EXPERIMENTAL INVESTIGATION OF A THERMOACOUSTIC-STIRLING
ENGINE ELECTRIC GENERATOR WITH GEDEON STREAMING
SUPPRESSION

A Thesis in
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Abstract

Thermoacoustic engines, or acoustic heat engines, can efficiently convert high temperature heat into acoustic power while maintaining a simple design with few moving parts. Previous experiments have demonstrated that a thermoacoustic engine based on ideally reversible heat transfer and traveling acoustic waves can convert thermal power to acoustic power with a conversion efficiency of 30%. However, these experiments on thermoacoustic engines have also relied on internal heater embedded hot zones placed directly inside the pressure vessel to apply heat to the working fluid. While this method of heat input is acceptable for examining the fundamental acoustic and thermodynamic cycles in a laboratory setting, it does not address the challenging design of a practical hot heat exchanger suitable for integration with a gas burner, which is required for a commercial product.

Recently, a thermoacoustic-Stirling engine electric generator, or TaSEG, consisting of a thermoacoustic engine coupled to a set of electrodynamic alternators has been developed and tested. This TaSEG was capable of producing 100 W of electrical output power with an overall thermal-to-electric efficiency of $\eta_{T-E}=7\%$. While the demonstrated overall efficiency has not been very high (for reasons that are mostly understood), the design has moved this technology away from laboratory prototypes and toward a commercially viable product configuration. One challenge left unaddressed in the aforementioned TaSEG is a method for suppressing so-called Gedeon streaming, or traveling-wave streaming, which is compatible with commercial mass-production. Previous tests on this TaSEG, as with other thermoacoustic engines, employed either an externally adjustable flow element (“jet pump”) or a latex barrier. The former involves an adjustment rod that passes through the pressure vessel via a sliding seal; the latter is a flexing part with an unknown lifetime and failure rate. This thesis describes the design and testing of a functional equivalent to the jet pump (“jet plate”), which has the durability of the jet pump, is just as compact as the latex barrier, and requires no sliding seal through the pressure vessel wall, all of which make it a viable option for a mass-produced TaSEG.

Additionally, several other experimental investigations of the TaSEG, including more accurately characterizing the heat flux into the TaSEG during steady state operation, ribbing of the power pistons, and exploring the TaSEG’s thermodynamic cycle phasing are briefly presented.

Based on the TaSEG’s experimental performance results, recommendations for future work that might improve the overall efficiency of the TaSEG are presented.
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List of Symbols

$\eta_{T-E}$  Thermal-to-Electric efficiency, p. iii

$\dot{W}$  Time-averaged Work Flow, p. 1

$T_C$  Thermal Source/Sink, Cold Temperature, p. 1

$T_H$  Thermal Source/Sink, Hot Temperature, p. 1

$\dot{Q}_C$  Time-Averaged Cold Heat Flow, p. 1

$\dot{Q}_H$  Time-Averaged Hot Heat Flow, p. 1

$\eta$  Efficiency p. 2

$\eta_C$  Carnot’s Efficiency, p. 2

$R_{alt}$  Alternator Resistance, p. 10

$X_{alt}$  Alternator Reactance, p. 10

$c$  Speed of Sound, p. 9

$f$  Frequency, p. 9

$\lambda$  Wave Length of Sound, p. 9

$T_m$  Mean Temperature, p. 11

$\rho_m$  Local Mean Density of the Working Fluid, p. 11

$U$  Volumetric Velocity, p. 11

$\tau$  Ratio of Absolute Temperatures at the Regenerator Ends, p. 11

$U_{1,h}$  Volumetric Velocity at the Hot End of the Regenerator, p. 11

$U_{1,c}$  Volumetric Velocity at the Ambient End of the Regenerator, p. 11

$p_{1,c}$  Pressure at Cold Side of Regenerator, p. 11

$U_{1,fb}$  Volumetric velocity in the Feedback Intertance, p. 11
\( R \)  Regenerator Resistance, p. 11  
\( L \)  Feedback Inertance, p. 11  
\( C \)  Feedback Compliance, p. 11  
\( \mu \)  Average Coefficient of viscosity, p. 11  
\( L_r \)  Length of the Regenerator, p. 11  
\( r_h \)  Hydraulic Radius of the Regenerator, p. 11  
\( A_r \)  Cross-sectional Area of the Regenerator, p. 11  
\( L_1 \)  Length of the Inertance, p. 11  
\( A_1 \)  Cross-sectional Area of the Inertance, p. 11  
\( V_c \)  Gas Volume of the Compliance, p. 11  
\( \gamma \)  Ratio of Isobaric to Isochoric Specific Heats of the Working Fluid, p. 11  
\( p_m \)  Mean Pressure of Working Fluid, p. 11  
\( \phi_{pU} \)  Phase Angle Between Acoustic Pressure and Volumetric Flow, p. 12  
\( p_1 \)  First-order Acoustic Pressure, p. 12  
\( \dot{E}_2 \)  Time-averaged Acoustic Power, p. 12  
\( \dot{E}_c \)  Acoustic Power Flowing into the Cold End of the Regenerator, p. 13  
\( \dot{E}_{fb} \)  Time-averaged Power Fed Through Inertance, p. 13  
\( p_{1,h} \)  Pressure at Hot Side of Regenerator, p. 13  
\( \dot{E}_{h} \)  Time-averaged Acoustic Power Flowing Out of the Hot Side of the Regenerator, p. 13  
\( \dot{E}_{alt} \)  Time-averaged Acoustic Power Flowing to Alternators, p. 13  
\( \dot{Q}_h \)  Heat Input to Hot Heat Exchanger, p. 13  
\( u \)  Velocity, p. 23  
\( \overline{u} \)  Time-Averaged Velocity, p. 23  
\( \rho \)  Density of the Working Fluid, p. 23  
\( \alpha \)  Phase Angle, p. 24  
\( \phi \)  Volumetric porosity of the regenerator, p. 23  
\( Re \)  Reynolds Number, p. 23  
\( D_h \)  Hydraulic Diameter, p. 23  
\( \rho_1 \)  Local First Order Mean Density, p. 23
\( c_p \)  Isobaric Heat Capacity per Unit Mass, p. 24
\( \Delta p_2 \)  Required Pressure Drop Across Regenerator to Suppress Gedeon Streaming, p. 25
\( \dot{M}_2 \)  Second Order Mass Flow, p. 24
\( \mu(T) \)  Viscosity of Working Fluid as a Function of Local Regenerator Temperature, p. 25
\( \mu_a \)  Viscosity of Working Fluid at Ambient End of the Regenerator, p. 25
\( T_a \)  Local Temperature at Ambient End of the Regenerator, p. 25
\( K \)  Minor-loss Coefficient, p. 27
\( K_{exp} \)  Expansion Minor-loss Coefficient, p. 27
\( K_{con} \)  Contraction Minor-loss Coefficient, p. 27
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Dedication

I would like to dedicate this thesis to my wife Kristina, who has followed me from one country to the next on my nomadic quest to develop thermoacoustic-Stirling engine technology, while providing constant love and support for me to pursue this dream.
I think we've got enough information now, don't you?

All we have is one 'fact' you made up.

That's plenty. By the time we add an introduction, a few illustrations, and a conclusion, it will look like a graduate thesis.
Chapter 1

Introduction

1.1 History and Governing Principles

Elementary thermodynamics courses often begin with the two basic types of heat engines shown in Figure 1.1. 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.[1]

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.

![Diagram of heat pump and prime mover](image)

Figure 1.1. The two types of heat engines: heat pumps often referred to as “refrigerators” and prime movers often referred to as “engines.”
Applying the law of conservation of energy to these heat engines leads provides a statement of the first law of thermodynamics, which states

\[ \dot{Q}_H - \dot{Q}_C - \dot{W} = 0. \tag{1.1} \]

The second law of thermodynamics governs the entropy generation of the engine, depending on its intended operation, and restricts the generated entropy to be positive or zero. During steady state operation of the engine, the net entropy increase in the reservoirs is of greatest interest:

\[ \frac{\dot{Q}_C}{T_C} - \frac{\dot{Q}_H}{T_H} > 0 \text{ Prime Mover,} \tag{1.2} \]

\[ \frac{\dot{Q}_H}{T_H} - \frac{\dot{Q}_C}{T_C} \geq 0 \text{ Heat Pump.} \tag{1.3} \]

Focusing solely on prime movers, the desired output divided by the required input is the so-called “first law efficiency”, \( \eta = \frac{\dot{W}}{\dot{Q}_H} \), often simply call the efficiency of the engine. Combining Eqs. (1.1) and (1.2) and eliminating \( \dot{Q}_C \), the efficiency then becomes

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

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:

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

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, known as a reversible process. While it is important to note this special efficiency, one should understand that nature dictates that no macroscopic thermal process is perfectly reversible, therefore 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.

It has been nearly 200 years since Reverend Robert Stirling unknowingly\(^1\) made use of these laws in his pursuit to invent a heat engine that provided a safer alternative to steam engines.[2, 3] The Stirling engine is a closed-cycle regenerative heat engine that employs an “economiser,” more commonly known as a regenerator. The primary effect of the regenerator in a Stirling engine is to considerably increase the thermal efficiency by internally recovering heat that would otherwise be lost to the inherent irreversibility of the engine.

The Stirling cycle has long fascinated engineers; the ideal Stirling cycle has Carnot’s effi-

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\(^1\)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.
efficiency, and being closed-cycle, a Stirling engine can run on any heat source.\cite{41} Despite the intrinsically high efficiency and fuel flexibility of the Stirling engine, the complexities of practical Stirling engine design 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, similar to that shown in Figure 1.2, 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.\cite{4, 5} These heat-generated sounds are termed thermoacoustic oscillations.

In the \textit{Theory of Sound}, by Lord Rayleigh (J. W. Strutt), published in 1878, Rayleigh not only confirmed the results obtained by Sondhauss, but expanded on them. Rayleigh explained that acoustic power could be created 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.”\cite{6} Though Rayleigh’s qualitative understanding was correct, a quantitatively accurate theoretical understanding was not achieved until Nikolaus Rott \textit{et al.} developed the mathematics which govern acoustic oscillations of a gas in a channel having an axial temperature gradient.\cite{8, 9, 10, 11, 12, 13, 14} The conclusions of their work laid the theoretical foundation applicable to fundamental experiments involving Stirling based thermoacoustic heat pumps and thermoacoustic prime movers.\cite{15, 16}

Not long ago, thermoacoustic prime movers or engines 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. 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 a structure known as the stack, usually made up of a collection of closely spaced plates, an array of channels, or a bundle of pins. While the imperfect thermal contact produces the required phase lag necessary for the engine to produce acoustic work output, it also generates entropy, resulting in a less efficient thermodynamic cycle. Numerous attempts were made to demonstrate the generation of electric power from heat energy by means of a standing wave thermoacoustic engines.
Figure 1.3. Simple coaxial 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 removed 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 inertance and compliance to the cold end of the regenerator. This acoustic power flow is indicated by the grey path, where the 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.

One such effort successfully demonstrated the coupling of a thermoacoustic standing wave engine to an electric generator (linear alternators), however it was only capable of generating tens of watts of electricity. In contrast to standing wave engines, thermoacoustic-Stirling engines, or TaSE’s, rely on acoustic pressure oscillations being virtually in phase with acoustic particle velocity, thus these types of engines are often referred to as “traveling wave engines.”

A basic sketch of an engine of this type is shown in Figure 1.3. Like traditional Stirling engines, the conversion of heat to acoustic power occurs in the regenerator through which the working fluid oscillates. The regenerator consists of numerous small, often random, channels which smoothly span a temperature gradient established by a hot and ambient heat exchanger. The channels in the regenerator must be much smaller, compared to those of a stack, to assure that the fluid contained within them is ideally at the same temperature as the regenerator solid. This intimate thermal contact permits heat transfer with minimal entropy production, nearly doubling the achievable efficiency over stack based standing wave engines.

In 1979, Peter Ceperley realized that the time phasing that exists between pressure and velocity in the regenerator of a Stirling engine is the same as for a traveling acoustic wave. Thus, the thermodynamic cycle undergone by the fluid in a traveling acoustic wave propagating through
a regenerator resembled that of the thermodynamic Stirling cycle.[19, 20, 21] He suggested that by using acoustics to govern the motion of the gas and gas pressure it would be possible to eliminate the pistons found in traditional Stirling engines to 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. Ceperly’s publications discuss the creation of a resonantly enhanced traveling wave fields 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 the low complex ratio of pressure $p_1$ to volume flow rate $U_1$,

$$Z = p_1/U_1,$$  \hspace{1cm} (1.6)

also known as acoustical impedance[17], of the working gas lead to inherently large viscous losses and unforeseen acoustic streaming.[22, 23, 25] Both Ceperley and Yazaki et al.[26] 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, but neither anticipated the presence of acoustic streaming.

In the same manner as conventional Stirling engines, a TaSEG can accept almost any form of heat which can then be converted into acoustic power. Waste heat from internal combustion engines, by-product heat created from industrial processes, like paper production, or heat produced by traditional pre-mixed air gas home heating systems are all examples of sources that can be accepted by TaSEG’s. The output power from these types of engines can be used to drive linear alternators, which in turn generate electricity.

Both pistonless acoustic-Stirling chillers and engines, comprises a regenerator situated at the location of traveling wave phasing in a lumped element acoustic feedback loop, have been developed and tested.[28, 27] The device under investigation in this thesis is a coaxial type thermoacoustic-Stirling engine which is similar to that developed by Backhaus[15] and Luo[30]. This engine is coupled to a pair of linear alternators in an effort to generate electrical power from thermal energy.

As the author was not part of the initial design process of the Thermoacoustic-Stirling Engine Generator experimentally investigated in this thesis, the specific details pertaining to the design are only briefly discussed in Chapter 2.

The main focus of this thesis is on the the fundamental principles of Gedeon streaming and the novel method used to suppress it within the TaSEG. Backhaus and Swift[15, 17] showed that it is possible to suppress Gedeon streaming in a thermoacoustic-Stirling heat engine by making use of asymmetric hydrodynamic end effects. However, there are several important differences that set the jet plate used to suppress Gedeon streaming in the TaSEG apart from the one implemented in the engine designed by Backhaus and Swift.[15]

The Backhaus and Swift jet pump had only two slots, which were adjustable from outside the pressure vessel. Their engine also had relatively large cross-sectional areas (between 71 and
92 cm² with gradual transitions between them, gentle bends and smooth surfaces to minimize dissipation due to flow separation and viscosity.[15, 17]

Due to compact size constraints placed on the TaSEG design, to make it a viable option for a micro combined heat and power appliance[7], having large cross-sectional areas with gradual transitions was not an option. Additionally, the TaSEG’s jet plate was designed to be a passive element that is not adjustable from outside the pressure vessel. The advantage of the non-adjustable design is that it aids in proving the serial production validity of the TaSEG.

![Figure 1.4. Backhaus and Swift[28] engine design(left); TaSEG design(right). The suppression devices (jet pump and jet plate) are highlighted in orange.](image)

Another difference between the two suppression devices is their physical location in each engine. This is shown in Figure 1.4. In the Backhaus engine, the jet pump is located just above the cold heat exchanger after the feedback inertance and compliance. Because of space constraints within the TaSEG, the jet plate had to be placed below the bottom flow straightener of the thermal buffer tube before the feedback inertance and compliance.

### 1.2 Thermoacoustic-Stirling Engine-Generator Overview

The Thermoacoustic-Stirling Engine Generator or TaSEG, seen in Figure 1.5, consists of an approximately 11,700 cubic centimeter pressure vessel filled with 33 bar mean pressure helium gas. An annular stacked screen regenerator located between a hot and ambient heat exchanger surrounds the thermal buffer tube. Below the thermal buffer tube there is a device that suppresses Gedeon streaming; it is the main topic of this thesis.[23] Further down are the feedback inertance tubes, which connect the compression space in front of the two power pistons to the compliance space located below the ambient heat exchanger. These components form a lumped element feedback loop that if correctly tuned creates acoustic traveling wave phasing in the regenerator.

Additionally, this engine makes use of a pair of balanced linear alternators to convert acoustic power into electricity.[33] The use of two alternators (or power pistons) instead of a single one has the advantage of producing dynamic balancing and vibration cancellation without requiring
a separate mass balancer, as found in almost all traditional free-piston Stirling engines. The downside to the twin motor configuration is an increase in the blow-by loses in the clearance seals that can significantly reduce the overall thermal-to-electric efficiency.[31] The operating frequency is established by the impedances of the engine components and alternators. To facilitate the connection of the TaSEG to an electrical grid, its operating frequency was chosen to be 50 Hz. In the end, operating the TaSEG at this frequency required an excessive moving mass to be placed on the linear alternators suspension, which in turn caused problems with the seal gap.[24]

![Diagram of TaSEG components](image)

**Figure 1.5.** Schematic section views of the TaSEG. The compliance is an annular volume that surrounds the collimator and the inertance is the silver pipes that are outside the main engine body. These two components provide the acoustic power feedback path within the TaSEG that is required to maintain the acoustic oscillation. The acoustic power flow is indicated by the red arrows.

Figure 1.5 shows a section of the TaSEG, which provides a glimpse of the aforementioned components that form the lumped element feedback loop. This feedback loop is primarily comprises an annular “compliance” volume, or gas-spring volume (compliance being the inverse of spring stiffness), the inertance (mass) of the gas in the inertance tubes, and the primarily resistive impedance of the regenerator.
Figure 1.6. Exterior view of the acoustic-Stirling engine-generator showing the important features.

The white tube protruding from the ambient heat exchanger in Figure 1.6 carries water into the ambient exchanger, while a similar tube, hidden from view 180° from the inlet, allows water to exit the exchanger. Three pressure sensors, located on the side of the engine, measure the pressure at important locations, and two displacement sensors (only one visible in photo) monitor the stroke of the alternators. 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. This prototype TaSEG produces acoustic power from a high temperature provided by an external electrical heat source. A fraction of the acoustic power is drawn off during each “trip” around the feedback loop at the alternator-inertance junction to drive the alternator pistons producing electricity. The remainder of the acoustic power feed back into the regenerator to be re-amplified.

It should also be observed that this TaSEG has a realistic hot heat exchanger designed to accept heat from a gas burner, opposed to numerous laboratory engines which have electrically generated heat applied directly to the working fluid from inside the pressure vessel. 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 electric heater designed to provide heat to the exchanger from outside the pressure vessel.
1.3 Power Generation by Engine

All thermoacoustic heat engines are monofrequency devices and use the same basic method to amplify acoustic power. A single frequency acoustic wave is comprised 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[6], oscillatory thermal expansion and contraction of a gas can produce 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 a TaSEG, the gas oscillates with traveling wave time phasing in a regenerator with a steep temperature gradient along but opposite the direction of acoustic power flow. A regenerator is a porous material whose characteristic dimensions are small enough to ensure intimate thermal contact between its walls 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 portion of the regenerator. However, in reality there does exist a finite difference between the gas parcel temperature and the adjacent regenerator temperature, which is a loss mechanism. In the TaSEG, the regenerator consisted of an annular stack of 84 wires per inch, 0.0035” wire diameter, metal screens, stacked around the thermal buffer tube and sandwiched between the two heat exchangers. 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.

1.4 Basic Lumped Element Circuit Analysis

This section presents a general explanation of how the TaSEG’s feedback geometry is sized to ensure that the phasing between gas motion and pressure oscillations is set so that the amplitude of the wave propagating through the regenerators axial temperature gradient is amplified. This traveling wave phasing is critical if the acoustic power entering the regenerator is to be amplified.

First, 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 \( \frac{1}{4} \) of the wavelength of sound at its operating frequency. Eq. 1.7 states that the wave length of sound is the ratio of the speed of sound in the working gas \( c \) and the frequency of the sound \( f \).[40] In the case of the TaSEG, the operating frequency is \( f = 50 \) Hz and the working fluid is helium, which has a sound of speed of \( c = 965 \text{ m/s} \) at 25 °C.

\[
\lambda = \frac{c}{f}
\]
Figure 1.7. A cutaway view of the TaSEG and its electrical equivalent lumped element circuit. The TaSEG components are much shorter than \( \frac{1}{4} \) 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.

Using these values in Eq. 1.7 yields, \( \lambda = 19.3 \text{ m} \) and one quarter \( \lambda = 4.8 \text{ m} \). Comparing \( \frac{1}{4} \lambda \) to the TaSEG’s overall height of 0.34 m, length of 0.54 m and width of 0.42 m, it is clear that all of the components that make-up the TaSEG are far smaller than a quarter wave length 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 first demonstrated by Backhaus and Swift.\[28\]

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.\textit{et al.}\[27\] and patented by C.M. de Blok\[44\]. In order to easily analyze the engine, its components are replaced by idealized electrical equivalent components.\[32\] This electric circuit analog to the acoustic circuit is responsible for setting the traveling wave phasing between pressure oscillations and the motion of the working fluid within the regenerator.

Figure 1.7 shows a cross section of the TaSEG and the electric circuit analog of the simplified lumped-element acoustic feedback loop. The two feedback tubes act like lumped masses (i.e. are dominated by inertia) with a parallel combined inertance \( L \), while the volume below the cold heat exchanger acts as a gas spring (i.e. a compliant volume) of compliance \( C \), much like the neck and the bulb of a Helmholtz resonator. Hence, the feedback tubes are often called “inertance tubes” or “inertances,” and the gas-spring volume is called a “compliance”. The alternators can be modeled as a simple resistance, \( R_{alt} \), in parallel with a reactive impedance, \( iX_{alt} \). Ignoring the spatial temperature dependence of viscosity across the regenerator, and its compliance, the
regenerator and adjacent heat exchangers can be modeled as a volume velocity source and a single resistor in the simplified equivalent electrical circuit. As the regenerator is made up of tightly packed stacked screen, the flow in the regenerator is viscously-dominated, and the regenerator as a lumped element appears resistive rather than inertial or compliant. The local temperature $T_m$ in the regenerator has a linear profile from $T_h$ to $T_c$. Thus $\rho_m$ changes according to $\rho_m \propto \frac{1}{T_m}$. Ignoring the compressibility of the gas within the regenerator, conservation of first order mass flux dictates that $\rho_m U_1$ remains constant throughout the regenerator. The decrease in $\rho_m$ with increasing local temperature means that the temperature gradient across the regenerator serves to increase the $U_1$ by the ratio of the absolute temperatures at the regenerator ends, $\tau = \frac{T_h}{T_c}$. 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.

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. Applying Kirchhoff’s junction rule, which states that the algebraic sum of the currents flowing into a junction is equal to the sum of the currents flowing out of the same node, to the lumped element circuit’s compliance node yields

$$-i \omega C p_{1,c} = U_{1,c} - U_{1,fb}, \quad (1.8)$$

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

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

$$-i \omega L U_{1,fb} = U_{1,c} R. \quad (1.9)$$

The $R$, $L$, and $C$ in Eqs. (1.8) and (1.9) are

$$R = \frac{6 \mu L_r}{A_r r_h^2}, \quad L = \frac{\rho_m L_l}{A_l}, \quad C = \frac{V_c}{\gamma \rho_m}, \quad (1.10)$$

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, $L_l$ is the length of the inertance, $A_l$ is the cross-sectional area of the inertance, $V_c$ is the compliant gas volume, $\gamma$ is the ratio of isobaric to iso-thermal specific heats, $\tau$ is the ratio of the pressures at the two ends of the regenerator, $i$ is the imaginary unit, $\omega$ is the angular frequency, $U_1$ is the volumetric velocity, $\rho_m$ is the mean gas density, and $T_m$ is the mean temperature. The subscript “m” signifies a mean, time invariant value of a variable, “1” signifies a variables which is first order in acoustic amplitude and “2” signifies a variable that is second order in acoustic amplitude.\[2\] Analogous to volume velocity in acoustic domain.
isochoric specific heats of the working fluid, and $p_m$ is the mean pressure.[17] For the TaSEG discussed in this thesis $R=370,000 \frac{Pa\cdot s}{m^3}$, $L=9,600 \frac{kg}{m^4}$, and $C=0.00036 \frac{m^3}{Pa}$.

Combining Eqs. (1.8) and (1.9) and solving for the volume velocity entering the regenerator yields Eq. (1.11):

$$U_{1,c} = \frac{\omega^2 LC p_{1,c}}{R(1 + \frac{\omega L}{R})}. \tag{1.11}$$

If the magnitude of the feedback inerance impedance $4\omega L$, is small compared to the regenerator resistance $R$, Eq. (1.11) can be further simplified yielding

$$U_{1,c} \approx \frac{\omega^2 LC p_{1,c}}{R} \tag{1.12}$$

which shows that the volume velocity and the pressure at the cold end of the regenerator are significantly in-phase, corresponding to traveling wave phasing at the regenerator.[20] As Eq. 1.12 shows, the magnitude of $U_{1,c}$ is set by the magnitudes of all three impedances: $R$, $\omega L$, and $\frac{1}{C\omega}$. Rearranging Eq. 1.12 yields

$$Z_{1,c} = \frac{p_{1,c}}{U_{1,c}} \approx \frac{R}{\omega^2 LC}, \tag{1.13}$$

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

For a pure traveling wave $Z = \frac{\rho_m c}{A}$ 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 inerance $L$, compliance $C$, and regenerator resistance $R$, should be selected so that $|\frac{p_{1,c}}{U_{1,c}}|$, at the cold side of the regenerator, is approximately 15 to 30 times $\frac{\rho_m c}{A}$. In the case of the TaSEG discussed in this thesis $|\frac{p_{1,c}}{U_{1,c}}| = 81 \frac{MPa\cdot s}{m^3}$, while $Z = 1.5 \frac{MPa\cdot s}{m^3}$.

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.[28] Most introductory acoustics textbooks define acoustic power as

$$\dot{E}_2(x) = \frac{\omega}{2\pi} \int Re[p_1(x)e^{(i\omega t)}]Re[U_1(x)e^{(i\omega t)}]dt = \frac{1}{2} Re[p_1\dot{v_1}] = \frac{1}{2} |p_1||U_1|\cos \phi_{PU}, \tag{1.14}$$

where the tilde represents the complex conjugate, $\phi_{PU}$ 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. Making use of this definition and multiplying Eq. 1.12 by $p_{1,c}$ gives the acoustic power flowing into the cold end of the regenerator:

$$\dot{E}_c = \frac{1}{2} p_{1,c} U_{1,c} \approx \frac{\omega^2 LC |p_{1,c}|^2}{2R}, \tag{1.15}$$

$^4$See Appendix A for further discussion of $\omega L << R$. 


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 cold side of the regenerator, $E_c = E_fb$. 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 $p_{1,c} \approx p_{1,h}$.

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

$$\dot{E}_h \approx \tau \dot{E}_c.$$  \hfill (1.16)

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

$$\dot{E}_{alt} \approx (\tau - 1) \dot{E}_c.$$  \hfill (1.17)

To determine $\tau$ to establish how the engine parameters set the operating temperatures, further equivalent circuit analysis can be performed. Substituting Eq. (1.15) into Eq. (1.17) and knowing that $\dot{E}_{alt} = \frac{|p_{1,c}|^2}{2R_{alt}}$, $\tau$ then becomes

$$\tau \approx \frac{1}{\omega^2 LCR_{alt}}.$$  \hfill (1.18)

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 fact combined with Eqs. (1.15) and (1.16) yields $|p_{1,c}|^2$ as a function of $\dot{Q}_h$:

$$|p_{1,c}|^2 \approx \frac{2R\dot{Q}_h}{\tau \omega^2 L C}.$$  \hfill (1.19)

Eqs. (1.18) and (1.19) 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 (alternators) 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 which 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 components have on these phasors. Research has shown that they can be very sensitive to non-ideal behavior of the various engine components. Also, while the lumped element regenerator model presented above is adequate for a simplified fundamental analysis of the TaSEG, a more complex model should be used in practice 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 $\delta_\nu$, to change within the regenerator as a function of length.\textsuperscript{[17]} Finally, in the “real world” the interaction between the load presented by the linear electrodynamic alternators
to the engine portion of the TaSEG requires a more in-depth evaluation.[42, 36]

1.5 Thesis Overview

Chapter 2 discusses Gedeon streaming, with theory and techniques to suppress it.[27, 23] In Chapter 3, the sensors and systems required to simultaneously measure the TaSEG’s performance are examined, including discussions of their calibrations. Chapter 4 outlines the Gedeon streaming suppression, jet plate design and its experimental performance. Chapter 4 also discusses several experiments that examine the TaSEG’s thermodynamic cycle phasing resistance, the heat flux into the TaSEG and the seal losses associated with the TaSEG’s linear alternators. Chapter 5 draws conclusions based on the outcome of the TaSEG experiments, makes recommendations for improving the TaSEG’s overall thermal-to-electric efficiency and explores applicable future work.
Chapter 2

Design Overview & Gedeon Streaming

2.1 Introduction

This chapter provides an overview of the design constraints and processes behind the TaSEG prototype, including the balance between the necessary attributes that make this engine a viable “real world” product and the technical goals that advance the fundamental understanding of acoustic-Stirling engines. Due to the fact that the TaSEG was a proof-of-concept unit, numerous manufacturing techniques which would be implemented in a manufactured production unit were not utilized in this design. Despite this, there are several design parameters which clearly distinguish this engine from other previously designed laboratory acoustic-Stirling engines.[15, 30, 35]

2.2 Physical Constraints and Simulation Results

Many acoustic-Stirling engines to date have been based on the toroidal geometry established by Backhaus and Swift[15] and de Blok,[44]. Most of these engines make use of an electric heater element placed inside the pressure vessel that applies heat directly to the working fluid. While this method of heat input is acceptable for examining the fundamental acoustic and thermodynamic cycles in a laboratory setting, it does not address the challenging design of a practical hot heat exchanger that is required in a commercial product. One of the main points of the design of the TaSEG prototype was to provide the attributes of a real micro combined heat and power (mCHP) product. Therefore, the requirement of accepting heat from an external heat source was placed on the hot heat exchanger design. This means that 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, in a commercial micro combined heat and power appliance, was a dominant factor in the overall physical layout of the TaSEG prototype.
Figure 2.1 shows both the Backhaus thermoacoustic-Stirling engine geometry (left)[15] 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 much greater power density, even if the resonator volume in the Backhaus engine is neglected.[15]

The need for a hot heat exchanger having the ability to accept external heat, coupled with the volume constraints placed on the TaSEG in order for it to compete with internal combustion and free-piston Stirling engines, restricted the sizing of components used within the design. Since the TaSEG prototype served as a baseline design for a practical commercial product, a radical departure from all previous work was required.

In addition to the aforementioned restrictions, several more design constraints were placed on the initial prototype. This included the choice of an operating frequency of 50 Hz. Although a higher operating frequency would have ultimately been much better for the performance of the alternators and the TaSEG, the 50 Hz operating frequency was selected to allow a direct tie-in with the electrical grid. It was also decided that a pair of QDrive’s 132 STAR [33] resonant linear reciprocating alternators would be used. By using a pair of matched alternators, it is possible to get double the power of one alternator, resulting in a combined output capacity of 400 W at 50 Hz (i.e. 200 W per alternator). This selection also resulted in dynamic vibration
balancing during operation and eliminated the need to design and implement a vibration mass balancer. This gave the design an advantage over similar free-piston Stirling engine generators. The 132-size alternators are limited to a 6 mm zero-to-peak stroke and a current of 3.5 and 7 A in the two-wire stator and four-wire stator configuration, respectively. Figure 2.2 shows one of QDrive’s 132 STAR resonant linear reciprocating alternators capable of outputting 250 W at 60 Hz. Before the first design iteration, it was realized that the TaSEG’s alternator piston areas and stroke limitations would require it be optimized for maximum power density.

**Figure 2.2.** QDrives’s 132 STAR alternator. STAR alternators are linear reciprocating devices that combine a unique flexing axial suspension, with plunger-mounted high-energy magnets and robust coil-over-iron stator. They have electrical-to-acoustical efficiencies of 80–90%.

Additionally, operating temperatures of 550 °C and 60 °C were establish to set the allowable hot metal and ambient outlet water temperatures of the TaSEG’s heat exchangers. Other additional constraints represent either desirable goals or typical values stemming from past practice and included a mean pressure of 3.0 MPa with a 10% pressure wave amplitude, hairpin hot heat exchanger tubes, a streaming suppression device, either a jet plate or latex barrier, a 3” (72.6 mm) piston diameter and a total alternator moving mass of 4 kg for each alternator.

These constraints that established that baseline design envelope were revised several times throughout the design process, before the TaSEG prototype was constructed. The choice of the coaxial engine geometry, a realistic engine hot-zone, the 50 Hz operating frequency and the use of twin alternators, led to the initial design goal of a combined 400 W electrical output at a thermal-to-electrical efficiency, \( \eta_{\text{Ther} - \text{Elec}} = 15\% \).

The primary software tool used to establish baseline engine geometries and simulate engine performance was the Design Environment for Low-amplitude Themroacoustic Energy Conversion, also known as DELTAEC[38]. This software is a free computer program for modeling one-dimensional thermoacoustic devices. It numerically integrates the one-dimensional wave equation in the low-amplitude acoustic approximation through a series of user defined segments such as ducts, cones, transducers, heat exchangers, etc. DELTAEC can be used for devices ranging from a simple Helmholtz resonator[40], up to full thermoacoustic prime movers and refrigerators. This software was developed by the condensed matter and thermal physics group at Los Alamos
Figure 2.3. Preliminary design sketch of TaSEG, the inertance “feedback” tubes are not pictured. Printed with permission from QDrive.

National Laboratory in Los Alamos, NM.

Figure 2.3 shows a conceptual illustration of the TaSEG, which reflects the aforementioned design criteria. This sketch served as the baseline mechanical design, specifying critical dimensions and ensured that the DELTAEC simulations were kept within the realistic limits.

Once the baseline DELTAEC model and mechanical design were established, an iterative process was used to optimize the TaSEG design to meet the initial design goals. The first iteration yielded mixed results due to the extra oscillating flow demands required to achieve the electrical output power goal. To reach the output demands, the design had to be altered to either increase the power density so that more power was produced with less oscillating flow or the power piston diameter would have to be increased. Increasing the power density would result in an increased pressure. Whether the diameter of the piston became larger or the pressure wave amplitude is increased, the total reactive spring force would have to increase to maintain the 50 Hz operating frequency, resulting in a heavier piston. As the initial constraint on the moving alternator mass of 3 kg was already considered a heavy load for the 132 alternator frame size, it became obvious that the 50 Hz operating frequency was creating difficulties in achieving the design goals.

Additionally, the volume surrounding the alternators in the DELTAEC simulation, termed back volume, was allowed to be whatever was necessary to reduce the pressure wave in the back
space. This resulted in an estimated back volume of 4 liters per side, meaning a total back volume of 8 liters would be required. This amount of back volume was simply unacceptable due to the space constraints of a commercially viable product.

Another problem uncovered during the design iteration process was the acoustic-to-electric conversion efficiency of the alternators. The DELTAEC model suggested an efficiency of 70% at full stroke which is lower than the typical efficiencies (80% to 90%) seen when QDrive 132 alternators are used in cryogenic systems.[33] The reduced efficiency was partly due to seal losses associated with the power pistons’ seal gap between the pistons and their housings. The normal clearance seal gap achieved by QDrive for a 3” to 4” diameter piston is on the order of 25 microns. However, due to the large, 4 kg moving mass required to meet the 50 Hz operating frequency, the seal gap on the TaSEG proved to be difficult to maintain. This was because of the loading of the 132 alternator suspensions. Decreasing the seal gap and its associated losses can have a dramatic effect on the TaSEG performance. This fact is illustrated in Figure 2.4.

Based on the alternator loading concerns and because the DELTAEC simulations suggested that an increase in the operating frequency would increase both the electrical power output and thermal-to-electric efficiency, it was suggested to increase the operating frequency of the TaSEG. However, the 50 Hz operating frequency was deemed the most important constraint and was therefore not able to be changed. Since the 50 Hz operating frequency was to remain unchanged, the effects of operating the TaSEG with a higher dynamic pressure wave on top of a higher mean pressure was investigated. After several iterations it became clear that either increasing the mean

![Figure 2.4. Influence of the Seal Gap on the TaSEG’s Performance](image-url)
pressure or increasing the piston size yielded similar effects on the estimated TaSEG performance. The advantage of increasing the pressure is that it leaves the piston diameter unchanged. This has a positive effect since it is easier to achieve a tighter nominal seal gap on a smaller piston diameter.[41] However, increasing the pressure wave amplitude also increases the seal loss and at a faster rate than increasing the piston diameter. How to carefully balance these effects was critical as the scope of the project switched from simulations to mechanical design.

As the simulations progressed it was noticed that adding an acoustic resistance in the form of a “minor loss” at the junction between the thermal buffer tube and alternator increased the TaSEG’s power output. This was counterintuitive. One would expect that an additional resistance would add extra dissipation resulting in a decrease in the overall power output. It was believed that the dissipation in the added resistance is offset by the resulting phase change between volume velocity and pressure it causes. This phase change results in more acoustic power being delivered to the ambient end of the regenerator and less dissipation per unit acoustic power within the regenerator. This unusual, but beneficial acoustic resistance begs one to ponder the nature of cycle phasing in acoustic engines versus true Stirling displacer engines and the essential differences between the inertial (massy) feedback found in acoustic engines and the springy feedback found in traditional Stirling engines.[57, 17]

At this point it was decided that a higher hot-heat exchanger temperature and inclusion of the beneficial acoustic resistance would result in the best outcome. Therefore, the overall goal of 400 W of electrical output was reduced to 300 W. Additionally, the power piston seal length was increased from 0.5” to 1” in an effort to reduce the seal losses and the hot heat exchanger material was changed to Incoloy to increase the maximum allowable hot heat exchanger temperature.

As the design constraints and simulations started to converge on basic component dimensions and operating conditions for the TaSEG, the realism of the model needed to be addressed. Both the flow straighteners and the hot heat exchanger tubes were too short for practical production. However making them longer had a negative impact on the engine performance. This was especially true of the hot heat exchanger length, as adding length to the tubes would introduce additional inertance into the acoustic circuit, shifting the cycle phasing in an undesirable direction.

The regenerator length was also a particularly difficult issue mechanically, as any thermal conduction from a short hot heat exchanger to the ambient exchanger would be detrimental to the TaSEG’s performance. Optimizing and increasing the regenerator length to a more realistic design to mitigate conduction concerns resulted in a further reduction of the predicted electrical power output and lead to the use of Inconel for the regenerator shell, with its added cost.

The ambient heat exchanger design was also addressed during the simulation period. Initially, a water-to-helium tube-and-shell exchanger was to be used. While very effective, these types of

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1These types of feedbacks have no resistance, only pure reactance of either the inertial variety or the springy variety. While these types of feedback are of opposite sign in the complex plane, the dot product of pressure and volume velocity is still in the same direction and the center of the regenerator does not know the difference [41]. This is mentioned here as the identification of fundamental differences between inertial and springy feedback has important implications on future work and should be investigated.
heat exchangers can be costly to produce. Given the relatively narrow annulus of the thermal core, it was decided that a radial channel heat exchanger that relies on the thermal conduction of radial metal fins within the annulus to pull heat out of the acoustic space would be used instead. This resulted in a conduction length that was short enough to keep any radial thermal gradients low, with channels that could be made narrow enough to guarantee a small temperature difference between the metal and helium. The metal-to-water side of the heat exchanger would be a second set of circumferential circular fins. This would enable water to flow around the outer diameter of the ambient heat exchanger. DELTAEC can estimate the temperature difference between the internal helium and an isothermal surface, but not the conduction penalty for moving the heat radially outward to the water jacket. To analyze this, supplementary calculations were performed. These calculations validated the assumption that conduction across the metal fins of the ambient exchanger were not a major concern. In the end, the helium-to-metal temperature difference was the larger concern. Using realistic fin dimensions for the ambient exchanger in the DELTAEC model, resulted in a further reduction of the predicted electrical output power.

The iterative simulation and design process used to establish the TaSEG design and operating conditions serve as a reminder that thermoacoustic engines are a viable option for micro combined heat and power appliance’s electrical power module, but care must be taken during all steps to optimize the performance. Neglecting one “small” detail that may at first glance seem trivial, could be catastrophic to the overall engine performance.

2.3 Mass Streaming

In acoustics, a second order steady mass flux density or velocity superimposed on and driven by a larger first-order oscillating mass flux density or velocity is commonly referred to as “streaming.” Simply put, during the acoustic cycle streaming can be thought of as taking 106 steps forward and only 100 backwards, equivalent to the superposition of a steady forward drift term of 6 steps during each cycle of an oscillating motion having a peak-to-peak amplitude of 103. The importance of streaming in thermoacoustic devices is its ability to convectively transfer heat. This can result in either undesirable losses or deliberate heat transfer.

Various thermoacoustic devices, including orifice and feedback pulse tube refrigerators, cascaded thermoacoustic engines, thermoacoustic-Stirling engines, and traditional Stirling engines and refrigerators, have several types of undesirable first-order driven acoustic streaming phenomena. Time averaged toroidal circulation driven by boundary layer effects, which can occur in pulse tubes or thermal buffer tubes, is known as “Rayleigh streaming.” A second type of toroidal circulation, which can occur in pulse and buffer tubes of thermoacoustic devices, but is driven by insufficient flow straightening at the ends of the tube is known as “jet-driven streaming.” A third type, known as “internal acoustic streaming” within the regenerator or stack of thermoacoustic devices can also occur. The most critical form of streaming pertaining to the TaSEG, is a net time-averaged mass flow $\dot{M}$ in the vertical direction through the regenerator. This type is known as “Gedeon Streaming.” Gedeon streaming is particularly...
Figure 2.5. 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.

of interest because it convects a time averaged enthalpy flux down the regenerator (i.e. from hot to cold) wastefully removing high temperature heat from the hot heat exchanger without generating acoustic power.

2.3.1 A Closer Look at Gedeon Streaming

One of the drawbacks of the TaSEG design is the essential closed-loop feedback flow path. This path is required for the TaSEG to fundamentally function, but it also introduces the potential for a DC gas flow, illustrated in Figure 2.5.

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. [23] 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. In this case it is the TaSEG’s regenerator.

Assuming that the flow through the porous regenerator in the TaSEG occurs at a low Reynolds number, the pressure drop $\delta p$ is a linear function of the velocity $u$, or

$$u \propto \delta p. \quad (2.1)$$

This type of flow is known as Darcy flow and based on Darcy’s law, which is a proportionality relationship between the viscosity of a fluid, the pressure drop over a given distance and the instantaneous flow rate through a porous medium (i.e. regenerator).[51] Eq. 2.1 is valid for Reynolds numbers up to approximately 10.

Defining the time-averaged velocity within the regenerator $\bar{u}$, as:

$$\bar{u} = \frac{|U|}{\phi A_r}, \quad (2.2)$$

where $\phi$ is the volumetric porosity of the regenerator and the standard Reynolds number equation $Re[53]$, as:

$$Re = \frac{\bar{u} D_h}{\nu}, \quad (2.3)$$

where $\nu$ is the kinematic viscosity and $D_h$ is the hydraulic diameter. Using Eqs. 2.2 and 2.3, as well as taking the appropriate input parameters from the DELTAEC, the time-averaged Reynolds number within the regenerator is found to be around 7. While assuming Darcian flow in the TaSEG’s regenerator is not entirely true, it does serve to illustrate the generation of unbalanced DC flow.

Multiplying both sides of Eq. 2.1 by the density of the working fluid $\rho$ yields:

$$(\rho u)_{\text{mean}} \propto \rho \delta p. \quad (2.4)$$

This equation has the advantage that the quantity on the left hand side becomes a mass flow rate per unit cross-sectional area, which is a conserved quantity. If both the pressure drop across a given flow resistance and the fluid density within vary sinusoidally, as is the case in the regenerator of the TaSEG, then $\delta p$ and $\rho$ can be rewritten as,

$$\delta p = \rho_m + \rho_1 \sin(\omega t), \quad (2.5)$$

and

$$\rho = \rho_m + \rho_1 \sin(\omega t + \alpha), \quad (2.6)$$

where $\rho_m$ is the mean density of the working fluid, $\alpha$ is some phase angle, and $\rho_1$ is the amplitude of the oscillating density.

It is useful to mention the trigonometric identity
\[2 \sin(\omega t) \sin(\omega t + \alpha) \equiv \cos(\alpha) - \cos(2\omega t + \alpha),\]  
(2.7)

which states that the product of two in-phase sine waves produces a cosine wave with twice the frequency plus a constant. When the phase angle \(\alpha\) is zero, the \(\cos(\alpha)\) term is equal to one, and when \(\alpha\) is an odd multiple of \(\frac{\pi}{2}\) this term is zero. The constant term from Eq. 2.7 is the source of DC flow.[23]

Substituting Eqs. (2.5) and (2.6) into Eq. (2.4) and expanding it using the trigonometric identity defined in Eq. (2.7) yields

\[
\rho u \propto \rho_m p_m + \rho_m p_1 \sin(\omega t + \alpha) + \rho_1 p_m \sin(\omega t + \alpha) + \frac{\rho_1 p_1}{2} \cos \alpha - \frac{\rho_1 p_1}{2} \cos 2\omega t + \alpha.
\]

(2.8)

Taking the time average of Eq. (2.8) yields the DC flow rate:

\[
\rho u \propto \rho_m p_m + \frac{\rho_1 p_1}{2} \cos \alpha.
\]

(2.9)

Thus, as long as the density fluctuations are in phase with the pressure drop, a DC flow or DC pressure drop will exist across the regenerator. This DC flow, also known as “Gedeon Streaming,” is a net time averaged mass flow through the regenerator. David Gedeon showed in detail how this time averaged second order mass flux can exist in systems which contain a closed loop path such as acoustic traveling wave engines.[23]

The acoustic network of the TaSEG investigated in this thesis\(^2\), clearly provides such a path. Hence, Gedeon streaming is a major problem that must be addressed in the TaSEG. Due to the negative consequences associated with time averaged mass flow \(\dot{M}_2\) within the TaSEG, it should be suppressed to zero. Doing so will prevent a large time averaged convective enthalpy flux:

\[
\dot{Q}_{\text{Loss}} \sim \dot{M}_2 c_p (T_H - T_C),
\]

(2.10)

from flowing from the hot heat exchanger to ambient heat exchanger. Left unaddressed, this unwanted heat leak will wastefully remove heat from the hot heat exchanger without producing any acoustic power. This will result in a significant decrease of the TaSEG’s overall thermal-to-electric conversion efficiency.

Gedeon reasons that \(\dot{M}_2\) is

\[
\dot{M}_2 = \frac{1}{2} \text{Re}[\rho_1 \dot{U}_1] + \rho_m U_2,
\]

(2.11)

where \(U_2\) is the time independent second order volumetric velocity.[15] Because \(\rho_1 \propto p_1\), the first term on the right hand side of Eq. (2.11) will be nonzero whenever acoustic power flux is nonzero. This just so happens to be the case in the closed loop feedback path that exists in the TaSEG. To ensure that the two terms on the right hand side of Eq. (2.11) are balanced, enforcing \(\dot{M}_2=0\),

\(^2\)Figure 1.7
a nonzero $U_2$ must flow around the closed feedback loop and through the regenerator. Swift, et al. [27] have shown that the required pressure drop across the regenerator in an engine to drive the correct $U_2$ up through the regenerator in the low-Reynolds-number limit is

$$\Delta p_2 \approx \frac{6}{A_r r_h^2 \rho_m} \int_{\text{regenerator}} \dot{W}(x) \mu_m(x) dx,$$  

(2.12)

where $\dot{W}(x)$ and $\mu_m(x)$ are spatially dependent coefficient of acoustic power flow and viscosity in the regenerator and $r_h$ is the hydraulic radius of the regenerator, respectively. Assuming that when Gedeon streaming is suppressed there exists a linear temperature gradient across the regenerator and $\dot{W}(x) = \frac{W_c T_m T_c}{T}$, the integral of Eq. (2.12) yields

$$\Delta p_2 = \frac{6 \mu_c L_r}{(b + 2) A_r r_h^2 \rho_m} \frac{\tau^{b+2} - 1}{\tau - 1} = \frac{W_c R_o (\tau + 1) f}{2p_m},$$  

(2.13)

where $\tau = \frac{T_h}{T_c}$, $L_r$ is the length of the regenerator, $b$ is a constant which accounts for the temperature dependence of $\mu$ which is defined by $\mu(T) = \mu_a \left( \frac{T}{T_a} \right)^b$ [15], and $R_o \approx \frac{\rho_c L_r}{\tau} S_\tau^2$ is the low Reynolds number limit flow resistance of the regenerator at constant $T$ over its entire length. Literature suggests that this required pressure drop is somewhere between a few hundred to a few thousand pascals.[15, 17]

### 2.3.2 Gedeon Streaming Suppression

Several methods have been used in the past to suppress Gedeon streaming within thermoacoustic devices.[42, 50, 27] Perhaps the most common and easiest method implemented is the insertion of a diaphragm within the closed loop feedback path. This method of streaming suppression was initially implemented within the TaSEG. The diaphragm is a very simple solution as it provides a physical barrier that makes it impossible for there to exist a nonzero mass flux within the feedback path, and because it is acoustically transparent (i.e. has a very small acoustical impedance) it dissipates negligible acoustic power. Although it has been demonstrated [52] that a latex diaphragm can be used to suppress Gedeon streaming in an acoustic refrigerator and last for several million cycles without failure, this is not considered a viable solution for a “real world” hermetically sealed TaSEG which is to be mass produced. Therefore, there is a need for another method which is low cost and consists of non-moving, non lifetime-limited components.

To remove the diaphragm from the TaSEG, there is a need for a second method to impose a $\Delta p_2$ across the regenerator to suppress Gedeon streaming. One way impose such a $\Delta p_2$ is to take advantage of asymmetric hydrodynamic end effects.[28] Both jet flow and turbulence accompany the transition of high-Reynolds number flow making abrupt transitions from a small cross-sectional area to a larger area, and generates an additional pressure drop and power dissipation. This physical phenomenon is referred to as “minor loss” in most literature.[53, 54, 17, 15]

---

3“through the regenerator” here means from the hot to the ambient end of the regenerator.

4The diaphragm is an isothermal surface.
The additional pressure drop due to minor loss $\Delta p_{ml}$ is

$$\Delta p_{ml} = \frac{K \rho v^2}{2} = \frac{K \rho U^2}{2A^2},$$

(2.14)

where $v$ is a positive directional component of velocity depending on the coordinate plane assignment, $K$ is the minor loss coefficient depending on the geometry of the transition and depends on the flow conditions.

Figure 2.6. Asymmetry of outward and inward high Reynolds number flow at small tube to large space transition.

Figure 2.6 shows an example of flow asymmetry depending on the flow direction. When a gas steadily flows out of a “small” tube connected to an essentially infinite volume at a high enough Reynolds number, a jet usually occurs and kinetic energy is lost as a result of downstream turbulence caused by the jetting of the gas. In this case the minor loss coefficient $K_{out}$ is approximately 1 and independent of the geometry of the transition.[53, 54] However, as gas flows into the tube, the streamlines in the infinite volume are smoothly convergent from a wide range of directions. In this case the minor loss coefficient $K_{in}$ is strongly dependent on geometry of the edge of the hole entrance. If the edge of the entrance is sharp, $K_{in} \approx 0.5$. As the entrance is rounded $K_{in}$ will decrease until the radius of the rounding, $r$, is such that $\frac{r}{D} \geq 0.15$. Here $D$ is the diameter of the circular opening.[53, 54] For values of $r$ and $D$ which satisfy the aforementioned equation, $K_{in} \approx 0.04$. By taking advantage of these geometry effects it is possible to create a situation where the flow resistance of an abrupt transition displays an asymmetry with respect to flow direction.

The through-hole schematic shown in Figure 2.7 illustrates two different flow patterns which represent the acoustic velocity during the two halves of the acoustic cycle. In order to estimate the time-averaged pressure drop across the through-hole it is assumed that the flow at each instant in time has little memory of its time dependence, inferring that the minor loss coefficients $K_{exp}$
and $K_{\text{con}}$ have the same values in oscillating flow as they do in steady state flow. Allowing the downward direction to be positive, the second order the time-averaged pressure drop due to the through-hole is

$$\Delta p_{\text{ml}} = \frac{\omega \rho}{2\pi} \left[ \int_0^{\frac{\pi}{2}} \left( K_{\text{con},L} v_s^2 + K_{\text{exp},s} v_s^2 \right) dt - \int_{\frac{\pi}{2}}^{\pi} \left( K_{\text{exp},L} v_s^2 + K_{\text{con},s} v_s^2 \right) dt \right]. \quad (2.15)$$

Taking the instantaneous velocities to be $v_s = |v_s| \sin(\omega t)$ and $v_L = \frac{a_s}{a_L} v_s$ yields

$$\Delta p_{\text{ml}} = \frac{\omega \rho}{2\pi} \left[ \int_0^{\frac{\pi}{2}} \left( K_{\text{con},L} \left( \frac{a_s}{a_L} \right) |v_s| \sin(\omega t) \right)^2 + K_{\text{exp},s} \left( |v_s| \sin(\omega t) \right)^2 dt \right. \right.$$

$$- \left. \int_{\frac{\pi}{2}}^{\pi} \left( K_{\text{exp},L} \left( \frac{a_s}{a_L} \right) |v_s| \sin(\omega t) \right)^2 + K_{\text{con},s} \left( |v_s| \sin(\omega t) \right)^2 dt \right]. \quad (2.16)$$

Integrating Eq. 2.16 in time yields

$$\Delta p_{\text{ml}} = \frac{\rho}{8a_s^2} \left[ \left( \frac{a_s}{a_L} \right)^2 (K_{\text{con},L} - K_{\text{exp},L}) + (K_{\text{exp},s} - K_{\text{con},s}) \right], \quad (2.17)$$

where $U_{1,\text{jp}}$ is the first-order volumetric velocity amplitude through the through-hole.

Such clever control of $M_2$ to suppress Gedeon streaming is not without penalty. In addition to generating the required $\Delta p_{\text{ml}}$, the through-hole will also dissipate acoustic power. Averaging the instantaneous power dissipation, given by $\Delta p_{\text{ml}}(t) U_{\text{jp}}(t)$, over one acoustic cycle results in
the time-averaged acoustic power dissipation per hole

$$\dot{E}_{jp} = \frac{\rho}{3\pi a_s^4} \left[ \left( \frac{a_s}{a_L} \right)^2 (K_{con,L} + K_{exp,L}) + (K_{exp,s} + K_{con,s}) \right].$$ \hspace{1cm} (2.18)

Eq. 2.18 suggests that the best way to produce the desired $\Delta p_{ml}$ is to insert a jet plate having numerous through-holes, at a location within the TaSEG’s feedback loop where $|U_1|$ is small. However, care must be taken to assure that the local volumetric velocity is substantial enough to generate the required $\Delta p_{ml}$, while keeping the resulting acoustic power dissipation to a minimum.

Assuming that the jet plate impedance can be characterized as a pure acoustic resistance $R$, the time-averaged power dissipation is given by $\dot{E} = \frac{R |U_1|^2}{2}$. [15] Using this definition, the minor loss resistance of the jet plate can be defined as

$$R_{jp} = \frac{2\rho}{3\pi a_s^4} \left[ \left( \frac{a_s}{a_L} \right)^2 (K_{con,L} + K_{exp,L}) + (K_{exp,s} + K_{con,s}) \right],$$ \hspace{1cm} (2.19)

which is in addition to that due to viscous and thermal relaxation losses.

To address the lifetime concerns of the latex diaphragm while at the same time suppressing Gedeon streaming, a “jet plate” which makes use of asymmetric hydrodynamic end effects was designed using the equations discussed above to create the required $\Delta p_{ml}$ across the regenerator. In order to minimize required modifications to the current TaSEG, the existing geometrical constraints were taken into consideration and used to design the jet plate to fit nicely into the space occupied by existing latex diaphragm, as can be seen in Figure 2.8.

Simulation data from the DELTAEC model of the TaSEG[42], to which experimental performance data was matched to better than 1% for all important operational parameters, except the required heat flux\(^5\), served as the source of input variables to calculate the required pressure drop across the regenerator needed to enforce $\dot{M}_2=0$ and suppress Gedeon streaming. Using the nec-

\(^5\)Simulations predicted 25% less input heat flux than was initially measured.
Figure 2.9. Machined Aluminum Jet Plate (Top) Rapid Prototype Jet Plate (Bottom). Both jet plates are 92.0 mm in diameter and have a thickness of 18.3 mm.

Essary DELTAEC simulation parameters in Eq. 2.13, the calculated required pressure drop across the regenerator is 58 Pa. Based on this and making use of Eq. 2.17 a jet plate that can provide the required 58 Pa pressure drop can be designed. However, Swift[17] cautions that, while the calculations are qualitatively correct, there exists poor quantitative agreement between Eqs. 2.13 and 2.17 and often the actual resistance required to suppress streaming in a real TaSEG has to be several times greater than the calculated value. Because of this fact it was decided to design the jet plate with the ability to cover a large range of pressure drops and flow resistances. Based on this desire a jet plate was designed that allows simple changes to the geometry of the jet plate through-holes such as, shaping/rounding hole edges, increasing hole diameters, and/or plugging holes. This in turn allows a wide range of flow resistances and pressure drops to be obtained with the jet plate. By making use of an experimental trial-and-error method it is possible to make simple modifications to the geometry of the jet plate between TaSEG operation tests to achieve the appropriate jet plate geometry required to suppress Gedeon streaming within the TaSEG.

While it may seem doubtful that a commercial “real world” product can make use of an element that requires in situ adjustments by experimental trial-and-error, it is important to note that the final design configuration of the jet plate that suppresses Gedeon streaming is expected to apply to all copies of a given TaSEG design. Therefore, individual jet plates will not have to be iteratively machined or honed to suppress Gedeon streaming for each TaSEG that rolls off an assembly line.
Chapter 3

Engine Instrumentation and Sensor Calibration

3.1 Introduction

This chapter outlines the equipment required to measure and record the TaSEG operational parameters including static and dynamic pressures, alternator displacements, and temperatures, during experimental testing. This chapter also discusses how these measurements will be utilized to gauge the performance of the TaSEG and effectiveness of the jet plate in suppression of Gedeon Streaming.

3.2 Sensors and Calibration

3.2.1 Pressure Sensors

Endevco model 8510B-500\(^1\) piezoresistive pressure transducers were used in the experiments. 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.

<table>
<thead>
<tr>
<th>Sensor Serial Number</th>
<th>Sensitivity [mV/psi]</th>
<th>Nonlinearity [% Full Scale Output]</th>
</tr>
</thead>
<tbody>
<tr>
<td>21107</td>
<td>0.54</td>
<td>0.12</td>
</tr>
<tr>
<td>21109</td>
<td>0.53</td>
<td>0.04</td>
</tr>
<tr>
<td>21110</td>
<td>0.53</td>
<td>0.03</td>
</tr>
</tbody>
</table>

Table 3.1. Endevco Pressure Sensor Sensitivities

Table 3.1 shows the as-delivered sensitivities of the 8510B-500 pressure sensors used during experimentation. Because the sensors were shipped directly from the supplier, an independent

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\(^1\)Endevco Corporation, www.endevco.com
calibration was not carried out. However, a comparison test was performed to compare the Endevco sensor readings to a calibrated Swagelok PGI-63B-PG600 industrial pressure gauge.

Table 3.2. Pressure Reading Comparison Test. All values shown in psig.

<table>
<thead>
<tr>
<th>Gauge Pressure</th>
<th>SN21104</th>
<th>SN21109</th>
<th>SN21110</th>
</tr>
</thead>
<tbody>
<tr>
<td>75</td>
<td>75.8</td>
<td>74.4</td>
<td>74.8</td>
</tr>
<tr>
<td>160</td>
<td>160.9</td>
<td>160.4</td>
<td>160.2</td>
</tr>
<tr>
<td>200</td>
<td>201.0</td>
<td>200.8</td>
<td>201.2</td>
</tr>
<tr>
<td>370</td>
<td>370.8</td>
<td>371.6</td>
<td>370.6</td>
</tr>
<tr>
<td>445</td>
<td>447.1</td>
<td>446.2</td>
<td>446.8</td>
</tr>
</tbody>
</table>

Figure 3.1 shows the test set-up of the comparison test, while Table 3.2 shows the results of the comparison test. Because the largest percentage difference between the pressure gauge and the Endevco transducers was on the order of 1% it was assumed that the calibration constants provided by Endevco were valid.

Both the required 10 V\textsubscript{DC} supply voltage and the full scale output of the Endevco pressure sensors were managed by an Endevco Model 136 DC amplifier. This amplifier is a three-channel signal conditioner specifically designed for use with Endevco piezoresistive pressure transducers. The unit provides an AC output voltage proportional to the voltage input and was configured to provide the \( \frac{m\text{V}}{\text{psi}} \) values listed in Table 3.1 at the BNC outputs on the back of the unit.

---

2Gauge accuracy is in accordance with ASME B40.1 Grade B and is \( \pm 1.5\% \) of the 600 psi gauge span.
4The helium supply tank is disconnected in the photo.
3.2.2 Displacement Sensors and Calibration

Micro-Epsilon, model VIP-50-GA-SA7-1 sensors 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 process, a measuring system interprets the physical relationship between the sensor housing and the measuring ring. The ring is attached to the moving alternator back piston, while the sensing rod is statically attached to the ends of the pressure vessel. Inside the sensor rod there is a segmented coil which 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. While the sensors are shipped from the supplier with a mA/mm calibration certificate, the data acquisition system could only accept 0 to 10 V DC voltage signals.
Displacement Sensor Sensitivity (SN2027)

\[ y = 0.06822x + 1.93060 \]

\[ R^2 = 0.99997 \]

\[ y = 0.06817x + 1.93038 \]

\[ R^2 = 0.99997 \]

---

Displacement Sensor Sensitivity (SN2028)

\[ y = 0.06816x + 1.92204 \]

\[ R^2 = 0.99994 \]

\[ y = 0.06793x + 1.92326 \]

\[ R^2 = 0.99989 \]

Figure 3.3. The slopes of the lines above establish the \( \frac{\text{mV}}{\text{mm}} \) calibration constant for each displacement sensor. Part (a) shows the displacement plot for the sensor with serial number 2027 and part (b) shows the displacement plot for the sensor with the serial number 2028.
As the sensor output range is 4 to 20 mA, a 250 Ω potentiometer was wired across the output as is illustrated in Figure 3.2. The potentiometer was then adjusted to provide the data acquisition system with a 1 to 5 V\textsubscript{DC} output that corresponded to the displacement of the power piston. While the voltage output was easily measured with the data acquisition system, the \( \frac{\text{mV}}{\text{mm}} \) calibration constant was unknown. In order to find the calibration constant a DC voltage was applied to the linear alternator. This causes the alternator piston to move 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 Mitutoyo Series 2330S-10 dial indicator\(^5\), over a range of applied DC voltage, it is possible to find the \( \frac{\text{mV}}{\text{mm}} \) calibration constant.

From Figure 3.3 it is clear that the calibration constants are 0.0681 ±0.0001 \( \frac{\text{mV}}{\text{mm}} \) and 0.0681 ±0.0002 \( \frac{\text{mV}}{\text{mm}} \) for sensor SN 2027 and SN 2028, respectively. The relative uncertainty of each calibration constant was found using a formula from Higbie.\(^{[55]}\)

### 3.2.3 Water Flow Measurements

In order for the TaSEG to operate, the temperature profile across the regenerator must be maintained after the thermodynamic cycle has started. To maintain the temperature at the ambient heat exchanger, located at the bottom of the regenerator, water was pumped through it. The flow of this water was controlled with an in-line B4B-6WU-02 Lake Monitors variable flow meter\(^6\) having a flow accuracy of ±2.5% of the 7.6 \( \frac{\text{l}}{\text{min}} \) span. The inlet and outlet water temperatures were measured with standard type K thermocouples discussed in the next section. The Lake Monitors B4B-6WU-02 variable flow meter was set to 4 \( \frac{\text{l}}{\text{min}} \) which was more than ample to maintain the ambient heat exchanger to between approximately 17° and 20°C.

### 3.2.4 Thermocouples and Calibration

Two types of type K thermocouples were used during the experimental investigation of the TaSEG. Several commercial Omega\(^7\) CASS-116G-12 Chromega-Alomega type K thermocouple probes were used to make temperature measurements in the water flow and to measure the helium temperature inside the TaSEG’s feedback compliance. Omega specifies that these thermocouples have an accuracy of ±1.1 °C.

Figure 3.4 shows the second type K thermocouples used, which were produced in-house. These thermocouples consisted of 30 AWG chormel and alumel wires each 2 meters in length. The chormel and alumel wire joint on the sensing end was produced using an SR48 Tempcontrol\(^8\) capacitive discharge welder. This production technique resulted thermocouples that have an exposed sensing junction, which provide the fastest response time of any thermocouple junction. The other end of the thermocouple had a standard Omega flat 2 pin quick connect connector.

\(^5\)http://www.mitutoyo.com  
\(^6\)Lake Monitors, http://www.lakemonitors.com  
\(^7\)Omega, www.omega.com  
\(^8\)Tempcontrol Industrial Electronic Products, www.tempcontrol.nl/
Figure 3.4. A thermocouple produced in-house and used for experimental testing made of 30 AWG chromel and alumel with an exposed sensing junction/tip.

These thermocouples were welded to numerous locations on and around the regenerator shell between the hot and ambient heat exchangers, and on the hot heat exchanger itself. In total 37 thermocouples were used to monitor TaSEG temperatures during testing.

All the thermocouples used during the experiments were calibrated with a Fluke 9173 metrology well calibrator. Each thermocouple is calibrated at 50°C, 250°C and 500°C using the calibrator. The calibrator has a display accuracy of ±0.2°C between 50°C and 425°C and a temperature stability of ±0.005°C between 50°C and 100°C and ±0.001°C between 100°C and 425°C. Using this calibrator it was found that the Omega thermocouples had an accuracy of ±2°C, while the thermocouples built in-house had an accuracy of ±0.7°C.

An Omega DAQSCAN-2005/DASYLab data acquisition system combined with an Omega 56 channel DBK90 thermocouple module was used to measure and log the thermocouple readings during TaSEG operation. With this system the thermocouple temperatures were measured and logged at 1 ms channel intervals. This high sampling rate minimizes errors which could arise due to time delay between temperature readings.

3.2.5 dSPACE Data Acquisition System

To interface and record all the pressure, displacement and temperature data simultaneously with one system during each operation of the TaSEG, a DS1103 PPC Controller Board and Control Development Package from dSPACE\(^9\) was used in conjunction with a standard Windows PC. The dSPACE controller board is single-board hardware for rapid control prototyping. It allows users to run customized software generated with the Matlab/Simulink block diagram environment, for real time data acquisition or control applications.

\(^9\)dSPACE Incorporated, www.dspaceinc.com
The control board provides a large selection of interfaces, including 4 A/D converters each having 4 multiplexed channels\textsuperscript{10}, 4 synchronous parallel channels each equipped with one A/D converter, and 8 D/A channels. Multiple inputs and outputs (analog, digital) allow the acquisition of data from all different kinds of sensors (e.g. temperatures, displacements, pressure, etc), but also the control of components (e.g. pumps, valves, electric motors etc). Figure 3.5 shows one experimental setup of the TaSEG with the dSPACE system outlined in a red box. One of the nice features of the dSPACE system is that it offers 4 parallel A/D converters that can be used to synchronously log data from the three pressure sensors on the TaSEG. By doing this, the phasing between each sensor is preserved and can provide useful information about the performance of the TaSEG, particularly acoustic power information.

Figure 3.6 shows the block diagram that was written to simultaneously log the three pressure sensors and relative displacements of the alternator power pistons during TaSEG operation.

\subsection*{3.2.6 Miscellaneous Equipment}

The heat flux required to operate the TaSEG was provided by a custom resistive electrical heater\textsuperscript{11}.

Figure 3.7 shows a CAD drawing of the custom heater as well as several pictures which show the inner and outer Thermocoax 2 ZE Ac 30 heating elements. Each coil was powered by electricity coming from two 2660VA, 240 VAC, bench top variable voltage transformers\textsuperscript{12},

\textsuperscript{10}Four channels belong to one A/D converter, therefore, four consecutive samplings are necessary to sample all channels belonging to one A/D converter.

\textsuperscript{11}Thermocoax, www.thermocoax.com

\textsuperscript{12}McMaster-Carr, www.mcmaster.com, PN 6994K22
Figure 3.6. dSPACE block diagram showing how data from the three pressure sensors and two displacement sensors was logged to the PC. The dSPACE system is set up so that all data is synchronously logged.

highlighted in green in Figure 3.5. By controlling the voltage supplied to the heating coils the temperatures on the hot heat exchanger can be set.

The electrical power being dissipated in the heater elements as well as the electrical power output coming from the TaSEG during operation was measured with a Hioki Model 3331 power Hitestest\(^{13}\), highlighted by blue in Figure 3.5. This particular hi-tester has a measurement accuracy of ±0.2% of the full scale output.

Additional test equipment that was required for experimental operation of the TaSEG included a Agilent 33220A function/waveform generator, an Agilent N3301A DC load bank, an Agilent E3648A 100W dual output power supply\(^{14}\), and a McCrypt PA 12000 MKII performance amp\(^ {15}\).

\(^{13}\)Hioki Group, www.hioki.com
\(^{14}\)Agilent Technologies, www.agilent.com
\(^{15}\)www.conrad-uk.com
Figure 3.7. CAD drawing and photos of the Thermocoax resistance heater used to provide the thermal input to the TaSEG during experiments.
TaSEG Experimental Setup and Performance

4.1 Introduction

In this chapter, the experimental setup, operation and performance of the TaSEG will be discussed. While the acoustic and electrical power produced, the pressure amplitude, and the heater input power are all quantities that are essential to characterize the TaSEG’s performance, the main focus of this thesis is on the suppression of Gedeon streaming within the TaSEG. For more in depth details about the overall performance of the TaSEG please see [42].

This chapter starts by explaining the experimental set-up of the TaSEG, followed by how steady state operation was achieved. Once steady state was achieved, the performance measurements of the TaSEG making use of the latex diaphragm to suppress streaming were performed. The diaphragm performance data then served as the baseline to which the jet plate performance data was compared. This comparison determines the jet plate’s effectiveness at suppressing Gedeon streaming within the TaSEG. This chapter also briefly discusses several other experimental investigations undertaken related to the TaSEG. These include the measured heat flux, collimator cycle resistance and seal gap experiments that were performed.

4.2 Experimental Setup and TaSEG Operation

Experiments with the TaSEG were completed using two experimental setups. The only difference between the setups was the physical location and the different variable voltage transformers used at each location (i.e. 120 VAC vs 240 VAC). All other measurement and test equipment was the same. Figures 3.5 and 4.1\(^1\) show the two different TaSEG test setups.

\(^1\)dSPACE data acquisition system not shown in picture
Figure 4.1. TaSEG experimental setup making use of 240 VAC variable transformers.

The TaSEG’s alternators were wired in series\(^2\) and connected to the custom load circuit. This circuit consisted of balance capacitors, a 5:1 step down transformer, a AC to DC bridge rectifier circuit, and an Agilent DC load bank which was set to maintain constant voltage.[42] The DC load bank was used to force a constant voltage across the alternators; however, it should be noted that the circuit presents a non-zero load even when the load bank is essentially “open.” Nevertheless, the DC load bank did enable the engine to achieve stable state operation, by holding the voltage to a narrow 4.5 to 8 V\(_{DC}\) range.

The TaSEG load and control circuit is shown in Figure 4.2.[56] For all the results given in this report, the balance series capacitance was 300 µF. Additionally, the TaSEG required a 3.35 MPa charge pressure of 99.9% pure helium.

Under normal circumstances the TaSEG will not spontaneously go into onset. Therefore, in order to start the TaSEG the alternator leads were connected to a “buzz” circuit. This circuit consisted of an Agilent 33220A function/waveform generator\(^3\) which provided input to a standard McCrypt PA 12000 MKII performance amp\(^4\). The output of the amplifier was used to provide an AC “buzz” briefly to the TaSEG alternators. This buzz was at low voltage, usually around 24 VAC, with a frequency that was near the 52 Hz natural TaSEG operating frequency. The short “buzz” signal was accomplished by installing an on/off toggle switch between the output of the amplifier and one lead going to the alternators. By briefly toggling the switch on and off, 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 thermocouples brazed to the resistance heaters read around 450 °C. By buzzing the alternators below the required onset temperature

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\(^2\)or parallel

\(^3\)Agilent Technologies, www.agilent.com

\(^4\)www.conrad-uk.com
the helium was moved around the hot heat exchanger tubes, helping to even out the temperature from tube to tube. The buzzing was continued every few seconds, as the heater elements and hot heat exchanger temperatures rose. When 500–550 °C was indicated on the hottest hot heat exchanger thermocouple, the engine would normally go into onset providing the DC load was correctly set. Once the TaSEG started the temperatures 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. A “safe” value of TaSEG operation occurred when the thermocouples indicated average hot heat exchanger temperatures of 615–630 °C.

Ideally, when the TaSEG is run using the latex diaphragm to suppress Gedeon streaming the temperature profile across the regenerator should be nearly linear. This is shown by line “a” of Figure 4.3. Gedeons theory[23] suggests that streaming, in the absence of any suppression mechanism, will proceed against the thermal gradient, from the ambient to the hot side of the regenerator, in the same direction as acoustic power flow. Streaming of this sort will cause the regenerator’s temperature profile to become concave with cooler helium traveling “up” the regenerator, shown by curve “c” of Figure 4.3. An “over designed” jet plate would drive streaming in the opposite direction of the acoustic power flow or from the hot to the ambient side of the regenerator. This will result in a convex regenerator temperature profile, shown by curve “b” of Figure 4.3.

After the TaSEG was run with the latex diaphragm for establishing baseline, “no streaming” operation, the jet plate was substituted for the diaphragm and steady state operation was attempted. Alterations to the edges of each through-hole along with changes in the smaller hole diameter allowed “tuning” of the jet plate’s flow resistance, which in turn allowed the point of

![Figure 4.2. TaSEG DC load and control circuit diagram.](image)
streaming suppression to be reached.

As the TaSEG was initially designed without the ability to insert thermocouples directly into the regenerator through the pressure vessel wall and because the regenerator temperature profile is very sensitive to streaming\cite{23}, additional type K thermocouples were affixed to the regenerator wall as seen in Figure 4.4. Five thermocouples were each welded to the north, east, south and west sides of the regenerator wall at the locations shown in Figure 4.5. These

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure4.3.png}
\caption{Regenerator temperature profiles under three different streaming cases. (a) Streaming suppressed using the latex diaphragm, (b) Streaming over suppressed, (c) No streaming suppression.}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure4.4.png}
\caption{Thermocouples welded to east and west sides of the regenerator wall in order to monitor the temperature profile across the regenerator.}
\end{figure}
Figure 4.5. Approximate locations of thermocouples welded to the regenerator wall. The exact measured location varies by (± 3 mm) and depends on what side of the regenerator (i.e. North, East, South, West) is examined. As the figure indicates, the regenerator begins 20 mm from the hot heat exchanger and is 38.6 mm in length, ending 58 mm from the hot heat exchanger. In all the following temperature vs. thermocouple regenerator location figures the extent of the regenerator is from 20 mm to 58 mm, but may not be entirely shown.

thermocouples accurately portray the axial temperature profile of the regenerator. This allows comparison between the steady state baseline temperature profile across the regenerator with the latex diaphragm and the profiles present during TaSEG operation with the jet plate.

4.3 Experimental Streaming Results

To establish baseline operating conditions, the TaSEG was run with the latex diaphragm in place to suppress Gedeon streaming.

Two “baseline” plots of the temperature profiles across the regenerator using the latex diaphragm during operation at two different steady states are shown in Figures 4.6 and 4.7. These serve as the baseline TaSEG operation plots for comparison with the jet plate performance.

The top picture of Figure 2.9 in Chapter 2 shows the CNC machined aluminum jet plate, while the bottom picture shows the rapid prototyped plastic jet plate. Both were constructed and tested within the TaSEG in an experimental trial-and-error method.

This jet plate design makes use of minor losses by channeling the flow below the collimator through the 15 through-holes that abruptly open into a larger volume below the jet plate. If the through-holes did not have a cone angle leading down to the expansion side, then the minor loss
Figure 4.6. Temperature profile across regenerator with diaphragm installed in TaSEG.

Figure 4.7. Temperature profile across regenerator with diaphragm installed in TaSEG.
that takes place at each end would be the same and the jet plate would have little asymmetry. This cone angle increases the cross-sectional area at the large end, thus reducing the velocity. Recalling Eq. 2.14, it can be seen that a reduction in velocity will in turn reduce the pressure drop. For the orientation shown in Figure 2.8, \( \Delta p_{\text{ml}} \) will have the correct sign to suppress Gedeon streaming around the TaSEG feedback loop.\cite{27}

Initially, the 15 jet plate through-holes had a small diameter of \( D_s = 4.14 \) mm and a large diameter of \( D_L = 12.95 \) mm. These diameters yield total areas of \( a_s = 201.9 \) mm\(^2\) and \( a_L = 1975.7 \) mm\(^2\). Because both the large and small holes had sharp edges (i.e. \( r = 0 \)), \( k_{\exp,s} = k_{\exp,L} = 1.05 \) and \( k_{\con,s} = k_{\con,L} = 0.5 \). The jet plate has a height of 18.2 mm, with a full cone angle of 30°, running from the larger hole to a depth of 16.4 mm.\footnote{The mechanical drawing of the jet plate can be found in Appendix 1.} Making use of Eqs. 2.17, 2.18, 2.19 and the initial through-hole dimensions the time-averaged pressure drop across the jet plate should be \( \Delta p_{\text{ml}} = 418 \) Pa, with the time-averaged acoustic power dissipation being \( \dot{E}_{\text{jp}} = 7.0 \) W, and its resistance equal to \( R_{\text{jp}} = 300 \) kPa \( \cdot \) s/m\(^3\). Comparing these values to Eq. 2.13, this initial design should be capable of imposing a larger \( \Delta p \) across the regenerator than is required to suppress Gedeon streaming.

After the baseline measurements making use of the latex diaphragm were complete, the rapid prototype plastic jet plate was inserted into the TaSEG. Several attempts were made to achieve steady state operation of the TaSEG, but were unsuccessful. The TaSEG did achieve onset, but was only able to sustain operation for approximately 20 seconds. Temperature profiles across the regenerator at TaSEG onset and after 20 seconds of operation, shortly before it stopped running, are shown in Figure 4.8. The concave bending of the temperature profiles after 20 seconds of operation is clearly visible in this figure. This, along with the subsequent ceasing of engine operation, suggests that the plastic jet plate is undercompensating for the tendency of the regenerator to stream in the direction of \( \dot{E} \), from ambient to hot heat exchanger.

Based on these results, it was decided to increase the smaller through-hole diameter to 4.5 mm with sharp edges in order to reduce the jet plate flow resistance \( R_{\text{jp}} \) and acoustic power dissipation \( \dot{E}_{\text{jp}} \) while maintaining a similar time-averaged pressure drop across the jet plate \( \Delta p_{\text{ml}} \). This jet plate configuration only changes the ratio of total areas of Eqs. 2.17, 2.18, and 2.19, while maintaining \( k_{\exp,s} = k_{\exp,L} = 1.05 \) and \( k_{\con,s} = k_{\con,L} = 0.5 \). Thus, the time-averaged pressure drop across the jet plate should now be \( \Delta p_{\text{ml}} = 376 \) Pa, the time-averaged acoustic power dissipation \( \dot{E}_{\text{jp}} = 6.3 \) W, and its resistance \( R_{\text{jp}} = 270 \) kPa \( \cdot \) s/m\(^3\). Again, the TaSEG was able to achieve onset, but only able to run for 40 seconds before stopping. Examining the temperature profiles at onset and after 30 seconds of operation seen in Figure 4.9, it is again clear that the temperature profile after only a short period of operation has become concave. This suggests that the new jet plate geometry is still undercompensating the natural Gedeon streaming within the TaSEG.

The next logical step based on this outcome was to attempt to alter the through-hole geometries, specifically the edge geometries of the smaller holes. This was done in an attempt
Figure 4.8. Temperature profile across the regenerator with the TaSEG using the rapid prototyped plastic jet plate. The diameters of the small through-holes was $D_s = 4.14$ mm and diameters of the large through-holes were $D_L = 12.95$ mm.

to increase the jet plate’s time-averaged pressure drop while further reducing its flow resistance and acoustic power dissipation. However, due to the layering method used to produce the rapid prototype plastic jet plate, the edges of the holes were not clearly defined. This meant it would have been very difficult to machine a defined, uniform radius onto any of the through-hole edges. Based on this factor it was decided to stop testing with the plastic jet plate and move on to the aluminum CNCed jet plate.

In order to compare the two jet plates, the aluminum jet plate having the initial undercompensating through-hole dimensions of $D_s = 4.14$ mm and $D_L = 12.95$ mm both with sharp holes edges was tested in the TaSEG under comparable operation parameters. Initial results were encouraging as the TaSEG achieved onset and appeared to be running towards steady state operating conditions. However, after approximately 2 minutes of operation the TaSEG again stopped.

Temperature profiles across the regenerator at TaSEG onset and after 127 seconds of operation, are shown in Figure 4.10. While the aluminum jet plate did allow the TaSEG to run for a longer time, the concave temperature profiles across the regenerator at the time the TaSEG stopped operating, suggest that the jet plate is again undercompensating and not canceling the natural streaming.

As both the plastic rapid prototyped jet plate and the aluminum jet plate had the same large
Figure 4.9. Temperature profile across the regenerator with the TaSEG using the rapid prototyped plastic jet plate. The diameters of the small through-holes was $D_s=4.5$ mm and diameters of the large through-holes were $D_L=12.95$ mm.

and small hole diameters, it is believed that the difference in surface roughness and the inability to define through-hole edge conditions could be the reasons for the different results obtained when running the TaSEG. The plastic jet plate layering production process results in a much rougher surface finish and undefined sharpness at the through-hole edges. This roughness of the plastic jet plate could result in the enhancement of acoustic power dissipation at the jet plate due to turbulence generation.\cite{15, 17}

The TaSEG was run several more times under similar operating conditions with the same aluminum jet plate configuration and yielded similar results. Therefore, it was decided to increase the smaller through-hole diameter to 4.5 mm with sharp edges. This should yield $\Delta p_{ml}=376$ Pa, $\dot{E}_{jp}=6.3$ W, and $R_{jp}=270 \frac{\text{Pa}\cdot\text{s}}{\text{m}^3}$, similar to the modified plastic jet plate.\footnote{Again, this geometry change only affects the ratio of total areas of Eqs. 2.17, 2.18, and 2.19, while maintaining $k_{exp,s}=k_{exp,L}=1.05$ and $k_{con,s}=k_{con,L}=0.5$.}

Temperature profiles across the regenerator at TaSEG onset and after 42 seconds of operation when the TaSEG stopped running, are shown in Figure 4.11. In this case the aluminum jet plate did allow the TaSEG to achieve onset, but for a shorter length of time. The concave temperature profiles across the regenerator at the time the TaSEG stopped operating, coupled with the reduced operation time suggest that the jet plate is still undercompensating the natural streaming. This result also suggests that any geometry modifications made to the jet plate should be done to increase the time-averaged pressure drop across the jet plate.
Figure 4.10. Temperature profile across the regenerator with the TaSEG using the initial design of the aluminum jet plate. The diameters of the small through-holes was $D_s = 4.14$ mm and diameters of the large through-holes were $D_L = 12.95$ mm with all holes having sharp edges.

Based on this conclusion, the edges of the smaller holes were rounded to have a radius of $r = 0.8$ mm, while maintaining the large and small through-hole diameters. This jet plate geometry results in $k_{exp,s} = k_{exp,L} = 1.05$, $k_{con,s} = 0.05$ and $k_{con,L} = 0.5$, while maintaining the total area ratio $a_s/a_L$.

The rounding of the small holes, visible in Figure 4.12, was done using a CNC machine and required laser centering of each hole with respect to the end mill of the CNC machine prior to each hole being machined. This ensured a well defined edge radius from hole to hole.

Once the edges of the jet plate’s smaller holes were rounded, it was inserted back into the TaSEG to evaluate its streaming suppression performance. Once onset was achieved, the TaSEG continued to operate, achieving steady state operating conditions similar to the conditions achieved when running the TaSEG with the diaphragm. Once the TaSEG achieved steady state operating conditions with the modified aluminum jet plate it was possible to compare the temperature gradients across the regenerator and other performance data to the baseline diaphragm TaSEG runs.

Figures 4.13 and 4.14 show the temperature profiles across the west and east sides of the regenerator. Sudden contraction minor loss coefficient reduced as $\frac{d}{D} \approx 0.19$. Steady state here is defined as stabilization of the electrical input to the heaters and all the temperature measurements within a $\pm 2$ °C range. Steady state operation with the aluminum jet plate with rounded edges normally happened after the TaSEG was operational for approximately 35 minutes.
Figure 4.11. Temperature profiles across regenerator with modified aluminum jet plate. The diameters of the small through-holes was $D_s = 4.5\ mm$ and diameters of the large through-holes were $D_L = 12.95\ mm$ with all holes having sharp edges.

Figure 4.12. The rounded edges of the small diameter holes in the modified aluminum jet plate.

regenerator during steady state TaSEG operation with both the latex diaphragm and aluminum jet plate\textsuperscript{9}. While Table 4.1 and Figures 4.13 and 4.14 serve as an encouraging indication that the aluminum jet plate with the rounded edges is sufficiently suppressing Gedeon streaming within the TaSEG, direct comparison between the jet plate and diaphragm is not possible because

\textsuperscript{9}Various load voltages applied across the linear alternators.
Steady State West Side Regenerator Temperature Profiles
Diaphragm and Jet Plate w/ \( D_{\text{small}} = 4.5 \text{ mm}, r = 0.8 \text{ mm} \)

Figure 4.13. Temperature profiles across the regenerators west side when the TaSEG was run with the aluminum jet plate. The diameters of the small through-holes was \( D_{\text{small}} = 4.5 \text{ mm} \) and diameters of the large through-holes were \( D_{\text{L}} = 12.95 \text{ mm} \). Additionally, the edges on the smaller through-holes were rounded to a radius of \( r = 0.8 \text{ mm} \).

Table 4.1. TaSEG Performance Comparison, After Approximately 44 minutes of Operation.

<table>
<thead>
<tr>
<th>Method</th>
<th>( t ) [min]</th>
<th>( W_{\text{Input}} ) [W]</th>
<th>( W_{\text{Output}} ) [W]</th>
<th>( T_{\text{HHX}} ) [°C]</th>
<th>( T_{\text{AHX}} ) [°C]</th>
<th>( p_a ) [mV RMS]</th>
<th>( \text{Load}_{\text{DC}} ) [V DC]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diaphragm</td>
<td>42</td>
<td>1170</td>
<td>40.8</td>
<td>651</td>
<td>15.9</td>
<td>198</td>
<td>6</td>
</tr>
<tr>
<td>Al Jet Plate</td>
<td>46</td>
<td>1250</td>
<td>25.2</td>
<td>652</td>
<td>15.9</td>
<td>202</td>
<td>6</td>
</tr>
</tbody>
</table>

of noticeable wearing of the power piston/bore seals during extended operation. This wearing increases the seal gap height, which in turn increases the amount of acoustic power dissipated within the seal, reducing the electrical output power of the TaSEG.

In Table 4.1, “Method” refers to the streaming suppression method used within the TaSEG, \( t \) is the duration of TaSEG operation before measurements were made, \( W_{\text{Input}} \) is the total electrical input power supplied to the resistance heaters, \( W_{\text{Output}} \) is the electrical output power of the TaSEG, \( T_{\text{HHX}} \) is the average external temperature of the hot heat exchanger tubes, \( T_{\text{AHX}} \) is the average ambient heat exchanger temperature\(^{10}\), \( p_a \) is the pressure wave amplitude taken in the compression space between the power pistons, and \( \text{Load}_{\text{DC}} \) is the DC voltage of the load.

In order to allow direct comparison of the TaSEG performance using the latex diaphragm to

\(^{10}\)This value is found by taking the average of the inlet and outlet water temperatures.
Figure 4.14. Temperature profiles across the regenerators east side when the TaSEG was run with the aluminum jet plate. The diameters of the small through-holes was $D_{\text{small}}=4.5$ mm and diameters of the large through-holes were $D_{L}=12.95$ mm. Additionally, the edges on the smaller through-holes were rounded to a radius of $r=0.8$ mm.

that of using the jet plate having small hole rounded edges, the linear alternators with power pistons were removed from the TaSEG and the Rulon J babbitt-like coating\textsuperscript{11} on the pistons was replaced. This coating can be seen in Figure 5.2. Reapplying Rulon J to the pistons assures that the seal gap which exists between the power pistons and the bore is comparable from run to run. Any increase in the seal gap will cause a decrease in the TaSEG’s electrical output, therefore seal preservation from run to run is required to accurately evaluate the effectiveness of the rounded aluminum jet plate. By having similar seal conditions for TaSEG operation with both the diaphragm and the rounded aluminum jet plate, any reduction in the TaSEG performance between runs can be attributed to the use of the different streaming suppression methods.

Once Rulon J was reapplied to both power pistons, the TaSEG was set up and several runs were made on the same day one right after the other, switching between the diaphragm and the aluminum jet plate with rounded small hole edges.

Figures 4.15 and 4.16 show the temperature profiles across the east and south sides of the regenerator during steady state operation, when both the latex diaphragm and rounded aluminum

Figure 4.15. Temperature profiles across the regenerators south and east sides when the TaSEG was run with both the aluminum jet plate having the rounded edges on the smaller through-hole diameters and the diaphragm. After the power pistons were re-coated.

Table 4.2. TaSEG Performance Comparison, After Approximately 42 minutes of Operation.

<table>
<thead>
<tr>
<th>Method</th>
<th>t [min]</th>
<th>(W_{\text{Input}}) [W]</th>
<th>(W_{\text{Output}}) [W]</th>
<th>(T_{\text{HHX}}) [°C]</th>
<th>(T_{\text{AHX}}) [°C]</th>
<th>(\Delta T_{\text{Regen}}) [°C]</th>
<th>(p_{\text{mean}}) [psi]</th>
<th>Load_{DC} [V_{DC}]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diaphragm</td>
<td>42</td>
<td>967</td>
<td>60.0</td>
<td>644</td>
<td>19.7</td>
<td>624</td>
<td>491</td>
<td>4.5</td>
</tr>
<tr>
<td>Al Jet Plate</td>
<td>43</td>
<td>962</td>
<td>51.5</td>
<td>644</td>
<td>19.7</td>
<td>625</td>
<td>486</td>
<td>4.5</td>
</tr>
</tbody>
</table>

jet plate were alternately used in the TaSEG. Tables 4.2 and 4.3 provide a comparison of the steady state operating conditions and TaSEG performance, where \(p_{\text{mean}}\) is the mean operating pressure and \(\Delta T_{\text{Regen}}\) is the average temperature difference established across the regenerator.

Both the figures and tables indicate that comparable performance between the streaming suppression methods was obtained. The largest difference in the performance of the TaSEG is a 5.5 to 8.5 W decrease in the electrical output. This drop in electrical output is to be expected, as insertion of the jet plate to generate the required \(\Delta p_{\text{ml}}\) to suppress Gedeon streaming also dissipates acoustic power.[15, 27] A reduction in acoustic power results in a decrease in the available

Table 4.3. TaSEG Performance Comparison, After Approximately 59 minutes of Operation.

<table>
<thead>
<tr>
<th>Method</th>
<th>t [min]</th>
<th>(W_{\text{Input}}) [W]</th>
<th>(W_{\text{Output}}) [W]</th>
<th>(T_{\text{HHX}}) [°C]</th>
<th>(T_{\text{AHX}}) [°C]</th>
<th>(\Delta T_{\text{Regen}}) [°C]</th>
<th>(p_{\text{mean}}) [psi]</th>
<th>Load_{DC} [V_{DC}]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diaphragm</td>
<td>60</td>
<td>904</td>
<td>45.8</td>
<td>648</td>
<td>18.9</td>
<td>629</td>
<td>484</td>
<td>4.5</td>
</tr>
<tr>
<td>Al Jet Plate</td>
<td>58</td>
<td>890</td>
<td>40.3</td>
<td>658</td>
<td>18.9</td>
<td>639</td>
<td>486</td>
<td>4.5</td>
</tr>
</tbody>
</table>
Steady State East and South Side Regenerator Temperature Profiles

Load= 4.5 Vdc

Figure 4.16. Temperature profiles across the regenerators south and east sides when the TaSEG was run with both the aluminum jet plate having the rounded edges on the smaller through-hole diameters and the diaphragm. After the power pistons were re-coated.

Unfortunately, malfunctioning pressure sensors resulted in erroneous pressure amplitude measurements making the examination of the pressure amplitude within the compression space between the pistons impossible. Even without this information it is still clear from other measurement data that the rounded aluminum jet plate is successfully working to suppress Gedeon streaming within the TaSEG.

Table 4.4. Sequential Run TaSEG Performance Comparison, After Approximately 44 Minutes of Operation.

<table>
<thead>
<tr>
<th>Method</th>
<th>$t$ [min]</th>
<th>$W_{Input}$ [W]</th>
<th>$W_{Output}$ [W]</th>
<th>$T_{HHX}$ [$^\circ$C]</th>
<th>$T_{AHX}$ [$^\circ$C]</th>
<th>$\Delta T_{Regen}$ [$^\circ$C]</th>
<th>$p_{mean}$ [psi]</th>
<th>$Load_{DC}$ [V dc]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diaphragm</td>
<td>46</td>
<td>1070</td>
<td>46</td>
<td>641</td>
<td>12.9</td>
<td>628</td>
<td>482</td>
<td>6.0</td>
</tr>
<tr>
<td>Al Jet Plate</td>
<td>42</td>
<td>1079</td>
<td>38.6</td>
<td>644</td>
<td>12.9</td>
<td>631</td>
<td>482</td>
<td>6.0</td>
</tr>
</tbody>
</table>

Figure 4.17 and Table 4.4 show the temperature profiles across the east and west sides of the regenerator and steady state operation data for the TaSEG operating with a higher voltage on the DC load. This measurement was performed to examine the maximum electrical output of the TaSEG as a higher voltage allows a higher stroke of the linear alternator, which should result in the production of more electricity. As the table indicates this was not the case.

This result led to an examination of the power piston bore in which a copious amount of
Figure 4.17. Temperature profiles across the regenerators west and east sides when the TaSEG was run with both the aluminum jet plate having the rounded edges on the smaller through-hole diameters and then the diaphragm. After the power pistons were re-coated.

Figure 4.18. Rulon dust in the bore of the alternator power pistons indicating wear of the seal.
Rulon dust was found, pictured in Figure 4.18. This indicates that the power pistons were rubbing against the bore, which almost certainly results in an increase of the seal gap. These in turn reduce the electrical output of the TaSEG. Regardless of the unexpected reduction in the electrical output, this data still supports the conclusion that the rounded aluminum jet plate successfully suppresses Gedeon streaming within the TaSEG, which was the main goal of this work.

Further support for the streaming suppression effectiveness of the rounded jet plate comes from examining the local helium temperature in the feedback compliance located below the ambient heat exchanger. A rapid increase in the local helium temperature is an obvious indicator of a DC flow of helium from the hot heat exchanger through the TaSEG and into the feedback compliance.

![Helium Temperature Evolution during TaSEG Operation](image)

**Figure 4.19.** TaSEG compliance helium temperature evolution after onset was achieved.

Figure 4.19 shows the helium temperature moments after the TaSEG achieves onset for several different jet plate geometries and the diaphragm. The local helium temperature evolution of the TaSEG operating with the diaphragm shows an increase of 3.3 °C over the first minute of operation without the presence of Gedeon streaming. This temperature increase is expected as the helium gas begins to experience acoustic oscillations and the TaSEG body temperature rises to steady state. After approximately 5 minutes of operation this temperature reached 25 °C and remained constant thereafter.

With either the plastic or aluminum jet plates with the 4.5 mm small hole diameters with sharp edges inserted in the TaSEG, the local helium temperature experienced an increase of
15 °C within the first 30 seconds of TaSEG operation. This rapid change in temperature was followed by the discontinuation of TaSEG operation and corresponds to the concave bowing of the regenerator temperature profiles, seen in Figures 4.9 and 4.11. This again suggests that the jet plate geometry was not “tuned” correctly to completely suppress the natural Gedeon streaming within the TaSEG.

The local helium temperature evolution for the TaSEG with the original aluminum jet plate that achieved TaSEG operation of 127 seconds was not recorded, but it is known that the temperature increased from 27.3 °C to 36.8 °C before the TaSEG stopped operating. This increase is represented by the dashed line in Figure 4.19 and again corresponds to the concave temperature profiles and the presence of Gedeon streaming during operation.

The triangle data points in Figure 4.19 correspond to the TaSEG operating with the aluminum jet plate with the 4.5 mm diameter small holes having rounded edges. As the figure shows, the local helium temperature increases 3.5 °C during the first minute of operation, similar to when the diaphragm was used within the TaSEG. Like the runs with the diaphragm, the temperature profile levels off to approximately 25 °C when the TaSEG achieves steady state operation. This further supports the belief that the aluminum jet plate with the 4.5 mm diameter small holes with rounded edges suppresses Gedeon streaming within the TaSEG.

### 4.4 Other Experimental Results

In addition to the experiments performed with the TaSEG to investigate a Gedeon streaming suppression method, several additional experiments were carried out. This section briefly discusses each of the extra investigations and how each pertains to the TaSEG.

#### 4.4.1 Collimator Resistance Experiments

Initially, the collimator was included in the TaSEG design to occupy volume between the alternator power pistons and the thermal buffer tube because the manifolding of the inertance tubes required moving the engine parts further away from the power pistons. Another disadvantage of using the latex membrane to suppress streaming is that any extra flow resistance required to optimize the TaSEG cycle phasing can not be included in the membrane design. Therefore, it was left to modifying the collimator element just above the membrane to approximate the proper flow resistance that results in the best TaSEG performance. Because the jet plate will add an additional flow resistance, optimizing the collimator in combination with the jet plate seemed logical.

Additional experimental work was completed to optimize the TaSEG thermodynamic cycle phasing with the rounded aluminum jet plate suppressing Gedeon streaming. This was done by operating the TaSEG in succession and varying the number of plugs inserted into the collimator, as well as changing the through-hole geometry of the plugs, both of which change the amount of flow resistance present in the collimator. The flow resistance in the collimator was changed
because this was easier and faster than further altering the rounded aluminum jet plate. The adjustment of the collimator flow resistance could not be done \textit{in situ}, but required repeated disassembly of the TaSEG between successive runs. The method used to change the total resistance of the collimator involved inserting completely new brass plugs with different through-hole diameters, lengths and entrance edge angles.

![Diagram of plug dimensions](image)

\textbf{Figure 4.20.} Important dimensions of the plugs tested in the TaSEG’s collimator.

<table>
<thead>
<tr>
<th>Plug</th>
<th>d [mm]</th>
<th>L [mm]</th>
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\textbf{Table 4.5.} Plug Dimensions of the TaSEG’s Collimator

Figure 4.20 and Table 4.5 provide a numerical description of the four different plug geometries tested, while Figure 4.21 shows a view of the collimator from the jet plate side, with one set of brass plugs inserted.

By changing the dimensions of the plugs and monitoring the electrical power output of the TaSEG, it was possible to determine the optimum flow resistance of the collimator in combination with the rounded jet plate.

Figure 4.22 shows the peak TaSEG electrical output versus its DC load voltage (stroke control) with the TaSEG having the rounded jet plate and varying flow resistance (plug geometries) in the collimator. As Figure 4.22 illustrates, the TaSEG performance is best when the diaphragm is used in combination with the original aluminum plugs. However, as the ultimate goal is to remove the latex diaphragm from the TaSEG due to lifetime concerns in a real world product, the resistance optimization was only done with the rounded aluminum jet plate. The figure also illustrates that the TaSEG performance was similar for the rounded jet plate with the original aluminum plugs or brass plugs 1, 3, or 4. Interestingly enough, all of the plugs tested which were symmetric about their center line yielded comparable results, while the one asymmetric plug (plug 2) yielding the worst performance. This data also suggests that the flow resistance
Figure 4.21. One geometry of brass plugs inserted into the TaSEG’s collimator.

Figure 4.22. DC load voltage vs. TaSEG electrical output.
provided by the original aluminum plugs was already very close to the optimal value for the best TaSEG cycle phasing.[42]

Furthermore, this result also supports the conclusion that the resistance of the jet plate is much less than what is required for the optimum phasing between $p_1$ and $U_1$. This means that while the jet plate resistance is a part of the total cycle phasing resistance, it is not the dominant controlling factor. Thus, a jet plate used for streaming suppression is a viable option for small scale thermoacoustic engines.

It should be noted that the electrical output is also affected by the seal gap which exists between the linear alternators and piston housing. Every effort has been made to do the plug tests in succession, without disassembling the alternator subassembly, in an attempt to preserve the consistency of the seal gap.

4.4.2 Heat Flux

While the resistance heater was able to provide adequate heat flux to successfully run the TaSEG at steady state, data taken during operation from numerous thermocouples located around the hot heat exchanger suggested that the heater did not evenly heat the hot heat exchanger tubes, as was hoped. As the resistance heater provides predominantly radiant heating of the hot heat exchanger tubes, with relatively little convection to assist the heat transfer, there also existed large difference between the resistance heater element temperatures and those of the hot heat exchanger tubes. This temperature difference between the resistance heater’s external metal and the TaSEG’s hot heat exchanger tubes was about 200 °C. Consequently, in spite of a large amount of insulation being placed around the heater, significant heat leak into the room was expected with the resistance heater metal running so hot.

However, it is possible to obtain a reasonably accurate estimate of the heat leak, without calculating it from first principles, or even knowing the exact thermal mass of the heater. This is done by making careful transient temperature measurements during the heat-up and cool-down cycles of the TaSEG. To do this, the custom resistance heater was placed on the TaSEG, thoroughly insulated and set-up to mimic steady state operating conditions, but with the alternator leads shorted together to prevent the TaSEG from operating. Temperature measurements of the heater at several locations around the hot heat exchanger were then made as a function of time. The heater power was then shut off and the drop in temperatures as a function of time was recorded.

Figure 4.23 shows the resulting data points plotted as a function of time with curve-fits which are used to determine the rate of change in the temperature as a function of time, $\frac{\Delta T}{\Delta t}$. Assuming that while the assembly is heating up without the TaSEG running, the hot end follows $c_p \frac{\Delta T}{\Delta t} = (W_{Elec} - \dot{Q}_{Lost})$ and that when the electrical power provided to the heater is cut this equation becomes $c_p \frac{\Delta T}{\Delta t} = -\dot{Q}_{Lost}$, at the temperatures of interest $\dot{Q}_{Lost} = 210$ W, corresponding to 1250 W of electrical input power. As this 210 W loss includes conduction and heat leak to the room, a subtlety in how the conduction loss is handled in the DELTAEC model must be
Figure 4.23. TaSEG hot end temperature vs. time.

considered to evaluate the total heat flux into the TaSEG.

Usually, the thermal conduction down the regenerator wall is included in the DELTAEC model by calculating the effective solid fraction of the regenerator, including the cross-sectional area of the regenerator wall, using this value for “KsFrac” input for the regenerator. However, the conduction then calculated by the stacked screen regenerator segment uses estimated temperatures for the matrix at the ends, which are marginally different from the estimated gas temperatures and greater still than the heat exchanger temperatures. Thus, any temperature drop between the hot heat exchanger metal and the helium is left out of the estimate of the conduction loss that occurs outside the pressure envelope and between hot heat exchanger metal temperatures. In the case of the TaSEG, the hot heat exchanger metal temperature difference is much larger than the internal regenerator temperature difference. Because of this, the actual conduction loss is almost twice the loss predicted by DELTAEC.

Accounting for this difference in the conduction loss for the point analyzed above and in reference [42], the estimated input heat flux to the TaSEG cycle is 1250 W minus the 210 W lost to both conduction and heat leak to the room, plus the 75 W of parasitic loss predicted by DELTAEC. This comes to 1115 W, compared to the 945 W that DELTAEC predicts is necessary to operate the TaSEG at steady state. While there still exists an error of 15%, this is better than the 25% error between DELTAEC and the initial estimated power in reference [42], where
the input heat flux was estimated from the heat rejected to the ambient heat exchanger water stream. This reduction in the estimated input heat flux error addresses the one open point from the initial TaSEG evaluation between the predicted TaSEG performance from the DELTAEC simulations and the measured experimental results. This also serves as evidence that DELTAEC is a valuable simulation tool that can guide the future development of TaSEG’s.

4.4.3 Direct-Contact Resistance Heater

As the initial custom resistance heater relied primarily on radiantly heating the hot heat exchanger tubes, an investigation was done to evaluate the heating of the tubes with a resistance wire that was affixed directly to the hot heat exchanger tubes, creating line contact between the heat source and exchanger tubes. Figure 4.24 shows the direct-contact resistance heater wire, affixed to the hot heat exchanger tubes. This heater provided enough heat flux to operate the TaSEG at steady state and provided the most uniform heating of the hot heat exchanger tubes. Data taken from operating the TaSEG with the direct-contact resistance wire heater indicates that the overall performance of the TaSEG was indifferent to the heating means, for a given average temperature of the hot heat exchanger tubes. Thus, the non-uniform temperatures on the tubes experienced during the Thermocoax radiant heater runs have no impact on the TaSEG’s performance. It should be noted that while the performance of the TaSEG was independent of the heating method, the direct-contact resistance heater which relied on conduction reached it operating temperature much faster and was easier to control to maintain at a constant temperature than the radiant heater.
4.4.4 Seal Gap Experiments

While the TaSEG continued to operate well and give significant electrical output power during its demonstration phase, its electrical output power gradually declined. This decline was a result of the “feathering” to the Rulon of the power pistons in an attempt to close the seal gap.[42] To do this, the Rulon in the seal is cut at several locations with a parting tool, but to no specific dimensions\(^\text{12}\). These “feathers” are then worn down by running the seals in for an extended period of time. However, it was believed that the large masses required to tune the alternators to 50 Hz resulted in the “feathers” were wearing out with operation time due to an effect similar to the bending of a cantilever beam\(^\text{58}\) occurring at the maximum alternator stroke length. This in turn allowed the feathers to be worn away and the seal gap to gradually open up. With that thought in mind, a different feathering technique was attempted using a parting tool on the lathe to plastically deform the Rulon coating of the pistons into ribs, or ridges with more substance than the “feathers”. The ribs were machined to have the dimensions shown in Figure 4.25.

![Figure 4.25. Dimensions in mm, of the rid machined into the Rulon of the power pistons.](image)

This results in the outer diameter of the power pistons having a 0.025–0.051 mm interference fit with the bore. The idea was that the Rulon should plastically deform, not compress appreciably during turning; therefore, the ribs should be higher than the undeformed surface. When the ribs wear in, after enough running using electric power to drive the alternators like motors if necessary, they should not “wear out” like the feathers.

Figure 4.26 shows one of the power pistons after the ribbing machining was completed, with a close up of the ribbed piston. While it seems logical that this method should work, it did not in fact result in appreciably lower seal losses as measured in flow tests as the TaSEG was unable to recapture or exceed the previously achieved 100 \(W_{\text{Elec}}\) output. However, the seal loss is the single biggest source of inefficiency in the system, so improving the seals would be one of the top priorities in developing a second generation TaSEG.

\(^{12}\)In tests on 132-STAR alternators with smaller pistons, Qdrive has found that this method is effective at reducing the seal loss while not contributing significant rubbing, which increases contact between the seal and the piston bore resulting in increased losses of another kind while reducing the seal leakage loss.[42]
Figure 4.26. One of the power pistons after ribbing, with closeup of the ribbed piston surface.
Conclusions and Future Work

A thermoacoustic-Stirling engine generator or TaSEG, based on the traveling-wave principles described by Ceperley\[19, 20, 21\], Backhaus\[15\] and a Stirling thermodynamic cycle\[57\] has been built and tested. This TaSEG makes use of a realistic hot heat exchanger and QDrive\(^1\) linear alternators to convert heat into electricity via sound, delivering 100 W at a thermal-to-electric efficiency of 7\%\.[42] A novel method of Gedeon streaming suppression that addresses the lifetime concerns of the oscillating diaphragm option in a 15 year hermetically sealed micro combined heat and power appliance\[7\] has also been successfully demonstrated. In designing and implementing the jet plate within the TaSEG, the basic theory described by Backhaus and Swift\[15\] to suppress Gedeon streaming by relaying on the asymmetry of hydrodynamic end effects has been used. However, there are several distinguishing characteristics that set this work apart from theirs, including the limited installation space available for the jet plate within the TaSEG, the number of jet plate through-holes, the jet plate’s location within the TaSEG’s torus, and the jet plate’s inability to be adjusted from outside the pressure vessel.

While the reduction in the electrical output of the TaSEG with the aluminum jet plate suppressing Gedeon streaming is 12\% to 16\% of the total electrical power output, a reduction of 5 to 8 W in a TaSEG capable of outputting 1 kW of electrical power is less than 1\% and deemed commercially acceptable. Thus, the designed and evaluated aluminum jet plate is the Gedeon streaming suppression solution of choice for a commercially produced TaSEG. However, an in-depth investigation into how the reduction in TaSEG output due to the inclusion of the jet plate to suppress Gedeon streaming scales as the overall electrical output is increased from 100 W to 1 kW should be carried out. This scaling effect should be investigated both computationally using DELTAEC or SAGE\[61\] and experimentally.

While it may seem doubtful that a commercial product can make use of an element that must be adjusted by a trial-and-error method, it is important to note that the final form of the jet plate that suppresses Gedeon streaming should be applicable to all copies of a given TaSEG design.

Figure 5.1. Sankey diagram of power flow within the TaSEG as indicated by DELTAEC simulation. The simulation matched all important experimental operation parameters to better than 1%.

Therefore, individual jet plates will not have to be iteratively machined or honed to suppress Gedeon streaming for each TaSEG that rolls off the assembly line.

Although the TaSEG’s thermal-to-electric efficiency is relatively low compared to traditional Stirling engines, the experimental research presented in this thesis, in conjunction with other research[42], provides clear paths to improved performance. All experimental data has been matched with the simulation results to better than 1% for all the important operating parameters, with the exception of the heat flux. Additionally, there still remains disagreement between Gedeon streaming suppression Eqs. 2.17 and 2.13. The simulation predicts 15% less heat flux than what has been measured, while the aforementioned equations show a discrepancy on the order of hundreds of pascals. Other than these two points of contention, the TaSEG operates as expected, with no major technical unknowns or risk factors remaining, while the means to improve the overall thermal-to-electric efficiency are fairly straightforward.

As the simulation was well-verified by experiments, it is worth looking at a loss inventory from within the DELTAEC simulation. Figure 5.1 shows a Sankey diagram of one particular TaSEG operating point where the TaSEG produced 78-watts of electric output power. As Figure 5.1
indicates, the losses associated with the power piston seals (i.e., seal gap) are the single largest source of lost work, eating up 48 W per side (98 W total) of the 214 W of useful acoustic power that flows into the alternator segments.

What is a little surprising is that the next largest loss comes from the TaSEG’s collimator, which dissipates 88 W, nearly as much as the power piston seals. This value is more difficult to corroborate through first-principles calculations than the seal loss, as 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 this is indeed accurate, this suggests that the flow resistance that makes the TaSEG thermodynamic cycle work, also robs it of nearly a third of its useful power.

5.1 Paths to Improved TaSEG Performance

Summarizing what has been learned in this work and looking forward to the next phase of development, a list of areas that will yield the most improvement in a next-generation TaSEG includes:

The slightly convex shape of the temperature profiles across the regenerator during steady state operation suggests that the modified aluminum jet plate might be a bit “over tuned.” Further, experimentation should be done to get the exact geometry required to maintain a linear temperature profile across the regenerator during steady state operation. Additionally, the measurements performed on Gedeon streaming suppression only provide qualitative evidence of the theory discussed in Chapter 2. To allow for quantitative jet plate design in future TaSEG’s, further research into minor losses in oscillating flow is required to ease the incorporation of these losses into the TaSEG design and to provide better agreement between Eqs. 2.17 and 2.13, which was not found in this research.

The work by Petculescu and Wilen could prove a good starting point to investigate the source of this disagreement.[59, 60] Their findings suggest that as the cone angle of the jet plate through-hole is decreased, the point at which the flow separates from the wall of the through-hole occurs at a larger diameters, leading to a reduced minor loss.[60] Applying their conclusion to the jet plate in the TaSEG, it can be assumed that \( k_{\text{con},s} = 0.05 \) used in Eq. 2.17 is under estimated due to the large, 30° cone angle. Any increase in \( k_{\text{con},s} \) due to flow separating from the through-hole wall earlier than expected will decrease \( \Delta p_{ml} \). The decrease in \( \Delta p_{ml} \) results in better agreement between Eqs. 2.17 and 2.13. For example, setting \( k_{\text{con},s} = 0.9 \) results in Eqs. 2.17 and 2.13 having the same order of magnitude. The best solution to resolve this discrepancy would be to investigate if and how much \( k_{\text{con},s} \) is increased in the case of the TaSEG’s jet plate design.

Another option to consider is to allow the TaSEG to operate at whatever frequency results in the highest thermal-to-electric efficiency. This is one of the most important lessons learned in all the collective work. If the 50 Hz operating frequency is required to directly connect to the electrical grid, then higher power alternators should be used, a single alternator design if necessary, having a low enough natural frequency that the gas-spring forces in the back volume
can bring the alternator resonant frequency up to 50 Hz without requiring any additional moving mass. This also has a direct bearing on the piston seal question, as it is difficult to machine power pistons to fit the seals if the alternator spring suspension cannot support the moving mass properly, which is thought to be the case in the TaSEG discussed in this thesis.

Figure 5.2. Wear of Power Piston Rulon J Babbitt-Like Material.

It is believed that the heavy piston mass required caused a deflection in the gravitational axis as the pistons oscillated from equilibrium to maximum stroke. This in turn resulted in the pistons moving off center, changing the nominal seal gap during operation. This theory is supported by wear marks seen on the pistons after the TaSEG was run for an extended period. These wear marks were on the front and back edge of both the top and bottom of the pistons. Figure 5.2 shows the top of one piston with the wear areas on the front and back edge circled in red.

The higher TaSEG operating frequency option also offers the benefit that it is possible to get more power out of smaller alternators by running them faster. By running faster, it is conceivable to preserve the current TaSEG’s twin alternator configuration for intrinsic vibration cancellation, and still have a higher power density than traditional displacer Stirling engines, which cannot be so easily adapted to higher operating frequencies. Assuming that the moving mass on the alternators is appropriate for its suspension, a tighter effective seal gap should be possible. As the flow losses in the seal go as the cube of the seal gap, this is a major source of efficiency benefit. Additionally, having a lighter power piston mass on the alternator(s) and/or ribbing the Rulon may succeed in a further reduction of the seal gap.

In addition to a higher TaSEG operating frequency, flexure seals or bellows, which completely eliminate the power pistons seal gap should also be investigated as this is a viable option given a peak-to-peak stroke of 12 mm.\[62, 63\]

The hairpin-tube heat exchanger used on the TaSEG was a source of design difficulty and lost performance, compared to the standard internally finned hot heat exchanger designs used in most traditional free piston Stirling engines.\[57\] Based on the experimental results, there
is an enormous temperature difference, over 200 °C, between the tube metal and the working gas. Additionally, the sparse, long tubes also add inertance which hurts the thermodynamic cycle phasing. A more open finned hot heat exchanger design with more effective internal heat transfer will improve the overall efficiency greatly by more efficiently transferring heat from the hot surface to the helium and reducing the unwanted inertance of the tubes.

The collimator segment is a deliberate flow resistance inserted into the TaSEG to optimize the flow resistance and improve the cycle phasing, while costing lost work. Analysis of the TaSEG model and the Sankey diagram shows that this lost work is a very significant part of the total acoustic work generated by the TaSEG. In a generation II TaSEG analysis, understanding the trade-off between the required optimum flow resistance and its lost work, will be a very important precursor to a new design.

While the resistance heater provided both a convenient way to measure the TaSEG’s overall thermal-to-electric efficiency and enough flux to successfully run the TaSEG at steady state, data taken during operation from numerous thermocouples suggests that the heater did not heat the hot heat exchanger tubes uniformly. There was also a large temperature difference between the resistance element and the external surface of the hot heat exchanger tubes, since the latter were being radiantly heated, with no contact, and relatively little convection to assist the heat transfer. The temperature difference between the resistance heater’s external metal and the TaSEG hot heat exchanger tubes was about 200 °C. Even with a large amount of insulation around the heater, significant heat leak into the room was expected with the resistance heater metal running so hot. A nominal wire-mesh gas burner heat source similar to those found in traditional wall mounted condensing boilers would be better and should be seriously considered for future TaSEG’s to maintain a more uniform hot heat exchanger temperature.

While clearly there is much work to be done to optimize TaSEG technology for residential micro combined heat and power appliances, this research project has shown that this technology can be made practical for this application, and has provided insight on the paths to follow to reach competitive performance. Work on a second generation TaSEG prototype making use of a stationary jet plate, built to a specific micro combined heat and power product specification having an increased electrical output and thermal-to-electric efficiency, is currently on-going.
A.1 $\omega L \ll R$ Discussion

Using the calculated values of $R = 370,000 \text{ Pa} \cdot \text{s}^2 \cdot \text{m}^3$, $L = 9,600 \text{ kg} \cdot \text{m}^4$, and $C = 0.00036 \text{um}^3\text{Pa}^{-1}$ for the TaSEG, $\omega L = 3 \times 10^6 \text{ Pa} \cdot \text{s}^2 \cdot \text{m}^3$. Thus, for the TaSEG $\omega L$ is actually an order of magnitude larger than $R$, which is the opposite of what Backhaus and Swift state is required to assure that $U_{1,c}$ and $p_{1,c}$ are nearly in phase at the regenerator.[15, 28] Further insight can be gained by examining the following equation that appears in Ref. [15, 64].

$$U_{1,c} = \frac{p_{1,c}}{1 + \frac{R_{fb}}{R_{St}} + \frac{i\omega L}{R_{St}}} \left[ \frac{\omega^2 LC}{R_{St}} \left( 1 - \frac{iR_{fb}}{\omega L} \right) + \frac{i\omega R_0 C_0 g}{2R_{St}} \right]$$  \hfill (A.1)

where $R_{fb}$ is the resistance of engine’s feedback path, $R_{jp}$ is the resistance of a jet pump (if included in the engine), $R_{st}$ is the effective series resistance of the jet pump and regenerator referred to $U_{1,c}$. For all other variable definitions please see Ref. [15].

Based on the authors understanding of Eq. A.1, there is no specific approximation made with respect to the relative sizes of the feedback inerance and compliance. If $R_{St}$ is much larger than $i\omega L$, then the imaginary part of the denominator of the first term in Eq. A.1 is small (assuming $R_{fb}$ is also small). Here however, $\omega L$ is being compared with the combined resistance of the regenerator and jet plate $R_{St}$, not the resistance of the regenerator only. One might assume that the regenerator resistance would dominate $R_{St}$ and perhaps in a “good” engine design this is the case, but in the TASEG it does not. The DELTAECC model shows that the pressure drop across everything in the thermal branch (i.e. flow straighteners, heat exchangers, regenerator, jet plate and collimator) is greater than that across the inerance, with the majority of the thermal core pressure drop occurring across the collimator that was the deliberately added resistance.

In a Backhaus and Swift type engine design[28], where the engine feedback is dominated by the inerance and compliance in terms of size, the inerance impedance $L = \frac{\omega L}{A_l}$, will naturally be smaller than the regenerator resistance. As long as the relative cross-sectional area of the inerance is larger than that of the regenerator, there is a good chance that $L\omega \ll R$, and the engine does not need nearly as much deliberately added resistance to optimize its cycle phasing. It is surmised that the selection of a more compact thermoacoustic-Stirling engine design may
be a compromise of the Backhaus and Swift design because of what is feasible with compact inertances.[64]

As this topic has only been vaguely investigated and what is discussed here is only speculation, a deeper understanding of this aspect of the TaSEG is definitely required.

A.2 Jet Plate Mechanical Drawing
A.3 Matlab Script for Jet Plate Calculations

% This code calculates the pressure drop and jet pump resistance of the jet plate gas diode I designed.

rh = 6.8e-5; % m from deltaEC model
area = 3.70e-3; % m^2 X-sect area of regenerator from deltaEC model
pm = 3.358e6; % Pascal from deltaEC model
regenL = 3.8569e-2; % m Regenerator Length from deltaEC model

% Edot = ((9859.2.*x)- 89.173); % Acoustic Power as a function of Position from file delta pml calc.xls
% mux = ((3.8939e-4.*x)+ 4.0331e-6); % Viscosity through regenerator as a function of position from file delta pml calc.xls

% deltap = (6/area*rh^2*pm)*(integral of Edot*mux over regeneratorlength)) equation 7.74 from Swift for the pressure drop across the regenerator required to drive correct U2,0 (second order time independent volume velocity). % in low Reynolds number limit.

x = sym('x'); % Defines x as a symbol

integral = double(int(((9859.2.*x)- 89.173)*(((3.8939e-4.*x)+ 4.0331e-6)),x,4.12e-2,7.98e-2));
deltap_regen = (6/(area*rh^2*pm))*(integral)

b = .68; % from Backhaus ASA 2000 for helium
Tc = 299; % K cold end of regen temperature
Th = 664; % K hot end of regen temperature
muc = 1.9884e-5; % kg/m*s cold end viscosity
Wc = 315.50; % W time average acoustic power flowing into regenerator cold end
tau = Th/Tc; %
deltap_regen2 = (6*muc*regenL)/((b+2)*area*rh^2*pm ))*(((tau^(b+2))-1)/(tau-1))*Wc; %

% Resistance of designed jet plate

rhom = 5.3962; % kg/m^3 mean density of helium
U1jp = 6.8159e-3; % m^3/s taken at end of collimator from DeltaEC model rb-mb-shortoff-Mar28c.out
holes = 15; % Number of holes in designed jet plate
rs = (4.25/2)*(1e-3); % m as designed values that Henny used on aluminum prototype tarq=pi*(holes);
rh = (12.95/2)*(1e-3); % m as designed values that Henny used on aluminum prototype tarq=pi*(holes);

% Area of small holes in jet plate
as = holes*pi*rs^2;
% Area of large holes in jet plate
ab = holes*pi*rb^2;

Kexps = 1.05; % Expanding Minor loss coefficient
Kexpb = 1.05; % Expanding Minor loss coefficient - Both Kexp values are assumed for an abrupt transition where steady flow expands into a much larger cross-sectional area as in the case for flow exiting the holes.
Kcons = 0.05; % Contracting Minor loss coefficient small holes
Kconb = 0.5; % Contracting Minor loss coefficient large holes, 0.5 is assumed for sharp edges. Kcon values strongly depend on the geometrical details of the holes, including edge effects such as rounding and or sharpness. ang = 15; % degrees angle of taper between large and small holes in jet plate.

% taper height from small to large holes
h = ((rhom*U1jp^2)/(8*as^2)) % (Kexp = Kcons) % Contracting Minor loss coefficient small holes

% pressure drop across the jet plate Pascal
E_diss = ((rhom*U1jp^3)/(3*pi*as^2)) % (Kexp = Kcons) %

% time averaged acoustic power dissipation by jet plate
Rjp = ((2*rhom*U1jp)/(3*pi*as^2)) % (Kexp = Kcons)

% D. Wilcox March 2010 close all; clear all; clc; format short;
A.4 Instrumentation Data Sheets

Industrial, General-Purpose

Part No.
PQI-63B-PG600-LAOX-J

Part Description:
Industrial Pressure Gauge, Adj Pointer, 63 mm, 0 to 600 psi, bar secondary, Lower Mount, 1/4 in. MNPT

Product Specifications

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Caution: Do not mix or interchange valve components with those of other manufacturers.

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8510B -200,-500,-2000
Piezoresistive pressure transducer

Features

• 100 to 2000 psi, 300 mV full scale
• Rugged, miniature

Description

The Endevco® Model 8510B is a rugged, miniature, high sensitivity piezoresistive pressure transducer. It has a 10-32 UNF-2A threaded case, 0.438 inch (11.12 mm) long, and is available in ranges from 1 psi to 2000 psi. It is highly reliable, combined with high noise margin at ideal for measuring dynamic pressure.

Endevco piezoresistive pressure transducers feature a four active arm strain gage bridge for maximum sensitivity and a wideband frequency response. This combination of high sensitivity and wideband frequency response provides stable performance over the temperature range of -65°F to +250°F (-54°C to +121°C), and also feature excellent linearity over 3X range, high shock resistance, and negligible sensitivity to temperature transients.

The model 8510B is designed for a wide variety of aerospace, automation, and industrial environments, where it must be expected to operate at maximum pressure. It is intended for use in reference and transducer calibration. The model 8510B is also suitable for high sensitivity measurements.

The Endevco model 8510B is available with H11 mounting format as 8510B-H11R on special order.

Endevco model 8510B Three-Channel System, model 136B, or 4430A signal conditioner, or OASIS 2000 computer-controlled system are recommended as signal conditioner and power supply.

Specifications

- Dynamic Performance Specifications: All specifications conform to ISA-RP-37.2 (1964) and are typical values, referenced at +75˚F (+24˚C), 100 Hz and 10 Vdc.
- Frequency response: 0.02 Hz to 1 MHz.
- Power supply: 10.0 Vdc recommended, 18.0 Vdc maximum.
- Range: 8510B-200 -500 -2000
- Mounting thread: 10-32 UNF-2A threaded case, 0.15 inch (3.8 mm) face diameter and is compatible with 1/8-27 NPT and BSPT thread as 8510B-XXM5 on special order.
- Mounting torque: 0.5 lbf-in (6.1 Nm).
The Endevco® model 136 is a three-channel, DC amplifier that is manually or computer program- mable. Feature control is accomplished through a soft-key panel, push buttons, three DC channels, LED readouts, a "select function" push button, the "function LED", a four-character LED display, the state of each function/channel, and/or "indie" push buttons to change the entries in the LED display. There are three LEDs used as fault status indicators for the auto zero function. Computer control is accomplished using the standard RS-232 port and optional application software.

There are two modes of operation: Normal and Programming. Both modes of operation utilize the front panel LED display in the Normal mode. There are two states: Monitoring mode and Programming. In the Monitoring mode, the LED display indicates the RMS reading of the signal present at the output of the selected channel. The Normal Programming mode turns off the LED display. For lower noise applications and to minimize power consumption, in the Programming mode, the LED display is blanked. The LED display is blanked when in the Programming mode and/or the "monitoring state" button is pushed.

The rear panel contains a 4-pole 9-pin "D" connector, a 9-pin "D" connector, and the input power connector. Three models of the 136 unit may be configured in a 19-inch rack mount setup. The units will automatically return to the Normal state with the "monitoring state" push button while the "monitoring state" function LED is flashing.

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The rear panel contains a 4-pole 9-pin "D" connector, a 9-pin "D" connector, and the input power connector. Three models of the 136 unit may be configured in a 19-inch rack mount setup. The units will automatically return to the Normal state with the "monitoring state" push button while the "monitoring state" function LED is flashing.

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The model 136 units may be configured in a 19-inch rack mount setup. The models will automatically return to the Normal state with the "monitoring state" push button while the "monitoring state" function LED is flashing.
Basic In-line Liquid Flow Rate Monitors

**FEATURES**

- **Choice of Three Materials of Construction**
- **Unrestricted Mounting**
- **Superior Exterior Design**
- **Good Viscosity Stability**
- **Rugged and Reliable**
- **High Pressure Operation**
- **24 Different Ports Available**
- **Bi-Directional and Reverse Flow Option Offered**

**MONITOR SPECIFICATIONS**

- **Measuring Accuracy**:
  - ±2.5% of range accuracy in center third of scale;
  - ±4% in upper and lower thirds.

- **Flow Measuring Range**:
  - .05-150 GPM (0.2-560 LPM)

- **Repeatability**:
  - ±1% of full-scale

- **Pressure Differential**:
  - See graphs on the right for typical pressure differentials. For specific differential graphs, refer to Lake data sheet PDDS-404.

- **Maximun Operating Temperature**:
  - 240ºF (116ºC)

- **Filtration Requirements**:
  - 74 micron filter or 200 mesh screen minimum

**TYPICAL PRESSURE DIFFERENTIALS**

**20 SERIES**

**30 SERIES**

**40 SERIES**

**60 SERIES**

**SUBCIRCUITS**

- Ideal for monitoring case drain flows, pump performance and media flows through hydraulic circuits and sub-circuits

- Designed as a hydraulic service tool, this monitor will provide years of maintenance-free performance.

- Standard selection of NPT, SAE and BSP ports reduces the amount of adapters required for installation.

- Selection of NPT, SAE and BSP ports provides excellent measurement stability for viscous flows from (0-500 SSU).

**ENGINEERING SPECIFICATION**

- **The In-line Flow Rate Monitor Shall:**
  - Use the annular-orifice technique with a stainless steel sharp-edged orifice.
  - Not require inlet or outlet straight plumbing, or vertical pipe mounting.
  - Have a measuring accuracy of ±2.5% of full scale in the center third of the measuring range, and ±4% in upper and lower thirds.
  - Have stainless steel sharp-edged orifice.
  - Have a weather-tight external construction.
  - Be Lake Monitors No. B _ _ – _ _ _ – _ _ _.

- **Materials of Construction**
  - Wetted Components: Aluminum, Brass or Stainless Steel.
  - Non-wetted Components: Polycarbonate, Teflon®, Viton® or Kalrez®.

- **Pressure Casing**:
  - Aluminum, Brass or Stainless Steel.

- **Window Tube**:
  - Polycarbonate, Teflon® coated Alnico.

- **Transfer Magnet**:
  - Viton® or Kalrez® coated Alnico.

- **Window Seals**:
  - Buna-N, Teflon®, Viton®.

- **All Other Internal Parts**:
  - Stainless Steel.

- **Port Sizes**:

- **High-Pressure Casing**:
  - Aluminum, Brass #303 Stainless Steel.

- **Sub-Circuits**:
  - Designed as a hydraulic service tool, this monitor will provide years of maintenance-free performance.

- **Mounting**:
  - Unrestricted mounting.

- **Materials of Construction**:
  - Wetted Components: Stainless Steel, Stainless Steel, Stainless Steel.
  - Non-wetted Components: Stainless Steel, Stainless Steel, Stainless Steel.

- **External Construction**:
  - Weather-tight for use outdoors and/or on systems where wash downs are required.

- **Superior Exterior Design**:
  - Allows the designer to install the monitor in any orientation - horizontal, vertical, or inverted.

- **Good Viscosity Stability**:
  - A sharp-edged stainless steel orifice provides excellent measurement stability for viscous flows from (0-500 SSU).

- **Low Cost Accuracy**:
  - ±4% in upper and lower thirds.

- **Engineering data sheet**
  - See our High Temperature data sheet.

- **For specific differential graphs, refer to Lake data sheet PDDS-404.

- **Contact Lake Monitors for more information.**

**CONTACT**

Lake Monitors
Basic In-line Liquid Flow Rate Monitors

**AW-LAKE COMPANY INC.**

262.884.9800 / Fax: 262.884.9805 / 800.850.6110

2863 ERASER STREET
Franksville, WI 53126

www.lakemonitors.com
VIP series: sensors with measuring ring and integral electronics

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

Parallel mounting:
The optimum ratio of measurement range to installed length of the sensor reduces the installation space needed for the VIP series. The parallel connection of the measurement object and measuring ring facilitates completely new construction and installation options. Whereas with conventional sensors with an axial measurement path, the length of the plunger must be added to the actual housing length, with the VIP series only the housing length has to be considered during the design.

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Basic Performance of the 3331/3332

- Evaluation of electric equipment such as inverters
  - High basic accuracy of ±0.4%
  - Measurement range and high accuracy for a wide range of power measurement from standby to normal usage.
- Simultaneous measurement of active power and reactive power.
  - Stability for transient power fluctuations.
- Simultaneous measurement of current and power at a 3-phase high-sensitivity data.
  - Measurement of 45Hz to 66Hz or ±0.1%rdg. or ±0.2%f.s. (±1.0%f.s. for reactive power and ±2.0%f.s. for active power).
- Systems can be easily constructed
  - A compact design that fits a half-inch rack-mountable size available at a special order.
  - GF/IR/RS-232C: Data can be transmitted to a printer or computer for efficient data management.
- Accurate evaluation of consumption power of electrical products
  - Power measuring instruments 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.

The 3332 covers a wide range of power measurement capacity.

<table>
<thead>
<tr>
<th>Model</th>
<th>3331</th>
<th>3332</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage and I</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>Voltage</td>
<td>100V</td>
<td>200V</td>
</tr>
<tr>
<td>I</td>
<td>15A</td>
<td>30A</td>
</tr>
</tbody>
</table>
| Accuracy | ±0.4% | ±0.2%

The 3331 is compatible from single-phase to 3-phase devices

<table>
<thead>
<tr>
<th>Range Table 3331</th>
<th>3332</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage</td>
<td>500V</td>
</tr>
<tr>
<td>I</td>
<td>1A</td>
</tr>
</tbody>
</table>
| Accuracy | ±0.4% | ±0.2%

For apparent power measurement, values of voltage and current are multiplied together to give the value of apparent power.

The 3332: Single-phase, 3-wire and three-phase, 3-wire type for measuring power of large-scale equipment

Accurate evaluation of consumption power of electrical products

As efficient use of energy for household and office equipment becomes more and more essential, the new HIROKI POWER TESTER 3331 does the job by offering a wide range of power measurement from standby to normal usage. The POWER TESTER 3331 is capable of evaluating 2-phase, 3-phase, and tri-phase air conditioners and refrigerators, and is also compatible with a broad band of devices. The 3331 and 3332 are ideal in a measuring range for a wide range of purposes from research and development to equipment evaluation.

- Baseline performance of ±0.1%rdg. ±0.2%f.s.
- Current waveform peak measurement function
- Voltage accuracy: ±0.5% (within 1 hour of warm-up time) ±0.4% (within 7 hours of warm-up time) ±0.3% (within 16 hours of warm-up time)
- Input voltage range: 100V to 300V (±3%)
- Input current range: 20A (±2%)
- Frequency range: 2Hz to 10kHz
- Resolution: 10mA (±0.5%)
- Response time: 0.1s (±2%)
- Communication terminal: RS-232C (for computer connection), GPIB (for printer connection)
- Battery life: Approx. 10 hours (45% to 66% of maximum voltage, current, and frequency)
- Dimensions: Approx. 210W × 100H × 261D mm

For apparent power measurement, values of voltage and current are multiplied together to give the value of apparent power.
### DS1103 PPC Controller Board

#### Highlights
- **Single-board system** with real-time processor and comprehensive I/O.
- **Cells interface and serial interfaces** suited to automotive applications.
- **High I/O speed and accuracy**.
- **Easy access** for accurate I/O state selection.

#### Application Areas
- The controller board is designed to meet the requirements of modern rapid control prototyping and is highly suitable for applications such as:
  - **Automotive controls**
  - **Industrials user control**
  - **Positioning systems and robot control**
  - **Active vibration control**

An integrated feature: Cells interface controller makes the board a attractive tool for automotive and automation applications.

#### Key Benefits
- The DS1103 is an on-board, digital controller board for automotive applications. It provides a great selection of features, including:
  - **Automation controls**
  - **Industrials user control**
  - **Positioning systems and robot control**
  - **Active vibration control**

#### Recording and Output of I/O Values
The control of electrical drives requires accurate recording and output of I/O values, to position the A/D-channels and D/A-channels, and the position of the controller board in the automotive applications. The combination of signal converters and digital interfaces is suitable for a high-quality input to achieve absolutely accurate I/O state selection.

### Technical Details

<table>
<thead>
<tr>
<th>Feature</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>CPU  clock</td>
<td>10 MHz</td>
</tr>
<tr>
<td>Cols</td>
<td>5000 10/100 Mbit/s</td>
</tr>
<tr>
<td>Data  rate</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>Bus frequency</td>
<td>100 MHz</td>
</tr>
<tr>
<td>Memory</td>
<td>128 MB (2x 64 MB)</td>
</tr>
<tr>
<td>On-board memory</td>
<td>128 MB (2x 64 MB)</td>
</tr>
<tr>
<td>Processor</td>
<td>32-bit (32-bit)</td>
</tr>
<tr>
<td>On-board memory</td>
<td>128 MB (2x 64 MB)</td>
</tr>
<tr>
<td>Processor</td>
<td>32-bit (32-bit)</td>
</tr>
<tr>
<td>Internal memory</td>
<td>128 MB (2x 64 MB)</td>
</tr>
<tr>
<td>External memory</td>
<td>128 MB (2x 64 MB)</td>
</tr>
<tr>
<td>JTAG interface</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>JTAG interface</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>AVI converter</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>D/A converter</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>A/D converter</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>Digital I/O channel</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>Analog I/O channel</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>Local Bus</td>
<td>100 Mbit/s</td>
</tr>
<tr>
<td>Local Bus</td>
<td>100 Mbit/s</td>
</tr>
</tbody>
</table>

### Order Information

**Ordering Code:** DS1103C-XXXX

**Software and Hardware**

- **Software:**
  - LABVIEW® Real-Time Design
  - MotionDesk (p. 202)
  - dSPACE RTI (p. 126)
  - dSPACE Softwares (p. 116)

- **Hardware:**
  - DS1103C-XXXX
  - DS1103C-XXXXA

### Relevant Software and Hardware

**Software**

- **MotionDesk (p. 202)**
- **Real-Time Interface (RTI) (p. 126)**
- **Real-Time Interface CAN Blockset (p. 136)**
- **DS1103 Real-Time Library – CLP1103**
- **CLIB**
- **MLIB/MTRACE**

**Hardware**

- **DS1103C-XXXXA**
- **DS1103C-XXXXB**
- **DS1103C-XXXXC**

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**Block Diagram**

[Diagram of DS1103 PPC Controller Board]

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**Catalog 2011 • dSPACE GmbH • Rathenaustraße 26 • 33102 Paderborn • Germany • info@dspace.de • www.dspace.com**
Agilent dc Electronic Loads
Models N3300A-N3307A

Increase your manufacturing test throughput with Fast electronic loads

- Increase test system throughput
- Lower cost of ownership
- Decrease system development time
- Increase system reliability
- Increase system flexibility
- Stable operation down to zero volts
- dc connection terminal for ATE applications

Supplemental Characteristics

<table>
<thead>
<tr>
<th>Series</th>
<th>N3302A</th>
<th>N3303A</th>
<th>N3304A</th>
<th>N3305A</th>
<th>N3306A</th>
<th>N3307A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Command processing time (using discrete commands)</td>
<td>3 ms</td>
<td>3 ms</td>
<td>3 ms</td>
<td>3 ms</td>
<td>3 ms</td>
<td>3 ms</td>
</tr>
<tr>
<td>Measurement time</td>
<td>0.1 ms</td>
<td>0.1 ms</td>
<td>0.1 ms</td>
<td>0.1 ms</td>
<td>0.1 ms</td>
<td>0.1 ms</td>
</tr>
<tr>
<td>Line-drop characteristics</td>
<td>Range 1: 0 - 15 V, Range 2: 15 - 30 V, Range 3: 30 - 60 V, Range 4: 60 - 120 V</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inrush current</td>
<td>38 A at 115 Vac, 36 A at 230 Vac</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table A-2 lists the supplemental characteristics, which are not warranted but are described as of typical performance determined either by design or testing.

Increase Test Throughput

Today’s high volume manufacturing requires optimization of test systems throughout, in response to product volume without increasing footprint. The N3300A family electronic loads can help you in a number of ways to achieve this goal:

- Reduced command processing time: Commands are processed from 3 times faster than previous electronic loads.
- Automatically assign command sequence: “Lists” of preprogrammed command sequences can execute independently of the computer, greatly reducing the time needed for test setup and computer interaction time during production testing.
- Programmable delay allows for either simultaneous or sequential load change: This is the most efficient way to conduct testing of multiple output dc power supplies simulating stress test conditions, with a choice of programming options.

Safety measurement data: Voltage, current and power measurements can be displayed to allow feedback for the computer, reducing computer interaction.

Control measurement speed vs. accuracy: Determine the number of samples in accordance with modified measurement speed, or calculate the number of samples to achieve higher measurement accuracy. You can calculate your measurements for each test.

Control ringing and line-shake: Separately: Reduce rate of loading or unloading to avoid ringing or line-shake under multiple conditions, but otherwise change load values at maximum rate.

Increase System Flexibility…for Both Present and Future Requirements

Most power supply and battery charger test systems designed today need to test a variety of products and/or assemblies. In the future, additional products or assemblies may be needed. A flexible family of electronic loads, standard design and future growth make sense.

Test low voltage power supplies: The N3300A series electronic loads operate with full stability in zero volts. Many other electronic loads available today have been found to become unstable in the operating range below zero volts. When designing power supply test platforms, the trend towards lower voltage test systems will continue, making the N3300A series the logical choice.

Supplemental Characteristics

<table>
<thead>
<tr>
<th>Series</th>
<th>N3302A</th>
<th>N3303A</th>
<th>N3304A</th>
<th>N3305A</th>
<th>N3306A</th>
<th>N3307A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Command processing time (using list commands)</td>
<td>20 ms</td>
<td>20 ms</td>
<td>20 ms</td>
<td>20 ms</td>
<td>20 ms</td>
<td>20 ms</td>
</tr>
<tr>
<td>Measurement time</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
</tr>
<tr>
<td>Line-drop characteristics</td>
<td>Range 1: 0 - 15 V, Range 2: 15 - 30 V, Range 3: 30 - 60 V, Range 4: 60 - 120 V</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
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<td>Inrush current</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table A-2 (continued)

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<td>20 ms</td>
<td>20 ms</td>
<td>20 ms</td>
</tr>
<tr>
<td>Measurement time</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
<td>100 ms</td>
</tr>
<tr>
<td>Line-drop characteristics</td>
<td>Range 1: 0 - 15 V, Range 2: 15 - 30 V, Range 3: 30 - 60 V, Range 4: 60 - 120 V</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table A-2 (continued)
Agilent 33220A 20 MHz Function/Arbitrary Waveform Generator

Measurement Characteristics (continued)

### Measurement Characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phase Noise</td>
<td>≤ 7° RMS</td>
</tr>
<tr>
<td>Slew Rate</td>
<td>≤ 400V/μs</td>
</tr>
<tr>
<td>Frequency Stability</td>
<td>± 1 ppm/°C</td>
</tr>
<tr>
<td>Accuracy for the life of those products</td>
<td></td>
</tr>
<tr>
<td>Linearity</td>
<td>&lt; 0.1% of peak output</td>
</tr>
<tr>
<td>Min. Rise/Fall Time</td>
<td>35 ns typical</td>
</tr>
<tr>
<td>Common Characteristics</td>
<td></td>
</tr>
<tr>
<td>Bandwidth</td>
<td>9 MHz typical</td>
</tr>
<tr>
<td>Overshoot</td>
<td>&lt; 2%</td>
</tr>
<tr>
<td>Jitter (RMS)</td>
<td>1 ns +</td>
</tr>
<tr>
<td>Asymmetry (@ 50% duty)</td>
<td>100 ppm of period</td>
</tr>
<tr>
<td>Rise/Fall time</td>
<td>&lt; 13 ns</td>
</tr>
<tr>
<td>Total harmonic distortion[2],[3]</td>
<td></td>
</tr>
<tr>
<td>Measurement Characteristics</td>
<td></td>
</tr>
<tr>
<td>Power Supply</td>
<td>±10 V DC</td>
</tr>
<tr>
<td>Power Supply frequency</td>
<td>50 or 60 Hz</td>
</tr>
<tr>
<td>Power Supply current</td>
<td>1 A maximum</td>
</tr>
<tr>
<td>Power Supply voltage</td>
<td>100 V maximum</td>
</tr>
<tr>
<td>Power Supply capacity</td>
<td>150 W maximum</td>
</tr>
<tr>
<td>Operating Temperature</td>
<td>-30°C to 70°C</td>
</tr>
<tr>
<td>Storage Temperature</td>
<td>-30°C to 70°C</td>
</tr>
<tr>
<td>Humidity</td>
<td>5% to 95%</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>23°C ± 5°C</td>
</tr>
<tr>
<td>Altitude</td>
<td>2000 m (6562 ft)</td>
</tr>
<tr>
<td>Linearity</td>
<td>&lt; 0.1% of peak output</td>
</tr>
<tr>
<td>Min. Rise/Fall Time</td>
<td>35 ns typical</td>
</tr>
<tr>
<td>Common Characteristics</td>
<td></td>
</tr>
<tr>
<td>Bandwidth</td>
<td>9 MHz typical</td>
</tr>
<tr>
<td>Overshoot</td>
<td>&lt; 2%</td>
</tr>
<tr>
<td>Jitter (RMS)</td>
<td>1 ns +</td>
</tr>
<tr>
<td>Asymmetry (@ 50% duty)</td>
<td>100 ppm of period</td>
</tr>
<tr>
<td>Rise/Fall time</td>
<td>&lt; 13 ns</td>
</tr>
<tr>
<td>Total harmonic distortion[2],[3]</td>
<td></td>
</tr>
</tbody>
</table>

### Ordering Information

**Agilent 33220A**

**20 MHz Function/Arbitrary Waveform Generator**

**Part Number:** 33220A

**Supplied accessories:**

- AC power cord
- USB cable
- CD containing software

**Supplied software:**

- Agilent N9913A Test and Measurement Desktop Software
- Agilent N9910A Test and Measurement Desktop Software
- Agilent N9914A Test and Measurement Desktop Software

**Modular options:**

- Agilent N9916A Test and Measurement Desktop Software
- Agilent N9917A Test and Measurement Desktop Software

**Options:**

- Agilent N9918A Test and Measurement Desktop Software
- Agilent N9919A Test and Measurement Desktop Software
- Agilent N9920A Test and Measurement Desktop Software

**Rack mount:**

- 212.8mm x 88.3mm

**Bench top:**

- 261.1 mm x 103.8 mm

**Linearity:**

- < 0.1% of peak output

**Min. Rise/Fall Time:**

- 35 ns typical

**Common Characteristics:**

- Bandwidth: 9 MHz typical
- Overshoot: < 2%
- Jitter (RMS): 1 ns +
- Asymmetry (@ 50% duty): 100 ppm of period
- Rise/Fall time: < 13 ns
Agilent Technologies

E3640A – E3649A Programmable DC Power Supplies

Data Sheet

Features

- Single and dual outputs
- Dual range output
- 30 W to 100 W output power
- Front and rear output terminals
- Over-voltage protection
- Remote diagnostics
- GPIB and RS-232 standard
- Save and recall functions

Great Performance, Outstanding Price

With the output power of 30 to 100 W, the Agilent E364xA series programmable DC power supplies provide excellent regulation and accuracy at a competitive price. All four models deliver clean power, with low noise and low disturbances. They also specify less than 90 msec injection. Agilent E364xA power supplies are able to maintain a steady output when power line and load changes occur. They also specify 0.01 percent load and line regulation for all models. They also specify accurate time for output to start and drop after an OVP condition occurred.

Steady Output

With 30 W to 100 W output power, the Agilent E364xA series programmable DC power supplies are able to maintain a steady output when power line and load changes occur. They also specify normal mode voltage noise and low disturbances. They also specify 0.01 percent load and line regulation for all models. They also specify accurate time for output to start and drop after an OVP condition occurred.

Remote Interface

The E364xA Series employs a cooling flange kit that requires Agilent Support Rails-E3663AC or customer support rails. The rear panel and screw-type terminals on the front panel and versatile binding posts on the rear panel. New front panel binding posts allow you to easily connect a GPIB interface. The output on/off button sets the output to off. If you press a dual output model, you can view two voltages or currents that are displayed simultaneously.

Versatile Power

Agilent E364xA power supplies give you the flexibility to select dual output ranges. Output load is protected against overcurrent, which can be safely monitored and adjusted from the front panel and remote interface. Remote monitoring is available in the rear panel to sélection errors caused by voltage drop on the line. Selecting the output model, you can view two voltages or currents that are displayed simultaneously.

Broad Support

Excellent error specification is available at minimum 10 mV and limited by front panel resolution. The SCPI Reference Guide includes sufficient information on programming and simple programming procedures. Besides, the user manual provides detailed information on programming all the commands. You can also use the SCPI Reference Guide to program the E364xA Series power supplies. The E364xA Series employs a cooling flange kit that requires Agilent Support Rails-E3663AC or customer support rails. The rear panel and screw-type terminals on the front panel and versatile binding posts on the rear panel. New front panel binding posts allow you to easily connect a GPIB interface. The output on/off button sets the output to off. If you press a dual output model, you can view two voltages or currents that are displayed simultaneously.

Ordering Information

Agilent E364xA Series Power Supplies

<table>
<thead>
<tr>
<th>Model Number</th>
<th>E3640A</th>
<th>E3641A</th>
<th>E3642A</th>
<th>E3643A</th>
<th>E3644A</th>
<th>E3645A</th>
<th>E3646A</th>
<th>E3647A</th>
<th>E3648A</th>
<th>E3649A</th>
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<td>30 W</td>
<td>50 W</td>
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<td>0 to 20 V</td>
<td>0 to 20 V</td>
<td>0 to 20 V</td>
<td>0 to 20 V</td>
<td>0 to 20 V</td>
<td>0 to 20 V</td>
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<td>Current Range</td>
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<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
<td>1.5 A</td>
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Rackmount Kits

To rackmount two instruments side by side:

<table>
<thead>
<tr>
<th>E3640A-E3649A</th>
<th>E3644A-E3649A</th>
</tr>
</thead>
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<tr>
<td>18U Rackmount Kit (P/N 5063-9256)</td>
<td>18U Rackmount Kit (P/N 5063-9256)</td>
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</table>

Standard Shipped Accessories

- Unit & Earring guide
- Product Reference Guide
- SCPI Reference Guide
- RS-232 Interface
- Windows 98® and NT® drivers are supported under Microsoft® Windows 98® and NT®.

Acknowledgement

Microsoft Windows 98 and Windows NT are US registered trademarks of Microsoft Corporation.
B.1 ΔELTAEC Schematic

B.2 ΔELTAEC Model

Please note that this model has been simplified for printing purposes.
0 BEGIN the setup

3.3580E+06 a Mean P Pa
55.509 b Freq Hz
299.56 c Tbeg K
2.6319E+05 d \(|p|\) Pa
0.0000 e Ph(p) deg
0.0000 f \(|U|\) m\(^3\)/s
0.0000 g Ph(U) deg

Optional Parameters

helium Gas type

1 INSULATE Insulate engine segments
2 TBRANCH split to Stirling

4.8779E+07 a Re(Zb) Pa-s/m\(^3\)
-5.2661E+07 b Im(Zb) Pa-s/m\(^3\)

299.56 c HtotBr W

479.09 c HtotBr W

479.09 b HtotBr W
-327.88 G HdotBr W

3 DUCT Change Me

Same 5a 3.5530E-03 a Area m\(^2\)
1.0000E-04 b Perim m
1.0000E-07 c Length m

3.6002E-03 C |U| m\(^3\)/s

479.09 b Htot W

479.09 b Htot W
327.88 G Hdot W
299.56 G TBeg K
299.56 H TEnd K

ideal Solid type

4 JOIN dq=0 to dT=0 interface

2.6319E+05 A \(|p|\) Pa
-2.8732E-08 B Ph(p) deg
3.6665E-03 C |U| m\(^3\)/s

479.09 b Htot W
327.88 G Hdot W

3 STKDUCT expansion space

Same 5a 3.5510E-03 a Area m\(^2\)
0.45077 b Perim m
1.0000E-04 c Length m
1.0000 a WallA m\(^2\)

2.6348E+05 A \(|p|\) Pa
-8.9709E-02 B Ph(p) deg
3.2415E-03 C |U| m\(^3\)/s
42.354 D Ph(U) deg

-681.15 e HeatIn W
-202.06 E Htot W
315.51 F Hdot W
299.56 G TBeg K
299.56 H TEnd K

ideal Solid type

315.51 G Hdot W

5 STKSCREEN regenerator

Same 5a 3.5510E-03 a Area m\(^2\)
0.45077 b Perim m
1.0000E-04 c Length m
1.0000 a WallA m\(^2\)

2.6119E+05 A \(|p|\) Pa
-8.9709E-02 B Ph(p) deg
3.2415E-03 C |U| m\(^3\)/s
42.277 D Ph(U) deg

-202.06 E Htot W
315.51 F Hdot W
299.56 G TBeg K
299.56 H TEnd K

stainless Solid type

299.56 H TEnd K
<table>
<thead>
<tr>
<th>Page</th>
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<tbody>
<tr>
<td>117</td>
<td>stainless Solid type 299.45 H TEnd K</td>
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<tr>
<td>118</td>
<td>14 SX Ambient Flow Straightener</td>
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<td>119</td>
<td>Same 11a 2.1470E-03 a Area m² 2.5851E+05 A</td>
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<td>120</td>
<td>b VolPor -0.50593 B Ph(p) deg</td>
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<td>121</td>
<td>c Length m 6.0233E-03 C</td>
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<td>122</td>
<td>d rh -35.51</td>
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<td>123</td>
<td>e HeatIn W</td>
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<td>124</td>
<td>15 DUCT FS retainer holes 637.72 E Hdot W</td>
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<td>125</td>
<td>Possible targets 299.45 G GasT K</td>
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<tr>
<td>126</td>
<td>stainless Solid type 299.45 H SolidT K</td>
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<td>127</td>
<td>6.4300E+05 A</td>
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<td>130</td>
<td>b Perim m -0.53876 B Ph(p) deg</td>
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<td>c Length m 6.2324E-03 C</td>
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<td>134</td>
<td>e HeatIn W</td>
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<td>135</td>
<td>16 DUCT Collimator 636.51 F Edot W</td>
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<td>136</td>
<td>Optional Parameters 636.51 F Edot W</td>
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<td>137</td>
<td>6.0000E-04 a Area m² 2.5825E+05 A</td>
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<td>138</td>
<td>b Perim m -0.57134 B Ph(p) deg</td>
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<td>139</td>
<td>c Length m 6.2911E-03 C</td>
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<tr>
<td>140</td>
<td>d Srough -39.104 B Ph(U) deg</td>
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<tr>
<td>141</td>
<td>Master-Slave Links 695.48 B Htot W</td>
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<tr>
<td>142</td>
<td>17 MINOR FS retainer orifi minor loss 695.48 B Htot W</td>
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<td>143</td>
<td>Optional Parameters 635.45 F Edot W</td>
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<td>Stainless Solid type 695.48 B Htot W</td>
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<td>145</td>
<td>6.5000E-05 a Area m² 2.4140E+05 A</td>
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<td>146</td>
<td>b Perim m 2.6896 B Ph(p) deg</td>
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<td>147</td>
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<td>d Srough -39.104 D Ph(U) deg</td>
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<td>149</td>
<td>Master-Slave Links 695.48 B Htot W</td>
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<td>150</td>
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<td>Optional Parameters 654.55 F Edot W</td>
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<td>Same 17a 6.5000E-05 a Area m² 2.4082E+05 A</td>
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<td>b Perim m 2.7114 B Ph(p) deg</td>
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<td>Master-Slave Links 695.48 B Htot W</td>
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<td>157</td>
<td>Stainless Solid type 695.48 B Htot W</td>
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<td>158</td>
<td>Optional Parameters 564.55 F Edot W</td>
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<td>159</td>
<td>19 DUCT Stainless Solid type 564.55 F Edot W</td>
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<td>160</td>
<td>Same 19a 4.3800E-03 a Area m² 2.4080E+05 A</td>
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<td>161</td>
<td>b Perim m 2.7060 B Ph(p) deg</td>
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<td>162</td>
<td>c Length m 6.8159E-03 C</td>
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<td>163</td>
<td>d Srough -43.853 D Ph(U) deg</td>
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<td>164</td>
<td>Master-Slave Links 695.48 B Htot W</td>
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<tr>
<td>165</td>
<td>Optional Parameters 564.26 F Edot W</td>
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<td>166</td>
<td>Stainless Solid type 695.48 B Htot W</td>
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<tr>
<td>167</td>
<td>20 SOFTEND end of engine 695.48 B Htot W</td>
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<tr>
<td>168</td>
<td>Possible targets 654.26 F Edot W</td>
</tr>
<tr>
<td>169</td>
<td>19.358 G Re(z) 20.441 H Im(z)</td>
</tr>
</tbody>
</table>
21 COMPLIANCE compliance

2.6400E-02 a SurfAr m^2 2.6319E+05 A |p| Pa
1.7600E-04 b Volume m^3 0.0000 B Ph(p) deg

Master-Slave Links
6.1182E-03 C |U| m^3/s
-114.29 D Ph(U) deg
-479.09 E Htot W

22 stainless Solid type
-331.2 F Edot W

23 MINOR Inertance top-end minor loss
Same 23a 6.7000E-04 a Area m^2 2.6323E+05 A |p| Pa
5.0000E-02 b K+ 1.9891E-02 B Ph(p) deg
1.0000 c K- 6.1182E+05 C |U| m^3/s
-114.29 D Ph(U) deg
-479.09 E Htot W
-331.51 F Edot W

24 CONE Oversized bore to get to compression space
6.7000E-04 a Area m^2 Mstr 2.6290E+05 A |p| Pa
0.12978 b Perim m 23a 5.3159E-02 B Ph(p) deg
2.0600E-02 c Length m 6.3284E+03 C |U| m^3/s
5.0000E-04 d Srough -113.46 D Ph(U) deg

Master-Slave Links
-479.09 E Htot W
Optional Parameters
-331.86 F Edot W

25 stainless Solid type

26 CONE To reduce minor loss resistance
6.7000E-04 a AreaI m^2 Mstr 2.6240E+05 A |p| Pa
0.12978 b PerimI m 24a 0.10173 B Ph(p) deg
1.6000E-02 c Length m 6.4271E+03 C |U| m^3/s
5.0000E-04 d Srough

Master-Slave Links
-479.09 E Htot W
Optional Parameters
-332.05 F Edot W

27 stainless Solid type

28 DUCT To reduce minor loss resistance
6.7000E-04 a AreaI m^2 Mstr 2.4208E+05 A |p| Pa
0.12978 b PerimI m 25a 2.5274 B Ph(p) deg
0.3315 c Length m 7.4064E+03 C |U| m^3/s
5.0000E-04 d Srough -110.2 D Ph(U) deg

Master-Slave Links
-479.09 E Htot W
Optional Parameters
-346.29 F Edot W

29 stainless Solid type

30 CONE Oversized bore to get to compression space
6.7000E-04 a AreaI m^2 Mstr 2.4151E+05 A |p| Pa
6.8550E-02 b PerimI m 26a 2.5914 B Ph(p) deg
1.6000E-02 c Length m 7.4993E+03 C |U| m^3/s
6.7000E-04 d AreaF m^2 Mstr -109.92 D Ph(U) deg
0.12978 e PerimF m 24d -479.09 E Htot W
5.0000E-04 f Srough -346.67 F Edot W

Master-Slave Links
Optional Parameters
-347.26 F Edot W

31 stainless Solid type

32 DUCT To reduce minor loss resistance
6.7000E-04 a AreaI m^2 Mstr 2.4074E+05 A |p| Pa
6.8550E-02 b PerimI m 25a 2.6682 B Ph(p) deg
3.8100E-02 c Length m 7.8607E+03 C |U| m^3/s
5.0000E-04 d Srough -108.86 D Ph(U) deg

Master-Slave Links
-479.09 E Htot W
Optional Parameters
-347.26 F Edot W
stainless Solid type

28 MINOR Inertance bottom-end minor loss

Same 26d 6.7000E-04 a Area m^2 2.4080E+05 A |p| Pa

Same 22c 1.0000 b K+ 7.8607E-03 C |U| m^3/s

Possible targets

29 UNION rejoin

20 a SegNum 2.4080E+05 A |p| Pa

30 ANCHOR Change Me

31 COMPLIANCE compression space

1.8283E-02 a SurfAr m^2 2.4080E+05 A |p| Pa

Master-Slave Links 1.4406E-02 C |U| m^3/s

stainless Solid type 214.45 F Edot W

32 BRANCH mirror other alternator

4.1336E+06 a Re(Zb) Pa-s/m^3 2.4080E+05 A |p| Pa

Master-Slave Links 3.1728E-04 C |U| m^3/s

Optional Parameters

33 TBRANCH split to alternator clearance seal

7.5879E+08 a Re(Zb) Pa-s/m^3 2.4080E+05 A |p| Pa

Master-Slave Links 3.1728E-04 C |U| m^3/s

Optional Parameters

34 HX seal gap

7.6793E-06 a Area m^2 5.7799E+04 A |p| Pa

Master-Slave Links -1.6341E+07 b Im(Zb) Pa-s/m^3

Possible targets

35 SOFTEND end of clearance seal

stainless Solid type 299.56 G SolidT K

Possible targets

36 TBRANCH mirror other alternator

7.5879E+08 a Re(Zb) Pa-s/m^3 2.4080E+05 A |p| Pa

Master-Slave Links 3.1728E-04 C |U| m^3/s

Optional Parameters

37 HX seal gap

7.6793E-06 a Area m^2 5.7799E+04 A |p| Pa

Master-Slave Links -1.6341E+07 b Im(Zb) Pa-s/m^3

Possible targets

38 HX seal gap

7.6793E-06 a Area m^2 5.7799E+04 A |p| Pa

Master-Slave Links -1.6341E+07 b Im(Zb) Pa-s/m^3

Possible targets

39 HX seal gap

7.6793E-06 a Area m^2 5.7799E+04 A |p| Pa

Master-Slave Links -1.6341E+07 b Im(Zb) Pa-s/m^3

Possible targets

40 HX seal gap

7.6793E-06 a Area m^2 5.7799E+04 A |p| Pa

Master-Slave Links -1.6341E+07 b Im(Zb) Pa-s/m^3

Possible targets
36 IESPEAKER 1S132M - 2 wire

5.5800E-03 a Area m^2 5.7790E+04 A |p| Pa

0.4600 b R ohms -169.89 B Ph(p) deg

1.1650E-02 c L H 7.1768E-03 C |U| m^3/s

23.800 d BLProd T-m -82.743 D Ph(U) deg

4.1060 e M kg 10.310 E Htot W

4.5700E+04 f K |p| Pa -169.89 B Ph(p) deg

5.7790E+04 A |p| Pa -169.89 B Ph(p) deg

0.2000 a SurfAr m^2 5.7790E+04 A |p| Pa

2.0000E-03 b Volume m^3 -169.89 B Ph(p) deg

1.1578E-06 C |U| m^3/s -64.901 D Ph(U) deg

-8.6542E-03 B Htot W -8.6542E-03 B Htot W

stainless Solid type -8.6542E-03 B Htot W

37 UNION rejoin seal and piston stainless Solid type

38 COMPLIANCE back volume 39 HARDEND The End

5.7790E+04 A |p| Pa -169.89 B Ph(p) deg

1.1578E-06 C |U| m^3/s -64.901 D Ph(U) deg

-8.6542E-03 B Htot W -8.6542E-03 B Htot W

Possible targets Possible targets

-8.6542E-03 B Htot W -8.6542E-03 B Htot W

1.9059E-05 H I(z/s)

40 RPN A=AC power, B=th-to-AC eff, C=alter eff, D=engine eff

79.600 a G or T 79.599 A Welec

8.7240 B Effy1

37.117 C ChngeMe

23.504 D ChngeMe

50 F 912.41 / 100 * 36G -2 * 31F / 100 * 36G -2 * 912.41 / 100 * 36G -2 *

41 RPN ChngeMe

0.0000 a G or T 0.0000 A ChngeMe

31F 912.41 / 100 * 36G -2 * 31F / 100 * 36G -2 * 912.41 / 100 * 36G -2 *
Bibliography


[34] http://www.lanl.gov/thermoacoustics/


[38] http://www.lanl.gov/thermoacoustics/DeltaEC.html


