynamic pressure in the back space, \( P_m \) is the mean pressure, \( A_p \) is the piston area, \( K_M \) is the alternators intrinsic (“mechanical”) stiffness, and \( \gamma \) is the ratio of isobaric to isochoric specific heats of the working fluid (helium). In the present TaSEG design, at the full-power (340 watt output per alternator) point, the formula predicts a forward drift of over 3.5 mm, out of a total zero-to-peak stroke amplitude of 5 mm. In other words, the alternator could lose over half of its available stroke to drift. There are a number of techniques that have been developed to deal with this. For instance, the piston can be shaped such that the portion of it that establishes the clearance seal length, \( L \) in Figure 3.11, deforms with the oscillating pressure so that the clearance seal gap, \( \delta \) in Figure 3.11, is modulated in such a way to form a weak check valve that pumps helium to the front side. This requires a fair amount of iterative design work in itself, and furthermore, requires that the pistons not be solid but have a thin enough “shell\(^7\)” that the acoustic pressure oscillations can deform. While this is a time-intensive solution to addressing piston drift, it is felt that this is the best way to address the drift problem.

The design of the anti-drift piston involves selecting a piston shape, running it through the finite element analysis (FEA) add-on in SOLIDWORKS simulation that yields the deflection of the piston along its outer edge at the two maximum peak pressure differences that exist across it during operation, then seeing how that deflection affects the mass flux/leakage through the seal. In the case of the TaSEG, ten piston shapes were evaluated before the an anti-drift shape that mitigates piston drift was achieved. Figure 3.12 shows the FEA anti-drift piston clearance seal length deflection results for the final piston shape, while Figure 3.13 shows the plotted clearance seal length deflection along the red dashed line shown in Figure

\(^7\)Shell here refers to the radial wall thickness of the piston edge that establishes the clearance seal length. In many cases it’s the web or cone that connects the shell and the shaft that does the most deformation.
The Finite Element Analysis deflection results showing the deflection of the anti-drift piston’s clearance seal length at the two extremes of pressure cycle (left). The undeformed anti-drift piston (top right) and a sectioned view of the undeformed piston (bottom right) for reference. The x,y axis defining the (0,0) coordinate found in Figure 3.13 is shown for reference. The red dashed line in the section piston view shows the clearance seal length whose deflection is plotted in Figure 3.13.

3.12. The deflection values are taken from the FEA results. These results are analyzed using a custom designed Excel spreadsheet and suggest that the piston design should have almost no drift up to a 5 bar peak working pressure amplitude, which is above the expected peak working pressure amplitude of 3.6 bar predicted by the DeltaEC model.[101] The mechanical drawing for the anti-drift piston, part number 2S132E-20293-C, along with the Excel spreadsheet used to analyze the piston performance, can be found in Appendix B.
Figure 3.13. Anti-drift piston clearance seal length deflection taken from SOLIDWORKS Simulation FEA results along the dashed red line shown in Figure 3.12.

3.2.2 Resultant TaSEG Mechanical Design Geometry

Now that the TaSEG’s pressure vessel design is deemed safe and the anti-drift piston design is complete, the mechanical design of the TaSEG is “finished” and procurement and construction of the TaSEG prototype can begin. The performance of this TaSEG design was simulated with DeltaEC. Table 3.4 shows predicted simulation results. The DelatEC simulation assumes that the TaSEG’s alternators will have fixed 5 mm zero-to-peak displacement, $x_1$, and holds the ambient heat exchanger metal at 330°C and the hot heat exchanger metal at 873°C. The DeltaEC simulation representing the TaSEG’s mechanical design is shown in Appendix B.

<table>
<thead>
<tr>
<th>Model</th>
<th>RGR Type</th>
<th>HHX K</th>
<th>AHX K</th>
<th>Freq Hz</th>
<th>$x_1$ mm</th>
<th>$\frac{p_l}{p_m}$ %</th>
<th>$Q_h$ W</th>
<th>$pV$ W</th>
<th>Power W</th>
<th>$\eta$</th>
</tr>
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<tbody>
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<td>DeltaEC</td>
<td>Screen</td>
<td>873</td>
<td>330</td>
<td>141</td>
<td>5</td>
<td>9.1</td>
<td>1668</td>
<td>513</td>
<td>332</td>
<td>0.20</td>
</tr>
</tbody>
</table>
Figure 3.14 shows a scaled, labeled, sectioned view of the dual 132 TaSEG. The “flow direction plate” in the middle of the figure serves as the central part in which one TaSEG half is mirrored to establish the dual design. In addition to establishing the mirroring plane, the plate also helps direct the traveling acoustic wave from the hot heat exchanger into the thermal buffer tube, which requires the wave be redirected 180°. Flow straighteners are placed at both ends of the thermal buffer tube to help establish plug flow within. Next, going outward along the axis, is the collimator and diaphragm that can be used to add flow resistance and “tune” the cycle phasing in addition to suppressing Gedeon streaming. These components face the alternator’s anti-drift piston and their boundaries define the compression space. Also in communication with the compression space is the entrance to the “cochlea” acoustic feedback inerterence tube that terminates into the feedback compliance volume. This volume communicates with the entrance to the ambient heat exchanger that is wrapped around a portion of the collimator-diaphragm assembly and the thermal buffer tube. Next to the ambient heat exchanger is the concentric regenerator that spans the distance between the ambient and hot heat exchanger. The hot heat exchanger then connects to the flow direction plate that completes the coaxial TaSEG geometry and establishes the necessary acoustic feedback path within the TaSEG.
Figure 3.14. Sectioned view of the SOLIDWORKS mechanical model of the dual 132 TaSEG. The important components discussed in more detail in Section 3.3 are labeled.
3.3 Constructed Experimental TaSEG

This section outlines the construction details of the dual 132 TaSEG. The subsequent subsections discuss each component of the TaSEG in detail. The experimental TaSEG is shown in Figure 3.15. In this figure both the external and some of the internal components are visible. These include (moving from right to left) a displacement sensor, the displacement sensor adapter, the back volume alternator housing, the alternator, the inertance flange (piston sleeve), the cochlea acoustic feedback, the collimator with attached diaphragm, the ambient heat exchanger, the flow straighteners, thermal buffer tube, some regenerator screens and the flanged center vessel. The hot heat exchanger and flow direction plate are not visible as they were already brazed inside the center vessel when the “exploded” TaSEG photo was taken. The TaSEG has an overall length of 99 cm and height of 27 cm.

![Figure 3.15. Exploded view of one half of the experimental TaSEG that shows the internal components. The hot heat exchanger and flow direction plates are not visible in the exploded view as they are brazed inside the flanged center vessel.](image)

3.3.1 Flanged Center Vessel

The flanged center vessel, which was the subject of the pressure vessel analysis, is shown in Figure 3.16. It consists of two 316L stainless steel flange assemblies welded by a certified welder\(^8\) to the ends of a Inconel 625 (UNS N06625) center tube and has a flange-to-flange outer face length of 22.9 cm. It houses the flow direction plate, the hot heat exchangers, regenerators, ambient heat exchangers,

\(^8\)Northeast Precision Welding Inc., Castleton On Hudson, NY 12033
Figure 3.16. Flanged center vessel prior to “clean-up” machining. The visible weld joints show the connections between the stainless steel flanges and the Inconel center tube.

thermal buffer tube assemblies, and the collimator-diaphragm assemblies. It is considered the “heart” of the TaSEG. The flanges on either end of the assembly are flat faced flanges that have an 2-242 O-ring groove in them. This forms a helium tight seal between the mating sections.

In accordance with Section VIII, Div. 1, Part UG-27 of the ASME pressure vessel code, the required vessel wall thickness $t_r$ is defined as

$$ t_r = \frac{Pr}{SE - 0.6P}, \quad (3.2) $$

where $P$ is the maximum working pressure (650 psi), $r$ is the inside radius of the center tube (1.65 inches), $S$ is the maximum allowable stress value of Inconel 625 material (23,068 psi) and $E$ is the joint efficiency that is 0.85. $E$ comes from Section VIII, Div. 1, Part UW-12. Thus, according to Eq. 3.2, the minimum required wall thickness of this assembly is 0.0558 inch. However, Section VIII, Div. 1, Part UG-16(b) states that the minimum thickness of shells and heads “shall be 0.0625 inch,” which is what is implemented in the flanged center vessel.
3.3.2 Hot Heat Exchangers

Figure 3.17 shows the TaSEG’s hot heat exchangers. These exchangers are made from C10100 oxygen-free high thermal conductivity copper and are a finned block type exchanger. They have an outside diameter of 85.28 mm. They have fifty-two, 1.4 mm wide by 32.7 mm long channels that have a radial length of 16.8 mm. The gas channels are used to transfer heat from the solid material to the working helium gas. The fin length is approximately 1.33 times the peak-to-peak gas displacement amplitude predicted by the DeltaEC model.

3.3.3 Flow Direction Plate

The flow direction plate that defines the mirroring plane with the TaSEG is also shown in Figure 3.17. This plate is also made of C10100 oxygen-free high thermal conductivity copper. It has an outside diameter of 85.26 mm and a height of 16.46 mm. It is sandwiched between the two hot heat exchangers. The plate is intended to aid in redirecting the acoustic wave 180° from the hot heat exchanger into the thermal buffer tube so it has a curved symmetric profile that is shown in sectioned component view of Figure 3.18. It also has four through holes that allow the two halves of the TaSEG to communicate acoustically with one another. It is believed[45, 86] that these holes will aid in the mode-locking of the TaSEG’s two halves, although no in-depth mode-locking work has been carried out to validate this hypothesis.

The hot heat exchangers are arranged so that they sandwich the flow direction plate as shown in the bottom image of Figure 3.17. They are then nickel brazed by an outside vendor to the inside diameter of the flanged center vessel in order to increase the expected heat transfer occurring there. A custom designed alumina silicate ceramic tool is used to set the sandwich stack height within the vessel. Brazing stop-off is used between the fins of the hot heat exchangers to prevent brazing material from flowing into and possibly clogging them during the brazing process.

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9Solar Atmospheres, 255 Township Line Road, Hatfield, PA 19440
Figure 3.17. The TaSEG’s finished flanged center vessel (top left), its copper flow direction plate (top center) and its copper, finned, ring shaped hot heat exchangers (top right). The 73.5mm long, white alumina silicate ceramic tooling required during the brazing of these parts to set the height of the hot heat exchangers and flow direction plate within the flanged center vessel is also visible.
3.3.4 Ambient Heat Exchangers

The ambient heat exchangers are similar to the finned block type hot heat exchangers except they have a different fin profile and count and are surrounded by a water jacket. The exchanger assemblies consists of two components, an inner, axially-radially finned ring part and an outer, azimuthally-radially finned part. Figure 3.19 shows the ambient heat exchangers prior to having the water jacket fins and O-ring grooves machined in their outside diameter.

Both ambient exchanger components are made from C10100 oxygen-free high thermal conductivity copper. The inner, axially-radially finned ring part has an outside diameter of 75.4 mm. It has sixty, 0.51 mm wide by 40.6 mm axial length channels that have a radial length of 9.1 mm. These gas channels are used to transfer (reject) heat from the working helium gas, through the solid material, to the outside world. Their axial length is approximately the same as the peak-to-peak gas displacement amplitude predicted by the DeltaEC model. This component
Copper, finned, ring shaped ambient heat exchangers prior to having the water jacket fins machined in their outside diameter. Also has a 2-033 O-ring groove machined in it that is used as an internal helium seal between the ambient exchanger, compact feedback and compression space. The outer, azimuthally finned part serves as the exchangers water jacket cooling the inner exchanger and the helium within. It has an outside diameter of 86.9 mm. In order to increase the heat transfer between these two components, prior to machining the water channels and O-ring grooves into the outer component, the inner and outer components are vacuum silver brazed to one another.\textsuperscript{10} Again, brazing stop-off is used between the fins of the inner exchanger part to prevent brazing material from flowing into and possibly clogging them during the brazing process. Then the finishing details (fins, O-ring grooves, etc.) are machined into the outer piece. There are seven, 1.6 mm wide water channels, and two (a 2-152 and a 2-235) O-ring grooves machined into the outer component after brazing is complete. The O-rings seal the water space from the rest of the internal TaSEG components. Figure 3.26 shows the finished ambient heat exchangers prior to being assembled into the prototype.

\textsuperscript{10}Solar Atmospheres, 255 Township Line Road, Hatfield, PA 19440
3.3.5 Thermal Buffer Tubes and Flow Straighteners

The thermal buffer tube is a straight, 87.4 mm long open cylinder made from 304 stainless steel seamless tubing. It is intended to provide a thermal buffer between the hot heat exchanger and compression space that is at room temperature. It has an outside diameter of 47.0 mm and inner diameter of 44.2 mm, which is much larger than the helium’s thermal penetration depth. Its length is approximately 2.68 times the peak-to-peak gas displacement amplitude predicted by the DeltaEC model. A 45.7 mm diameter by 9.1 mm step is machined into one end of the buffer tube while a 45.7 mm by 6.9 mm step is machined into the other end. These steps will serve as the housings for the hot and ambient flow straightener, respectively.

The inside surface has a measured roughness average surface finish of $Ra = 12.7 \mu\text{in} \ (0.32 \mu\text{m})$. As described in ASME B46.1, $Ra$ is the arithmetic average of the absolute values of the profile height deviations from the mean line, recorded within the evaluation length. Another way to think about it is that $Ra$ is the average of a set of individual measurements of a surfaces peaks and valleys. Ideally, the surface roughness should be much less than the viscous and thermal penetration depths to avoid the generation of turbulence in the boundary layer that would result in additional unwanted losses.[27] Making use of Eq. 1.7 and 1.8, the average thermal penetration within the thermal buffer tube is $\delta_{\kappa} = 181 \mu\text{m}$ and the average viscous penetration depth within the thermal buffer tube is $\delta_{\nu} = 145 \mu\text{m}$. Thus, the measured surface finish satisfies the $Ra \ll \delta_{\kappa}, \delta_{\nu}$ specification.

Woven screen flow straighteners are inserted into each end of the thermal buffer tube. At the hot end, the end of the buffer tube that is press fit into the hot heat exchanger, the flow straightener that is inserted is made of 8 layers of 316 stainless steel, 14 x 14 mesh, 0.020 inch diameter wire, woven screen. At the ambient end, the end of the buffer tube that is press fit into the ambient heat exchanger, the flow straightener inserted is made of 13 layers of 316 stainless steel, 35 x 35 mesh, 0.011 inch diameter wire, woven screen. In order to produce the flow straighteners, 4 inch x 4 inch squares of the appropriate number of layers and screen size are stacked on a 45° alternating basis then compressed to a specified height. This clamped assembly is then run through a sintering process. The resulting sintered square screen beds are then wire electrical discharged machined to produce the final circular flow straighteners. In total four flow straighteners are produced for
the TaSEG, two of each kind—hot and ambient. The hot end flow straighteners have an average outside diameter of 45.7 mm and an average thickness of 6.2 mm, while the ambient end flow straighteners have an average outside diameter of 45.7 mm and an average thickness of 6.3 mm. An arbor press is used to press-fit the flow straighteners into the appropriate end of the thermal buffer tube. Flow straighteners are used to ensure that the flow entering the thermal buffer tube is spatially uniform ("plug" flow), not a jet flow that could result in jet-driven streaming.[18] Figure 3.20 shows the flow straighteners and thermal buffer tubes both before and after flow straighteners are inserted.

3.3.6 Collimator-Diaphragm Assembly

After the ambient end flow straightener comes the collimator-diaphragm assembly. The collimator is a machined piece of aluminum that has an outside diameter of 45.7 mm. The outside diameter on one end is threaded with $1\frac{3}{4}$-5 UNC-2A threads that allows it to be easily inserted and removed from the middle of the ambient heat exchanger. Wrench flats are machined into the other end to aid in the insertion and removal process. It also has 19 evenly spaced 6.35 mm through holes. These large through holes minimize the initial flow resistance and serve as insertion holes for "tuning" plugs. The idea is that the collimator’s flow resistance can be adjusted
3.3.7 Regenerator

Two different regenerators were fabricated for use within the TaSEG. The first was a 3.8 cm stack of 540, 316 stainless steel 240 x 240 wire mesh screens, having a 0.036 mm (0.0014 inch) wire diameter. A custom designed punch is used to produce a concentric 85.85 mm outside diameter, 46.48 mm inside diameter regenerator piece. The second regenerator is a 3.8 cm stack of 48, 0.84 mm thick, 316 stainless steel...
Figure 3.22. The machined aluminum collimators, the fabricated latex diaphragms and a few of the “tuning” plugs. The “tuning” plugs can be inserted into the collimator’s through holes to change its flow resistance while the latex diaphragm is attached to the threaded holes in the collimator to suppress Gedeon streaming.

random fiber discs. The random fiber sheets that are punched to produce these discs have a manufacturer\textsuperscript{12} specified porosity of 84\% and a fiber diameter of 40 µm. The punching process produces a disc that also has a 85.85 mm outside diameter and a 46.48 mm inside diameter. Both regenerators are designed to have a 254 µm (10 mil) interference fit with the regenerator wall inside diameter and the thermal buffer tube outside diameter to prevent regenerator blow by that would be detrimental to the TaSEG performance. Both regenerator types were tested in the prototype TaSEG. As the regenerator housing is the thermally active portion of the TaSEG pressure vessel no thermocouples are integrated through its wall. Therefore, no direct measurement of the axial or azimuthal temperature profiles in the regenerator are made. However, several type K thermocouples are attached to the outside diameter of the regenerator-housing/pressure-vessel wall to allow

\textsuperscript{12}Fuji Filter Mfg. Co., LTD, Japan, per Chart drawing number 2S132E-23396-C.
Table 3.5. Calculated Regenerator Porosities

<table>
<thead>
<tr>
<th>RGR Type</th>
<th>Weight g</th>
<th>$V_m$ CC</th>
<th>$\phi$ %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Screen</td>
<td>310.5</td>
<td>38.8</td>
<td>75.1</td>
</tr>
<tr>
<td>Fiber</td>
<td>200.2</td>
<td>25.0</td>
<td>83.9</td>
</tr>
</tbody>
</table>

the measurement of the temperature profile across the regenerator during testing. These thermocouples are shown in Figure 4.5 of Chapter 4.

The parameters of the regenerators are examined in more detail in order to compare what was built and tested to the DeltaEC simulation. The porosity of the regenerator can be calculated in a couple of different ways. One way is to weigh a certain number of screens/discs, then divide by the theoretical density of the 316 stainless steel, $\rho_{316SS}=7.87$ g/cc, to obtain the volume of the metal $V_m$. By making these measurements it is then possible to determine the porosity $\phi$ by the formula

$$\phi = 1 - \frac{V_m}{V_t},$$  \hspace{1cm} (3.3)

where $V_t$ is the total occupied regenerator volume. Both regenerators occupy the same volume within the TaSEG, $V_t=155$ cc. Table 3.5 gives the calculated regenerator porosity for both the 316 stainless steel woven screen and random fiber regenerators based on this methodology.

A second method to compute the porosity of the regenerator involves a different method to determine the volume of the stainless wire. This relies on the accurately knowing the screen’s wire diameter and mesh size. Expressions for the hydraulic radius of stacked-screen regenerators as a function of porosity and wire diameter are derived by Organ.[37] The total volume of metal in one screen equals the cross sectional area of one wire, times the length of the wire, times the number of wires, times two because there are two crossed wires in each mesh. As the regenerator has a concentric shape each wire length will be different. Therefore, it is easiest to compute the amount of wire in a square geometry and multiply by the ratio of the area of a circle and square, $\frac{\pi}{4}$. Assuming that the aforementioned ratio is that of a square having a side length $D$ and a circle of diameter $D$, then the above method simplifies into the following relationship
\[ \phi = 1 - \frac{\pi n d_w}{4}, \quad (3.4) \]

where \(d_w\) is the diameter of the regenerator wire and \(n\) is the number of wires per distance in the screen, also referred to as the screens mesh number.[18] Substituting the appropriate values into Eq. 3.4 for the 316 stainless steel 240 x 240 wire mesh screen (having a 0.0014 inch wire diameter) regenerator material yields a calculated porosity of 73.6\%, which is within 1.9\% of the previous calculation for this regenerator. As the random fiber does not have a defined mesh size (because it is random) this calculation method can not be applied to it. It is important to know the porosity of the regenerator material used as this is a critical user defined parameter required in the DeltaEC and SAGE simulations.

Another important regenerator parameter required in the DeltaEC and SAGE simulations is the hydraulic radius \(r_h\) that is defined as the ratio of the volume of the fluid divided by the wetted surface area of the solid. In the case of a wire mesh, the volume of the fluid is the total volume \(V_t\) minus the volume that is occupied by wire, \(V_w = V_t (1 - \phi)\). The solid area is \(V_t (1 - \phi) \frac{4}{d_w}\), where \(\frac{4}{d_w}\) is the ratio of a cylinder’s (wire) surface area to its volume. Combing these relationships results in

\[ r_h = d_w \frac{\phi}{4(1 - \phi)}. \quad (3.5) \]

Using the two porosity values calculated with Eq. 3.3 and Eq. 3.4, the hydraulic radius of the woven screen regenerator is 26.7 \(\mu\)m and 24.8 \(\mu\)m respectively, which differ by 7\%. When matching the experimental TaSEG performance data with the DeltaEC simulation the averages of the porosity and hydraulic radius will be used. Using the porosity found with Eq. 3.3, 83.9\%, the hydraulic radius of the random fiber regenerator is 52.1 \(\mu\)m. These values will be used in the SAGE simulation when matching the experimental TaSEG performance.

### 3.3.8 132 Alternator and Anti-drift Piston

Two Chart 1S132M/A motor-alternators are used by the TaSEG. These are linear reciprocating devices that combine a unique flexing axial suspension with plunger mounted magnets and robust coil-over-iron stators. These alternators convert the
Figure 3.23. Exploded SOLIDWORKS model of the Chart 1S132M/A alternator used in the TaSEG. The major components that make up the alternator are visible. The “piston” in the model is replaced by the anti-drift piston.

The acoustic power generated by the TaSEG into useful electricity by the following process[38]:

1. Alternating pressure is applied to the piston on which high-energy magnets are mounted. The force of the pressure (and associated gas flow) cause the piston to mechanically oscillate.

2. The oscillatory movement causes changing flux links between the magnets moving with the piston and the stationary iron stator.

3. The reversing flux changes in the iron induce voltage in the windings. If connected to a load resistance, the voltage causes current to flow and electrical power to be delivered from the alternator.

4. The power factor (phase of voltage and current) can be corrected by use of an appropriate balance capacitor wired in series with the alternator and load.
Figure 3.24. The front (left) and back (right) of the anti-drift alternator piston that was designed to mitigate piston drift during operation of the TASEG. The outer piston shell, shaped such that it deforms with oscillating pressure to modulate the clearance seal gap, is visible on the front of the piston.

Figure 3.23 shows an exploded view of a 1S132M/A alternator in which all of the major components are visible. The alternators used in the TaSEG differs from that shown in Figure 3.23 in that the solid “piston” in the figure is replaced by the aforementioned anti-drift piston that is pictured in Figure 3.24. The machined aluminum pistons have an outside diameter of 97.0 mm, resulting in a piston area of 75.4 cm² and a seal length L of 31.8 mm. It should be pointed out that the inside profile of the alternator housing shown in Figure 3.14, which encapsulates the alternator and establishes the TaSEG’s back volume, is designed to mimic a weld cap having a ratio of radius to depth of 2:1.

3.3.9 An Interesting Aside: 1S132 Alternator vs Motor Operation Mode

Chart’s linear reciprocating devices can be used as both an alternator as described in Section 3.3.8 or as a motor. When used as a motor alternating voltage is
converted into acoustic power delivered to some load in the following process[38]:

1. Alternating voltage is applied to the linear motor from an alternating current (AC) power source. Current flows through the motor windings, causing a magnetic field in the iron that they surround.

2. The applied field, which is oscillating due to the reversing electrical input from the AC power source, pushes and pulls on permanent magnets mounted on the moving plunger/piston assembly.

3. When the applied voltage frequency is close to the resonant frequency of the combined motor and presented load, the piston oscillates and power is delivered to the acoustic load by phase separation of the current and piston displacement.

4. Typically, when connected to an acoustic load like a cryogenic coldhead, helium gas is cyclically compressed and expanded relative to the mean pressure by the piston of the driver to power a thermodynamic cycle for refrigeration.

All of the measured 1S132M/A data required as inputs for the DeltaEC TaSEG simulations are done from the motor mode perspective as Chart’s (and others) interests lie principally in using these devices as drivers of acoustic loads. However, in the case of the TaSEG, when these devices are used in alternator mode there is a penalty that must be applied to the transduction coefficient $B_l$ that is a required input in the DeltaEC simulations.[103, 89] This reduction can be explained by the following$^{13}$:

“Chart alternators/motors are Lorentz force machines$^{14}$, in which the cross product of B-field and current in the wound coil wire is proportional to force. When the input is the current (i.e. the device is operating in motor mode), the field it generates is confined closely to the air gap in which the permanent magnet resides, because the coils

$^{13}$The quote is given by the designer of the device and reflects their opinion of what exactly causes the change in performance when the device is used as an alternator rather than a motor. Further investigation is suggested to fully understand the physics of what is occurring in each case.

$^{14}$Much like all E-M motors.
are near the pole tips of the iron. The permeability coefficient of the magnet material is near unity (about 1.05), so there is little effective ‘bucking’ or back electromotive force (EMF) to oppose that electrically produced field. However, when these devices are used in alternator mode, with the moving magnets field as the input that varies the field in the iron thats surrounded by the coil wires, the current in those wires produces a back EMF, that opposes and diffuses the imposed field from the permanent magnets. So, in a motor, the EMF (force due to current) is the driving input, and produces the desired force on the magnet plunger/piston. In an alternator, the applied force on the plunger is the input, and the EMF due to the developed coil current opposes that developed current and diminishes the net output. The overall effect for a conventional Chart geometry (either 4 or 8 pole) is an approximate 20% reduction in the transduction coefficient, $Bl$ of alternator vs motor mode.”[102]

Thus, this reduction must be implemented in the DeltaEC simulation to accurately predict the performance of the TaSEG.

### 3.3.10 Compact “Cochlea” Acoustic Inertance Compliance Feedback

The last component used in the TaSEG prototype is the cochlea acoustic feedback, which consists of two separate sections, the inertance tube and compliance volume, combined into one compact component. The cylindrical feedback component, made out of aluminum, has an outside diameter of 18.40 cm and a height of 3.75 cm. A CNC machine is used to machine out the profile of the inertance tube in one side of the aluminum and the compliance volume in the other side. These sides are shown in Figure 3.4. These profiles create an inertance tube that has an overall center-line length of 9.27 cm and an initial perimeter of 6.23 cm at its entrance that transitions down to a perimeter of 5.11 cm over an entrance length of 1.01 cm and a compliance volume that is 147.2 cc in size. The perimeter of the inertance tube is given because its cross section is not circular due to the compact design.\textsuperscript{15} While

\textsuperscript{15}The mechanical drawing of this component can be found in Appendix B.
overall design is very compact, the CNC machining process used to create this assembly results in very sharp edges at the transitions between the compression space and the entrance to the inertance tube and between the compliance volume and the exit of the inertance tube. In order to reduce the minor losses associated with those transitions as much as possible from those of sharp edges[91], each transition is carefully rounded by hand with needle files.

Referring to Figure 3.4, in order to establish the inertance tube a plate must be friction stir welded over the profile machined into the front side of the component. Figure 3.25 shows the constructed compact “cochlea” acoustic feedback assemblies. The front (inertance) side of the machined assembly absent the plate is visible in the top right while the SOLIDWORKS model also showing the front side is visible in the top left. The dashed red lines indicate the welding seams used
when the plate is welded onto this side of the assembly, thus establishing the inertance tube portion of the acoustic feedback. The bottom left image shows the front side of the feedback assembly with the plate welded in place, while the bottom right image shows the back side of the feedback assembly that mates to the ambient heat exchanger to form the compliance volume. The inertance tube connects the TaSEG’s compression space in front of the alternators to the compliance volume. This assembly is then anodized in a black hard coat having a thickness of approximately 0.025 mm to aid in protecting its surfaces when the TaSEG is assembled and disassembled during experimental testing.

### 3.3.11 Assembly Process

The previous sections in this chapter have given a detailed description of all the internal components of the prototype TaSEG. This section outlines the methods by which these components are assembled to produce the prototype dual 132 TaSEG.

Figure 3.26 shows several key components of the TaSEG prototype. The flanged center vessel that has the TaSEG’s two hot heat exchangers and flow direction plate brazed to its inside diameter is visible in the top left. This figure also shows the TaSEG’s two Thermocoax resistance heating elements that are brazed to the central vessel’s outside and the TaSEG’s thermal buffer tubes that have the flow straighteners pressed-fit into their stepped ends (middle left), the custom designed regenerator insertion tool (middle right), the machined ambient heat exchangers (front left) and several woven screen regenerator elements are also pictured. A standard Zebra F-701 pen is in the foreground for scale.

The assembly process begins with the flanged center vessel that has the two hot heat exchangers and flow direction plate brazed to its inside diameter and the thermal buffer tubes having the flow straighteners pressed-fit into its stepped ends. Given the different thermal expansion coefficients of Inconel 625 $\alpha_{\text{UNS-N06625}} = 17.0 \, \mu\text{m}/(\text{m}\cdot\text{°C})$ and copper C10100 $\alpha_{\text{C10100}} = 12.8 \, \mu\text{m}/(\text{m}\cdot\text{°C})$, extreme care had to be taken in the sizing and fit of the mating components and several experimental runs through the brazing process were carried out supported by the brazing house, Solar Atmospheres.

A custom designed insertion tool is used to press fit the thermal buffer tube
Figure 3.26. The TaSEG’s flanged center vessel assembly. A standard Zebra F-701 pen is in the foreground for scale.

into the center through hole of the hot heat exchanger. Care is taken to ensure that the end of the thermal buffer tube having the 14 x 14 mesh screens is pressed up against the center knob of the flow direction plate. Once the buffer tube is in place, the regenerator screens (or fiber discs) are inserted, one at a time, then pressed into place with the custom insertion tool. The woven mesh regenerator screens naturally want to curl, so care is taken when inserting them to assure that they don’t become folded over on themselves during the insertion process. Figure 3.27 shows the regenerator of one half of the TaSEG being packed. The regenerator screens are inserted into the center vessel until a regenerator height of 3.8 cm ±0.05 cm is achieved.

Next, O-rings are installed on the ambient heat exchanger and it is pressed into the flanged center vessel via a hydraulic press. This ensures that the O-rings
Figure 3.27. A view looking down into the flanged center vessel, visible are a thermal buffer tube, a 35 x 35 sintered screen flow straightener, and a few inserted woven 240 x 240 wire mesh regenerator screens. Curling of the regenerator screen is visible on the left side. Care must be taken during the regenerator packing process to avoid folding these curled edges over on themselves.

are fully seated in and pressing against the wall of the flanged center vessel. With the ambient exchanger pressed into place, a 2-251 Buna-N O-ring is placed into the O-ring groove on the flange and a 5.3 mm thick, anodized aluminum retaining plate is placed on top of the flange. Four 10-32, 18-8 stainless steel hex drive flat head screws are then used to attach the retaining plate to the flange. The latex diaphragm assembly is attached to the bottom of the collimator with four 4-40, 18-8 stainless steel hex drive rounded head screws. The diaphragm-collimator assembly is then inserted into the threaded center portion of the ambient heat exchanger. Figure 3.28 shows the flanged center vessel having the regenerator screens, thermal buffer tube, flow straighteners, ambient heat exchanger, the collimator-diaphragm assembly and the retaining plate installed. Once the diaphragm-collimator assembly is tightened into place, the flanged center vessel is carefully flipped over and the entire process is repeated in the other half.

In parallel to the packing of the flanged center vessel, the TaSEG’s two alter-
Figure 3.28. The ambient heat exchanger is hydraulically pressed into the flanged center body that has already been packed with the regenerator screens and thermal buffer tube (left), the fully packed center vessel (right) with the ambient heat exchanger, retaining plate and collimator-diaphragm assembly installed.

Alternator assemblies are constructed. First, a 1.57 mm piece of Rulon J\(^{16}\), a blended PTFE material, is epoxied to the outside diameter of the anti-drift pistons. The stock alternator pistons are then removed from the off-the-shelf 1S132M/A alternators and the anti-drift piston is attached to it. The alternators stool cage\(^{17}\) and the outside diameter of the Rulon J covered anti-drift piston are then turned in a lathe to allow them to mate with the inertance-compliance flange. This turning process is a critical step as it establishes the alternator’s seal gap, which will be discussed in more detail in Chapter 5. The turned motor assembly is then installed in the inertance flange\(^{18}\) and held in place with four 10-32, alloy steel socket head screws. The alternator housing is then installed over top of the alternator, establishing the back volume, and bolted to the inertance-compliance flange with four \(\frac{3}{4}\)-10, high-

\(^{17}\)Refer to Figure 3.23.
\(^{18}\)Refer to Figure 3.14.
The TaSEG’s alternator assemblies. The alternator housing having the displacement sensor adapter is placed over on top of the inertance adapter, covering the alternator (left). The alternator housing having the displacement sensor adapter is placed behind the inertance adapter, showing the attached alternator (right).

strength, Grade 8 Steel hex head bolts and four $\frac{3}{4}$-10, high-strength, Grade 8 Steel hex nuts. The displacement sensor adapter is then attached to the alternator housing via eight $\frac{1}{4}$-20, alloy steel socket head screws. A $\mu\epsilon$ brand displacement sensor, which measure the piston motion, is then threaded into the adapter. Figure 3.29 show the assembled alternator assemblies minus the displacement sensors.

Each alternator assembly is then attached to the completely packed flanged center vessel via eight $\frac{3}{8}$-16, Grade 8 steel screws tightened into threaded holes in the inertance flange. After the specified torque for all of the screws, bolts and nuts has been double checked, the experimental TaSEG prototype is ready for testing.

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19 Refer to Figure 3.14.
Figure 3.30. Experimental TaSEG (with dimension shown for scale). The green helium fill valves (A) are visible along with the brass hose barbs (B) that supply cooling water to the ambient heat exchangers. The water tube (C) connecting the ambient heat exchangers in series is visible in the middle of the background.

Figure 3.30 shows the fully assembly TaSEG Prototype. It should be noted that the assembly process is exactly the same when the 316 stainless steel random fiber regenerator discs are installed in the TaSEG.
Chapter 4

TaSEG Experimental Set up

4.1 Introduction

This chapter outlines the equipment required to measure and record the TaSEG performance during experimental testing. This includes static and dynamic pressure amplitudes and phases, alternator displacements, temperatures at various locations, input and output power, and so on. It also outlines the procedure used to calibrate the pressure transducers and displacement sensors used to measure the performance of the TaSEG.

4.2 Experimental Equipment

4.2.1 Pressure Sensors and Calibration

Several different Endevco\(^1\) piezoresistive pressure transducers models were used to measure the static and dynamic pressures at various locations throughout the TaSEG. This included two 8530B-500 sensors, one 8530B-1000 sensor, one 8510B-500 sensor and one 8530BM37-500-1 sensor. These transducers incorporate a piezoresistive bridge doped on a silicon membrane that deflects in response to pressure changes. As the membrane flexes, a strain is produced, which in turn changes the resistance of the bridged resistors. These transducers feature excellent linearity, have a wideband frequency response and provide stable performance.

\(^{1}\)Endevco Corporation, www.endevco.com
Table 4.1. Endevco Pressure Sensor Sensitivities

<table>
<thead>
<tr>
<th>Serial Number</th>
<th>Model</th>
<th>Sensitivity [mV/psi]</th>
<th>Uncertainty [mV/psi]</th>
</tr>
</thead>
<tbody>
<tr>
<td>32519</td>
<td>8530B-500</td>
<td>0.624</td>
<td>0.0015</td>
</tr>
<tr>
<td>35030</td>
<td>8530BM37-500-1</td>
<td>0.740</td>
<td>0.0015</td>
</tr>
<tr>
<td>32959</td>
<td>8530B-1000</td>
<td>0.249</td>
<td>0.0023</td>
</tr>
<tr>
<td>30103</td>
<td>8510B-500</td>
<td>0.486</td>
<td>0.0020</td>
</tr>
<tr>
<td>23514</td>
<td>8530B-500</td>
<td>0.662</td>
<td>0.0008</td>
</tr>
</tbody>
</table>

over the wide temperature range of -54°C to +121°C. The sensors require a 10 V supply voltage that is provided by a Chart 2S000K-19043-A Pressure Signal Processor\(^2\). This processor is a four-channel signal conditioner specifically designed for use with Endevco pressure transducers. In addition to the sensor supply voltage, the processor has BNC outputs that provide measurable sensor output voltages. These output voltages consist of an AC voltage that is proportional to the oscillating pressure amplitude and a DC output voltage that is proportional to the mean pressure applied to the sensors.

Each sensor is supplied with a calibration certificate that gives the sensor sensitivity to two significant figures (e.g. two digits right of the decimal point) when expressed in mV/psi. This sensitivity and the sensor’s initial DC voltage offset can be affected by the torque applied to the sensor during installation. Therefore, an *in situ* calibration is carried out on the sensors. In order to carry out this calibration, the pressure sensors are installed in the TaSEG prototype, which is in turn connected to a high pressure helium bottle. An ultra-high-accuracy analog temperature-compensated 0-400 psig Ashcroft\(^3\) pressure gauge is connected inline with the helium bottle. The gauge is capable of measuring 1 psi increments and is calibrated by the manufacturer in accordance with ASME B40.1 Grade 3A, resulting in an accuracy of ±0.25% over the entire gauge scale. A Fluke 287 multimeter is connected to the BNC outputs of the Endevco pressure signal processor and used to measure the sensor DC output voltage. The TaSEG is then incrementally pressurized via the helium bottle. At each pressure step the DC output voltages of each sensor and the Ashcroft pressure are recorded.

Figure 4.1 shows the measured Endevco output voltages and mean gauge pres-

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\(^2\)See Appendix A for more details on this device

\(^3\)http://www.ashcroft.com/
Figure 4.1. Endevco Pressure Transducer \textit{in situ} calibration.

The slope of each line divided by the 10x gain of the processor provides the \textit{in situ} sensitivity of each pressure sensor. Table 4.1 provides a summary of the \textit{in situ} pressure sensor sensitivities and offsets used during the TaSEG experiments. The relative uncertainty of each calibration constant is found using a formula from Higbie.\cite{85} It should be noted that all of the pressure sensors used during TaSEG testing are basically the same thing. The differing model numbers represent optional features. For example, the 8530BM37-500-1 sensor is a 0-500 psi sensor that has a detachable cable, hence the slightly different model number.

### 4.2.2 Displacement Sensors and Calibration

Two $\mu$e model VIP-50-GA-SA7-1 sensors powered by a BK Precision\footnote{http://www.bkprecision.com} 1902, 15-amp switching DC power supply were used in the experiments to measure the
displacement of the linear alternator pistons. These sensors consist of an aluminum measuring ring and a rod-shaped sensor housing. In a non-contacting inductive process, the measuring system interprets the physical relationship between the sensor housing and the measuring ring. The ring is attached to the alternator’s axis screw in the back volume\(^5\), while the sensing rod is statically attached to the alternator housings via an adapter piece. This can be seen in Figure 4.2.

Inside the sensor rod there is a segmented coil that is supplied with an alternating current. The alternating electromagnetic field caused by the alternating current induces eddy currents in the aluminum ring, which in turn influences the coil sections inside the sensing rod. As the measuring ring moves concentrically along the sensing rod, the voltage drops of the individual segments within the rod are picked up and added in an internal amplifier. This movement produces a mA current output that is proportional to the position of the measuring ring.[92]

As the sensor output range is 4 to 20 mA, a 470 Ω potentiometer is wired across the output as is illustrated in Figure 4.3. The potentiometer\(^6\) is adjusted

\(^5\)Refer to Figure 3.14.

\(^6\)Precision resistors are preferred as potentiometers can change their resistance as the wiper or resistance element oxidizes or moves slightly. However, precision resistors were not available when the sensors was implemented in the experimental setup.
Figure 4.3. VIP-50-GA-SA7-1 displacement sensor output signal $V_{out}$, that is monitored with a voltmeter/LabVIEW DAQ and load resistor/potentiometer. Further details related to the wiring of the sensors can be found in these sensors instruction manual.

to provide the data acquisition system with a 1 to 5 $V_{DC}$ output that corresponds to the displacement of the alternator’s piston. The sensors are shipped from the supplier with a mA/mm calibration certificate, but the data acquisition system used can only accept 0 to 10 $V_{DC}$ voltage signals. Thus, the unknown mV/mm calibration constant for each sensor must be found. In order to find the calibration constant, a DC voltage was applied to the linear alternator. This causes the alternator’s piston to move from a fixed distance from equilibrium. The larger the applied DC voltage, the farther from equilibrium the piston moves. By measuring the DC voltage output of the sensor and the displacement from equilibrium of the alternator piston using a Starrett Model 799A-6/150 electronic caliper\(^7\) attached to the piston, over a range of applied DC voltages, it is possible to find the sensor’s mV/mm calibration constant.

From Figure 4.4 it is clear that the calibration constants are $20.94 \pm 0.010$ mm/mV and $18.43 \pm 0.016$ mm/mV for sensor SN 2063 and SN 2064, respectively. The relative uncertainty of each calibration constant was found using a formula from Higbie.\(^8\)

\(^7\)http://www.starrett.com/metrology/metrology-products/precision-measuring-tools
Figure 4.4. The slopes of the lines above establish the mm/mV calibration constant for each displacement sensor.

4.2.3 Water Flow Measurements

In order for the TaSEG to operate, a temperature profile across the regenerator must be maintained after the thermodynamic cycle has started. To maintain the temperature at the ambient heat exchanger water is pumped through it. The flow of this water is controlled by a ball valve placed in series with a gate valve on the lab wall. The inlet and outlet water temperatures are measured with standard KTSS-316G-12 chromega-alomega OMEGA Engineering\(^8\) type K thermocouples. The flow rate was measured using a stop watch and a 1 liter graduated cylinder. A flow rate of 4.8 to 5.1 liter/min was established for all experiments and was more than ample to maintain the ambient heat exchanger to between approximately 17\(^\circ\) and 24\(^\circ\)C.

\(^8\)http://www.omega.com/
4.2.4 Thermocouples

In addition to the aforementioned OMEGA Engineering KTSS-316G-12 type K thermocouples, several additional commercially available type K thermocouple types are used during the experimental investigation of the TaSEG.

The TaSEG was designed with the intention to combine it with a custom gas burner appropriate for a µCHP appliance. As such, the ability to insert thermocouples directly into the regenerator through the pressure vessel wall was not an option. Because the regenerator temperature profile is very sensitive to Gedeon streaming[26, 29], eleven 5TC-GG-K-20-56 chormega-alomega OMEGA type K thermocouples (shown in Figure 4.5) are welded using a capacitive discharge welder to the outside wall of the regenerator housing. Five thermocouples span the regenerator of side A and six span the regenerator of side B of the TaSEG. OMEGA Engineering specifies that both the KTSS-316G-12 and 5TC-GG-K-20-56 thermocouples have an accuracy of ±1.1 °C. Because the regenerator elements are sized to have a slight interference fit with the regenerator housing, it is assumed that when the regenerator wall is well insulated these thermocouples approximate the axial temperature profile of the regenerator. A nearly linear profile across the regenerator means that Gedeon acoustic streaming has been prevented within the TaSEG. This temperature profile will be used to confirm Gedeon streaming suppression with the latex barrier in place. These profiles can also be used in future work to develop⁹, design, implement and test a static jet plate suppression device.[93]

Additionally, two Thermocoax TKI 10/50 type K thermocouples are brazed onto the outside wall of the regenerator housing/flanged center vessel and used to measure the local temperature of the heating elements used to heat each stage/side of the TaSEG. Thermocoax specifies that these thermocouples have an accuracy of ±1.5 °C.

Two eight channel Measurement Computing¹⁰ USB-TC data acquisition devices are used to measure the thermocouple voltage outputs and convert them into a °C value. These are interfaced with a custom LabVIEW virtual instrument (VI) that records the temperature values during TaSEG operation. This system (VI plus

⁹Initially, the development of a static streaming suppression device was intended to be part of this work. However, time constraints prevented it from being completed.

¹⁰https://www.mccdaq.com/
Figure 4.5. Eleven 5TC-GG-K-20-56 type K thermocouples are welded to the outside regenerator wall of the TaSEG.

USB-TC DAQ) was set up to measure and log the thermocouple temperatures at intervals of 1 ms per channel (i.e. when all eight channels are used, the VI reads and writes the temperatures from all 8 channels in 8 ms). This sampling rate minimizes errors that could arise due to the time delay between temperature readings.

4.2.5 TaSEG Heat Source

The heat flux required to operate the TaSEG is provided by two custom, mineral-insulated Thermocoax ZEZ Ac 20/1-4.3-1(m) resistive electrical heaters\textsuperscript{11}. These heaters are wrapped around the outside diameter of the TaSEG pressure vessel in the vicinity of the hot heat exchangers and brazed in place.

Figure 4.6 has a CAD drawing showing the placement of the brazed-on Thermocoax resistance heaters, a sectioned TaSEG thermal core view showing the location of the hot heat exchangers, and a photo of the constructed TaSEG thermal core.

\textsuperscript{11}http://www.thermocoax.com/
Figure 4.6. A CAD drawing (top left) showing the placement of the brazed on Thermocoax resistance heaters, sectioned TaSEG thermal core view (top right) showing the location of the hot heat exchangers and photo of the constructed TaSEG thermal core with the resistance heaters brazed on (bottom). These heaters are used to provide the thermal input to the TaSEG during experiments.

with the brazed on resistance heaters. Each (of the two) TaSEG engines makes use of one heater element each to provide the thermal input to the TaSEG during experiments. These heater elements are wired in series with one another and powered by a Belhman BL4500 programmable AC power supply\textsuperscript{12}, pictured in the bottom right of Figure 4.9. By controlling the voltage supplied to the heating coils, the external skin temperatures on the TaSEG’s hot heat exchangers can be set. One of the nice features of this power supply is that it displays its electrical

\textsuperscript{12}http://www.belhmanpower.com/blhp.htm
output voltage, current and frequency.

4.2.6 Data Acquisition System

In order to interface and record all the pressure, displacement and temperature data simultaneously during testing of the TaSEG, a DELL Inspiron All-in-One Intel Core i3-2130 PC running Windows 7 is used in conjunction with LabVIEW 2014\textsuperscript{13} and two National Instruments USB-6001 Multifunction I/O devices\textsuperscript{14}. This system allows users to create and run customized “virtual instruments” (VI’s) generated with LabVIEW. Each VI has two main components: a block diagram and a front panel. The front panel is built using controls and indicators. Controls are user inputs that allow a user to supply information to the VI. Indicators are outputs that indicate, or display, the measured results based on the inputs given to the VI. The back panel is a block diagram that contains the graphical source code. All of the objects placed on the front panel appear on the back panel as terminals. The back panel also contains structures and functions that perform operations on controls and supply data to indicators. Controls, indicators, structures, and functions are connected to one another using wires, e.g., two controls and an indicator can be wired to the addition function so that the indicator displays the sum of the two controls, et cetera. Thus, a virtual instrument can be run as a program, with the front panel serving as a user interface. Multiple inputs and outputs (analog, digital) allow the acquisition of data from all different kinds of sensors (e.g. temperatures, displacements, pressure, etc). Figure 4.7 and 4.8 show the block diagrams and front panels of the two VI’s that were written to simultaneously log the static and dynamic operating pressures, alternator displacements, temperatures at various locations and the output power generated by the TaSEG during experimental testing.

\textsuperscript{13}\url{http://www.ni.com/labview/}
\textsuperscript{14}\url{http://www.ni.com/en-us/support/model.usb-6001.html}
Figure 4.7. LabVIEW virtual instrument used in conjunction with two Measurement Computing USB-TC data acquisition devices and several type K thermocouples to log various temperatures during TaSEG testing.
**Figure 4.8.** LabVIEW virtual instrument used in conjunction with two National Instruments USB-6001 data acquisition devices to log operating static and dynamic pressures, alternator displacements, and output power of the TaSEG during testing.
4.2.7 Additional Equipment

The electrical power output generated by the TaSEG’s two alternators during operation is measured with a Hioki Model 3167 power meter\textsuperscript{15} that makes use of a 9272-10 current clamp, highlighted by blue in Figure 4.9. This particular power meter has a measurement accuracy of $\pm 0.2\%$ of the full scale output.

Additional test equipment that is required for experimental operation of the TaSEG includes an Agilent 33220A function/waveform generator\textsuperscript{16}, a Stanford Research Systems SR830 lock-in amplifier\textsuperscript{17}, a Fluke 287 multimeter\textsuperscript{18} and a Behringer Europower EP4000 power amplifier\textsuperscript{19}, a four-channel Chart 2S000K-19043-A pressure signal processor, two 100 $\mu$F capacitors, and an incandescent light-bulb load bank. A complete summary of all the test equipment used during experimental testing can be found in Table 4.2. Information related to all the equipment used in the TaSEG experimental setup can be found in Appendix A.

\textsuperscript{15}http://www.hiokiusa.com/
\textsuperscript{16}http://www.keysight.com
\textsuperscript{17}http://www.thinksrs.com/products/SR810830.htm
\textsuperscript{18}http://www.fluke.com/
\textsuperscript{19}https://www.music-group.com/brand/behringer/home
### Table 4.2. TaSEG Test Equipment Summary

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### 4.2.8 Experimental Test Setup

Figure 4.9 shows the experimental setup of the dual TaSEG in an explosion-resistant test room. This room is designed to handle any type of prototype test-
Figure 4.9. TaSEG experimental setup making use of the LabVIEW data acquisition system outlined in green, the Belhman power supply providing the electrical power input to the resistance heater outlined in red and the Hioki power meter used to measure the TaSEG’s electrical output outlined in blue.

ing that involves high temperatures and high pressures, which is the case for the TaSEG. The TaSEG is placed on a flame-resistant G10 slab that prevents any thermal damage to the test bench. Additionally, a custom impact-resistant polycarbonate blast shield is placed around the TaSEG. The TaSEG is charged to approximately 40 bar of 99.999% pure helium.

The TaSEG’s alternators are wired in parallel and connected to a custom load circuit. This circuit consists of capacitors used to balance the induction of the linear alternators and a set of ten 100 W light bulbs that can be wired in any configuration with one another. An electrical wiring schematic of the TaSEG load and control circuit is shown in Figure 4.10. For all the results given in this report, the balance capacitance consists of two 100 µF capacitors wired in parallel with each other in series with the light bulb load.
4.3 A note on TaSEG Load Interaction

In the case of a TaSEG (power generator) and load (power absorber), stable operation at a fixed alternator stroke is achieved only if there exists a stable crossover point between the power produced and the power absorbed as a function of alternator stroke. In a resonant system, much like the TaSEG, power produced by the cycle is a function of the square of pressure-wave amplitude and, roughly speaking, the square of the alternator’s piston stroke. The TaSEG’s alternator, which converts that power into electricity, produces a voltage that is proportional to stroke; and power depends on voltage or stroke squared. A fixed resistive load also absorbs power proportional to the voltage squared via
Figure 4.11. TaSEG alternator output power versus stroke plots showing the interaction between the TaSEG and the load that is presented to its alternators.

\[ P = \frac{V^2}{R}, \]  

(4.1)

where \( P \) is power, \( V \) is voltage and \( R \) is resistance. Because this voltage is proportional to stroke, the absorbed power is also proportional to stroke squared, like the power being produced by the alternator. This means that the combined system (TaSEG plus load) has no single solution.

If the system is perfectly balanced (i.e. \( P_{TaSEG} = P_{Load} \) at any operating point), it will happily operate at any stroke or voltage as long as the power generated matches the power absorbed. However, if the TaSEG’s output power is slightly lower because of added friction or a cooler hot end temperature or a warmer ambient end temperature, \textit{et cetera}, then the TaSEG’s output power will always be lower than the loads absorbed power (i.e. \( P_{TaSEG} < P_{Load} \)) regardless of the alternator’s stroke and the TaSEG will stall and stop running. On the other hand,
if the TaSEG’s output is slightly higher because it is running at a higher hot end temperature or with lower friction per se, its output power will always be higher than the absorbed power (i.e. $P_{TaSEG} > P_{Load}$), and the TaSEG’s stroke will run away possibly causing damage to its alternators. These three TaSEG-load operating conditions are shown in Figure 4.11.

Yet another case can exist in which the square dependency of the load is replaced with an approximately cubic relationship so that the load curve begins lower than the TaSEG’s output at low stroke, then crosses over and becomes higher than the TaSEG’s output at high strokes. Should the TaSEG’s stroke continue to increase higher than the stability point, the excess load will reduce it. Should the TaSEG’s stroke become lower than the stability point, the insufficient load would allow it to increase back toward the stability point. This case is illustrated in the bottom, right plot of Figure 4.11.[86]

Care must be taken when using normal resistors (much like the light-bulb load bank) as they act like the $P_{TaSEG} > P_{Load}$ plot of Figure 4.11 as their resistance will increase as more current is delivered to them from the TaSEG. In its intended application the TaSEG power module of a $\mu$CHP appliance would be connected to the electrical grid, which provides a strong voltage lock, resulting in stable operation. However, care must be taken in the design of the appliance to prevent power from flowing from the grid into the TaSEG when it’s operating with a low alternator stroke. At a higher stroke the power would flow out of the TaSEG into the grid, which can absorb a large amount even if the current is higher than that of the mains. In the case of a stand-alone appliance or TaSEG testing in the lab, stable operation could be achieved with a real load used in conjunction with a parasitic load that has the ability to be switched on and off rapidly, synthesizing a linear load in effect. However, this would involve an active controller that monitors the voltage and actively modulates the total load seen by the alternators. In order to avoid the added complexity of developing a load of that type and to make use of the simplified normal resistor (light bulb) load, the thermal power being input to the TaSEG is controlled by controlling the electrical power supplied to the Thermocoax resistance heaters that establish the hot heat exchanger temperature during testing.
Experiments and Performance

5.1 Introduction

In this chapter, the experimental testing and operation of the TaSEG will be discussed. In addition to evaluating the overall performance of the TaSEG, including its acoustic and electrical power generation and its thermal-to-electric efficiency, component level testing that establishes safety and other inputs required for the DeltaEC simulation will be outlined.

This chapter describes testing done with elastomer barriers (e.g. latex diaphragms) to suppress the Gedeon streaming. The TaSEG was designed to accommodate a “jet plate” fluid-diode element as described by “the author[93]” which is a compact version of the jet-pump developed by Backhaus and Swift.[16] The end goal is to optimize the jet plate as part of the final TaSEG design. However, it is possible to insert a diaphragm into the assembly where the jet-plate would go. The diaphragm has limited displacement in this configuration before physical contact with neighboring components, so the TaSEG can only be run at relatively low amplitude before the diaphragm breaks. Nonetheless, there is tremendous value in running the TaSEG with a diaphragm in place first, to baseline its performance. The jet-plate, unlike laboratory-oriented jet-pumps featured in previous work [16, 35] is not adjustable, and it is virtually certain that iteration will be required to optimize the plate and successfully suppress the streaming.[93] Furthermore, streaming suppression with a fluidic diode is notoriously amplitude-sensitive, and even an optimized jet-plate may not work particularly well at low
amplitude—yet low-amplitude data is most likely to match simulations, where non-linear effects are small. Therefore, the TaSEG is run first with a diaphragm at low amplitude, to establish a zero-streaming, linear baseline, which can be later compared to the TaSEG with the jet plate.

For similar reasons, the TaSEG is tested first with a woven-screen regenerator, rather than the lower-cost random fiber that is preferred for an industrial product. Viscous flow and heat transfer are better characterized in woven screens than in random fiber[18, 107, 108], and screens also tend to be better documented and controlled in terms of critical dimensions. Thus, woven screens make a better base lining tool for understanding the TaSEG, at least initially.

Failure of the diaphragms at modest amplitudes is not unexpected; however, it is found during initial testing that the diaphragms fail sooner than expected. The acoustics of the compact feedback network and the thermal core are not exactly as predicted, and this results in higher volume velocity at the diaphragms than what was originally anticipated. Therefore, some iteration on diaphragm design is performed, in order to obtain the desired baseline testing with first woven screens and then random fiber in the regenerator.

Additionally, this chapter begins by discussing the component level TaSEG testing that is performed, followed by discussing the challenges encountered attempting to achieve steady-state operation. The TaSEG’s steady state performance is then measured and evaluated. Once the steady state performance measurements are complete, they are compared to the theoretical performance predicted by the DeltaEC simulation. The differences between the predicted performance and the actual performance are discussed. Using both the measured performance data and the simulations, the thermal-to-electric (first law) efficiency of the TaSEG is found. The fraction of Carnot, or the second law efficiency, is calculated using the measured temperatures at the ends of the regenerator during operation together with the first law efficiency.
5.2 TaSEG Component Testing

5.2.1 Alternator Assembly Testing

The TaSEG’s 1S132M/A alternators efficiently generate useful electricity from acoustic power. There are a number of important parameters that must be measured to ensure acceptable performance for the TaSEG. There are three basic tests that are carried out.[86] These include:

1. Static testing.

2. Free decay “ringdown” testing.

3. Dynamic testing.

The static testing is made up of two sub-tests: the mechanical stiffness $K_M$ test and the transduction coefficient $Bl$ test. The mechanical stiffness test measures the alternator’s mechanical suspension plus magnetic stiffness in N/m. This is done by moving the alternator’s plunger/piston a set distance while measuring the opposing force with a load cell. The transduction coefficient test measures the product of the magnet field strength times the length of the copper coil within the field in N/A.[103] This is done by physically restraining the plunger/piston from moving by pressing it against a stationary load cell and applying incrementally increasing amounts of known DC current to the motor while measuring the opposing force.

Two additional values, the alternator’s maximum piston displacement $S_{Max}$ (in mm) and the DC electrical resistance $R_e$ (in Ω), are also defined by the static testing. The maximum displacement comes from the stiffness testing as the alternator’s plunger is axially away from equilibrium until it contacts the enclosing metal frame (called the stool\(^1\)) of the alternator that serves as a hard stop. The DC electrical resistance is a byproduct of the transduction coefficient test if the current across the coils is measured at the same time as the DC current, then applying Ohm’s Law. Strictly speaking the electrical resistance has a weak dependence on the operating frequency and should be corrected based on the measured TaSEG operating frequency.[86, 104]

\(^1\)See Figure 3.23
The “free decay” test measures the transient response (voltage versus time) of the alternator by applying an impulse of DC current to the coils, causing the piston/plunger to displace suddenly and then freely ring down under open circuit conditions. The measured results are fit to an exponentially-decaying sinusoidal wave. From this curve-fit the natural frequency of free decay $f_0$ in hertz and the alternator’s damping coefficient (also referred to as its mechanical resistance) $R_m$ in (N·s)/m are found.

The dynamic testing of the alternator is carried out as a secondary check of the alternator’s free decay frequency $f_0$ and $R_m$ to verify that the alternator’s have no manufacturing flaws that could result in failure. In this test the alternator’s are run at their natural frequency that is found by maintaining a constant applied peak-to-peak voltage at some nominal stroke and then varying the input frequency until the lowest input current is achieved. At this operating point, a resonance of sorts, the frequency, voltage, current, power and piston stroke are recorded.

In addition to these tests, the alternator’s moving mass $M$ in kilograms and the stator induction $L_s$ in mH are found by inspection of the motor components. The moving mass is found by physically weighing all the alternator’s moving parts\(^2\) on a scale, and its inductance is found by connecting LCR meter to the bare stator coils before the alternator is assembled. Figures 5.1 and 5.2 are the test reports for both alternators (serial #132427 and #132433) that show the measured mechanical resistance $R_m$, stiffness $K_M$, free decay natural frequency $f_0$, moving mass $M$, inductance $L_s$, maximum (piston) zero-to-peak displacement $S_{\text{Max}}$, DC electrical resistance $R_e$ and transduction coefficient $B_l$.

The radial clearance seal gap $\delta$ that exists between the alternator’s piston and corresponding bore must also be determined as it is a required input to the DeltaEC simulation. This gap is indirectly measured for both alternator assemblies by measuring flow through the clearance seal as a function of pressure drop. This DC flow test involves pressurizing the alternator back volume with helium to several different mean pressures while measuring the flow through the seal gap, which is open to the atmosphere and is the only exhaust path within the assembly. The test setup consists of an Endevco 8510B-500 pressure sensor communicating with the

\(^2\)Only $\frac{1}{2}$ of the alternator’s flexure mass is used as one end of each flexure is riveted to its stool cage, hence it does not move.
Figure 5.1. 1S132 alternator serial #132427 test report showing the measured mechanical resistance \( R_m \), stiffness \( K_M \), free decay natural frequency \( f_0 \), moving mass \( M \), inductance \( L_s \), maximum zero-to-peak displacement \( S_{\text{Max}} \), DC electrical resistance \( R_e \) and transduction coefficient \( Bf \).
Figure 5.2. 1S132 alternator serial #132433 test report showing the measured mechanical resistance $R_m$, stiffness $K_M$, free decay natural frequency $f_0$, moving mass $M$, inductance $L_s$, maximum zero-to-peak displacement $S_{Max}$, DC electrical resistance $R_e$ and transduction coefficient $B_l$. 
alternator back volume, an Endevco 4428A signal conditioning box, an Aalborg He-GFM37 digital flow meter calibrated to measure the mass flow of helium, an Ashcroft 302084SD02L100# digital pressure gauge and a bottle of pressurized 99.999% helium. Figure 5.3 shows the test setup used to find the DC flow versus back pressure curve for the TaSEG’s alternator assemblies, which is then used to calculate the seal gap.

Treating the radial seal gap as two parallel plates spaced uniformly apart, in which pure Poiseulle flow occurs and assuming that the mass flux through the seal is conserved, P.S. Spoor[101] has shown that the seal gap as a function of the entering pressure $P_C$ and flow rate $\dot{V}$ is given by

$$\delta = \left[ \frac{24\mu L P_{\text{atmo}}\dot{V}}{(P_C^2 - P_{\text{atmo}}^2)\Pi} \right]^\frac{1}{3}, \quad (5.1)$$
Figure 5.4. The measured helium flow in standard liters per minute (SLM) through the alternator seals versus the mean pressure into the seals for both alternators.

where $\mu$ is the viscosity of the gas used during testing (helium in this case), $L$ is the seal length, $P_{\text{atmo}}$ is atmospheric pressure, $\dot{V}$ is the measured helium flow through the seal gap, $P_C$ is the mean pressure in the back volume causing the flow, and $\Pi$ is the perimeter of the seal gap. Notice that the flow through the seals does not have the normal linear dependence on the pressure difference across the seal, since the pressure into the seals can be several times higher than the ambient pressure, which affects the gas density. If the range of pressures is small in the measurement, the data will look locally linear, yet not appear headed for an intersection with zero (zero flow at zero pressure difference), which may not be intuitive. If data are collected over a large range of pressures, the quadratic nature of the curve is evident. Figure 5.4 shows the measured helium flow through the alternator seals versus the pressure into the seals for both alternators.
Table 5.1. Serial #132427 (right), SN #132433 (left). The first two columns of each table (the absolute pressure measured in the back volume with the digital Ashcroft Gauge and the flow rate measured with the Aalborg digital flow meter) are measured values while the last two columns (the absolute pressure and the seal gap) are calculated values based on the measured data and Eq. 5.1. A mechanical stop prevented the pistons from moving when the back volume was pressurized, thus the same section of the piston bore was measured for each data set.

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</table>

Average 0.72   Average 0.57

Based on past experience a radial seal gap of 0.025 mm (1 mil) or less is desirable to balance the flow losses with the other losses in the system. Smaller seal gaps are of course better, but they are challenging to manufacture. A much larger seal gap will result in a system dominated by seal loss. Table 5.1 shows the measured helium flow and back volume pressure, and gives the calculated seal gap based on Eq. 5.1.

As the table indicates, average seal gaps of 0.018 mm (0.72 mil) and 0.015 mm (0.57 mil) were measured for the TaSEG’s two alternators. The average value of these two measurements, 0.017 mm (0.65 mil) is implemented in the DeltaEC and SAGE simulations by approximating the seal as a single-passage, parallel plate

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3Stemming from my time working at Chart Inc. - Qdrive.
heat exchanger. In this way, the losses associated with the flow through the seals are accounted for in the performance predicted by the DeltaEC simulation.

### 5.2.2 Hydrostatic Pressure Vessel Testing

![Image of pressure vessel prototype under hydrostatic test]

**Figure 5.5.** The TaSEG pressure vessel prototype under hydrostatic test. A hydrostatic test is a way in which pressure vessels such as boilers, gas cylinders, etc. are tested for strength and leaks. The hydrostatic test involves filling the TaSEG pressure vessel prototype with a dyed water, submerging it in water, then pressurizing it to specific pressure values and letting it sit at that pressure for a duration of time. If no leaks or failures occur the vessel is deemed safe for operation.

Before any pressurized testing of the TaSEG can be performed, the TaSEG’s pressure vessel must be evaluated to verify its ability to safely handle the intending working pressure it will be subjected to during experimental testing. A hydrostatic test is carried out in accordance with the ASME boiler and pressure vessel code guidelines set forth in Section VIII, Div. 1, Part UG-99.\[71\] A hydrostatic test is a way in which pressure vessels such as boilers, gas cylinders, etc. are tested for strength and leaks. The chosen test is a hydrostatic one that involves filling an...
assembled TaSEG pressure vessel prototype with a dyed water, to help in visually detecting leaks, submerging it in clear water, then pressurizing it to a specific pressure value and letting it sit for a duration of time to prove that the vessel can withstand the forces. Figure 5.5 shows the hydrostatic test set up and the TaSEG prototype under pressure.

The maximum internal working pressure that the TaSEG vessel will need to handle during operation is 44 bar. Based on this and the ASME requirements, it was decided to hydrostatically test the pressure vessel up to a mean pressure of 66 bar, 1.5x the maximum working pressure. Dyed water was placed inside the vessel, completely filling the inside. Care was taken to assure that all air pockets were removed from the vessel prior to testing. The vessel was then submerged in a 50 gallon tank of clean, clear water and plumbed up to a pressurized helium tank. The vessel pressure, fed from a helium tank, was first increased to 27.6 bar and allowed to sit for 1 hour. The vessel welds and O-ring joints were inspected and no leaks were found. Thus, the vessel pressure was increased to 44 bar and allowed to sit for another hour. Again, the vessel welds and O-ring joints were inspected and no leaks were found. Finally, the vessel pressure was increased to 66 bar and allowed to sit for another hour. No leakage or failure was recorded during this test. Hence, the vessel was deemed safe for experimental testing.  

5.2.3 Heat Leak Estimate

By making use of careful transient measurements during the heat-up and cooldown of the TaSEG, it is possible to obtain a reasonably accurate estimate of the heat leak between the heater and the ambient environment, including the ambient heat exchangers, experienced during testing without trying to calculate it from first principles, or even knowing the exact thermal mass of the heater. This loss includes conduction loss as well as heat leak to the room and is subtracted from the overall thermal input to the TaSEG in order to calculate its thermal-to-electric

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4This statement is based on the vessel passing the hydrostatic test and Chart's past experience working with similar vessel loading. It is understood that when the vessel is heated during experimental testing there will exist a differential expansion stress that the hydrostatic test doesn't check. Therefore, extra safety measures are taken during testing. This vessel does not have a Boiler and Pressure Vessel Code ASME stamp, but it was evaluated per the Section VIII, Div. 1 based on its intended experimental operating conditions.
efficiency $\eta_{Th-Et}$.

The resistive heaters are thoroughly insulated and set up to mimic steady state operating conditions achieved during testing of TaSEG, but in this test the TaSEG is prevented from running by keeping the leads of its alternators shorted together. Temperature measurements of the hot heat exchanger skin and regenerator housing at several locations are recorded as a function of time. The heater power is then shut off and the drop in temperature as a function of time is recorded. The resulting data points are plotted and linearly curve-fit over a local temperature range of interest based on the TaSEG’s steady state operating point of interest. This is done in order to determine an approximate rate of change in the temperature as a function of time, $\delta T/\delta t$. This test is carried out for each steady state operating condition. Figure 5.6 shows one such result.

\[ y = 0.3786x + 45.206 \quad R^2 = 0.9978 \]

\[ y = -0.4547x + 1286.4 \quad R^2 = 0.9982 \]

**Figure 5.6.** Heat leak measurements from resistance heater on TaSEG.

In order to estimate the heat leak, it is assumed that while the TaSEG is heating up without running the hot end obeys
\[ C_p \frac{\delta T}{\delta t} = (\dot{W}_{Elec} - \dot{Q}_{Lost}), \] 

where \( C_p \) is the total heat capacity of the hot end, \( \sum \dot{c}_i \dot{m}_i \) where the \( c_i \) and \( m_i \) represent the different materials in the hot end, \( \dot{W}_{Elec} \) is the electrical power provided to the resistance heaters, and \( \dot{Q}_{Lost} \) is the estimated heat leak and with no electrical power being supplied to the resistance heaters. During cool-down this equation becomes

\[ C_p \frac{\delta T}{\delta t} = (-\dot{Q}_{Lost}). \] 

Figure 5.6 shows these curves for the TaSEG when the regenerator is packed with random fiber. The heat leak (at a hot end temperature of 550 K) is found to be approximately 300 W. So for a steady-state operating point at this temperature, when the total input to the resistive heater is 1120 W, the heat actually reaching the hot end of the engine is about 820 W. It should be noted that \( \dot{Q}_{Lost} \) is expected to be proportional to the temperature difference between the hot and ambient ends of the regenerator, hence the cool-down is modeled as a relaxing exponential.

### 5.3 Experimental TaSEG Testing

Under normal circumstances the TaSEG will not spontaneously go into onset, due to the load presented to the engines by the linear alternators. Therefore, in order to start the TaSEG the alternator leads were connected to a “buzz” circuit, as shown previously in Figure 4.10. This technique for starting the TaSEG is similar to that used to start the free piston Stirling engines found in commercially available \( \mu \)CHP appliances. This circuit consists of an Agilent 33220A waveform generator that provided a sinusoidal input to a standard EuroPower EP4000 performance amplifier. The output of the amplifier was used to breifly “buzz” the TaSEG alternators. This buzz was at low voltage and had a frequency that was near the 120 Hz natural TaSEG operating frequency. The short “buzz” signal was accomplished by installing a push button contact switch between the output of the amplifier and one lead going to the TaSEG’s alternators. By depressing and then releasing the switch, the AC signal coming out of the performance amplifier would...
be supplied briefly to the alternators. The buzz was not provided to the alternators until the desired temperature gradient was established across the regenerator. The buzzing then continued every few seconds, as the heater elements and hot heat exchanger temperatures rose, until onset was achieved. Once the TaSEG started the temperature of the hot heat exchanger would drop briefly before stabilizing and the electrical input to the heaters would have to be increased to maintain operation.

Initial testing was carried out with the TaSEG having the 316 stainless steel, 240 x 240 woven wire mesh screen regenerator and set up in the experimental configuration shown in Figure 4.9. The very first time the TaSEG was setup to run experimentally, it achieved onset and sustained operation running at low amplitudes ($p_1/P_m < 1\%$). At this operating point the TaSEG was able to achieve steady state operation, outputting electricity (5W), with a hot heat exchanger skin temperature as low as 230°C, corresponding to a temperature difference across the regenerator of 213°C. This is a very low operational temperature gradient for a thermoacoustic-Stirling engine generator of this type and is one accomplishment for this work.[16, 61, 106]

Table 5.2. Inputs of TaSEG with a woven screen regenerator

<table>
<thead>
<tr>
<th>$\frac{p_1}{P_m}$</th>
<th>$P_m$ [MPa]</th>
<th>$V_{in}$ [V]</th>
<th>$I_{in}$ [A]</th>
<th>$W_{Elec,in}$ [W]</th>
<th>$\text{H}_2\text{O}$ Flow [liters/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.3</td>
<td>3.96</td>
<td>45</td>
<td>10.1</td>
<td>454.5</td>
<td>16.1</td>
</tr>
<tr>
<td>3.9</td>
<td>3.98</td>
<td>56</td>
<td>12.3</td>
<td>688.8</td>
<td>14.5</td>
</tr>
</tbody>
</table>

Table 5.3. Outputs of TaSEG with a woven screen regenerator

<table>
<thead>
<tr>
<th>$\frac{p_1}{P_m}$</th>
<th>$\Delta T$ [°C]</th>
<th>Freq. [Hz]</th>
<th>$V_{out}$ [V]</th>
<th>$I_{out}$ [A]</th>
<th>$W_{Elec, out}$ [W]</th>
<th>$\eta_{\text{Th}−\text{El}}$ [%]</th>
<th>$\eta_C$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.3</td>
<td>241.9</td>
<td>120.6</td>
<td>37</td>
<td>0.21</td>
<td>10.1</td>
<td>0.0222</td>
<td>4.9</td>
</tr>
<tr>
<td>3.9</td>
<td>455.8</td>
<td>119.9</td>
<td>62.1</td>
<td>1.41</td>
<td>80.0</td>
<td>.1161</td>
<td>19.0</td>
</tr>
</tbody>
</table>

Numerous test runs were carried out on the TaSEG at different operating conditions to investigate its performance. Tables 5.2 and 5.3 show the inputs and outputs, respectively, for two runs of the TaSEG with a woven screen regenerator, at two different acoustic amplitudes ($p_1/P_m = 2.3\%$ and 3.9\%). The overall efficiency $\eta_{\text{Th}−\text{El}}$ is given by $W_{Elec, out}/W_{Elec, in}$. 
Figure 5.7. TaSEG 80 watt electrical output powering light bulb load.

Figure 5.7 shows the TaSEG running at steady state, providing the necessary electricity to power the light bulb load on the left. This figure corresponds to the $p_1/P_m = 3.9\%$ measured results presented in Tables 5.2 and 5.3.

Figure 5.8 shows the temperature gradient across the regenerator for the low amplitude $p_1/P_m = 2.3\%$ test shown in the first row of data in Tables 5.2 and 5.3. This temperature profile can be used to evaluate supression of Gedeon streaming, a steady mass flow along the regenerator, which can show itself as a nonlinear temperature profile along the regenerator.[18, 93] The $R^2$ linear fit to each of these profiles suggests that the latex diaphragms, physical barriers that oscillates with the acoustic wave and restricts DC flows from occurring, are adequately preventing Gedeon streaming from occurring in the TaSEG. The latex diaphragms held up nicely during low amplitude testing, allowing limitless, steady state operation of the TaSEG.

However, the compactness of the collimator-diaphragm assembly, which was only supposed to be used for initial baseline testing, proved problematic. There was insufficient space for the diaphragms to accommodate the necessary swept volume at higher acoustic amplitudes, resulting in repeated contact of the diaphragm with the collimator and eventual failure after a few minutes of running at $p_1/P_m=3.9\%$. 
Figure 5.8. TaSEG’s regenerator profile during steady state operation at $p_1/P_m = 2.3\%$.

Figure 5.9 shows the pressure wave amplitude versus time for the $p_1/P_m = 3.9\%$ run corresponding to the second row of data in Tables 5.2 and 5.3. The portion of the figure in orange corresponds to steady state operation. The two clear, sudden decreases in the pressure wave amplitude are attributed to the failure of the two diaphragms within the TaSEG. A decrease in the TaSEG electrical output and a sudden drop in the hot heat exchanger temperatures were also observed at these instances. Left to run, the hot heat exchanger temperature continued to drop until the TaSEG stopped operating. The failure of the diaphragms was confirmed by disassembling the TaSEG and inspecting the latex diaphragms. Figure 5.10 shows one of the two diaphragms *in situ* after the TaSEG alternator side B was disassembled.
Figure 5.9. Pressure amplitude versus time for the $p_1/P_m = 3.9\%$ TaSEG run. The portion of the figure in orange corresponds to steady state operation.

Figure 5.10. Latex diaphragm failure after $p_1/P_m = 3.9\%$ testing (top), broken diaphragms removed from each half of the TaSEG (bottom). The collimator’s through hole pattern is visible in the diaphragm on the left (side A) as the diaphragm was continually pressed against them during operation.
In an effort to find a longer-term solution without redesigning the insides of the TaSEG, several different diaphragm configurations were constructed and tested. Figure 5.11 shows several of the different diaphragm configurations that were produced and tested. These alternative configurations included latex diaphragms of varying thickness (0.10 mm, 0.15 mm, and a 0.25 mm thick diaphragms were built and tested), diaphragms constructed with various levels of tension (i.e. pulled taught/pre-tensioned and left limp/no tension), two diaphragms that were produced from the sealed ends of pre-shaped latex tubes (condoms) and two pre-shaped Bellofram elastomeric rolling diaphragms.

Testing each diaphragm configuration was a time intensive process as it involved constructing the new diaphragms, disassembling the TaSEG, installing the new diaphragms, reassembling the TaSEG, then preparing it for experimental testing. Despite the numerous diaphragms that were tested, none were able to achieve high amplitude runs longer than 5 minutes. All of the diaphragms that were made of latex, regardless of the level of the tension place on them, their shape or thickness, failed within the first two minutes of high amplitude testing. The Bellofram rolling elastomeric diaphragms were able to run at elevated hot heat exchanger temperatures, but the rigidity of their shaped construction provided a distinct maximum swept volume, which in turn limited the stroke of the alternator and the achievable pressure wave amplitude.

The original goal of the experimental work was to develop and use static jet plates to suppress Gedeon streaming, similar to those described by “the author[93]” which is a compact version of the jet-pump developed by Backhaus and Swift.[16]. The collimator-latex diaphragm assemblies and the woven screen regenerator were intended to be tools used to establish and optimize the baseline performance of the TaSEG. This is done as these options are easiest and cleanest to implement. Diaphragm failure, while unfortunate, was not a surprise as their inclusion in the TaSEG design was only to establish baseline performance data that could then be used in the design of jet plates. Unfortunately, diaphragm failure occurred at a much lower amplitude than originally anticipated. Rebuilding the diaphragm numerous times was done in order to get the best baseline performance data possible, but since it wasn’t the intended focus of the work, continued experiments of the

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5Marsh BelloFram Group of Companies, https://www.marshbellofram.com/
Figure 5.11. Alternative diaphragm configurations produced and tested. No tension, 0.254 mm thick latex diaphragms (top), *in situ* pre-shaped latex tube ends (middle) and Bellofram elastomeric diaphragms (bottom).

TaSEG with woven screen regenerators and rebuilt diaphragms was stopped and the testing focus shifted to the random fiber regenerator with the diaphragm.

Despite the diaphragm failure set back, the decision was made to proceed according to plan and investigate the low amplitude performance of the TaSEG with a diaphragm and a regenerator made of a stack of the 316 stainless steel random fiber discs with a specified porosity of 84% and a fiber diameter of 40µm. In order to carry out this testing the TaSEG was disassembled, then repacked with the random-fiber, punched regenerator discs. Because the failure of the latex diaphragms in previous testing appeared to be mechanical in nature, it was decided to construct and implement even more robust 0.51 mm thick latex diaphragms.
Table 5.4. Inputs of TaSEG with a random fiber regenerator

<table>
<thead>
<tr>
<th>$\frac{p_1}{P_m}$</th>
<th>$P_m$</th>
<th>$V_{in}$</th>
<th>$I_{in}$</th>
<th>$W_{Elec,in}$</th>
<th>$\text{H}_2\text{O Flow}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>[%]</td>
<td>[bara]</td>
<td>[V]</td>
<td>[A]</td>
<td>[W]</td>
<td>[liters min]</td>
</tr>
<tr>
<td>2.3</td>
<td>38.8</td>
<td>135</td>
<td>7.7</td>
<td>1039.5</td>
<td>4.8</td>
</tr>
<tr>
<td>3.3</td>
<td>39.4</td>
<td>195</td>
<td>11.0</td>
<td>2145.0</td>
<td>5.2</td>
</tr>
</tbody>
</table>

Table 5.5. Outputs of TaSEG with a random fiber regenerator

<table>
<thead>
<tr>
<th>$\frac{p_1}{P_m}$</th>
<th>$\Delta T$</th>
<th>Freq.</th>
<th>$V_{out}$</th>
<th>$I_{out}$</th>
<th>$W_{Elec,out}$</th>
<th>$\eta_{Th-El}$</th>
<th>$\eta_C$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>[%]</td>
<td>[°C]</td>
<td>[Hz]</td>
<td>[V]</td>
<td>[A]</td>
<td>[W]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.3</td>
<td>326.2</td>
<td>119.5</td>
<td>53.1</td>
<td>2.34</td>
<td>65.0</td>
<td>0.1050</td>
<td>20</td>
</tr>
<tr>
<td>3.3</td>
<td>456.3</td>
<td>122.1</td>
<td>85.1</td>
<td>3.62</td>
<td>132.2</td>
<td>0.0832</td>
<td>14</td>
</tr>
</tbody>
</table>

These diaphragms were the thickest diaphragms constructed and implemented in the TaSEG. While increasing the thickness undoubtedly increases the stress the diaphragm experiences during operation (flexing), it is believed that these will be more accepting of mechanical damage stemming from contacting the collimator during operation. This will afford the best chance of collecting good steady-state baseline data, which can be used to guide the refinement of the jet plate for long-term streaming suppression.

Tables 5.4 and 5.5 show the inputs and outputs, respectively, for two runs of the TaSEG with a random fiber regenerator, at two different acoustic amplitudes ($p_1/P_m = 2.3\%$ and $3.3\%$).

The thicker diaphragms allowed the TaSEG to achieve steady state operation, running for over a half hour during each run, but ultimately still failed. Figure 5.12 shows the 0.51 mm thick diagrams removed from the TaSEG after failure, the tears in each are circled in red. Their failure is no doubt caused by the contact made with the collimator during operation. However, this is a vast improvement, and calculations suggest that despite the thicker material, the diaphragms are still acoustically transparent in this environment. Thus it is possible to collect steady-state baseline data, which can be used to guide and refine the intended long-term streaming-suppression method (the jet plate).

At its most powerful steady state operating point to date, the TaSEG with the random fiber regenerator produced 132 watts of electrical power at a thermal-to-electric efficiency of 0.0832, corresponding to 14% of the Carnot efficiency. At its
Figure 5.12. The 0.51 mm thick diaphragms removed from the TaSEG after failure, the tears in each are circled in red. The pattern of the collimator’s thru holes is visible on each diaphragm. Their failure is no doubt caused by the contact made with the collimator during operation. This is one facet of the TaSEG design that should be revisited in future work.

most efficient steady state operating point to date, the TaSEG having the random fiber regenerator generated 65 watts of electrical power at a thermal-to-electric efficiency of 0.1050, corresponding to 20% of the Carnot efficiency. While the intended scope of work deviated from that which was originally planned (i.e. no jet plate was developed to statically suppress Gedeon streaming), achieving these steady state operating conditions with a diaphragm is another accomplishment of this work. Figure 5.13 shows the TaSEG powering the light bulb load with its 132.4 watt electrical output visible on the Hioki power meter.
5.3.1 DeltaEC Simulation Versus Experiment

In the interest of gaining a better understanding of the TaSEG’s performance, a thorough investigation of the DeltaEC simulation is undertaken to determine possible sources of inefficiency and guide possible improvements in its design. Several adjustments are first made to the DeltaEC simulation based on the outcome of the experimental testing carried out and conversations with several experts in the field.[86, 102] Measurement data suggest that the alternator electrical resistance will increase by approximately 25% as the operating frequency of the alternator increases from the nominal operating frequency of 60 Hz to the measured operating frequency of 122 Hz. There will also be a slight, but negligible decrease in the alternator mechanical resistance. Based on this, the alternator electrical resistance is changed from 2.1 Ω to 2.6 Ω and the mechanical resistance is decreased from 8.5 (N·s)/m to 8.25 (N·s)/m within the DeltaEC model. Additionally, the alternator’s transduction coefficient $B_l$ is decreased from 47.5 N/A to 38.0 N/A and the radial clearance seal gap is adjusted to match the experimentally measured
average gap, 0.017 mm (0.65 mil). Once these parameters have been updated, the internal geometry of the TaSEG is painstakingly checked against the SOLIDWORKS model that accurately reflects the dual 132 TaSEG. Once the geometry is updated, measured experimental parameters, including the pressure amplitudes in the compression space $p_{1,c}$ and back volume $p_{1,b}$, and the ambient heat exchanger temperature, are used as “targets” within the DeltaEC simulation to evaluate the agreement between the experiment and simulation.

Tables 5.6 and 5.7 show the agreement between the experimentally measured values of the TaSEG with the random fiber regenerator for the steady state operating having $p_1/P_m = 3.3\%$. The bold values are forced into agreement. The agreement is excellent for everything except the hot heat flux, and the thermal-to-electric efficiency, and the heat exchanger temperature. One possibility is that there is Rayleigh streaming occurring in the thermal buffer tubes. This streaming would both reduce the heat available to the cycle, and cool the HHX temperature below its predicted value. However, due to time constraints no in-depth investigation into the presence of Rayleigh streaming was carried out in this work. Another possibility is that the DeltaEC simulation program is not very good at estimating the hot metal temperature as a function of the heat flux into the heat exchanger. The metal temperature may need to be hotter than DeltaEC thinks it ought to be in order to get the heat into the helium. Unfortunately, there is no easy way to test this theory without constructing a separate test-rig dedicated to evaluating the

### Table 5.6. Measured Data vs. DeltaEC Simulation Results for steady state TaSEG Operation

<table>
<thead>
<tr>
<th></th>
<th>Freq. [Hz]</th>
<th>$Q_{in}$ [W]</th>
<th>$p_{1,c}$ [Bar pk]</th>
<th>$p_{1,b}$ [Bar pk]</th>
<th>$\text{Ph}(z_{b-c})$ [°]</th>
<th>$x_1$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Data</td>
<td>122.1</td>
<td>818.6</td>
<td>1.31</td>
<td>0.40</td>
<td>189.4</td>
<td>1.94</td>
</tr>
<tr>
<td>DeltaEC</td>
<td>122.3</td>
<td>501.6</td>
<td>1.31</td>
<td>0.40</td>
<td>186.9</td>
<td>1.93</td>
</tr>
</tbody>
</table>

### Table 5.7. Measured Data vs. DeltaEC Simulation Results for steady state TaSEG Operation

<table>
<thead>
<tr>
<th></th>
<th>HHX Temp. [K]</th>
<th>AHX Temp. [K]</th>
<th>$W_{Elec}$ [W]</th>
<th>$\eta_{Th-El}$</th>
<th>$\eta_C$ [%]</th>
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</thead>
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<td>Data</td>
<td>753.6</td>
<td>297.4</td>
<td>64.3</td>
<td>.0785</td>
<td>14</td>
</tr>
<tr>
<td>DeltaEC</td>
<td>763.2</td>
<td>297.4</td>
<td>63.7</td>
<td>.127</td>
<td>20</td>
</tr>
</tbody>
</table>
heat transfer occurring from the resistance heaters, through the Inconel pressure vessel wall, into the hot heat exchanger, then into the helium gas. Some insight into the discrepancies can be gained by reading the DeltaEC user’s guide that states that for heat exchanger segments “The temperature difference between the solid [heat exchanger] temperature $T_{\text{solid}}$ and the [helium] gas temperature $T_{\text{gas}}$ is proportional to $\dot{Q}$. The proportionality constant is not well validated, either theoretically or experimentally; we [the authors] find that it seems to be accurate within a factor of 2.”[89] Thus some error in the hot heat exchanger temperature is not unexpected. Given the excellent agreement between the simulation and the data, it is reasonable to use the simulation as a basis for taking an “inventory” of losses in the system, including those that are difficult to measure directly. Figure 5.14 shows an energy-flow diagram, or Sankey diagram, for the dual 132 TaSEG having a random fiber regenerator operating at a drive ratio of $p_1/P_m = 3.3\%$, while generating 132 watts of electricity.

Figure 5.14 suggests that the collimator is single biggest source of lost work, eating up 40 watts of useful acoustic power. This value is difficult to corroborate
through first-principles calculations since it involves oscillating flow through multiple orifices and depends on the effective local loss coefficients. Nonetheless, this is the loss indicated by the flow resistance that makes the simulation agree with the measurements. If accurate, it says that the flow resistance that makes the cycle work robs it of nearly 16% of its useful power! Further research and testing with various “tuning” plugs is required to better understand this loss.

5.3.2 A TaSEG based $\mu$CHP appliance

Building a gas burner prototype appropriate for a $\mu$CHP appliance to test with the TaSEG was also part of the initial plan. However, (again) due to time and project budget constraints this objective was abandoned. Instead, a TaSEG based $\mu$CHP appliance rendering was produced. Figure 5.15 show the $\mu$CHP appliance with an integrated, streamlined TaSEG power module. The TaSEG’s flange joints have been replaced with simple weld joints that decreased the overall outside diameter of the power module. A hypothetical gas burner and secondary heat exchanger are shown. Unfortunately, neither a prototype gas burner nor a full appliance were constructed.
Figure 5.15. A Scaled TaSEG Based $\mu$CHP appliance rendering.
Conclusions and Future Work

A dual 132 Thermoacoustic-Stirling Engine Generator, or TaSEG, has been designed, constructed and demonstrated. The TaSEG makes use of realistic hot-heat exchangers and proven alternators, all while delivering as much as 132 watts into an electrical load at a thermal-to-electrical efficiency as high as 8.3%. The TaSEG design, while experimental in nature, is deemed suitable for inclusion in a $\mu$CHP appliance. The prototype does require a “buzz” kick start, but it was able to achieve steady state operation the very first time it was attempted. Additionally, during steady state operation the two halves of the TaSEG phase lock despite no in depth work being carried out on this topic. The TaSEG can also achieve onset with a temperature gradient across the regenerator as low as 213°C, which is considered to be a very low temperature differential for a device of the type. The TaSEG, making use of an un-optimized collimator and latex diaphragm, achieved steady state operation with a low amplitude $p_1/P_m = 2.3\%$ pressure wave ratio, generating 65 W of electricity at a thermal-to-electric efficiency of 10.5%. This corresponds to 20% of the Carnot efficiency. At a higher amplitude $p_1/P_m = 3.3\%$ pressure wave ratio, the TaSEG generated 132 W of electricity at a thermal-to-electric efficiency of 8.32%, corresponding to 14% of the Carnot efficiency. At this operating point good agreement is found between the measured data and the performance predicted by the DeltaEC simulation. This gives confidence in further optimizing the TaSEG performance via the simulation before implementing changes experimentally, which should result in a more stream lined optimization process.
Although the efficiency is modest, there are clear paths to improvement. Most notable is the optimization of the flow resistance within the “collimator” and the replacement of the collimator and latex barrier with a static fluidic diode or jet pump. Both these options should increase the acoustic power available for conversion into electricity, but additional development work is required. The TaSEG achieved steady state operation at a low pressure amplitude ratio with no major technical unknowns, and the means to improve the efficiency and operate at higher pressure amplitudes are fairly straightforward. A streamlined TaSEG was also successfully integrated into a concept $\mu$CHP appliance rendering.

6.1 Paths to Improved TaSEG Performance

Based on the accomplishment of this work and the desire to increase the performance of the dual TaSEG, areas that will yield the most improvement in understanding and optimizing in the prototype are as follows:

1. **Optimize collimator flow resistance with latex diaphragm.**
   The “collimator” segment is a deliberate flow resistance inserted into the cycle that can be tuned to improve the phasing, at the expense of dissipating acoustic power. The work carried out to date has not included any tuning of the collimator’s resistance. Optimizing the collimator’s flow resistance through altering the dimensions of its tuning plugs in order to optimize the TaSEGs performance (emphasizing either thermal-to-electric conversion efficiency or electrical output) needs to be done. Understanding the trade-offs between the collimator’s lost work and that of the total acoustic work produced by the TaSEG is a very important precursor to a final $\mu$CHP TaSEG power module.

2. **More accommodating compact “cochlea” acoustic feedback.**
   The second compact cochlea acoustic feedback shown in Figure 6.1 has been designed to allow the insertion of a larger, 8.6 cm latex diaphragm, while having the same overall dimensions as the feedback already implemented in the TaSEG. Further optimization of this “drop-in” feedback replacement design within the DeltaEC simulation is required to define the most optimum
3. Tighter seal gap.
Based on past experience with Xylan\textsuperscript{1} coated, ribbed power pistons it might be possible to achieve a tighter effective radial clearance seal gap $\delta < 0.5$ mil. Since the flow losses in the seal gap go as the cube of the seal gap, a reduced seal gap should result in an increase in the TaSEG’s performance. Care should be taken when decreasing the seal gap as a reduced seal gap will decrease the losses associated with leakage, but it will increase the viscous losses associated with shearing of the working gas within the gap.[86, 109] Carefully balancing these losses to achieve optimized performance is a must.

4. Simulation versus experimental results.
Additional reconciliation of the experimental performance data with the DeltaEC computer simulation needs to be done. It would also be beneficial to produce a SAGE simulation that matches the TaSEG’s operating points

to see if further insight into its performance can be gained from a different viewpoint/simulation package.

5. **Better DeltaEC heat transfer model.**
Related to the task above, producing a better heat transfer model for the DeltaEC simulation program that can more accurately estimate the heat exchanger metal temperatures as a function of the heat flux into or out of the heat exchangers would be very beneficial and allow better comparison between the measured experimental results and the simulation. Constructing a separate test-rig dedicated to evaluating the heat transfer occurring from the resistance heaters, through the Inconel pressure vessel wall, into the hot heat exchanger, then into the helium gas might also provide more insight into the heat transfer mechanisms occurring within the TaSEG.

6. **Investigate Rayleigh streaming in thermal buffer tubes.**
Experimentally investigating if Rayleigh streaming is occurring within the TaSEG’s thermal buffer tubes might provide some insight into the unresolved discrepancy existing between the hot end heat exchanger temperature and input power experimentally measured and that predicted by the simulations.

7. **Jet pump design, testing, and optimization.**
Initially planned, but derailed by the failing of the latex diaphragms during baseline testing of the TaSEG operating at higher acoustic amplitudes, the implementation of a static, non-lifetime limited component that can suppress Gedeon streaming within the TaSEG is a must. Therefore, the design and implementation of a static jet pump is the most important future work that needs to be carried out.[105]

Clearly there is much work to be done to further improve the Thermoacoustic-Stirling Engine Generator for residential $\mu$CHP appliances. This work has shown that the technology can be made in a practical, cost effective, compact form that is appropriate for this application and established the paths forward necessary to reach competitive performance.

There were several general research related “lessons learned” from this work. The first was that the devil is in the details, but the trick is knowing which details
are critical to invest time into and which should only be looked over! Additionally, it is impossible to know everything, so one should seldom waver from relying on the experience, know-how and input of others. This will often result in the best outcome. The last lesson learned is that a good researcher/scientist should always be in a “hurry” to reach their next mistake and make forward progress.
Appendix A

A.1 Experimental Instrumentation Data Sheets
Endevco®

Piezoresistive pressure transducer
Model 8530B -200, -500, -1000

Key features
- ±200, ±500 and ±1000 psia ranges
- ±400 and ±1000 psig ranges
- ±1000 psi in 1200 psig input
- ±5000 psi in 5000 psi input

Specifications
- Endevco, synonymous with reliability
- Stringent Quality Control requirements, and compulsory corrective action procedures. These measures, together with conservative specifications have made the
- Continued product improvement necessitates that Meggitt reserve the right to modify these specifications without notice. Meggitt maintains a program of constant

Model 8530B is a miniature, high sensitivity piezoresistive pressure transducer for measuring absolute pressures. The sensor detects the diaphragm to be measured and glass bonding provides an absolute pressure reference. Full scale output is ±200, ±500 and ±1000 psi with very high overload capability and high frequency response. It is available in ranges from 200 psi to 5000 psi and model 8530B for lower pressure ranges.

Endevco brand pressure transducers feature an active four arm silicon bridge diaphragm mounted into a machined and ground aluminum case for maximum accuracy and high pressure capacity. The integrated hybrid silicon transducer provides stable performance over the temperature range of -65°F to +250°F (-54°C to +121°C). Endevco brand transmitter models are built on a unique self contained circuit board that provides full scale output and stability over a wide range of environments.

Endevco transmitters feature excellent accuracy from 20 to 100 emu, high shock resistance, and high stability over a range of environmental conditions.

Meggitt Sensing Systems
For measurement, monitoring and control.

Contact
Meggitt Sensing Systems
1210 South Wood Street
Des Plaines, IL 60018
P: +1 (847) 982-6732
F: +1 (847) 982-3833
www.meggitt.com

Endevco®

Piezoresistive pressure transducer
Model 8530B -200, -500, -1000

Key features
- ±200, ±500 and ±1000 psia ranges
- ±400 and ±1000 psig ranges
- ±1000 psi in 1200 psig input
- ±5000 psi in 5000 psi input

Specifications
- Endevco, synonymous with reliability
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F: +1 (847) 982-3833
www.meggitt.com
The Endevco® model 8510B is a rugged, miniature, high-sensitivity piezoresistive pressure transducer. It has a 10-32 mounting thread, 0.110 inch O.D. face diameter and is available in ranges from 200 to 2000 psi. Its high sensitivity is combined with high resistance, making it ideal for measuring dynamic pressures.

Essence pressure transducers feature low-active arm, strain gauge design and a unique multipiezoresistor element for maximum sensitivity and minimal frequency influence. The combined low thermal piezoresistor compensation provides stable performance over the temperature range of -50°F to 302°F (-45°C to 150°C). Endevco transducers also feature excellent linearity (less than ±0.1% full scale), high shock resistance, and negligible sensitivity to temperature transients.

The model 8510B is designed for a wide variety of applications, including static and dynamic pressure measurements, which require a transducer sensitive to low pressures and resistant to high excitation voltages. Its strain gauge design and unique multipiezoresistor element provide stable performance over temperature transients.

The model 8510B is available in ranges from 200 to 2000 psi, making it suitable for applications requiring high sensitivity and resistance to temperature transients. Its strain gauge design and unique multipiezoresistor element provide stable performance over temperature transients.

Endevco model 8510B is a piezoresistive pressure transducer that provides stable performance over the temperature range of -50°F to 302°F (-45°C to 150°C). Endevco transducers also feature excellent linearity (less than ±0.1% full scale), high shock resistance, and negligible sensitivity to temperature transients.
Model 8530BM37
Piezoresistive pressure transducer

Features
- Available full scale ranges: 200 psi (1.4 bar) 500 psi (3.5 bar) 1000 psi (6.9 bar)
- Full scale output: 300 and -1 K
- Absolute reference
- Detachable cable

Description
The Endevco® model 8530BM37 is a miniature, high sensitivity piezoresistive transducer for measuring dynamic pressure. The transducer is designed with a miniature receptacle to allow for detachment of the model 3027A-120 cable assembly. This pressure transducer is ideal for automotive brake line pressure measurements in sophisticated brake system (ABS) studies. With broad frequency response and excellent overload capability, the 8530BM37 suits many applications where a rugged, high performance sensor is needed.

Endevco strain gauge pressure transducers feature an active four-wire, stress gauge bridge diffused into a sandwiched silicon diaphragm for maximum sensitivity and wideband frequency response. Self-contained circuitry and temperature compensating provide reliable performance from -10°F to +175°F (-23°C to +80°C), with a wide operating temperature range from -40°F to +250°F (-40°C to +121°C). Endevco strain transducers also feature excellent stability, high shock resistance, and high reliability during temperature transients.

Endevco model 8530BM37 offers internal potentiometer, solder type connector or shunt on request at no additional cost.

Model number definition
- 8530BM37-XX-1
- Model number
- Full scale in psi
- +5 or -5 option
- Unit identification

Specifications
- Range: 200, 500, 1000, 2000 psi
- Resistance: 2000 ±800 ohms
- Supply voltage: 10.0 Vdc
- Burst pressure: 8000, 2000, 4000, 4000 psi
- Equiv. psi/g: 0.0003, 0.0002, 0.0002, 0.0002
- Thermal zero shift: ±100 ppm/°C over temperature range
- Zero measurand output: ±10 mV max ±10 mV max ±10 mV max ±10 mV max
- Pressure hysteresis % FSO typical: 0.1 0.1 0.1 0.1
- Non-repeatability % FSO typical: 0.1 0.1 0.1 0.1
- Non-linearity, independent % FSO typical: 0.2 0.2 0.2 0.2
- Combined: non-linearity, hysteresis, and non-repeatability % FSO typical: 0.50 0.50 1.0 1.0
- Photoflash response: >100,000 Hz
- ISA-S37.10, PARA. 6.7, Proc. II.
- Combined: non-linearity, hysteresis, and non-repeatability % FSO typical: 0.50 0.50 1.0 1.0
- Absolute reference
- Full scale output
- Protection option
- Burst pressure
- Zero measurand output
- Pressure hysteresis
- Non-repeatability
- Non-linearity, independent
- Photoflash response
- ISA-S37.10, PARA. 6.7, Proc. II.

Notes:
1. 1 psi = 6.895 kPa = 0.069 bar.
2. FSO (Full Scale Output) is defined as transducer output change from 0 to + full scale pressure.
3. Endevco model 3027A-120 cable assembly. This pressure transducer is ideal for automotive brake line pressure measurements in ABS brake system (ABS) studies.
4. Per ISA-S37.10, Para. 6.7, Proc. II.
5. Warm-up time is defined as elapsed time from excitation voltage “turn on” until the transducer reaches ± 1% of full scale pressure.
6. Use of excitation voltages other than 10.0 Vdc requires manufacture and calibration at that voltage.
7. Media in brake line systems ideal for automotive brake line pressure measurements.
8. WELD OR BRAZE attack epoxies.
9. Case pressure is the media containment pressure in the event of diaphragm rupture.
10. Units can be compensated over any 200°F (93°C) span from -65°F to +250°F (-54°C to +121°C) on request.

Model number definition
- 8530BM37-XX-1
- Full scale in psi
- +5 or -5 option
- Unit identification

Model 8530BM37 Piezoresistive pressure transducer
Specifications
<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Range</td>
<td>200 psi (1.4 bar) 500 psi (3.5 bar) 1000 psi (6.9 bar) 2000 psi</td>
</tr>
<tr>
<td>Resistance</td>
<td>2000 ±800 ohms</td>
</tr>
<tr>
<td>Supply voltage</td>
<td>10.0 Vdc</td>
</tr>
<tr>
<td>Burst pressure</td>
<td>8000, 2000, 4000, 4000 psi</td>
</tr>
<tr>
<td>Equiv. psi/g</td>
<td>0.0003, 0.0002, 0.0002, 0.0002</td>
</tr>
<tr>
<td>Thermal zero shift</td>
<td>±100 ppm/°C over temperature range</td>
</tr>
<tr>
<td>Zero measurand output</td>
<td>±10 mV max ±10 mV max ±10 mV max ±10 mV max</td>
</tr>
<tr>
<td>Pressure hysteresis % FSO typical</td>
<td>0.1 0.1 0.1 0.1</td>
</tr>
<tr>
<td>Non-repeatability % FSO typical</td>
<td>0.1 0.1 0.1 0.1</td>
</tr>
<tr>
<td>Non-linearity, independent % FSO typical</td>
<td>0.2 0.2 0.2 0.2</td>
</tr>
<tr>
<td>Combined: non-linearity, hysteresis, and non-repeatability % FSO typical</td>
<td>0.50 0.50 1.0 1.0</td>
</tr>
<tr>
<td>Photoflash response</td>
<td>&gt;100,000 Hz</td>
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<tr>
<td>ISA-S37.10, PARA. 6.7, Proc. II.</td>
<td>Combined: non-linearity, hysteresis, and non-repeatability % FSO typical</td>
</tr>
</tbody>
</table>

Notes:
1. 1 psi = 6.895 kPa = 0.069 bar.
2. FSO (Full Scale Output) is defined as transducer output change from 0 to + full scale pressure.
3. Endevco model 3027A-120 cable assembly. This pressure transducer is ideal for automotive brake line pressure measurements in ABS brake system (ABS) studies.
4. Per ISA-S37.10, Para. 6.7, Proc. II.
5. Warm-up time is defined as elapsed time from excitation voltage “turn on” until the transducer reaches ± 1% of full scale pressure.
6. Use of excitation voltages other than 10.0 Vdc requires manufacture and calibration at that voltage.
7. Media in brake line systems ideal for automotive brake line pressure measurements.
8. WELD OR BRAZE attack epoxies.
9. Case pressure is the media containment pressure in the event of diaphragm rupture.
10. Units can be compensated over any 200°F (93°C) span from -65°F to +250°F (-54°C to +121°C) on request.

Endevco model 3027A-120 (supplied)

Technical data
- Basic model number definition
- Full scale in psi
- +5 or -5 option
- Unit identification
- Burst pressure
- Zero measurand output
- Pressure hysteresis
- Non-repeatability
- Non-linearity, independent
- Photoflash response
- ISA-S37.10, PARA. 6.7, Proc. II.
The BL4500 High-Power Series delivers all the quality features our customers have come to expect from Behlman, clean sine wave output with excellent line and load regulation, high efficiency and low harmonic distortion. A multi-pulse input transformer offers low input harmonic distortion and high power factor as required by MIL-STD-784 and European standards. In addition to the usual features (Proper cooling, internal temperature protection, earth-ground protection and voltage fold-back during overloads to maintain undistorted waveform), Amplitude and Frequency adjustment, line drop compensation, phase-angle adjustment and output undistortion, these compact, reliable units are supplied with analog remote control and available optional RS-232 and IEEE-488 remote control interfaces. Other options include: Extended Frequency Range (up to 1000 Hz) and Motor Test option which has the capability to soft-start motors, pumps and compressors thereby eliminating the need for high power devices. The HIGH power factor and low harmonic distortion, makes the BL High Power series ideal for industrial product testing, precise acoustic test and power conditioning, ATE, bulk power and motor generator replacement.

**FEATURES**

- Multi-Pulse Input Rectification – Low Harmonic Distortion
- Analog Remote Control – Remote Programming
- Low cost per VA – Cost Savings
- Compact Size – Less rack space

**PROREV 1.0**

**OCTOBER 1997**

**MANUAL CONTROL OR PROGRAMMABLE AC POWER**

The BL4500 High-Power Series delivers all the quality features our customers have come to expect from Behlman, clean sine wave output with excellent line and load regulation, high efficiency and low harmonic distortion. A multi-pulse input transformer offers low input harmonic distortion and high power factor as required by MIL-STD-784 and European standards. In addition to the usual features (Proper cooling, internal temperature protection, earth-ground protection and voltage fold-back during overloads to maintain undistorted waveform), Amplitude and Frequency adjustment, line drop compensation, phase-angle adjustment and output undistortion, these compact, reliable units are supplied with analog remote control and available optional RS-232 and IEEE-488 remote control interfaces. Other options include: Extended Frequency Range (up to 1000 Hz) and Motor Test option which has the capability to soft-start motors, pumps and compressors thereby eliminating the need for high power devices. The HIGH power factor and low harmonic distortion, makes the BL High Power series ideal for industrial product testing, precise acoustic test and power conditioning, ATE, bulk power and motor generator replacement.

**PROTECTIVE CIRCUITS**

**Input**

- Fault sensing main circuit breaker
- Overload automatically causes voltage fold-back to provide additional margin for undistorting output waveform

**Options**

- Extended frequency range, 45-1000 Hz
- IEEE-488 Interface
- RS-232 Interface

**Environmental**

- Operating Temp: 32°F to 131°F (0-55°C)
- Humidity: 5-95% non-condensing
- Weight: 250 lbs (113.5 kgs)

**Input Voltage Table 1**

<table>
<thead>
<tr>
<th>Model</th>
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<th>Options</th>
</tr>
</thead>
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<tr>
<td>BL4500X</td>
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<td>BL4500XX</td>
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</tr>
<tr>
<td>BL4500XXX</td>
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</tbody>
</table>

**Ordering Information**

BL4500

- Model: BL4500
- Input: 230/400VAC, 3 phase 47Hz – 63Hz
- Optional Extended Frequency: 45-1000 Hz

Contact factory for additional options.
Accurate evaluation of consumption power of electrical products

Power measuring instruments

3332: Single-phase, 2-wire type that can accurately measure even standby power
3331: Single-phase, 3-wire and three-phase, 3-wire type for measuring power of large-scale equipment

The POWER HiTESTER 3332 does the job by offering a wide range of power measurement from standby to normal usage. The POWER HiTESTER 3331 is capable of evaluating 3-phase devices, such as industrial air conditioners and refrigerators, or single-phase, large-scale devices. Both power testers deliver high accuracy of ±0.2% (45 to 66Hz), direct input up to 50A, and a broad bandwidth from 1Hz (the 3331 from 10Hz) to 100kHz. System construction is made easy with these compact, lightweight and reasonably priced tools, which come equipped with an external interface as a standard feature. The 3331 and 3332 can be used as a measuring component for a wide range of purposes, from research and development to equipment evaluation.

### Measurement from minute single-phase power to large-scale 60 kW 3-phase equipment.

<table>
<thead>
<tr>
<th>Model</th>
<th>3332</th>
<th>3331</th>
</tr>
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<tbody>
<tr>
<td>Measurement line</td>
<td>1ø2W only</td>
<td>1ø2W to 3ø3W</td>
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<td>U Range</td>
<td>15 to 600V (6 ranges)</td>
<td>150 to 600V (3 ranges)</td>
</tr>
<tr>
<td>I Range</td>
<td>1mA to 50A (15 ranges)</td>
<td>500mA to 50A (7 ranges)</td>
</tr>
<tr>
<td>Frequency characteristics</td>
<td>1Hz to 100kHz</td>
<td>10Hz to 100kHz</td>
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<td>Basic accuracy</td>
<td>±0.1%rdg. ±0.1%f.s. (45 to 66Hz)</td>
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</tr>
<tr>
<td>Dimensions</td>
<td>Approx. 210W ✕100H ✕261D mm</td>
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</tbody>
</table>
SR810 and SR830 DSP Lock-In Amplifiers

The SR810 and SR830 DSP Lock-In Amplifiers provide high performance at a competitive cost. The SR810 contains analog-to-digital filters to provide high performance at a much lower cost than the SR830. The SR810 contains analog-to-digital filters in the digital signal processing (DSP) and replaces the multi-channel, output filters, and amplifiers based on conventional lock-ins. The SR830 contains additional performance that is available on an operating range of 1 mHz to 102 kHz and 100 dB of drift rejection.

Input Channel

The SR810 and SR820 have differential inputs with 14-bit A/D converters. The reference signal is acquired by using the input signal with a high precision 14-bit converter. Two input signal levels can be configured for current measurements with different curries for lower current inputs. The input stage is designed to allow the user to configure a variety of measurement applications.

Digitizing

The SR810 and SR830 contain 16-bit, high-speed digitizers that provide 5 MHz sampling rate with 16-bit resolution. The digitizers are designed for use with high input impedances and provide a very low noise, high dynamic range, and high signal to noise ratio. The digitizers are designed to allow the user to configure a variety of measurement applications.

Signal Channel

The SR810 and SR830 provide high performance at a competitive cost. The SR810 contains analog-to-digital filters to provide high performance at a much lower cost than the SR830. The SR830 contains analog-to-digital filters in the digital signal processing (DSP) and replaces the multi-channel, output filters, and amplifiers based on conventional lock-ins. The SR830 contains additional performance that is available on an operating range of 1 mHz to 102 kHz and 100 dB of drift rejection.

Reference Channel

The SR810 and SR830 have differential inputs with 14-bit A/D converters. The reference signal is acquired by using the input signal with a high precision 14-bit converter. Two input signal levels can be configured for current measurements with different curries for lower current inputs. The input stage is designed to allow the user to configure a variety of measurement applications.
Inductive and LVDT sensors for displacement, distance and position. Inductive displacement sensors, LVDT gaging sensors, for integration in machines and systems. Accurate displacement and position sensors to favourable conditions.

VIP series: sensors with measuring ring and integral electronics

Sensor Rod
Patented measurement principle

There is no mechanical contact between the measuring element (ring) and the sensor rod. The sensor therefore operates without any wear.

VIP series: SR750 ZA 2.5
Electrical output
SA7 = connector, axial (housing version GA)
SR7 = connector, radial (housing version ZA)

Electromagnetic compatibility (EMC)
Protection class
Current consumption
Power supply
Output load
Frequency response (-3 dB)
Resolution
Measuring range
Temperature stability
Linearity
Model

<table>
<thead>
<tr>
<th>Measuring range</th>
<th>VIP-50</th>
<th>VIP-100</th>
<th>VIP-150</th>
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</thead>
<tbody>
<tr>
<td>mm</td>
<td>50</td>
<td>100</td>
<td>150</td>
</tr>
<tr>
<td>Linearity</td>
<td>±0.5 % FSO</td>
<td>±0.25 % FSO</td>
<td>±0.15 % FSO</td>
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<tr>
<td>±20 mm</td>
<td>5.0 mm</td>
<td>5.7 mm</td>
<td>6.4 mm</td>
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<tr>
<td>Temperature</td>
<td>40 °C ≤ t ≤ 85 °C</td>
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</tr>
<tr>
<td>Temperature stability</td>
<td>±50 ppm / °C</td>
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<td></td>
</tr>
<tr>
<td>Frequency response (3 dB)</td>
<td>200 Hz</td>
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<tr>
<td>Output</td>
<td>0 - 20 mA</td>
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<td>Output load</td>
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<td>Power supply</td>
<td>18 - 30 VDC</td>
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<td>Current consumption</td>
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<td>Electromagnetic compatibility (EMC)</td>
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<td>Interference immunity</td>
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<td>Frequency range (3 dB)</td>
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<tr>
<td>Resolution</td>
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<tr>
<td>Measuring range</td>
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<tr>
<td>Temperature stability</td>
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<tr>
<td>Linearity</td>
<td>±0.5 % FSO</td>
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<td>±0.25 % FSO 0.125 mm 0.25 mm</td>
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<tr>
<td>±0.125 % FSO 0.25 mm 0.5 mm</td>
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<td></td>
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<tr>
<td>±0.063 % FSO 0.5 mm 1.0 mm</td>
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<tr>
<td>±0.032 % FSO 1.0 mm 2.0 mm</td>
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<tr>
<td>±0.016 % FSO 2.0 mm 5.0 mm</td>
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<tr>
<td>±0.016 % FSO 5.0 mm 10.0 mm</td>
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<td></td>
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<tr>
<td>±0.008 % FSO 10.0 mm 20.0 mm</td>
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</tr>
<tr>
<td>±0.004 % FSO 20.0 mm 50.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.002 % FSO 50.0 mm 100.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.001 % FSO 100.0 mm 200.0 mm</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>±0.0005 % FSO 200.0 mm 500.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.00025 % FSO 500.0 mm 1000.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.000125 % FSO 1000.0 mm 2000.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.0000625 % FSO 2000.0 mm 5000.0 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Parallel mounting
The optimum ratio of measurement range to selected length of the sensor is ensured in the installation space needed. For VIP series, the parallel connection of the measuring object and measuring ring facilitates compact new construction and installs suitable systems. Whereas with conventional sensors with an axial measurement path, the length of the plunger must be added to the actual housing length, with VIP series only, the housing length has to be considered during the design.

VIP-50 VIP-100 VIP-150
Electrical output
SA7 = connector, axial (housing version GA)
SR7 = connector, radial (housing version ZA)

Electromagnetic compatibility (EMC)
Protection class
Current consumption
Power supply
Output load
Frequency response (-3 dB)
Resolution
Measuring range
Temperature stability
Linearity
Model

<table>
<thead>
<tr>
<th>Measuring range</th>
<th>VIP-50</th>
<th>VIP-100</th>
<th>VIP-150</th>
</tr>
</thead>
<tbody>
<tr>
<td>mm</td>
<td>50</td>
<td>100</td>
<td>150</td>
</tr>
<tr>
<td>Linearity</td>
<td>±0.5 % FSO</td>
<td>±0.25 % FSO</td>
<td>±0.15 % FSO</td>
</tr>
<tr>
<td>±20 mm</td>
<td>5.0 mm</td>
<td>5.7 mm</td>
<td>6.4 mm</td>
</tr>
<tr>
<td>Temperature</td>
<td>40 °C ≤ t ≤ 85 °C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature stability</td>
<td>±50 ppm / °C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Frequency response (3 dB)</td>
<td>200 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Output</td>
<td>0 - 20 mA</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Output load</td>
<td>500 Ohm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Power supply</td>
<td>18 - 30 VDC</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Current consumption</td>
<td>max. 40 mA</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Protection class</td>
<td>IP 67</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electromagnetic compatibility (EMC)</td>
<td>EN 50 081-2 spurious emission, EN 50 082-2 interference immunity</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Interference immunity</td>
<td>EN 50 081-2, EN 50 082-2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Frequency range (3 dB)</td>
<td>5 Hz ... 44 Hz ± 2.5 mm; 44 Hz ... 500 Hz ±20 g</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>&lt;0.03 % FSO 0.015 mm 0.03 mm 0.045 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measuring range</td>
<td>50 mm 100 mm 150 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature stability</td>
<td>±0.25 % FSO 0.125 mm 0.25 mm -40° C...+85° C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Linearity</td>
<td>±0.5 % FSO</td>
<td></td>
<td></td>
</tr>
<tr>
<td>±0.25 % FSO 0.125 mm 0.25 mm</td>
<td></td>
<td></td>
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<tr>
<td>±0.125 % FSO 0.25 mm 0.5 mm</td>
<td></td>
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<tr>
<td>±0.063 % FSO 0.5 mm 1.0 mm</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>±0.032 % FSO 1.0 mm 2.0 mm</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>±0.016 % FSO 2.0 mm 5.0 mm</td>
<td></td>
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<tr>
<td>±0.016 % FSO 5.0 mm 10.0 mm</td>
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<tr>
<td>±0.008 % FSO 10.0 mm 20.0 mm</td>
<td></td>
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<tr>
<td>±0.008 % FSO 20.0 mm 50.0 mm</td>
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<tr>
<td>±0.004 % FSO 50.0 mm 100.0 mm</td>
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<tr>
<td>±0.004 % FSO 100.0 mm 200.0 mm</td>
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<tr>
<td>±0.002 % FSO 200.0 mm 500.0 mm</td>
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<tr>
<td>±0.002 % FSO 500.0 mm 1000.0 mm</td>
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<td>±0.001 % FSO 1000.0 mm 2000.0 mm</td>
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<tr>
<td>±0.001 % FSO 2000.0 mm 5000.0 mm</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>±0.0005 % FSO 5000.0 mm 10000.0 mm</td>
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<td></td>
<td></td>
</tr>
</tbody>
</table>

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The Option 001 phase lock/TCXO timebase for increased frequency stability lets you synchronize with up to three 33120As. An external clock input/output lets you perform phase changes on a computer, allowing precise phase adjustment. The Option 001 phase lock/TCXO timebase for increased frequency stability lets you synchronize with up to three 33120As. An external clock input/output lets you perform phase changes on a computer, allowing precise phase adjustment.

**Agilent 33120A Function/Arbitrary Waveform Generator**

- **Output Characteristics**
  - **Output Type**: 12-bit, 40 MSa/s, 16,000-point deep arbitrary waveforms
  - **Amplitude Resolution**: 12 bits (including sign)
  - **Waveform Length**: 8 to 16,000 points
  - **Arbitrary**: Standard sine, square, triangle, ramp, noise, sin(x)/x, exponential rise, exponential fall, cardiac, dc volts.
  - **Standard**: Sine, square, triangle, ramp, noise, sin(x)/x, exponential rise, exponential fall, cardiac, dc volts.

- **Frequency Characteristics**
  - **Frequency Range**: 10 Hz to 50 MHz
  - **Resolution**: 1 Hz (typical)
  - **Accuracy**: ± 1% (typical)
  - **Internal Rate**: 10 mHz - 50 kHz ± 1%
  - **Carrier -3dB Freq.**: 10 MHz (typical)
  - **Modulation Depth**: 0% - 120%
  - **Sub-Oscillator**: Fixed frequency
  - **Duty Cycle**: 1% - 99% (fixed)

- **Modulation**
  - **Frequency**: 10 Hz - 50 MHz
  - **Amplitude**: 1 Vpp into 50 Ω
  - **Phase Offset**: > 1 Vpp into 50 Ω
  - **Impedance**: 50 Ω

- **Phase Noise**
  - **Relative Phase Noise**: 100 Hz to 100 kHz ± 20 dBc

- **Spectral Purity**
  - **Harmonic Distortion**: 100 kHz to 1 MHz ± 1.5% (0.15 dB)
  - **Carrier -3dB Freq.**: 10 MHz (typical)

- **Protection**: Short circuit protected
- **Isolation**: 42 Vpk maximum to earth
- **Resolution**: 3 digits, amplitude and offset (fixed)
- **Accuracy**: ± 2% of setting + 2 mV
- **Temperature Range**: 0°C to 55°C
- **Storage Environment**: -40°C to 70°C
- **Power Line Frequency**: 45 Hz to 66 Hz and 360 Hz
- **Power Supply**: 110V/120V/220V/240V ± 10%
- **Vibration and Shock**: MIL-T-28800, Type III, Class 5
- **Electrical Specifications**:
  - **Input Resistance**: ≥ 100 kΩ
  - **Input Impedance**: ≤ 100 kΩ
  - **Output Impedance**: ≤ 100 Ω
  - **Output Voltage**: ± 15 Vpk overdrive < 1 minute
  - **Function Change**: 80 ms
  - **Configuration Data**: 0.5 minute
  - **Power Line Change**: 0.5 minute

- **Signal Characteristics**
  - **Quadrature**: Max. ± 15 Vpk
  - **Data Rate**: 500 kbps
  - **Data Quality**: ≥ 100 kΩ

- **Arbitrary Waveforms**
  - **Arbitrary Waveforms**: Standard sine, square, triangle, ramp, noise, sin(x)/x, exponential rise, exponential fall, cardiac, dc volts.
  - **Arbitrary Waveforms**: Standard sine, square, triangle, ramp, noise, sin(x)/x, exponential rise, exponential fall, cardiac, dc volts.

- **Operating Environment**: 0°C to 55°C
- **Storage Environment**: -40°C to 70°C
- **Humidity**: Non-condensing
- **Power Consumption**: 19 W
- **Dimensions**: 374.0 mm × 212.6 mm × 254.4 mm
- **Weight**: 5 kgs
- **Warranty**: 1 year
- **Ordering Information**:
  - **Part Number**: E0350A
  - **Options**: Opt. 001 Phaselock/TCXO Timebase

- **System Characteristics**
  - **Setability**: < 0.01 ppm
  - **Stability**: ± 1 ppm 0° - 50°
  - **Aging**: < 2 ppm/yr
  - **Setability**: < 0.01 ppm
  - **Stability**: ± 1 ppm 0° - 50°
  - **Aging**: < 2 ppm/yr

- **Support**: Agilent engineers and technicians worldwide can provide basic measurement assistance for the Agilent instruments you use. For the fastest return on investment of your Agilent instruments, return your unique technical and business needs. Solve problems. We strive to ensure that you get the full performance, troubleshooting, and application help you need. Our extensive support is available for at least five years beyond the warranty. Agilent's overall support policy: “Our support is available for at least five years beyond the warranty.”

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  - **Telephones**:
    - **United States**: 1-800-829-4444
    - **Canada**: 1-888-410-4444
    - **Latin America**: 0800 047 866
    - **Europe**: (tel) (65) 6375 8100
    - **Asia-Pacific**: (tel) (81) 426 56 7832
    - **Japan**: (tel) (81) 426 56 7840
    - **China**: (tel) 877 894 4414
    - **Canada**: (tel) 800 829 4444
    - **New Zealand**: (tel) 0800 829 4444
  - **Fax Numbers**:
    - **United States**: (fax) (65) 6375 8300
    - **Canada**: (fax) (65) 6375 8300
    - **Latin America**: (fax) (65) 6375 8300
    - **Europe**: (fax) (82 2) 2004 5115
    - **Asia-Pacific**: (fax) (65) 6379 8300
    - **Japan**: (fax) (81) 426 56 7840
    - **China**: (fax) 877 894 4414
  - **Mailing Addresses**:
    - **United States**: 1505 South 30th Street, Santa Clara, CA 95051, USA
    - **Canada**: 3999 York Mills Rd., Markham, ON L3R 9C4, Canada
    - **Europe**: Agilent Technologies EMEA, Agilent Technologies Campus, D-82156 Germering, Germany
  - **Online Assistance**:
    - **WWW**: www.agilent.com/find/emailupdates
  - **Online Assistance**:
    - **Worldwide Customer Support Online**: Online Assistance: www.agilent.com/find/emailupdates

- **Agilent Email Updates**
  - **For the latest information on products and applications, please visit www.agilent.com/find/emailupdates**
New Fluke 287
True-rms Electronics Logging Multimeter with TrendCapture

Specifications

<table>
<thead>
<tr>
<th>Function/Feature</th>
<th>Fluke 287</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage</td>
<td>0.01 V to 1000 V</td>
</tr>
<tr>
<td>Current</td>
<td>0.005 A to 10 A</td>
</tr>
<tr>
<td>Resistance</td>
<td>0.5 kΩ to 100 GΩ</td>
</tr>
<tr>
<td>Frequency</td>
<td>0.01 Hz to 500 kHz</td>
</tr>
<tr>
<td>Temperature</td>
<td>-20 °C to 60 °C</td>
</tr>
<tr>
<td>Storage</td>
<td>-40 °C to 60 °C</td>
</tr>
</tbody>
</table>

Equipment with new functionality

- **New** – Large 50,000-count 1/4 VGA display with white backlight. Multiple sets of measurement information can be simultaneously displayed at the same time.
- **New** – Backlit keys for easy viewing while freeing your hands to focus on the job.
- **New** – Multi-lingual interface enables you to document, store and analyze individual readings or a series of saved measurements.

Additional Functions/Features

- **FlukeView® Forms**
- **Temperature Probe**
- **Soft Case**
- **C280 Soft Case**
- **TPAK Magnetic Hanging Kit**
- **FVF-SC2 FlukeView® Forms Software with Cable**

General specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Fluke 287</th>
</tr>
</thead>
<tbody>
<tr>
<td>LCD display</td>
<td>1/4 VGA display with white backlight. Multiple sets of measurement information can be simultaneously displayed at the same time.</td>
</tr>
<tr>
<td>Battery life</td>
<td>170 hours in logging mode</td>
</tr>
<tr>
<td>Size (H x W x L)</td>
<td>22.2 cm x 10.2 cm x 4.8 cm (8.75 in x 4.03 in x 1.87 in)</td>
</tr>
<tr>
<td>Weight</td>
<td>870.9 g (28 oz)</td>
</tr>
<tr>
<td>Specifications subject to change without notice.</td>
<td></td>
</tr>
</tbody>
</table>

Fluke, keeping your world up and running.

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Printed in U.S.A. 5/2007 3032814 D-EN-N Rev A

Specifications subject to change without notice.

Limited lifetime warranty

For Fluke products information, technical support or service, please call 1-800-36-FLUKE (1-800-363-5853) in the U.S. or 1-800-913-2510 in Canada. In other countries, contact your local Fluke sales office or authorized distributor. In Europe, contact Fluke Europe B.V. PO Box 9090, 5602 BD Eindhoven, The Netherlands, tel +31 (0) 40 2675 222, fax +31 (0) 40 2675 223. The Fluke 287, when used with TrendCapture, is intended for electrical measurement applications without exceeding a DC voltage of 750 V and without exceeding an AC voltage of 500 V rms.

TrendCapture quickly documents design performance and TrendCapture electronics professionals can depend on. The advanced, logging digital multimeter with high accuracy to a PC to detect a trend. The Fluke 287 packs more accuracy and capabilities mean you no longer need to download logged readings by APP to achieve a trend. The Fluke 287 packs more accuracy and convenience into a handheld multimeter than ever before.

Technical Data

The Fluke 287 True-rms Electronics Logging Multimeter with TrendCapture apply documents design performance and graphically displays what happened. Its unique logging and graphing capabilities mean you no longer need to download logged readings by APP to achieve a trend. The Fluke 287 packs more accuracy and convenience into a handheld multimeter than ever before.
**MECHANICAL & ENVIRONMENTAL**

**Dimensions:**
- Single ø Input: 28.3” H x 31.6” D x 22.1” W
- Three ø Input: (2) 19” (48.3 cm) rack-mount chassis 7” H and 8.75” H x 22” D
- Weight: 250 lbs (113.5 kgs)

**Input Voltages Table 1**

<table>
<thead>
<tr>
<th>Model</th>
<th>Input</th>
<th>Options</th>
</tr>
</thead>
<tbody>
<tr>
<td>BL4500</td>
<td>47Hz – 63Hz</td>
<td>N/A</td>
</tr>
<tr>
<td></td>
<td>45-500 Hz</td>
<td>N/A</td>
</tr>
<tr>
<td></td>
<td>71.9 cm H x 80.3 cm D x 55.6 cm W</td>
<td>N/A</td>
</tr>
</tbody>
</table>

**Environmental / Connections**
- Operating Temp: 32°F to 131°F (0-55° C)
- Humidity: 0-95% RH non-condensing
- Operating Temp: 32°F to 131°F (0-55° C)
- Humidity: 0-95% RH non-condensing
- Voltage:
  - 120/208 VAC, 3 phase
  - 200 VAC, DELTA, 3 phase
  - 220/380 VAC, 3 phase
  - 230 VAC, 1 phase
  - 230/400VAC, 3 phase
  - 240/415 VAC, 3 phase
  - 277/480 VAC, 3 phase
  - 346/600 VAC, 3 phase
- Frequency:
  - 47Hz – 63Hz
  - 45-500 Hz
  - 0-10 VDC programming for
    - voltage and frequency
    - contact closure for output
  - external synch

**Ordering Information**

**Options**
- Extended frequency range, 45-1000 Hz
- IEEE-488 Interface
- RS232 Interface
- Locking nut
- Mounting Angle (2 per chassis)
- Motor test
- Ruggedized for use in shock and vibration

**www.behlman.com**

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**FEATURES**
- Multi-Pulse Input Rectification – Low Harmonic Distortion
- Analog Remote Control – Remote Programming
- Low cost per VA – Cost Savings
- Compact Size – Less rack space

**MANUAL CONTROL OR PROGRAMMABLE AC POWER**

The BL4500 High-Power Series delivers all the quality features our customers have come to expect from Behlman: clean sine wave output with excellent line and load regulation, high efficiency and low harmonic distortion. A multi-pulse input transformer offers low-input harmonic distortion and high power factor as required by MIL-STD-704B and European standards.

**INPUT**
- Voltage:
  - 4.350 V
- Frequency:
  - 47-63 Hz

**Output**
- Power:
  - 4,500 VA
- Voltage:
  - 0-270 V, single phases, isolated
- Resolution:
  - 1
- Accuracy:
  - +/- 0.2% of full scale
- Frequency:
  - 400 Hz
- Resolution:
  - 1 Hz
- Accuracy:
  - +/- 2 Hz
- Current:
  - 17 Amps / phase
- Resolution:
  - 0.1 Amp, +/-1 digit
- Accuracy:
  - +/- 0.2% of full scale
- Crest Factor:
  - 3:1
- Power Factor:
  - 100% of rated output into any power factor load
- Distortion:
  - 0.2% THD, measured at 115 volts, 60 Hz
- Line Regulation:
  - +/- 0.1% for +/- 10% line change
- Load Regulation:
  - +/- 0.2% no load to full load
- Efficiency:
  - 80% typical

**Protective Circuits**
- Input:
  - Fault sensing main circuit breaker
  - Overload automatically causes voltage fold back to provide
    - protection from heat damage and vibration
  - voltage fold-back during overloads to maintain
  - temperature protection, short-circuit protection and
  - unique protection circuitry provides for over-
  - required by MIL-STD-1399 and European standards.
  - input harmonic distortion and high power factor as
  - distortion. A multi-pulse input transformer offers low-
  - amplitude and frequency adjustment, line drop
  - uncompromised waveforms.

**Contact Factory for Other Input Voltages**

**www.behlman.com**

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**www.behlman.com**
Temperature sensors are manufactured with the same stringent selection of materials as for the heaters. THERMOCOAX supplies many different applications from 0.25 mm O.D. up to 2300°C. They are resistant to corrosive media, acids and chemicals.

Temperature range / uniformity:
- Operational power and voltage:
- Material / Mounting / design: stainless steel, aluminium with or without protection metal sheath.

Heating solution based on its long experience, THERMOCOAX provides solutions and temperature measurement, with high accuracies and small tolerances. Solutions for the automotive industry and the foundry industry for light metals are offered.

- Technological solutions and temperature measurement,
- Very good temperature uniformity, fast ramp up times and very reliable operational life.
- Thermal service, complete engineering and assembling.
- Customised design, two or three-dimensional, multi zone heating cooling plates.

Foundry Industry for light metals

- Industrial furnaces for the production of investment castings.
- Holding and bale-out furnaces
- Low pressure furnaces
- Immersion heaters for high pressure test
- Heating-cooling plates for production of exhaust gases.

Other applications:
- Heating of injection nozzles, aluminium feeder tubes, plastic Industry
- Heated extruder nozzles for pipe corrugators
- Heaters for extrusion and injection nozzles
- Heated bending tool for plastic parts
- Heated extruder nozzles for pipe corrugators

Special thermocouples and heaters

- High-Watt density heaters
- Heating-cooling plates for production of exhaust gases
- Ion source for ion implantation to modify surface properties
- Heating devices for high vacuum, oil
- Heating devices for high vacuum, oil
- Heating devices for high vacuum, oil
- Heating devices for high vacuum, oil

Preheating of sealings before welding

- Preheating of sealings before welding
- Brazing (torch and vacuum brazing)
- Welding (Laser, Plasma, TIG),

Welding / brazing equipment

- Ensures long life time
- Welded seam with high quality and utmost reliability
- Welding / brazing equipment

High temperature Thermocouples

- High temperature thermocouples
- Thermocouple for each heating zone
- Thermocouple for each heating zone
- Thermocouple for each heating zone
- Thermocouple for each heating zone

Special thermocouples and heaters

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- Heating devices for high vacuum, oil
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- Heating devices for high vacuum, oil
Overview
The USB-600x devices are ideal for low-cost, basic applications such as:
- \[ \text{Data logging} \]
- \[ \text{Portable measurements} \]
- \[ \text{Academic lab experiments} \]
- \[ \text{Instruments} \]

Application and Technology
The USB-600x devices are ideal for low-cost, basic applications such as:
- \[ \text{Data logging} \]
- \[ \text{Portable measurements} \]
- \[ \text{Academic lab experiments} \]
- \[ \text{Instruments} \]

Requirements and Compatibility
USB-600x devices have removable screw terminals for easy signal connectivity. For extra flexibility when handling multiple wiring configurations, NI offers the USB-600x Prototyping Kit, which provides space for adding more circuitry to the inputs of USB-600x devices.

Comparison Tables
<table>
<thead>
<tr>
<th>Feature</th>
<th>USB-6000</th>
<th>USB-6001</th>
<th>USB-6002</th>
<th>USB-6003</th>
</tr>
</thead>
<tbody>
<tr>
<td>Analog Input Sampling Rate (kS/s)</td>
<td>10</td>
<td>8</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Analog Input (SE/DI)</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>Digital Input/Output</td>
<td>25</td>
<td>25</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>Digital Input Sampling Rate (kS/s)</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
</tbody>
</table>

Support and Services
Technical Support
NI offers training and consulting services for the following national/international resources:
- \[ \text{Online Community} \]
- \[ \text{Instructor-led training} \]
- \[ \text{Classroom training} \]
- \[ \text{On-site training} \]

Repair
NI provides repair services for the following national/international resources:
- \[ \text{Worldwide service} \]
- \[ \text{Extended warranty} \]

OEM
OEM applications use the USB-600x devices for OEM applications such as:
- \[ \text{Low-cost basic applications} \]
- \[ \text{Data logging} \]
- \[ \text{Portable measurements} \]
- \[ \text{Academic lab experiments} \]
- \[ \text{Instruments} \]

Ordering Information
For a complete list of accessories, visit the product page on ni.com.
## View basic information about your computer

**Windows edition**
- Windows 7 Enterprise

**Copyright © 2009 Microsoft Corporation. All rights reserved.**
- Service Pack 1

### System
- Rating: [Windows Experience Index](#)
- Processor: Intel(R) Core(TM) i3-2120 CPU @ 3.30 GHz
- Installed memory (RAM): 4.00 GB
- System type: 64-bit Operating System
- Pen and Touch: Touch Input Available with 10 Touch Points

### Computer name, domain, and workgroup settings
- Computer name: TRODFSCWX1
- Full computer name: TRODFSCWX1-chart-incl.local
- Computer description: TRODFSCWX1-chart-incl.local
- Domain: chart-incl.local

### Windows activation
- Windows is activated
- Product ID: 5604L-011-2038857-86897

[Learn more online...](#)
EUROPOWER EP4000
Professional 4,000-Watt Stereo Power Amplifier with ATR (Accelerated Transient Response) Technology

- 2 x 2,000 Watts into 2 Ohms; 2 x 1,400 Watts into 4 Ohms; 4,000 Watts into 4 Ohms (Bridge mode)
- ATR (Accelerated Transient Response) technology for ultimate punch and clarity
- Switchable limiters offer maximum output level with reliable overload protection
- Detented gain controls for precise setting and matching of sensitivity
- Precise Power, Signal and Clip LEDs to monitor performance
- XLR and 1/4" TRS input connectors for compatibility with any source
- Professional speaker connectors and “touch-proof” binding posts support most speaker wiring systems
- Selectable low-frequency filters remove damaging infra-sound frequencies
- Independent DC and thermal overload protection on each channel automatically protects amplifier and speakers
- High-current toroidal transformer for ultra-high transient response and absolute reliability
- “Back-to-front” ventilation system including air filter for reliable operation
- "Built-like-a-tank," impact-resistant, all-steel 2U rackmount chassis
- 3-Year Warranty Program*
- Designed and engineered in Germany

© www.music-group.com/Categories/Behringer/Power-Amplifiers/Portable-Amplifiers/EP4000/p/P0A37

Tech Specs

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<tr>
<th>Parameter</th>
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<td>Watts/Side @ 4 ohms</td>
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<tr>
<td>Outputs</td>
<td>2 x speakerON, 4 x Binding Posts</td>
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Quantity Discounts

AC Electric Motor Run Capacitor RC0096 - 100 uf (mfd) 440 VAC Round HVAC

Be the first to review this product

For use with 1 and 3 phase motors. No PCBs.

Specifications

<table>
<thead>
<tr>
<th>TEMCo ID</th>
<th>RC0096</th>
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<td>Warranty</td>
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Product Description

TEMCo's AC Electric Motor Run Capacitors are designed specifically for replacement of OEM single and three phase motor capacitors. These units are housed in aluminum casing with a steel cap.

All units are PCB-free. Units are usually in stock and ship with Same Day Shipping from California. delivered within 2 to 3 Days.

Features

- No PCBs
- For use with 1 and 3 phase motors
- Heavy aluminum casing with steel cap
- Same Day Shipping - receive your order in 2 to 3 business days
- 10,000 AIC Protected

Support Information

Warranty

1 Year Capacitor Warranty (http://www.temcoindustrial.com/downloads/9/Rapids/9/product/51214187_year_capacitor_warranty.pdf) (Size: 3710.3 KB)
STAR Motors & Alternators: 1S175M/A

The 1S175M/A is a high efficiency, linear reciprocating device that can be used as a motor or alternator.

The shaft is supported by Qdrive's patented flexure design which provides a single degree of motion and eliminates contact between moving parts. This, by design, requires no maintenance and offers virtually infinite life. The 1S175M/A has a stroke limit of 17 mm and is driven by up to 750 Watts of AC power.

SPECIFICATIONS

- 750 W at 60 Hz, 15 mm stroke (600 W @ 60 Hz)
- Acoustic power as driven ~ 500W at 60 Hz
- Core impedance: 0.5 ohm DC, (7.5 @ 60 Hz)
- Stator Inductance: 68 mH
- Nominal BL product: 68 N/Ampere
- Rated Operating Voltage: 208 VAC 1a rms @ 60 Hz
- Rated Operating Current 6 Ampere rms (0.8 power factor) per motor
- Intrinsice Selfiness 50,000 N/m
- Damping, Nm 18 N-s/m (approximate)
- Moving mass 1.69 kg (approximate with 2.553” piston)
- Stroke limit 17 mm
- 175 mm diameter
STAR Motors & Alternators: 1S132M/A

The 1S132M/A is a high efficiency, linear reciprocating device that can be used as a motor or alternator.

The shaft is supported by Qdrive’s patented flexure design which provides a single degree of motion and eliminates contact between moving parts. This, by design, requires no maintenance and offers virtually infinite life. The 1S132M/A has a stroke limit of 14 mm and is driven by up to 250 Watts of AC power.

SPECIFICATIONS

- 250 W at 60 Hz, 12 mm stroke
- Acoustic power as driver ~ 220W at 60Hz
- Core impedance @ 110VAC winding: 2 ohm DC, (18 @ 60 Hz)
- Stator inductance @ 110VAC winding: 46mH
- Rated operating voltage/current: 110VAC 1a rms @ 60 Hz/4.0 A rms (0.85 power factor)
- Stroke limit 14 mm (12mm operation must be centered within 1 mm)
- Nominal BL product @ max voltage winding: 47 N/Ampere (approximate)
- Intrinsic Stiffness: 46 kN/m (approximate)
- Damping, Rm: 7.0 N-s/m (approximate)
- Moving mass 0.721kg (with 1.992" piston)
- 132 mm diameter, 80 mm long
Relevant TaSEG Documentation

B.1 Dual 132 Alternator TaSEG DeltaEC Simulation

Please note that this model has been simplified for printing purposes.
### 2S132E_22364_Matching_Solid_Model_6_for_PHD

**Engine**

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<td>P</td>
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<tr>
<td>-6.2727 e Ph(</td>
<td>P</td>
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<tr>
<td>0.0000 f</td>
<td>U</td>
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<tr>
<td>0.0000 g Ph(</td>
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**Optional Parameters**

- **helium**
  - Gas type

**1 TBRANCH** split to Stirling

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**Master-Slave Links**

| 5.5218E-03 C |U| m^3/s |
|------------|-----|
| 39.853 D Ph(|U|) deg |

**Optional Parameters**

- **757.01**
  - HtotBr W

**2 DUCT**

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**Master-Slave Links**

| 5.5218E-03 C |U| m^3/s |
|------------|-----|
| 39.853 D Ph(|U|) deg |

**Optional Parameters**

- **757.01**
  - Htot W

**3 JOIN**

| Gues | 3.9561E+05 A |P| Pa |
|-------|-------------------------------|
|       | -6.2727 B Ph(|P|) deg |
|       | 5.3933E-03 C |U| m^3/s |
|       | 39.853 D Ph(|U|) deg |
|       | 757.01 E HtotBr W |
|       | 757.01 F EdotBr W |
|       | -757.01 G EdotTr W |

**4 HX** Slotted Copper Ambient HX - Checked

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**Master-Slave Links**

| 344.31 G GasT K |
| 344.31 H TEnd K |

**5 STKDUCT**

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**Master-Slave Links**

| 344.32 H TEnd K |

**copper**

- Solid type

**6 STKSCREEN** Regenerator - Checked

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### COMPLIANCE Feedback compliance

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### DUCT Connecting Duct to Compression Space

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<td>Same 15b</td>
</tr>
<tr>
<td>144</td>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>148</td>
<td>Master-Slave Links</td>
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<tr>
<td>152</td>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>156</td>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>158</td>
<td>Master-Slave Links</td>
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<td>160</td>
<td>Master-Slave Links</td>
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### SOFTEND

<table>
<thead>
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<th>Text</th>
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<tr>
<td>165</td>
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<tr>
<td>168</td>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>172</td>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>174</td>
<td>Possible targets</td>
</tr>
</tbody>
</table>

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### MINOR Inertance top-end minor loss

<table>
<thead>
<tr>
<th>Page</th>
<th>Text</th>
</tr>
</thead>
<tbody>
<tr>
<td>173</td>
<td>Same 17a</td>
</tr>
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</table>
1.0000 c K- 1.2320E-02 C |U| m^3/s
-114.55 D Ph(U) deg
-757.01 E Htot W
-767.81 F Edot W

1.8040E-04 a Area m^2
3.6513E+05 A |p| Pa

5.1100E-02 b Perim m
-4.3996 B Ph(p) deg

-757.01 E Htot W

9.4190E-02 c Length m
1.3173E-02 C |U| m^3/s

5.7000E-04 a Area m^2
3.6535E+05 A |p| Pa

5.0000E-04 c K-
1.3173E-02 C |U| m^3/s

1.0000 D Ph(U) deg
-757.01 E Htot W

3.6513E+05 A |p| Pa

3.6513E+05 A |p| Pa

-89.028 D Ph(U) deg
-782.76 F Edot W

344.31 G T K

0.31107 A Eff

518.88 B Power

518.88 A Edot

1.7235E-02 a SurfAr m^2
3.6535E+05 A |p| Pa

5.2780E-05 b Volume m^3
-4.3099 B Ph(p) deg

1.0000 b K+
-4.3099 B Ph(p) deg

3.3462E-02 C |U| m^3/s

2.8688E-04 C |U| m^3/s

518.88 E Htot Br W

513.11 F Edot Br W

518.88 E Htot W

2.8688E-04 C |U| m^3/s

518.88 E Htot W

513.11 F Edot W

52.388 E Htot Br W

52.388 F Edot Br W

460.72 G Edot Tr W

7.6954E-06 a Area m^2
1.1635E+05 A |p| Pa

1.0000 b GasA/A
-179.01 D Ph(U) deg

3.0730E-02 c Length m
2.8643E-04 C |U| m^3/s

1.2500E-05 d y0 m
-4.0686 D Ph(U) deg

0.0000 e HeatIn W
52.388 E Htot W

2.8688E-04 C |U| m^3/s

52.388 E Htot W

460.72 G Edot Tr W

344.31 G GasT K
| 29 SOFTEND | 1.1635E+05 A |p| Pa |
|------------|--------------|------|
|            | -179.01 B Ph(p) | deg |
|            | 2.8643E-04 C |U| m^3/s |
|            | -4.0686 D Ph(U) | deg |
|            | 52.388 E Htot | W |
| Possible targets | -16.599 F Edot | W |
|            | -0.50997 G Re(z) |
|            | -4.5099E-02 H Im(z) |
|            | 344.31 I T K |

<table>
<thead>
<tr>
<th>10 IRSPkA/15132H/A Values based on SN 427 &amp; 433 Motors</th>
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<tr>
<td>7.5400E-03 a Area m^2</td>
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<tr>
<td>2.2000 b R ohms</td>
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<tr>
<td>0.0000 c L H</td>
</tr>
<tr>
<td>47.500 d BLProd T-m</td>
</tr>
<tr>
<td>0.97419 e M kg</td>
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<tr>
<td>4.1850E+04 f K</td>
</tr>
<tr>
<td>7.5000 g Rm N-s/m</td>
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<tr>
<td>3.3313 h</td>
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<tr>
<td>180.00 i Ph(I)</td>
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<td>4.8133E+05 K</td>
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<td>stainless Solid type</td>
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<table>
<thead>
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<th>31 RPN Piston 0-peak stroke (mm)</th>
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<tbody>
<tr>
<td>Targ 5.0000 a G or T</td>
</tr>
<tr>
<td>Ul mag w / 30a / 1000 *</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>12 RPN Axial load resistance, B=load reactance</th>
</tr>
</thead>
<tbody>
<tr>
<td>28.480 a G or T</td>
</tr>
<tr>
<td>-1.4952E-14 B ChngMe</td>
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</table>

<table>
<thead>
<tr>
<th>30H 30J 180 + sin * 30H 30J 180 + cos *</th>
</tr>
</thead>
<tbody>
<tr>
<td>32B RPN Load Reactance</td>
</tr>
<tr>
<td>Targ 0.0000 a G or T</td>
</tr>
<tr>
<td>32B</td>
</tr>
</tbody>
</table>

<table>
<thead>
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<th>34 UNION</th>
</tr>
</thead>
<tbody>
<tr>
<td>29 a SegNum</td>
</tr>
<tr>
<td>TargSame 29A 1.1635E+05 b</td>
</tr>
<tr>
<td>TargSame 29B -179.01 c Ph(p)</td>
</tr>
<tr>
<td>Possible targets</td>
</tr>
<tr>
<td>128.14 E Htot</td>
</tr>
<tr>
<td>16.001 F Edot</td>
</tr>
<tr>
<td>344.31 I T K</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>35 COMPLIANCE Back Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4883 a SurfAr m^2</td>
</tr>
<tr>
<td>2.1614E-03 b Volume m^3</td>
</tr>
<tr>
<td>Master-Slave Links</td>
</tr>
<tr>
<td>5.1309E-17 C</td>
</tr>
<tr>
<td>71.201 D Ph(U)</td>
</tr>
<tr>
<td>180.14 E Htot</td>
</tr>
<tr>
<td>stainless Solid type</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>36 HARDEND</th>
</tr>
</thead>
<tbody>
<tr>
<td>Targ 0.0000 a R(1/z)</td>
</tr>
<tr>
<td>Targ 0.0000 b I(1/z)</td>
</tr>
<tr>
<td>Possible targets</td>
</tr>
<tr>
<td>32B</td>
</tr>
</tbody>
</table>

<p>| Page: 5 |</p>
<table>
<thead>
<tr>
<th>No.</th>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>291</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>292</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>293</td>
<td>338.75</td>
<td>A AC Power</td>
</tr>
<tr>
<td>294</td>
<td>66.018</td>
<td>C ChangeMe</td>
</tr>
<tr>
<td>295</td>
<td>30.361</td>
<td>D ChangeMe</td>
</tr>
<tr>
<td>296</td>
<td>30.044</td>
<td>B Thr-Elec</td>
</tr>
<tr>
<td>297</td>
<td>20.044</td>
<td>A ChangeMe</td>
</tr>
<tr>
<td>298</td>
<td>20.044</td>
<td>A ChangeMe</td>
</tr>
<tr>
<td>299</td>
<td>0.2700</td>
<td>a G or T</td>
</tr>
<tr>
<td>300</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>301</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>302</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>303</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
<tr>
<td>304</td>
<td>0.0000</td>
<td>a G or T</td>
</tr>
</tbody>
</table>

Legend:
- **A**: AC power
- **B**: Thr-to-AC eff
- **C**: Alter eff
- **D**: Engine eff

RPN: 38 F 37 A / 100 * 30 G -1 * 26 F / 100 * 30 G -1 * 37 A / 100 * 30 G -1 *
B.2 Single 175 Alternator and Dual 132 Alternator TASEG Production Cost Estimates
<table>
<thead>
<tr>
<th>Description</th>
<th>Qty</th>
<th>Cost @ 50K each</th>
<th>Material</th>
<th>Function</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Vessel</td>
<td>1</td>
<td>54.59</td>
<td>Aluminum</td>
<td>Holds pressure of engine. Located around alternator and connects to piston housing.</td>
</tr>
<tr>
<td>Hot Heater Head Assembly</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heater Head</td>
<td>1</td>
<td>60.25</td>
<td>Inconel Alloy</td>
<td>Transfer heat into engine. Maintain helium pressure. Connect to regenerator.</td>
</tr>
<tr>
<td>Inner Fin</td>
<td>1</td>
<td>14.22</td>
<td>Copper</td>
<td>Transfer heat from head wall to working gas.</td>
</tr>
<tr>
<td>Outer Fin</td>
<td>1</td>
<td>26.64</td>
<td>Copper</td>
<td>Transfer heat from exhaust gas to head wall.</td>
</tr>
<tr>
<td>Thermocouple blocks</td>
<td>3</td>
<td>2.18</td>
<td>Copper</td>
<td>Monitor Head temperature.</td>
</tr>
<tr>
<td>Shield Plate</td>
<td>2</td>
<td>1.39</td>
<td>Copper</td>
<td>Shield engine seal from burner head.</td>
</tr>
<tr>
<td>Ambient Cooler Assembly</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooler Inner Fin</td>
<td>1</td>
<td>47.84</td>
<td>Copper</td>
<td>Transfer heat from working gas to cooler wall. Maintain helium pressure.</td>
</tr>
<tr>
<td>Cooler Outer Fin</td>
<td>1</td>
<td>38.87</td>
<td>Copper</td>
<td>Transfer heat from cooler wall to CH water. Equally guide CH water around cooler. Hold CH water pressure. Provide connection to water line. Connect to regenerator.</td>
</tr>
<tr>
<td>Water connection</td>
<td>1</td>
<td>0.86</td>
<td>SS 316</td>
<td></td>
</tr>
<tr>
<td>Regenerator</td>
<td>1</td>
<td>59.26</td>
<td>SS 316 Fuoror Stacked Screens</td>
<td>Store and Transfer Heat from Heater Head to Cooler</td>
</tr>
<tr>
<td>Thermal Buffer Tube</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Upper (Hot) Flow Straighter</td>
<td>1</td>
<td>8.60</td>
<td>SS 305 series Metal</td>
<td>Provide flow straightening at entrance and exit of Buffer Tube. Prevent jet-driven streaming.</td>
</tr>
<tr>
<td>Lower (Ambient) Flow Straighter</td>
<td>1</td>
<td>4.20</td>
<td>Bronze</td>
<td>Provide flow straightening at entrance and exit of Buffer Tube. Prevent jet-driven streaming.</td>
</tr>
<tr>
<td>Buffer Tube</td>
<td>1</td>
<td>14.23</td>
<td>SS 305 series or ???</td>
<td>Provide plug flow of “slug” of oscillating working fluid.</td>
</tr>
<tr>
<td>Gas Diode</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas Diode Plate</td>
<td>1</td>
<td>20.92</td>
<td>Aluminum</td>
<td>Suppress DC streaming. Provide correct resistance to enhance cycle phasing.</td>
</tr>
<tr>
<td>Membrane (if needed)</td>
<td>1</td>
<td></td>
<td>Copper</td>
<td>Suppress DC streaming.</td>
</tr>
<tr>
<td>Additional Resistance (if needed)</td>
<td>1</td>
<td></td>
<td>Aluminum</td>
<td>Provide correct resistance to enhance cycle phasing.</td>
</tr>
<tr>
<td>Linear Alternator</td>
<td>1</td>
<td>200.25</td>
<td>Qdrive</td>
<td>Houses Piston and connects back volume to engine head.</td>
</tr>
<tr>
<td>Piston Housing</td>
<td>1</td>
<td>20.30</td>
<td>Aluminum</td>
<td></td>
</tr>
<tr>
<td>Mass Balancer</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Balancer Springs</td>
<td>24</td>
<td>9.33</td>
<td>Copper</td>
<td>Allow vertical movement of the balancer mass.</td>
</tr>
<tr>
<td>Balancer Mass</td>
<td>1</td>
<td>24.74</td>
<td>Copper</td>
<td>Provide certain mass for balancing.</td>
</tr>
<tr>
<td>Balancer Ring</td>
<td>2</td>
<td>19.70</td>
<td></td>
<td>Maintain spring position.</td>
</tr>
<tr>
<td>Balancer Assembly Components</td>
<td></td>
<td>16.27</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Other Engine Parts</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Suspension System</td>
<td>1</td>
<td>10.56</td>
<td></td>
<td>Mitigate vibration from engine to appliance frame.</td>
</tr>
<tr>
<td>Engine Mounting frame</td>
<td>1</td>
<td>15.21</td>
<td></td>
<td>Align engine w.r.t. burner. Secure for transport.</td>
</tr>
<tr>
<td>Power Cable &amp; Power Cable Feed-through</td>
<td>1</td>
<td>172.21</td>
<td></td>
<td>Connect electrically to grid box. Provide electrical connection through pressure vessel wall. Maintain helium pressure.</td>
</tr>
<tr>
<td>Grid Box</td>
<td>1</td>
<td>96.30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Engine Burner</td>
<td>1</td>
<td>111.11</td>
<td></td>
<td>Seal combustion chamber. Prevent exhaust gas leakage to living space. Uniformly heat engine head.</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>1054.44</td>
<td></td>
<td></td>
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</table>
## Dual 132 TaSEG Cost Summary

<table>
<thead>
<tr>
<th>Descriptions</th>
<th>Qty</th>
<th>Cost @ 50K/engine/Year</th>
<th>Material</th>
<th>Function</th>
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</thead>
<tbody>
<tr>
<td>Pressure Vessel</td>
<td>1</td>
<td>54.59</td>
<td>Aluminum</td>
<td>Holds pressure of engine, located around alternator and connects to piston housing.</td>
</tr>
<tr>
<td>Hot Heat Head Assembly</td>
<td>1</td>
<td>68.25</td>
<td>Inconel Alloy</td>
<td>Type TBD. Transfer head into engine, maintains Helium pressure. Connect to regenerator.</td>
</tr>
<tr>
<td>Outer Fin</td>
<td>1</td>
<td>26.64</td>
<td>Copper</td>
<td>Transfer head from exhaust gas to head wall.</td>
</tr>
<tr>
<td>Thermocouple blocks</td>
<td>3</td>
<td>2.13</td>
<td>SiC</td>
<td>Monitor head temperature.</td>
</tr>
<tr>
<td>Shield Plate</td>
<td>1</td>
<td>1.39</td>
<td>Hi-temp silicon</td>
<td>Shield engine seal from burner head.</td>
</tr>
<tr>
<td>Scaling for size change</td>
<td></td>
<td>0.43</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ambient Cooler Assembly</td>
<td></td>
<td></td>
<td>SS 316</td>
<td>Store and transfer heat from heater head to cooler.</td>
</tr>
<tr>
<td>Cooler Inner Fin</td>
<td>1</td>
<td>47.84</td>
<td>Copper</td>
<td>Transfer heat from working gas to cooler wall. Maintains Helium pressure.</td>
</tr>
<tr>
<td>Cooler Outer Fin</td>
<td>1</td>
<td>38.87</td>
<td>Copper</td>
<td>Transfer heat from cooler wall to CH water.</td>
</tr>
<tr>
<td>Water connection</td>
<td>1</td>
<td>0.86</td>
<td>SS 316</td>
<td>Hold CH water pressure. Provide connection to water line.</td>
</tr>
<tr>
<td>Scaling for size change</td>
<td></td>
<td>0.43</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Regenerator</td>
<td>1</td>
<td>59.26</td>
<td>SS 316 or Stacked Screens</td>
<td>Store and transfer heat from heater head to cooler.</td>
</tr>
<tr>
<td>Scaling for size change</td>
<td></td>
<td>0.43</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lower (Ambient) Flow Straighter</td>
<td>1</td>
<td>4.30</td>
<td>Bronze</td>
<td>Provide flow straightening at entrance and exit of Buffer Tube. Prevent jet-driven streaming.</td>
</tr>
<tr>
<td>Buffer Tube</td>
<td>1</td>
<td>14.23</td>
<td>SS 300 series or SS 316</td>
<td>Provide plug flow of &quot;slug&quot; of oscillating working fluid.</td>
</tr>
<tr>
<td>Scaling for size change</td>
<td></td>
<td>0.61</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas Diode</td>
<td>2</td>
<td>41.84</td>
<td>Aluminum</td>
<td>Suppress DC streaming. Provide correct resistance to enhance cycle phasing.</td>
</tr>
<tr>
<td>Membrane (if needed)</td>
<td>1</td>
<td>1.39</td>
<td>Latex</td>
<td>Suppress DC streaming.</td>
</tr>
<tr>
<td>Additional Resistance (if needed)</td>
<td>1</td>
<td>1.39</td>
<td>Aluminum</td>
<td>Provide correct resistance to enhance cycle phasing.</td>
</tr>
<tr>
<td>Linear Alternator</td>
<td>1x132 (CFIC)</td>
<td>296.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piston Housing</td>
<td>2</td>
<td>59.59</td>
<td>Aluminum</td>
<td>Houses piston and connects back volume to engine head.</td>
</tr>
<tr>
<td>Piston Bore</td>
<td>2</td>
<td>59.59</td>
<td>Aluminum</td>
<td></td>
</tr>
<tr>
<td>Engine Mounting frame</td>
<td>1</td>
<td>15.21</td>
<td>Aluminum</td>
<td>Align engine or j.l. burner. Secure for transport.</td>
</tr>
<tr>
<td>Power Cable &amp; Power Cable Feed-through</td>
<td>1</td>
<td>172.21</td>
<td>Aluminum</td>
<td>Connect electrically to grid box. Provide electrical connection through pressure vessel wall. Maintain Helium pressure.</td>
</tr>
<tr>
<td>Grid Box</td>
<td>1</td>
<td>96.30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Engine Burner</td>
<td>1</td>
<td>111.11</td>
<td></td>
<td>Seal combustion chamber. Prevent exhaust gas leakage to living space. Uniformly heat engine head.</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>413.39</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
B.3 Dual 132 TaSEG Bolt Calculations
**Bolt-in-tension Stress Estimator**

**INPUT (all units inches):**

<table>
<thead>
<tr>
<th>Metric equiv</th>
<th>Bolt Material (ST,SS,AL,BR):</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
</tr>
</tbody>
</table>

- **Yield Stress (ksi):** 135
- **Ultimate Stress (ksi):** 135
- **Modulus (ksi):** 30000

<table>
<thead>
<tr>
<th>Bolt Geometry:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum Diameter (inch): 0.3073</td>
</tr>
<tr>
<td>Active Length (inch): 2.792</td>
</tr>
<tr>
<td>Applied Torque (in-lb): 236</td>
</tr>
<tr>
<td>Mean Pressure Force/Bolt, if Any (lbf): 169.16</td>
</tr>
<tr>
<td>Total Hook Length (inch): 2.792</td>
</tr>
<tr>
<td>Contact Length (inch): 0.625</td>
</tr>
<tr>
<td>Thread Major Diameter (inch): 0.375</td>
</tr>
</tbody>
</table>

**Dynamic Load:**

- **Frequency (Hz):** 65
- **Stroke (inch):** 0
- **Mass @ 1 End (lbm):** 0
- **Pressure Wave Force/Bolt, if Any (lbf): 169.16 |

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<th>Pressure Wave Force/Bolt, if Any (lbf):</th>
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<tr>
<td>169.16</td>
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**Clamped Material (ST,SS,AL,BR):**

- **Clamped Area (in²):** 0.88357293 |
- **Clamped Length (inch):** 1.14 |
- **Clamped Modulus (ksi):** 193 GPa

**Clamped Ultimate Stress (ksi):** 19

**OUTPUT**

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<th>Metric equiv</th>
<th>Bolt Stiffness (lb/in²):</th>
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<td>796931 139513171 N/m</td>
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<th>Bolt Preload Force (lb):</th>
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<td>3333.3 14833 N</td>
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<th>Bolt Preload Stress (ksi):</th>
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<th>Bolt Thread τ Stress (psi):</th>
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<th>Bolt Shear Safety:</th>
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<td>2.96</td>
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**Joint Separation Check: Clamped**

**Separation Safety: 2.13 corrected 101014 to include static pressure load**

**Bolt Load Share:**

- **15.8% of Total Dynamic Load**

**Bolt Alternating Stress (ksi):**

- **0.36 2.5 MPa**

**Bolt Fatigue Safety:**

- **2.96**

**Clamped Material (ST,SS,AL,BR):**

- **Clamped Area (in²):** 0.88357293 |
- **Clamped Length (inch):** 1.114 |
- **Clamped Modulus (ksi):** 28000 GPa

**Torque Force/Bolt, if Any (lbf):**

- **169.168173 753 N**

**Bolt Shear Safety:**

- **2.01**

**Torque Force/Bolt, if Any (lbf):**

- **169.168173 753 N**

**Bolt Shear Safety:**

- **2.01**

**Bolt Shear Stress (psi):**

- **4964**

**Bolt Shear Stress (psi):**

- **4964**

**Bolt Fatigue Safety:**

- **22.29**

**Bolt Fatigue Safety:**

- **22.29**

**o-ring outside diameter (inches):**

- **2.20**

**total pressure load on bolts (lbs):**

- **29486.46**

**Material Specs:**

- **303 SE Stainless Steel, annealed, bar.pdf**
- **303 SE Stainless Steel, cold drawn, bar.pdf**
- **A320A320M-SS bolt spec.pdf**
- **A582A582M-SS materials303&FreeMach.pdf**

**NOTE:** Fatigue safety figured on limit line at Su/2 max alt to Su max mean stress ratio of load point (mean, alt) to extension of load line to limit line

Safety Factors Should be more than 1.5, MUST be more than 1.2
### Input (All Units Inches)

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<thead>
<tr>
<th>Material</th>
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<td>Ultimate Stress (ksi)</td>
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<td>Yield Stress (ksi)</td>
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<tr>
<td>Bolt Material (ST, SS, AL, BR)</td>
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</table>

### Bolt Geometry:

| Minimum Diameter (inch) | 0.3299 |
| Active Length (inch) | 0.6 |
| Applied Torque (in-lb) | 335.012 |
| Mean Pressure Force/Bolt, if Any (lbf) | 13119 |
| Thread Engagement length (in) | 0.3 |
| Bolt Geometry: | |

### Dynamic Load:

| Frequency (Hz) | 60 |
| Stroke (inch) | 0.70866142 |
| Mass @ 1 End (lbm) | 0 |
| Mass @ 1 End (kg) | 0.00 |
| Pressure Wave Force/Bolt, if Any (lbf) | 294.80437 |

### Bolt Shear Safety:

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<tr>
<th>Material</th>
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<td>Copper C102 annealed</td>
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<td>Acetal (Delrin)</td>
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<tr>
<td>Alumina</td>
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### Output

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<th>Bolt Material (ST, SS, AL, BR)</th>
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<tr>
<td>Bolt Shear Force (lbf):</td>
<td>2948.0437</td>
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<td>Bolt Shear Stress (ksi):</td>
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<td>Bolt Shear Stress (psi):</td>
<td>5.3 MPa</td>
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<td>Bolt Shear Safety:</td>
<td>2.95</td>
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</table>

### Clamped Material (ST, SS, AL, BR)

| Clamped Area (in^2) | 0.88357293 |
| Clamped Length (inch) | 0.00 |
| Clamped Modulus (ksi) | 1000 |
| Hole Thread Stress (psi): | 9858 |
| Hole Thread Shear Safety: | 18.26 |

### Make Entries in Boxes Only

**Material Specs**
- 203 SE Stainless Steel, annealed, bar.pdf
- 303 SE Stainless Steel, cold drawn, bar.pdf
- A320A320M-SS bolt spec.pdf
- A582A582M-SS materials303&FreeMach.pdf

**NOTE:** Fatigue safety figured on limit line at Su/2 max alt to Su max mean stress ratio of load point (mean, alt) to extension of load line to limit line

Safety Factors Should be more than 1.5, MUST be more than 1.2
B.4 Dual 132 TaSEG Anti-Drift Piston Drawing
NOTES:

1. MATERIAL: 6061-T6 ALUMINUM

2. FINISHED PARTS SHALL BE CLEAN AND FREE FROM FOREIGN MATTER AND OILS.

3. PARTS SHALL BE PACKED AND BAGGED. LEGIBLY MARK BAG WITH PART NUMBER.

ANTI-DRIFT PISTON

CHAMFER 10 X 45°

BREAK ALL SHARP EDGES R .005 MAX.

BAG AND TAG ALL PARTS UNLESS OTHERWISE NOTED.

THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF CFIC, INC. ANY REPRODUCTION IN PART OR WHOLE WITHOUT THE PRIOR WRITTEN PERMISSION OF CFIC IS PROHIBITED.

FINISH: NONE

SCALE 1:1

DIMENSIONS ARE IN INCHES

TOLERANCES:
- FRACTIONAL: ± .005
- TWO PLACE DECIMAL: ± .001

SURFACE FINISH:
- 63

INTERPRET GEOMETRIC TOLERANCING PER ASME Y14.5M.

REV: S

Sheet 1 of 1

DATE: 2/19/2015

APPROVED: DEC

ZONE REV. DESCRIPTION DATE APPROVED
A RELEASE FOR MANUFACTURE 2/19/2015 DEC

DRAWN: 7/22/2005

QUALITY ASSURED: 3/12/2005

ENG APVD: DATE

BAG AND TAG ALL PARTS UNLESS OTHERWISE NOTED.
PWG drift calcs:

Target Pm: 40 bara

\( \gamma = 1.67 \) Shape 10 target \( P_1 \) 3.6558 bar, pk

\( \Delta P = 3.857 \) in radial gap = 1 mils, or 0.0000254 meters

Pressure flex \( \frac{df}{dp} = -0.0045 \) mil/bar

FEA \( \Delta P = 4.8193 \) bar, pk

\( A_p = 0.00754 \) m\(^2\)

\( \mu = 2.00 \times 10^{-4} \)

\( V_c = 1.355 \) liter

\( \frac{dc}{dx} = 0 \) (mil/mm)

\( \frac{frac \ off \ ctr}{L/2} = 0 \)

\( L_{\text{seal}} = 31.75 \) mm

\( \frac{V_b}{2} = 2.12850 \) liter (actually half, cuz one side)

\( \frac{\Delta P}{dP} = 0 \) mil/mm

\( K_m = 41.85 \) kN/m

\( \frac{p_1}{p_b} = 3.14207134 \)

\( \frac{p_1}{p_m} \times x_1 \approx 0.009 \) mm\(^{-1}\)

Assumed (one side) \( \frac{p_b}{p_m} \times x_1 \approx 0.00295 \) mm\(^{-1}\)

Predicted \( p_1/p_b \), pred. 3.14

Inputs

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The calculations use these values to find the theoretical drift--if you know

-2

0

1

2

3

4

5

Calculated Drift (mm)

Compression Space Dynamic Pressure (bar, peak)

Predicted drift of pistons

Theory (basic)

sagging ~50% off-center

sag + 2 || bypass

no sag + 2 || bypass
1. ANODIZE BLACK HARD COAT
THICKNESS = .0005 TO .001 INCH
DIMENSIONS APPLY AFTER COATING
2. FINISHED PARTS SHALL BE CLEAN AND FREE FROM FOREIGN
MATTER AND OILS.
3. PARTS SHALL BE PACKED AND BAGGED. LEGIBLY MARK BAG
WITH PART NUMBER.
4. MATERIAL: 6061-T6 ALUMINUM

NOTE:
Bibliography

[1] Microgen Engine Corporation, 


[8] Lord Rayleigh (J. W. Strutt), The Theory of Sound (Dover, New York, 1945) 2nd ed., 2, Section 322g.


[63] Los Alamos National Laboratory, www.lanl.gov/thermoacoustics/DeltaEC


Vita
Douglas A. Wilcox Jr.
Douglas A. Wilcox II
4490 Pleasant Woods Drive
Cumming, GA 30028

EDUCATION

Doctor of Philosophy in Acoustics, The Pennsylvania State University. August 2017

Master of Science in Acoustics, The Pennsylvania State University, December 2011

Bachelor of Science in Mechanical Engineering, Magna cum laude, University of Hartford, Concentration in acoustics; minor in mathematics, May 2005

PROFESSIONAL EXPERIENCE

Development Engineer, Chart Industries - Qdrive, Troy, NY, August 2012 – Current

- Principal investigator on the development of a thermoacoustic engine including analysis, mechanical design and experimental validation.
- Principal engineering and project leader for a low cost acoustic-Stirling cryocooler for use in a residential oxygen liquefier appliance.
- Mentor to numerous undergraduate summer interns from various institutions.

Senior Pre-Development Engineer, Bosch Thermotechnology, Deventer, The Netherlands, November 2009 – July 2012

- Project manager and lead developer of energy research subsidy project.
- Wall-hung condensing boiler combustion management system development project leader.
- Corporate advisor to Master’s and PhD graduate students.
- Responsible for pre-development of next generation Free-Piston Stirling engine based micro-combined heat and power appliance.

AWARDS, PATENTS, PUBLICATIONS AND PRESENTATIONS POTPOURRI

Awarded €1 million subsidy from the Dutch Ministry of Economic Affairs, Agriculture and Innovation to investigate thermoacoustic energy conversion technology appropriate for micro-combined heat and power appliances.

Commendation design award from the Technical Committee on Architectural Acoustics of the Acoustical Society of America and the National Council of Acoustical Consultants for the design of a drama theater complex located in an urban mixture development, 2005.

