

Design and Control of a Free-Liquid-Piston Engine Compressor for Compact Robot Power

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This paper presents the design and control of a free-liquid-piston engine compressor (FLPEC). The FLPEC produces low temperature, high-pressure air for use as a high energy density power supply for untethered, compact robots. Internal combustion with a propane-air mixture in the engine combustion chamber drives a high inertance, low mass liquid piston, which compresses air into an onboard reservoir. A real time logical controller is implemented in Matlab to regulate the timings of the air and fuel injection, spark, and exhaust valve and to compensate for misfires. Experimental pressure data is overlaid with the logical state of the injectors, spark plug, and exhaust valve to demonstrate the efficacy of the controller to account for cycle-to-cycle pressure variations in the combustion chamber.

INTRODUCTION AND MOTIVATION

Free piston engines (FPEs) differ from traditional crankshaft internal combustion engines due to the constraints placed upon piston motion. Pistons in conventional engines are kinematically linked, i.e. their position is constrained at both ends by a rotating crankshaft. The piston in a FPE is “free” since its motion is dictated by the interaction of load forces and forces of combustion gases in the chamber. This enables FPEs to have variable compression ratios and to operate with fewer moving components relative to their kinematically-linked counterparts. FPEs are not novel technology; R. P. Pescara is credited with the development of the original free-piston engine with a patent describing a single piston air compressor in 1928 [1]. The German company, Junkers developed free-piston air compressors in 1936, which were subsequently used by the German Navy to launch torpedoes [2]. The Societe Industrielle Generale de Mecanique Appliquee (SIGMA) designed and constructed free-piston, diesel powered gas generators in 1944, which had a power output of 850 kW at 613 rpm, and were used in more than 20 minesweepers of the French Navy [3]. The US Maritime Administration vessel, the GTS William Patterson carried a power plant using 6 free-piston gas generators developed by General Motors in the late 1950’s [4]. Fortunately, recent efforts have successfully scaled down the size and bulk of FPEs. In 1998, Clark et al. at West Virginia University developed a spark ignited free piston engine prototype with a power output of 316 W [5]. In 2002, Van Blarigan et al. at Sandia National Laboratories designed and validated a homogeneous charge compression ignition (HCCI) dual free-piston engine generator with a power output of 40 kW [6].

Progress in FPE development has not only been confined to research laboratories; industries have taken advantage of the mechanically simpler design and lower fuel consumption of FPEs. Sunpower, Inc. manufactures several free-piston Stirling engines, which range in power output from 42 W to 1 kW. Miniaturization of FPEs has drastically expanded its potential to be used in smaller-scale applications: one such field is untethered robots (i.e., robots with onboard power supplies).

Current untethered, compact robots are usually powered by lithium ion batteries (10-100 W) and actuated by DC motors. Advantages of this power source include low cost of lithium ion and nickel-metal hydride batteries and ease of DC servo motor control. However, these benefits are outweighed by the limited useful mechanical work output between charges, limited operating time duration, and bulky electromagnetic actuators, all of which contribute to low energy and power densities. Modern battery-powered, DC motor actuated robots include the Honda Asimo and the Aldebaran Robotics Nao. Asimo is equipped with a 6 kg, 51.8 V lithium ion battery, which takes 3 hours to fully charge and can provide 1 hour of operation. Similarly, Nao is powered by a 21.6 V lithium ion battery, which takes 2 hours to fully charge and provides 1.5 hours of operation. Though progress has been made in increasing the energy density of DC batteries and efficiency of electromechanical systems, it is worthwhile to explore other forms of power to prolong the operating duration and reduce the charging time for such devices.

One such alternative for this application involves the use of hydraulic pumps driven by miniature engines. Such systems can operate with almost

10 times the volumetric power density of their electro-mechanical counterparts (battery powered, motor actuation) and can be coupled with a variety of actuators. BigDog developed by Boston Dynamics is a four-legged robot capable of walking, running, and climbing and is powered by a two-stroke internal combustion engine (delivering 11.2 kW), which drives a hydraulic pump and provides an operation time of up to 2.5 hours [7]. Hydraulic actuation is an effective power supply source for BigDog, which weighs 109 kg, but scaling down this power source is difficult since hydraulic systems require an assortment of components, such as reservoirs, control valves, hosing, return lines, and accumulators, which make it impractical for use in sub-human scale robotics.

Pneumatic power is a favorable option for such smaller scale applications due to the low weight of actuators and the ability to expel exhaust gases to the environment (eliminating the need for return lines). Furthermore, pneumatic systems are more power dense than comparable electromechanical systems and can be used with a variety of actuators.

Therefore, the free-liquid-piston engine compressor (FLPEC) attempts to provide a more energy and power dense alternative for use in sub-human-scale robots in the form of pneumatic energy (low temperature, high pressure air) than comparable electromechanical systems.

OPERATION

Traditional internal combustion engines operate on a 2-stroke or 4-stroke cycle. A 4-stroke cycle involves a piston completing four distinct strokes: intake, compression, combustion, and exhaust, all of which occur during two revolutions of the crankshaft. A 2-stroke cycle manages to implement these strokes in a single crankshaft revolution usually by combining the intake and exhaust strokes.

Though each operating cycle of the FLPEC consists of 2 strokes, the intake, compression, and expansion are combined into one stroke and combustion gases are expelled in the second stroke. This “inject and fire” cycle is possible due to the presence of an onboard pressure reservoir to provide compressed air and fuel, and a low mass, high inertia liquid piston. The compressed fuel source for the FLPEC is propane, which has a relatively high vapor pressure (obviating the need for a fuel pump). The piston is a tube of water trapped between two elastic diaphragms or membranes (Fig. 1). The incompressibility of water and tube design lend to the low mass and high inertial nature of the piston.

At top dead center, the elastic diaphragms are bound to the combustion and compressor heads (Fig. 1a). In the first stroke, compressed propane and air are injected into the combustion chamber, which is the volume bounded by the combustion head and the elastic diaphragm. The inertia of the liquid piston prevents the piston from moving too quickly as the combustion chamber is pressurized. A spark plug then ignites the propane-air mixture and the resulting forces of the rapidly expanding combustion gases act upon the combustion head elastic diaphragm, which moves the piston causing the elastic diaphragm on the compressor head to expand. This expansion forces the air in the compressor head into the reservoir (Fig. 1b).

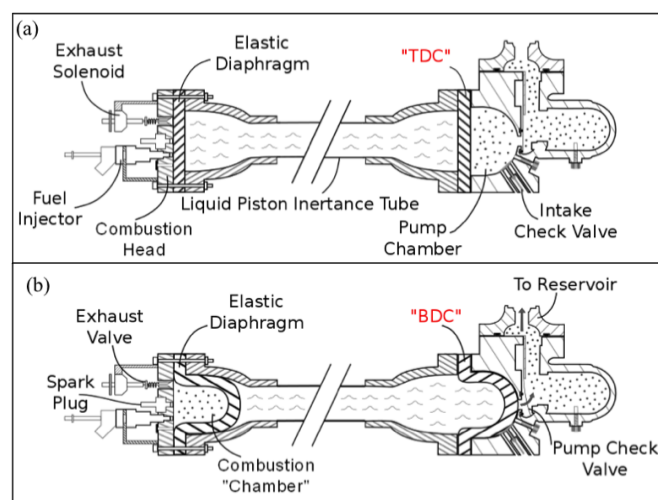


Figure 1: FLPEC at (a) top dead center and (b) bottom dead center

Once the combustion gases fully expand, an exhaust valve in the combustion head opens (Fig. 1b). The second stroke begins as the elastic membranes return the piston to its equilibrium position. The pump check valve then closes and atmospheric air refills the compressor chamber through another check valve (not shown in Fig. 1). The second stroke ends when the exhaust valve closes and the cycle repeats after a certain amount of time when compressed air and fuel are once again injected into the combustion chamber.

COMPONENTS

The FLPEC (Fig. 2) is a continuation of the work done by Riofrio [8] and Willhite [9-10]. Improvements to earlier designs are discussed, the most salient of which include the figure-eight liquid piston configuration, efficient compressor head check valve, and integration of onboard electronics.

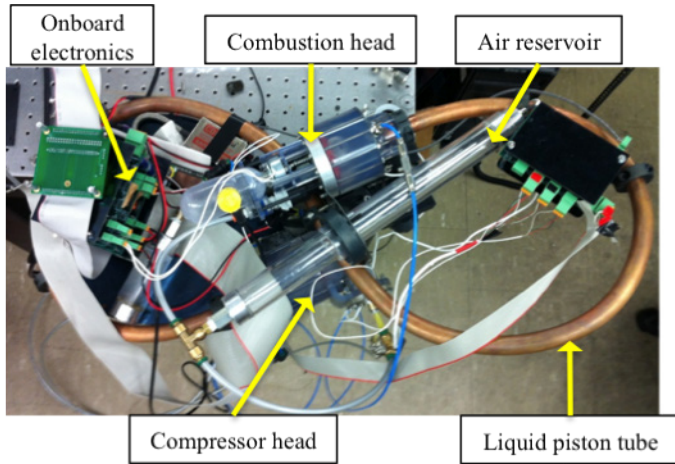


Figure 2: Photograph of the FLPEC

Figure-eight liquid piston configuration

During the first stroke, either the piston must possess significant inertia or the elastic membranes must be relatively stiff in order to maintain compression while air and fuel are being injected. However, increasing piston inertia usually requires increasing piston mass, which would decrease the energy density of the system. Additionally, stiffening the elastic membranes would cause irrecoverable energy losses (from stretching the membrane) each cycle, leading to decreased overall efficiency. A liquid piston in a figure-eight configuration is therefore used to address this concern since it possesses high inertia at relatively low mass. Furthermore, the use of the figure-eight configuration also serves to partially negate engine vibration. Since water is (essentially) incompressible, it will have the same acceleration at every point within the tube and because the tube doubles back on itself, the external bracing will have no net force or moment acting on it. This will effectively serve to neutralize engine vibration, without compromising efficiency or energy density.

Simulation results of the FLPEC indicated that the optimal piston inertia to balance compression and weight was 6.4 kg/mm^3 , which was implemented using a tube (1.52 m long, 1.9 cm in diameter) containing 0.63 kg of water. As Fig. 2 illustrates, there is a nozzle at both ends of the tube (with a diameter of 5.1 cm). This expansion in cross section was necessary to leave room for the injectors and spark plug on the combustion head.

Compressor head check valve

The compressor head functions to pump air into the high-pressure reservoir via a check valve, the dynamics of which will be discussed in this section. Be-

fore doing so, it is worthwhile to elaborate on how an ideal check valve would operate in a FLPEC. A check valve is essentially a mechanical valve which allows fluid to flow through it in a single direction. In the context of a FLPEC, an ideal check valve would allow a large mass flow rate of air to the reservoir as soon as the pressure inside the compressor head exceeded that of the reservoir. Furthermore, to prevent loss of compressed air, the check valve would ideally instantaneously close when the compressor pressure drops below the reservoir pressure. The proposed check valve is a thin metal cantilevered beam as shown in Fig. 3.

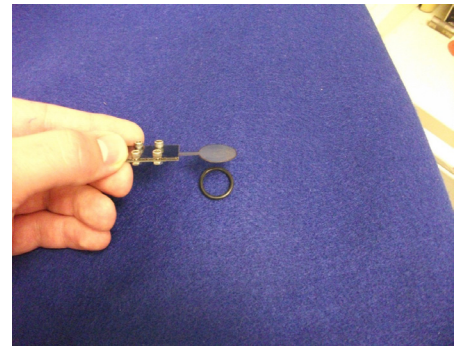


Figure 3: Check valve with O-ring

The check valve was modeled as a mass attached to a spring and damper, which responds to external forces caused by the pressure difference across the compressor head and the reservoir. The force acting on the valve is equivalent to the product of the pressure difference across the check valve and the area of the check valve itself. Simulation of the valve indicated that upon reaching saturation, there was no change in flow restriction. The rise time of a step response to a critically damped first order system was used to develop a metric to evaluate the effectiveness of the check valve by assessing the time taken to reach saturation (i.e., the ideal check valve would instantaneously reach saturation upon an infinitesimal pressure difference). Specifically, the time to reach saturation ($t_{\text{saturation}}$) was related to the mass (m) and stiffness (k) of the check valve as follows:

$$t_{\text{saturation}} \propto \sqrt{mk}$$

Therefore, it is desirable to reduce the check valve's mass and stiffness to decrease its saturation time. This intuitively makes sense since a heavy or stiff check valve would not respond quickly to a small pressure difference across the compressor head and reservoir. The mass of the check valve was reduced by using a thin piece of metal and the stiffness was reduced

by utilizing a long cantilevered beam. The model developed by Willhite [9-10] predicts that the maximum pumping pressure permitted by the proposed check valve is 2170 kPa (315 psi), which is higher than the maximum pressure observed during combustion. This indicates that the check valve does not impede the FLPEC from reaching its maximum pumping pressure.

Onboard components

Previous iterations of the FLPEC were not self-contained and so would inhibit its use as a power source for an untethered, compact robot. Fortunately, improvements in the current design of the FLPEC have aided functionality while minimizing electrical power consumption and weight.

An Athena single board computer manufactured by Diamond Systems was used to provide output signals, monitor the output from the pressure transducers, and implement commands from the controller. The Athena board was integrated (Fig. 4b) to the compressor and operated at a sampling rate of 1 kHz. The electronics were mounted on printer circuit boards and stacked in columns to save space as shown in Fig. 4a.

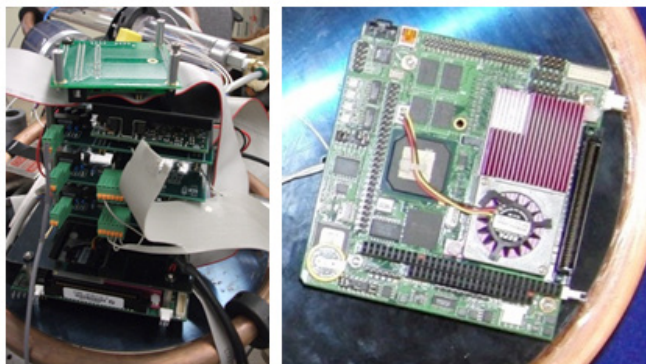


Figure 4: (a) Onboard electronics (left) (b) Athena Single Board computer (right)

The pressure transducers were manufactured by Kulite Semiconductor Products and were used to monitor the pressures in the compressor head and reservoir. The exhaust valve constructed by O.S. Engine is motivated by a solenoid, which is powered by a servoamp. The spark plug requires high voltage to produce a strong enough electrical discharge for instigating combustion and is powered by an ignition box. Two Bosch fuel injectors were used for independent air and fuel injection into the combustion chamber. To expedite the fuel injector response time and decrease overall power consumption, an injector drive controller from Texas Instruments was used. The reservoir is a high-pressure resistant glass tube and is connected to the compressor

head via tubing.

CONTROLLER

A controller was created in Matlab using Stateflow to monitor the air and propane injection timings, spark and to regulate the exhaust valve based on pressure values in the compressor chamber. The injection timings for the air and propane were occasionally adjusted for different trial runs but did not vary across cycles.

Misfires

It was empirically observed in successful cycles that combustion occurred within 20 ms after the spark was ignited and that the pressure of the air in the compressor head rose well above 100 psig. However, occasionally the FLPEC misfired and combustion did not occur after the spark was ignited. The cause of these misfires was ascertained to be the failure of the exhaust valve to completely expel all combustion gases, i.e. the combustion chamber still contained gas after the cycle terminated. Therefore, in the ensuing cycle, when propane and air were injected into the combustion chamber and the gas was ignited, the adulterated mixture led to a misfire. This was corrected by implementing a safeguard in the controller. The controller continuously monitors the pressure in the compressor head, and keeps track of the time taken for the pressure to rise above 50 psig after the spark. If the pressure stays below 50 psig in 50 ms following the spark, that cycle was deemed a misfire and consequently, the exhaust valve was opened for a certain amount of time to expel the propane-air mixture from the combustion chamber to end the cycle.

Regular Operation

In a typical combustion cycle, propane is first injected into the combustion chamber, followed by air. The timings of these injections are predetermined to create an optimal air-to-propane ratio mixture in the combustion chamber. The spark then initiates combustion and the controller monitors the rising pressure in the compressor head every millisecond. Following the spark, the controller monitors two values: the current pressure (the output from the pressure transducer at a given time) and the maximum value (the largest pressure value recorded by the controller in a cycle). Initially the controller makes the current pressure (CP) value equal to the maximum pressure (MP). In the next iteration, this maximum pressure is compared to the current pressure in the compressor head. If the current pressure is greater than the maximum pressure, the value of MP

is replaced with CP. This process repeats as the pressure increases till the combustion peak (at which point the pressure reaches a maximum and MP is constant for the rest of the cycle). The controller computes the difference between MP and CP and if that difference exceeds 2 psi (i.e., which indicates that the pressure in the compressor head is decreasing), the controller opens the exhaust valve to expel the combustion gases.

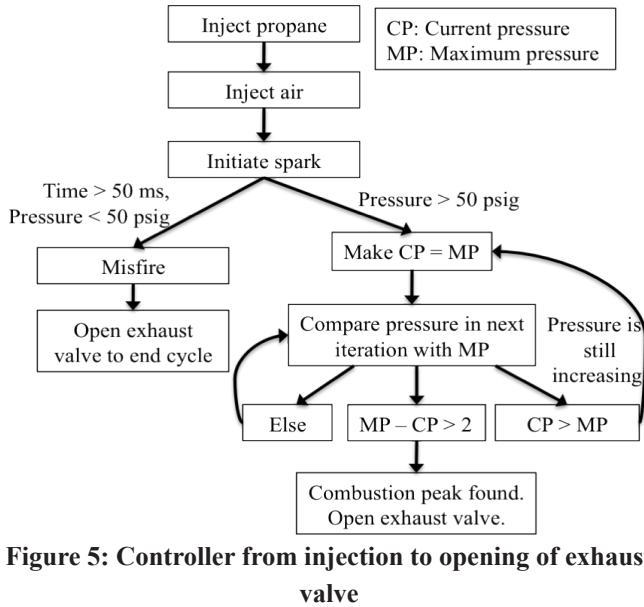


Figure 5: Controller from injection to opening of exhaust valve

This ensures that the exhaust valve is opened right after the combustion peak and that the propane-air mixture has completely combusted. The workings of the controller from the fuel injection till the exhaust valve is opened are illustrated in Fig. 5.

The use of 2 psi as a comparison between consecutive pressure values in the controller (Figures 5 and 6) has proven experimentally reasonable since the value is large enough to ensure that pressure fluctuations were not caused by noise in the pressure transducers and small enough for the controller to effectively regulate the exhaust valve without too much delay.

As the gases are expelled from the combustion head during the second stroke, the elastic membranes, which were expanding during combustion, retract to their equilibrium position. However, due to the high inrtance of the liquid piston, the membrane overshoots before returning to its equilibrium position, i.e. the liquid piston exhibits transient oscillatory motion, which force the adjacent membranes to also move in a similar manner. This spring-like behavior of the elastic membranes manifests into pressure spikes in the compressor head even after the exhaust valve is opened. These progressively smaller pressure peaks consistently follow

each combustion pressure peak (Fig. 7). In order for transient membrane motion to not affect air and fuel injection in the following cycle and to minimize the possibility of future misfires, it was found that the exhaust valve would need to be closed during the rise of the third pressure peak. This is achieved by finding three local minima and two local maxima (after the combustion pressure peak) using similar controller techniques used to find the pressure peak during combustion. Fig. 6 illustrates the aforementioned process of finding a local pressure minimum.

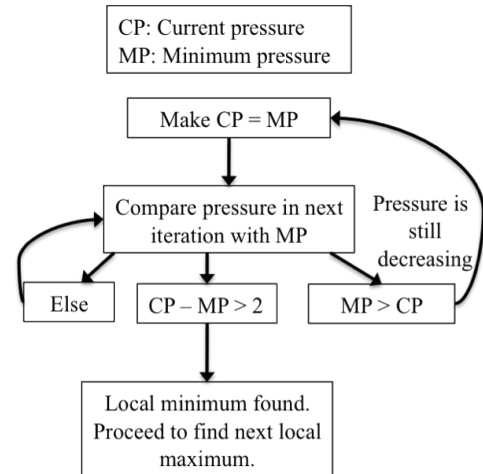
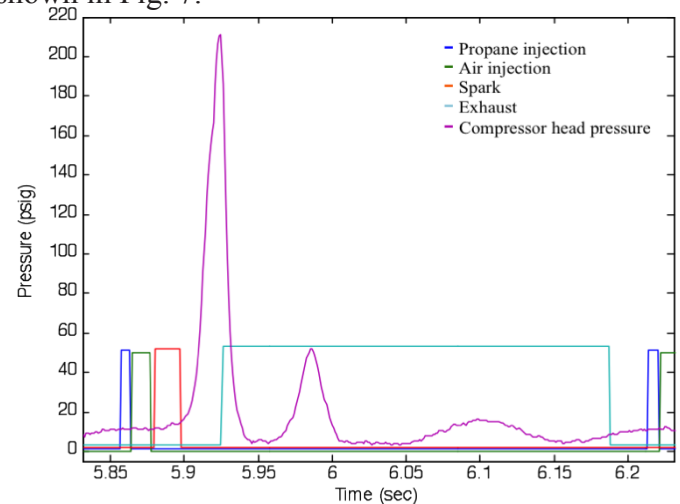


Figure 6: Partial controller for finding local minimum

After the third minimum is found, the controller waits for a few milliseconds and then closes the exhaust valve. It then waits a preset amount of time before re-starting the cycle.

RESULTS

Experimental data from the compressor head pressure transducers indicates that compressed air is consistently produced up to 1034 kPa (210 psig) as shown in Fig. 7.



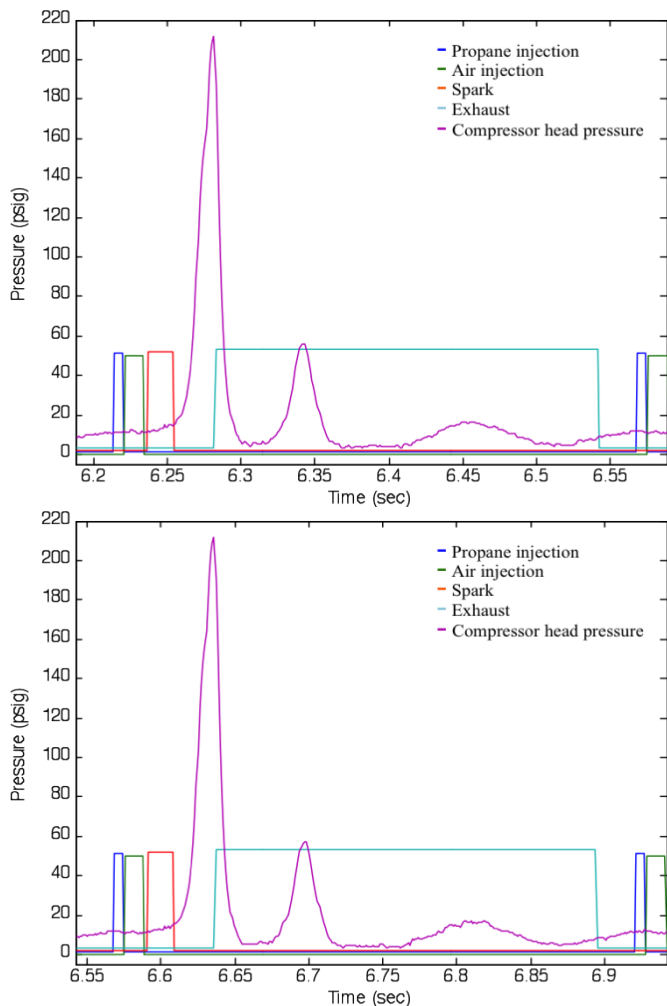


Figure 7: Compressor head pressure graph of three consecutive cycles

The logical high states correspond to injection, spark, and the opening of the exhaust valve, while the logical low states indicate the corresponding off-states. The combustion pressures are remarkably consistent across cycles and the controller works reasonably well in terms of identifying combustion peaks (to open the exhaust valve) and the rise of third maximum peaks (to close the valve). Though it may appear that the exhaust valve timings are constant across all cycles, this is in fact not true and inspection of the exhaust timings corroborate this. Figure 8 overlays the pressure in the compressor head and the reservoir. Consecutive cycles are shown to indicate the increase in reservoir pressure over a relatively short period of time.

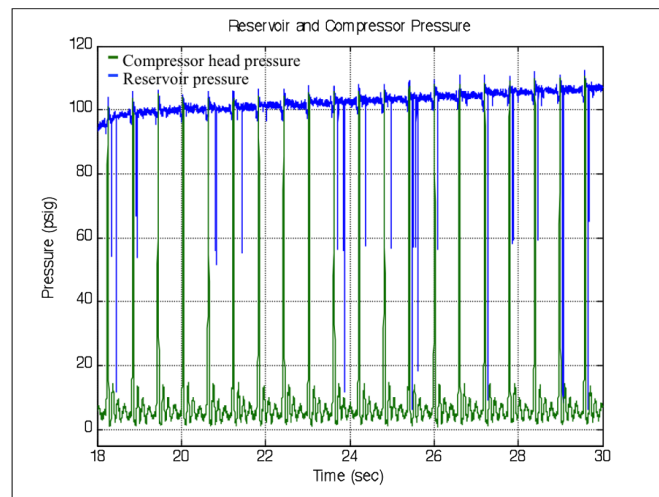


Figure 8: Reservoir and compressor head pressure

In the trial run shown above, the combustion pressure peaks at around 105 psig and the reservoir pressure rises to this value from around 95 psig in less than 12 seconds. By modifying the air-propane ratio through changing injection timings, the peak combustion pressure can be altered and the reservoir can be pumped to any intermediary pressure (under 210 psig).

CONCLUSION

The free-liquid-piston engine compressor is a continuation of work done by previous researchers and the prototype discussed in this paper is an improvement in terms of functionality and performance. The incorporation of a figure-eight piston configuration serves to minimize net force and moment, a redesigned check valve optimizes the flow rate into the reservoir, and on-board electronics aid in the control of the FLPEC. The controller has proven effective in regulating the injection timings, spark, and crucially in operating the exhaust valve. Experimental pressure data indicates that due to robust real time logical control of the FLPEC, consistent combustion upto 210 psig is observed and the reservoir can be pressurized in a relatively short time, making the FLPEC a promising pneumatic power source for untethered, compact robots.

ACKNOWLEDGEMENTS

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