

5-kWe Free-Piston Stirling Engine Convertor

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Abstract. The high reliability, long life, and efficient operation of Free-Piston Stirling Engines (FPSEs) make them an attractive power system to meet future space power requirements with less mass, better efficiency, and less total heat exchanger area than other power convertor options. FPSEs are also flexible in configuration as they can be coupled with many potential heat sources and various heat input systems, heat rejection systems, and power management and distribution systems. Development of a 5-kWe Stirling Convertor Assembly (SCA) is underway to demonstrate the viability of an FPSE for space power. The design is a scaled-down version of the successful 12.5-kWe Component Test Power Converter (CTPC) developed under NAS3-25463. The ultimate efficiency target is 25% overall convertor efficiency (electrical power out over heat in). For the single cylinder prototype now in development, cost and time constraints required use of economical and readily available materials (steel versus beryllium) and components (a commercially available linear alternator) and thus lower efficiency. The working gas is helium at 150 bar mean pressure. The design consists of a displacer suspended on internally pumped gas bearings and a power piston/alternator supported on flexures. Non-contacting clearance seals are used between internal volumes. Heat to and from the prototype convertor is done via pumped liquid loops passing through shell and tube heat exchangers. The preliminary and detail designs of the convertor, controller, and support systems (heating loop, cooling loop, and helium supply system) are complete and all hardware is on order. Assembly and test of the prototype at Foster-Miller is planned for early 2008, when work will focus on characterizing convertor dynamics and steady-state operation to determine maximum power output and system efficiency. The device will then be delivered to Auburn University where assessments will include start-up and shutdown characterization and transient response to temperature and load variations. Future activities may include testing at NASA GRC.

Keywords: Free-piston Stirling engine, space power

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INTRODUCTION

From 1984 through 1996 under NASA contract, Mechanical Technology Inc. (MTI) conducted a two-phase program to develop a 25-kWe free-piston Stirling engine (FPSE) with a hot end temperature of 1050 K and a temperature ratio of 2.0 (525 K cold end temperature). The goal of Phase I was to demonstrate the feasibility of using a FPSE directly coupled with a permanent magnet linear alternator to meet the requirements of a space power generating system. The resulting Space Power Demonstrator Engine (SPDE) was successfully tested in October 1986, configured as a double-ended FPSE with half the total power generated per side. A lower head temperature of 650 K was selected while maintaining the temperature ratio of 2.0 to reduce development time and cost. A pumped molten-salt loop provided heat to the engine and a conventional water/glycol loop was used for the cold end. The goal of Phase II was to demonstrate engine operation at design temperatures of 1050 K/525 K and to perform component development. Between April 1990 and September 1996, the Component Test Power Converter (CTPC) was successfully tested (Dhar, 1999). The CTPC used a sodium-filled heat pipe heated by electric radiant heaters to supply heat to the engine and a pumped mineral oil loop to extract heat from the engine.

Since that time, requirements for the power convertor were changed based on a proposed power system that uses a lower temperature reactor (Mason, 2006). The power level was reduced from 25 kWe to 10 kWe per convertor and the hot end temperature was reduced to 830 K, but the temperature ratio of 2.0 was maintained. In 2007, Foster-Miller was contracted by Auburn University Space Research Institute to develop a 5-kWe single-cylinder FPSE demonstrator convertor. The design of this Stirling Convertor Assembly (SCA), described herein, is based on the successful designs of the SPDE and CTPC with certain differences due to cost and time constraints.

DEMONSTRATOR CONVERTOR DESIGN

The SCA is designed to operate at a hot end temperature of 830 K, a cold end temperature of 415 K, with a nominal power output of 5 kWe, a peak power output of 6 kWe with overstroke, and a design life of 5 years (44,000 hr) of continuous operation at 100% power level. High pressure (15 MPa) helium serves as the working gas, and the convertor operates at a resonant frequency of 85 Hz. The final design goal is to achieve a minimum of 25% overall efficiency (electrical power out over heat in) and a specific mass of 6 kg/kWe. Substitution of materials such as alloy steel for beryllium, the use of a commercially available linear alternator, and inclusion of bolted joints for convenience will preclude achievement of the efficiency and specific mass goals at this time. Facilities limitations will also constrain testing of the hot end to 617 K at Foster-Miller and 650 K at Auburn University.

As described below, the SCA consists of three primary subassemblies: the hot end assembly, the displacer drive assembly, and the cold end assembly (see Figure 1).

Hot End Assembly

The hot end assembly consists of the three heat exchangers: the heater head, regenerator, and cooler (see Figures 2 through 4). The heater head and cooler are both shell and tube heat exchangers, each with 1800 small diameter tubes (0.89 mm ID, 1.65 mm OD). The ends of each tube are electron-beam welded to the tube sheets. The heater shell, cooler shell, and tubes are constructed of Inconel 718. The regenerator is constructed of a porous metal matrix with 80% porosity (20% density) brazed to an inner and outer metal liner. The matrix is a sintered structure of randomly arranged metal fibers 0.022 mm in diameter. The matrix is type 316 stainless steel; the inner and outer liners are Inconel 718.

Displacer Drive Assembly

The displacer drive (Figure 5) pumps the helium through the heat exchangers. It consists of the displacer (dynamic component) and the post and flange (stationary component). The displacer consists of a central rod, a displacer dome and its support on the hot end, and the gas spring piston on the cold end. Two gas springs, one on each end, provide the spring stiffness needed to establish resonance. The forward spring is relatively weak (24% of stiffness) versus the aft spring (76% of stiffness). The rod provides the bearing support for the displacer drive. An internally pumped hydrostatic gas bearing is used in the SCA design for long life. All components in the displacer drive are alloy steel, with the exception of the displacer dome, which is Inconel 718 due to exposure of the dome to the hot expansion space. In the CTPC, the equivalent steel parts were fabricated from beryllium for its low density and high stiffness. Because of the additional losses in the gas springs suspending a heavy steel displacer, a penalty of approximately 3.5 points in efficiency is predicted. Since the bearings are not active during initial engine start-up, a compliant wear couple is implemented on the bearing and seal surfaces to tolerate transitory contact. The inside diameters of the cylinders are plated using Poly-Ond® and the outside diameters of the rod and pistons are coated with Xylan® 1054.

Cold End Assembly

The cold end assembly consists of the alternator assembly, pressure vessel, and joining ring (see Figure 6). The alternator assembly includes the linear alternator, power piston and cylinder, and the suspension system. A flexure support is used instead of a gas bearing to support the piston and plunger. The flexure provides radial and circumferential stiffness which resists the forces between the magnets and stator iron. The entire alternator assembly, including power piston and cylinder, will be supplied as a single assembled unit by Qdrive, Inc. (Troy, NY). The alternator is designed to deliver 5 kWe at a nominal stroke of 22 mm and 6 kWe at an overstroke of 24 mm, both at 85 Hz. To reduce lead time and expense, an off-the-shelf size was customized to obtain the power output, voltage level, and temperature capability of the design requirements. These modifications included slightly larger magnets, Hipercob laminations in the stator, samarium cobalt magnets, and custom windings. A performance penalty was accepted in lieu of an optimized custom alternator design. The current design is calculated to provide 88% efficiency at 5 kWe, versus approximately 93% efficiency with an optimized design.

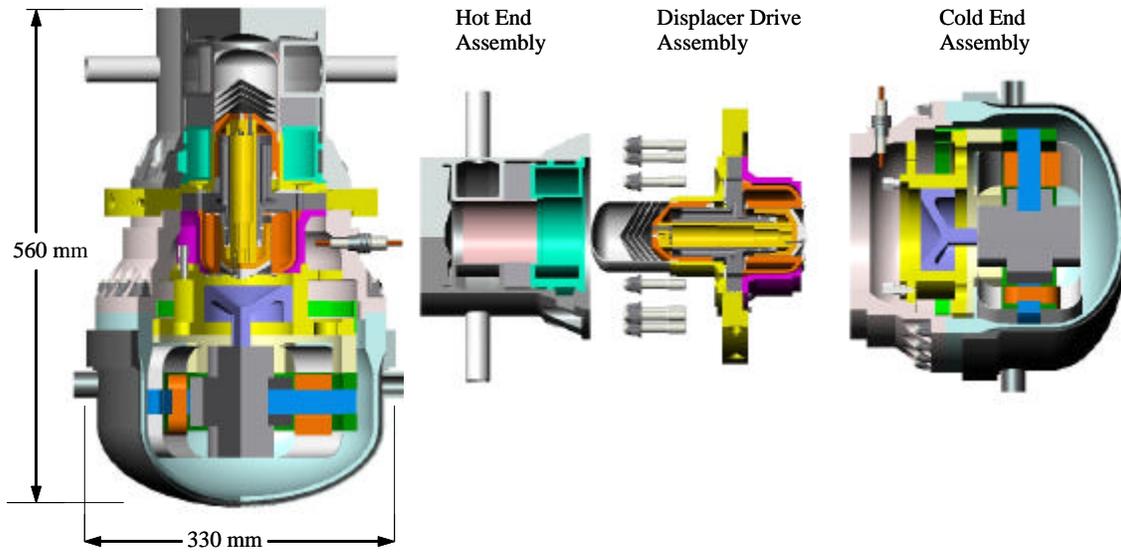


FIGURE 1. Demonstrator Engine.

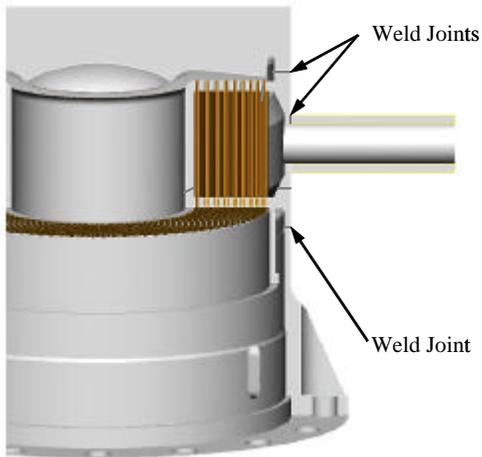


FIGURE 2. Heater Head Assembly.

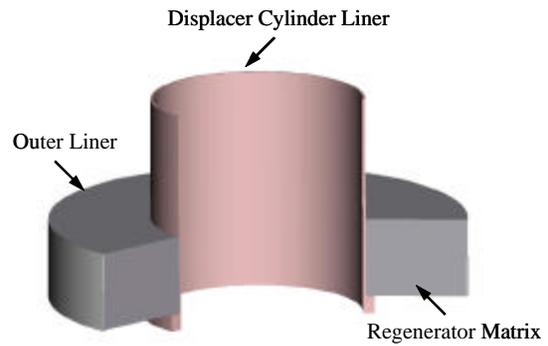


FIGURE 3. Regenerator Assembly.

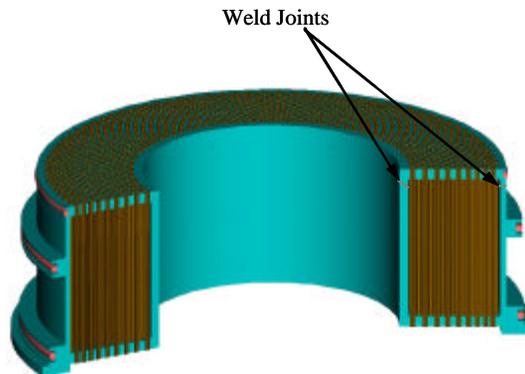


FIGURE 4. Cooler Assembly.

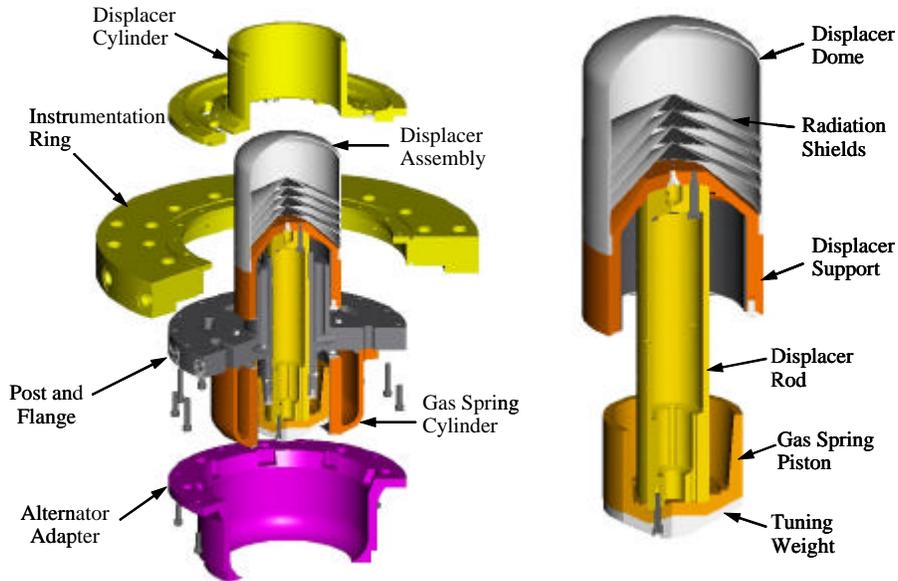


FIGURE 5. Displacer Drive Assembly (left) and Displacer Assembly (right).

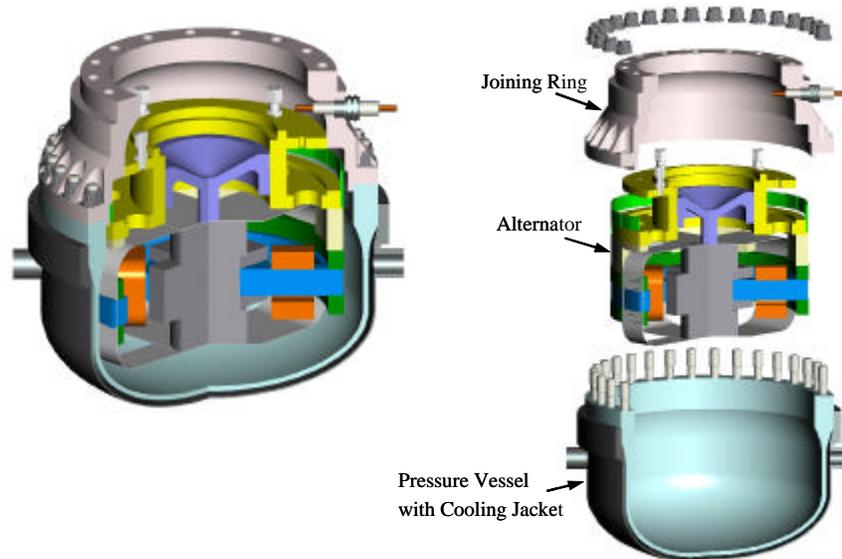


FIGURE 6. Cold End Assembly.

Bearings and Porting

The displacer is supported with a hydrostatic gas bearing located on the displacer rod. The bearing is internally pumped, pulling gas from the aft gas spring when its amplitude is high, charging a volume internal to the rod when a port opens at roughly 50% stroke. Two rows of 12 holes, each 0.30 mm in diameter, feed the bearing from this volume. The gas then flows along the rod and drains to the forward gas spring. A centering port located approximately at the mid-stroke position returns the bearing gas flow back to the aft gas spring, along with any net flow occurring due to seal leakage. To correct for seal leakage along the power piston seal, a centering port connects the compression space to the piston gas spring at approximately the mid-stroke position of the power piston.

Convertor Operation and Data Monitoring

The output power of the SCA is a function of the hot end temperature, cold end temperature, mean pressure, frequency, displacer stroke, piston stroke, and phase angle. Heat will be provided via an electrically heated pumped loop using Therminol® 72 heat transfer fluid. The heating loop will have a separate controller to maintain a set temperature value up to 617 K. Heat will be rejected through a water/glycol pumped loop connected to an air-cooled radiator. A separate controller on the cooling loop will maintain a set temperature by operating a mixing valve. A pair of capacitance gap sensors monitors both displacer and power piston positions. The sensors are located 180° apart and their signals averaged to eliminate variations due to eccentricities, thermal growth, and so forth. Since the sensors have a small range (~2 mm) compared to the stroke (22 mm), a taper is machined into the target to turn down the displacement. The controller monitors both displacement signals to safeguard against an overstroke condition. It also uses the power piston stroke as the process variable in the control scheme. A separate control load is used in parallel with the user load, and by pulse width modulation (PWM) switching of the control load, a change in total load resistance is induced to maintain the piston stroke set point (Kirby and Vitale, 2008). Alternator voltage and current will be measured to determine the output power. Flow rate and temperature drop will be measured across the heat exchangers to determine total heat in and rejected heat. Overall engine mean pressure and dynamic pressures of the cycle and displacer and piston gas springs will be monitored to help evaluate engine performance. All measurements will be displayed and recorded using LabVIEW (National Instruments). Figure 7 presents the instrumentation and data acquisition display screen.

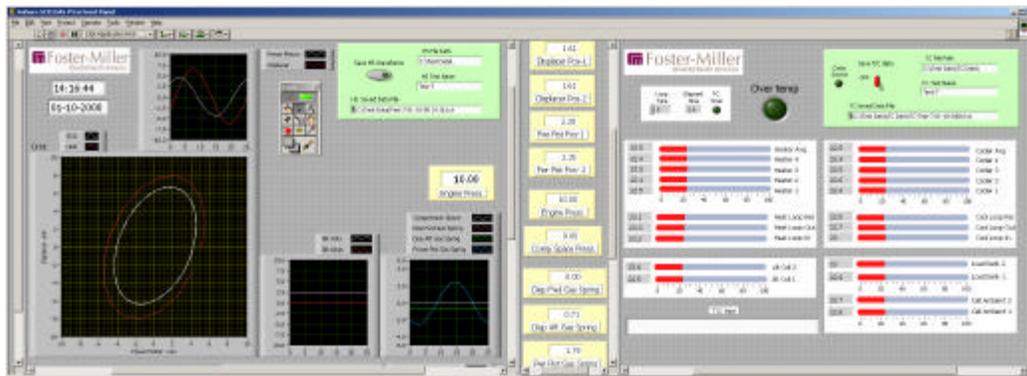


FIGURE 7. Instrumentation and Data Acquisition Display Screen.

DESIGN ANALYSIS

The design process for the SCA included thermodynamic analysis of the Stirling cycle, dynamical analysis of the complete mechanical system, and structural analysis of the pressure vessel.

Thermodynamic Analysis

Thermodynamic analysis of the Stirling cycle was performed using the HFAST (version 2.00) computer code developed by MTI under NAS3-25330. Optimization studies were performed on several engine parameters to determine the best design when considering output power, overall efficiency, and specific mass. These parameters included: heat exchanger length, tube diameter, and number; regenerator length, wire size, and porosity; engine frequency and phase angle. Figure 8 presents the power balance diagram for the SCA at the design operating temperature.

Dynamical Analysis

Another code developed by MTI called STIRDYN was used to model the dynamical behavior (interaction between the displacer, power piston, and various volumes within the engine) of the SCA. This code calculates the gas pressure variations in the volumes and the resulting forces on the displacer and power piston and can extract data

from the HFAST output. Five dynamical forces act on the displacer: compression space pressure force, expansion space pressure force, forward gas spring pressure force, aft gas spring pressure force, and inertial force associated with displacer mass. Six dynamical forces act on the power piston: compression space pressure force, piston gas spring force, damping force representing compression space leakage losses, alternator magnet spring, alternator shaft force, and inertial force associated with power piston mass.

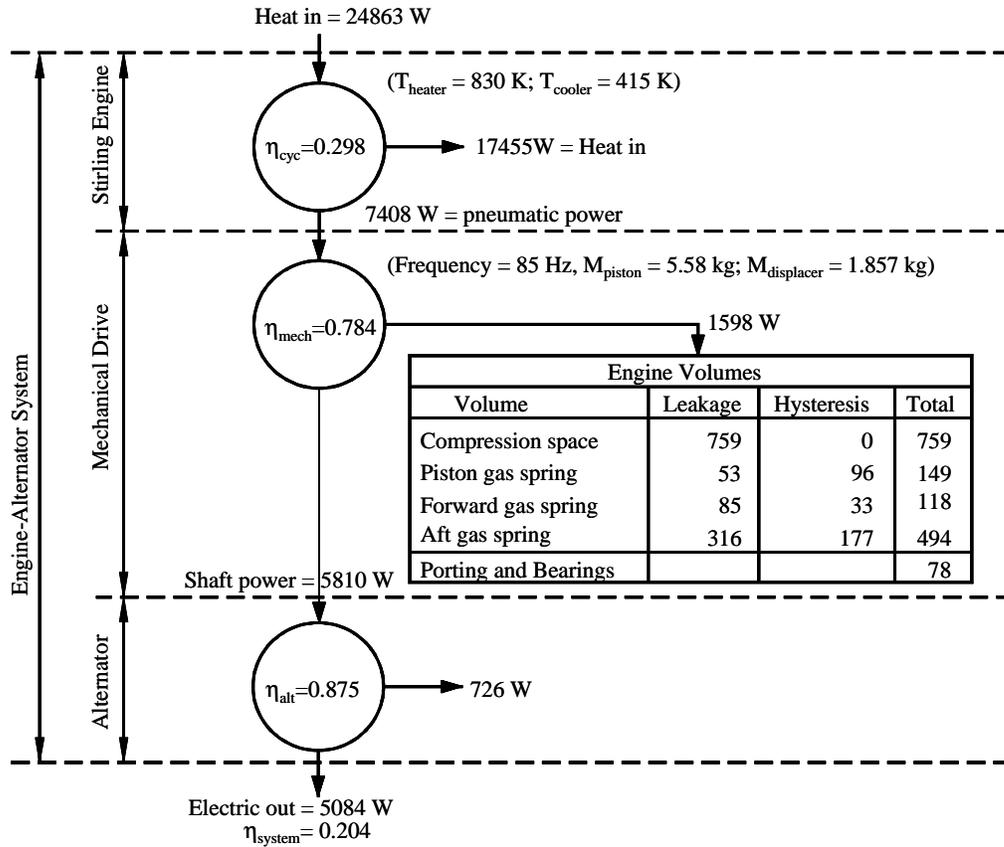


FIGURE 8. Power Balance Diagram of the 5-kWe SCA at Design Conditions.

Structural Analysis

Because the internal pressure of the vessel is high, a complete structural analysis was performed. Since the cold end of the vessel is essentially isothermal, the analysis of that section is straightforward. The heater head, however, is more challenging due to the somewhat unique combination of pressure and thermal stresses, both static and cyclic in nature. The hot section is initially sized to provide ample creep life at the design temperature. The length of the outer wall, which carries the temperature gradient, is then sized to minimize the thermal stresses. The optimum regenerator length tends to be short in order to keep the pumping losses down, which improves the pressure stresses but tends to increase the thermal stresses. To allow use of a shorter regenerator without negatively impacting the outer heater head wall, a separate inner wall is implemented. The cavity between the inner and outer walls is pressurized to engine mean pressure. Since the inner wall does not see the absolute pressure, higher thermal stresses can be accommodated.

Table 1 summarizes stresses for critical locations around the heater head. Figure 9 shows the contour plot of the mean stress component for the combined pressure and temperature stresses. Figure 10 shows the contour plot of the alternating stress component for the combined pressure and temperature stresses.

TABLE 1. Heater Head Stress Summary. Stress Values are in MPa.

Location	Mean Stress	Alt. Stress	Stress Ratio	Fatigue Safety Factor
Upper Heater Head Wall (H1)	211	25.8	0.123	4.00
Middle Heater Head Wall (H2)	383	13.2	0.0344	2.71
Lower Heater Head Wall (H3)	475	8.8	0.0185	2.33
Closure Plate Wall (H4)	427	57.9	0.136	1.89
Upper Regenerator Wall (H5)	283	112	0.396	1.92
Middle Regenerator Wall (H6)	44.3	144	3.248	2.70

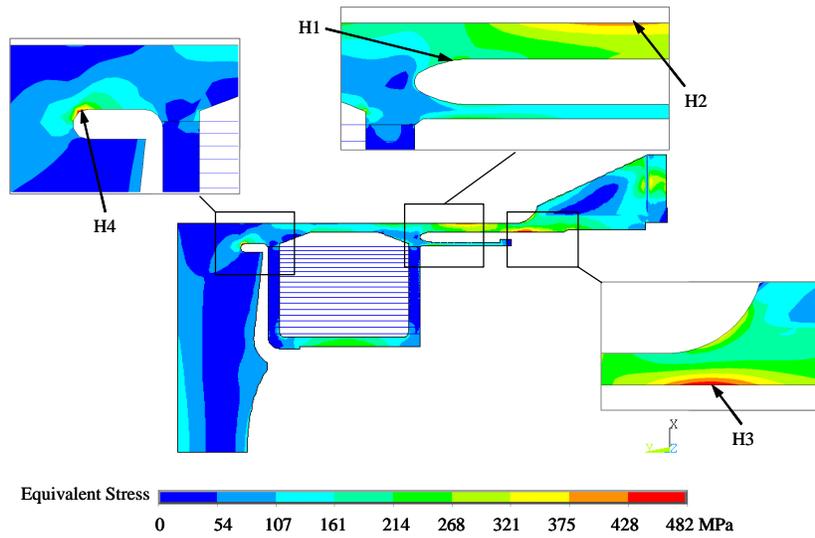


FIGURE 9. Heater Head Mean Stress due to Pressure and Temperature.

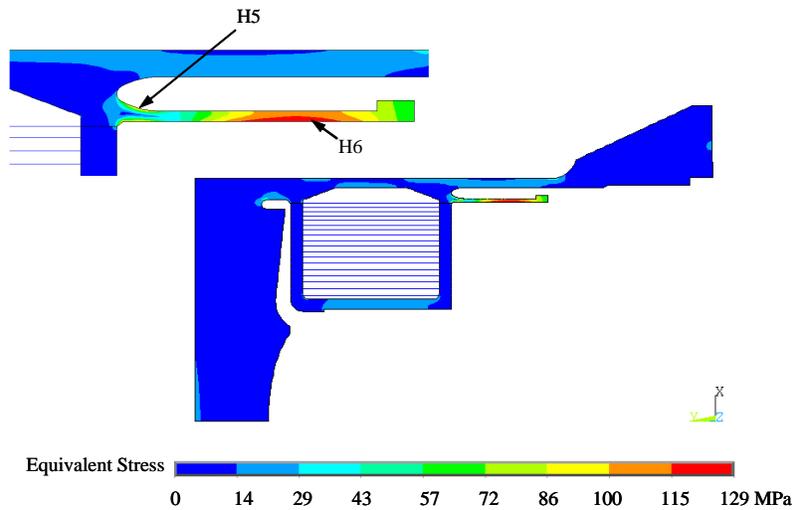


FIGURE 10. Heater Head Alternating Stress due to Pressure and Temperature.

CONCLUSION

The efforts conducted continue to support the concept that a Stirling cycle convertor is viable for nuclear space power generation. It offers high efficiency, low specific mass, and long life at the power levels and temperatures identified in the system studies. The technology has been demonstrated at lower power levels (100 W per cylinder) as well as higher power levels (12.5 kWe per cylinder). This design suggests that these goals are still achievable at the 5-kWe power level. Testing in the near future will validate this.

ACKNOWLEDGMENTS

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