Proceedings of IMECE'03
2003 ASME International Mechanical Engineering Congress
Washington, D.C., November 15–21, 2003

IMECE2003-42606

DESIGN, CONSTRUCTION, AND EXPERIMENTAL EVALUATION OF A MONOPROPELLANT POWERED FREE PISTON HYDRAULIC PUMP

Justin W. Raade[1]
Timothy G. McGee[2]
H. Kazerooni[3]

Department of Mechanical Engineering
University of California, Berkeley
Berkeley, CA 94720

ABSTRACT

A monopropellant powered free piston hydraulic pump (FPHP) was designed as a human scale (1.0 to 3.0 kW) mobile robotics power supply. The FPHP utilized high concentration hydrogen peroxide, which decomposes into hot gas when exposed to a catalyst, as the monopropellant energy source. Energy was extracted from the hydrogen peroxide and transferred directly to hydraulic fluid by expanding the hot decomposition gas in an integrated piston/cylinder arrangement. The prototype FPHP successfully produced 50 W of hydraulic power by pumping hydraulic fluid at an average pressure of 6.5 MPa (940 psi) and flow rate of 0.48 liters/min (0.13 gallons/min).

1 INTRODUCTION

Mobile robotic systems require energetic autonomy with no tether connecting the machine to its power supply. Hence, the power supply must be portable and have a long operation time. These requirements make the power supply performance the limiting factor in mobile robotic autonomy.

A free piston internal combustion engine represents a possible solution to the need for a potent power supply. Various studies have produced theoretical simulations of both diesel and gasoline powered free piston engines [1-5]. However, Achten [6] describes the only known free piston engine capable of practical operation. Free piston engines are not widespread due to the significant design problems presented by the creation of an operational free piston engine. With no rigid link to a crankshaft, the stroke length of the free piston can vary, making the compression ratio variable. The spark timing that produces the maximum torque is dependent upon the compression ratio and therefore must be precisely controlled in a free piston engine [7]. Fuel injection must also occur at the correct time to ensure proper fuel/air mixing and burning rates. Without good compression, thorough fuel/air mixing, and properly timed ignition, a free piston engine will stall. The lack of a flywheel makes continuous operation of the free piston engine difficult since there must be a way to store energy for a compression stroke after the power stroke. Additionally, free piston engines cannot be started by a simple electric motor, since there is no rigid crankshaft to turn. The problems of compression and startup in a free piston engine require an accumulator to store pressurized fluid that can push the piston back and forth to initiate motion of the engine. This necessity adds complexity to the engine.

Monopropellants enable a unique solution to the characteristic difficulties of a free piston engine, eliminating the problems of fuel/air mixing, compression and ignition, startup, and idling. Previous work on monopropellants includes several U.S. patents describing hydrogen peroxide power conversion systems [8,9]. Researchers have recently explored the use of hydrogen peroxide to move pneumatic actuators [10]. The free piston hydraulic pump (FPHP) described in this work integrates a monopropellant based robotic power supply and a free piston pump, two concepts not previously realized in a single system.

The FPHP represents a concept that may be developed into a feasible mobile robotic power supply capable of energetic autonomy. A monopropellant powered FPHP is capable of operation independent of the atmosphere or separate oxidizer, making operation possible in such anaerobic environments as underwater, space, and oxygen deprived buildings. This paper first describes the concept of a monopropellant powered FPHP as well as the energetics of monopropellants. Next, the specific design problems are presented along with their solutions. Finally, the experimental results of the FPHP are described and evaluated.

[1] Funded by NSF Fellowship
[2] Funded by NDSEG Fellowship
[3] Professor of Mechanical Engineering

2 DESIGN CONCEPT

In spite of its limited practicality to date, the free piston concept is a simple, elegant format for a hydraulic power supply. In contrast to the standard reciprocating engine, a free piston engine extracts work from hot gas by directly harnessing linear motion of the piston and pressurizing the hydraulic fluid, resulting in an integrated engine/pump design. Figure 1 shows a cross sectional diagram of the final design concept of the FPHP. The FPHP is symmetric about its centerline and has a double acting free piston and three cylinders: two hot gas cylinders and one for hydraulic fluid. Its overall length from catalyst bed to catalyst bed is approximately 68 cm (27 inches). The catalyst beds were manufactured by General Kinetics, LLC, and use fine silver mesh to decompose the hydrogen peroxide. The FPHP has no actuated hydraulic or exhaust valves, only passive one-way check valves for the hydraulic fluid and exhaust ports for the hot gas.

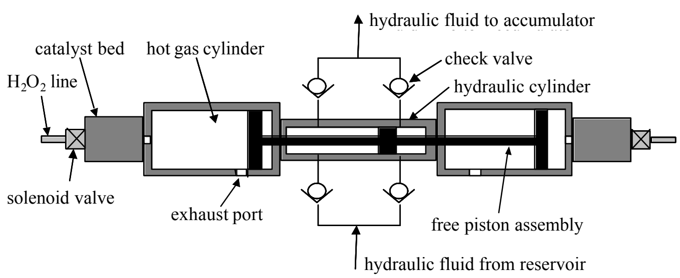

Figure 1: Double acting free piston concept.

The operation of the FPHP begins with the injection of high-pressure hydrogen peroxide into the left catalyst bed by opening the left solenoid valve. The hydrogen peroxide decomposes into steam and oxygen as it passes through the left catalyst bed. These hot gases expand within the left hot gas cylinder and force the free piston assembly to the right. The free piston's movement pumps hydraulic fluid at high pressure from the right chamber of the hydraulic cylinder while simultaneously drawing in low pressure fluid from the reservoir into the left chamber of the hydraulic cylinder. During this stroke the upper right and lower left check valves are open while the upper left and lower right check valves remain closed. The hot gases expand until the hot gas piston uncovers the left exhaust ports, at which point the gases vent to the atmosphere and the free piston stops moving. The right solenoid valve then opens to begin the cycle on the opposite side of the FPHP, moving the left hot gas piston from the exhaust port toward the cylinder head and compressing the gas in the left cylinder. This cycle is the same as the left-hand process. During operation, the free piston assembly moves right and left, resulting in a pulsating flow of hydraulic fluid from the reservoir to the accumulator.

The overall performance goal for a proof-of-concept FPHP was to achieve an average continuous power output of 2.2 kW (3.0 horsepower) by pumping hydraulic fluid at 6.9 MPa (1000 psi) and 19 liters/min (5.0 gallons/min). All pressures are expressed as gauge pressure. The mass of the FPHP was not optimized since its purpose was to demonstrate the concept of a novel mobile robotic power supply, not to embody a field-ready device. In order to achieve the desired hydraulic power output and flow rate, the FPHP was designed to operate at ten cycles per second. One cycle was defined as a complete stroke of the free piston assembly to the right followed by a complete stroke to the left. To minimize monopropellant consumption, the FPHP needed to maximize the computed conversion efficiency, η_{conv}, defined as the output hydraulic work divided by the chemical energy put into the system:

$$\eta_{conv} = \frac{W_{out}}{E_{in}} = \frac{P_{avg} V}{Q_{LHV} m_{HP}} \tag{1}$$

where P_{avg} is the average hydraulic pressure over one stroke, V is the volume of the hydraulic fluid expelled during one stroke, Q_{LHV} is the lower heating value of the hydrogen peroxide, and m_{HP} is the mass of the hydrogen peroxide injected for one stroke.

The final dimensions of the FPHP, driven by the aforementioned performance goals, were determined from an iterative process with the theoretical model and simulation of the FPHP as developed by McGee [11]. For example, a smaller hot gas cylinder bore resulted in slower peak free piston speeds, which eased hydraulic sealing but also resulted in lower power output. During the design process, possible bore and stroke sizes of the hot gas cylinder as well as relative diameters of the hot gas and hydraulic pistons were evaluated with the simulation in order to arrive at the final values. The final design theoretically produced the required power output while maintaining a relatively low peak free piston speed of 8.0 meters per second (m/s). The area ratio between the hot gas piston and the hydraulic piston was 6.5, resulting in a pressure amplification in the hydraulic fluid. The pressure amplification facilitated the venting of the hot gas at a lower pressure, extracting more work from the hot gas while still maintaining an average hydraulic pressure near 6.9 MPa.

In addition to the pumping unit of the FPHP previously described, a hydraulic load system and a high-pressure hydrogen peroxide delivery system were devised. Since these systems were constructed to prove the concept of the FPHP, no attempt was made to minimize their mass. The hydraulic circuit was constructed with commercially available hardware and assembled on a bench top. The system consisted of an accumulator operating at an average hydraulic pressure of 6.9 MPa (1000 psi) and a reservoir pressurized to 310 kPa (45 psi). A relief valve set to approximately 6.9 MPa remained closed until the pressure in the accumulator reached 6.9 MPa, at which point it opened and allowed the fluid to flow to the reservoir. In this way the relief valve simulated an approximately constant maximum load for the FPHP. The hydrogen peroxide system consisted of monopropellant lines, solenoid valves, and a 2.3 liter stainless steel tank containing the liquid hydrogen peroxide and pressurized with inert nitrogen gas. As the solenoid valves intermittently opened during FPHP operation, the pressure in the tank remained constant as the hydrogen peroxide level dropped by keeping the tank connected to the high-pressure gas source.

3 MONOPROPELLANT ENERGETICS

Monopropellants spontaneously decompose into hot gas when brought into contact with a catalyst, requiring no oxidizer to release energy. This characteristic provided a unique solution to the typical problems of a free piston engine. There was no need for compression or mixing with air to provide power, eliminating the need to properly control a spark and regulate a compression ratio. The lack of a flywheel, a disadvantage for a gasoline or diesel powered free piston engine, provided the benefit of power on demand for the monopropellant powered FPHP. The FPHP could be started and stopped intermittently, since the monopropellant could be injected and decomposed at any desired time. Continuous operation was achieved by injecting enough monopropellant each cycle to move the free piston the length of its stroke.

Hydrogen peroxide was selected as the monopropellant for this project since it was readily available in a pure, highly concentrated form from several commercial sources. Additionally, hydrogen peroxide produced only steam and oxygen as decomposition products, which are non-toxic. Hydrogen peroxide decomposes according to the following reaction [12].

$$H_2O_2 \xrightarrow{CATALYST} H_2O + \frac{1}{2}O_2 + \text{heat} \tag{2}$$

Energy density was defined as the total energy content of a fuel or a power supply system divided by its mass, expressed in terms of megajoules per kilogram (MJ/kg). The reaction in Eq. (2) releases 2.9 MJ/kg of energy at standard temperature and pressure conditions. This energy density is the higher heating value of hydrogen peroxide, which indicates that the water in the products is in liquid form. The lower heating value indicates that the water is in vapor form in the exhaust, and provided a more realistic measure of the available energy in the monopropellant since the exhaust was hot gas containing water vapor.

Propellant grade 90% hydrogen peroxide was used for the testing of the FPHP. Purified water constitutes the balance of the mixture for hydrogen peroxide concentrations less than 100%. Table 1 shows heating values and decomposition temperatures for 100% and 90% mass concentrations of hydrogen peroxide [12,13].

Percent H_2O_2 by mass	Higher heating value	Lower heating value	Adiabatic decomposition temperature
100%	2.9 MJ/kg	1.6 MJ/kg	1000°C
90%	2.6 MJ/kg	1.2 MJ/kg	740°C

Table 1: Heating value data for hydrogen peroxide.

4 DESIGN CHALLENGES AND SOLUTIONS

4.1 HIGH TEMPERATURE SEALING AND LUBRICATION

It was apparent from the beginning of the design process that the FPHP would need to function like an internal combustion (IC) engine mated to a piston hydraulic pump. Standard pneumatic piston/cylinder actuators can operate in conditions up to approximately 250°C. The FPHP had to withstand hydrogen peroxide decomposition gases at a temperature exceeding 700°C. Polymer seals and traditional petroleum lubricants combust at this temperature, so the technology used to construct a high temperature resistant piston/cylinder assembly was borrowed from IC engine design. Internal combustion engines use steel rings fitting in grooves on the hot gas piston to provide a seal against the bore of the cylinder, a design that reliably withstands peak combustion temperatures in excess of 2000°C [7]. This design archetype was adapted for use in the FPHP.

The hot gas pistons in an IC engine are usually made from aluminum while the cylinder bore is traditionally iron. This arrangement improves performance by reducing reciprocating mass and providing favorable wear characteristics between the piston and cylinder bore, since the softer aluminum wears in to the harder iron. Aluminum, however, has a coefficient of thermal expansion approximately 35% higher than iron. The hot gas piston must be designed with the appropriate diametrical clearance in the cylinder bore to avoid seizure at high temperatures. To estimate the required clearance, the circumference of a cylindrical object was assumed to expand linearly with increasing temperature. This assumption is valid for thin-walled cylinders in which the circumference is much greater than the thickness, since the amount of radial expansion of the wall is negligible compared to the circumferential expansion. For a thick-walled or solid cylinder (like the hot gas piston), the analysis may underestimate the amount of diametrical expansion since the radial thickness is not negligible. However, it is presumed that the assumption of linearity is adequately accurate for moderate temperature changes. Therefore, the change in length due to thermal expansion equals the product of the coefficient of expansion, the change in temperature, and the original length. The difference in the piston and cylinder diametrical expansions C_{dia} is given by

$$C_{dia} = (a_P - a_C) D \Delta T \tag{3}$$

where a_P and a_C are the linear thermal expansion coefficients for the piston and cylinder, respectively, D is the nominal bore of the cylinder and diameter of the piston, and ΔT is the difference between the operating temperature of the FPHP and room temperature.

Equation (3) was applied to the design of the FPHP. The bore of the FPHP hot gas cylinder was 47 mm (1.85 inches), and the operating temperature was estimated at 260°C, a temperature commonly achieved by the pistons in internal combustion engines [7]. This temperature produced an increase of approximately 235°C from room temperature. Evaluating

Eq. (3) under these conditions showed that the aluminum piston would expand approximately 0.08 mm (0.003 inches) more than the steel bore. The bore therefore was made about 0.13 mm (0.005 inches) larger than the piston at room temperature in order to maintain proper sliding clearance at operating temperature.

In addition to allowances for thermal expansion, the heat from the hot gases needed to be dissipated to avoid excessively high temperatures. Aluminum alloy 6061, used for the hot gas pistons of the FPHP, becomes soft at a temperature of 580°C, well below the temperature of the hydrogen peroxide decomposition gases. Traditional IC engines use an oil-filled sump to provide a constant bath of oil to the wall of the cylinder. The oil provides a lubricating film to avoid metal-to-metal contact between the piston and cylinder, while also cooling the piston and cylinder to prevent seizure or meltdown. The FPHP could not be designed with an oil sump since it was comprised of the free piston geometry and had no crankcase to hold oil. Therefore, several unique ceramics manufactured by Techline Coatings were utilized to ensure adequate heat dissipation and lubrication. Thermal dispersant ceramic coating was applied to the exterior of the hot gas cylinder walls to improve heat dissipation, producing a black coloration. A dark gray lubricating ceramic layer was applied to both the hot gas piston sleeve and the hot gas cylinder bore. This ceramic coating reduced the friction between the two components and provided a very hard surface layer and longer wear life. A thermal barrier ceramic was applied to the hot gas piston heads to reduce heat conduction from the hot gases into the piston. Finally, solid lubricating powder was buffed onto the piston and cylinder bore contact areas.

4.2 ALIGNMENT AND ASSEMBLY

It was crucial to maintain a precise alignment of the hot gas and hydraulic cylinders to ensure smooth movement of the free piston assembly. Proper alignment was achieved by the use of press fit interfaces between the three cylinders. Figure 2 shows a close up view of the press fit. A shoulder on the hot gas cylinder was pressed into the bore of the hydraulic cylinder with a light interference fit, ensuring that all three cylinders were concentric.

It was also necessary to assemble the hot gas piston concentrically with the connecting rod and hydraulic piston. A press fit in this location was infeasible since this would make disassembly very difficult. The hot gas piston also needed to withstand very high loads. With a calculated peak pressure near 1.7 MPa (250 psi) in the hot gas cylinder, there would be a load of approximately 3.0 kN (670 lbf) on the hot gas piston [11]. The solution used was a conical bore in the hot gas piston that fit over a taper on the end of the connecting rod. The taper ensured a high level of concentricity and a large contact area between the hot gas piston and connecting rod to support high loads and prevent surface damage between the two. A crown nut preloaded the hot gas piston onto the connecting rod, as shown in Fig. 3. The crown nut was assembled with thread locking compound and safety wired onto the connecting rod to avoid loosening during high speed, high temperature operation.

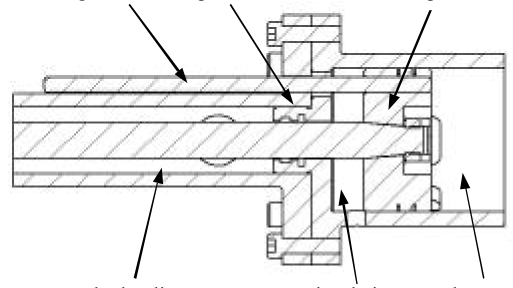

Figure 2: Press fit between hot gas and hydraulic cylinders.

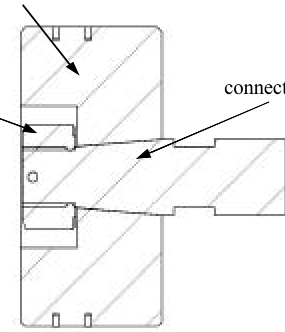

Figure 3: Cross-section view of hot gas piston assembly.

4.3 EXHAUST PORT DESIGN

The FPHP exploited the fact that a monopropellant does not require mixing with fresh air by eliminating intake ports and incorporating simple exhaust ports machined into the hot gas cylinder wall, which the hot gas piston uncovered at the end of each stroke. This feature allowed the hot gases to expand as much as possible before venting to the atmosphere. In contrast to traditional IC engines, there was no cam necessary to actuate exhaust valves as in four-stroke engines, nor proper timing to coordinate between the intake and exhaust ports as with two-stroke engines. The gas in each cylinder was compressed slightly before the hydrogen peroxide was injected at the start of each cycle, but the energy used to compress the gas was largely returned during expansion. The gas experienced less than a 5:1 compression ratio, not including catalyst bed internal volume which significantly reduced the effective compression ratio.

The use of exhaust ports in the hot gas cylinder wall necessitated the use of locking pins to prevent the rotation of the hot gas piston rings in their grooves. Should a ring rotate so that the seam of the ring passed over the exhaust port, the ring could spread into the port and catch on its edge, potentially seizing the piston within the bore. The locking pins were press fit into the ring groove, and each piston ring was notched at its ends to fit over the head of the locking pin and prevent rotation, as shown in Fig. 4. In a two-stroke IC engine, the piston itself is constrained from rotating by the connecting rod. The pistons in the FPHP needed to be constrained similarly to prevent the entire free piston assembly from rotating. A guide shaft, shown in Fig. 2, was designed to be press fit into one of the hot gas pistons and passed through the base of the hot gas cylinder via a bronze bushing. The guide shaft, moving with the free piston assembly, effectively constrained rotational motion of the piston assembly while allowing linear motion. Using Eq. (3), the diametrical clearance between the guide shaft and bronze bushing was estimated to be sufficient to avoid seizure at high temperatures due to differing thermal expansion rates.

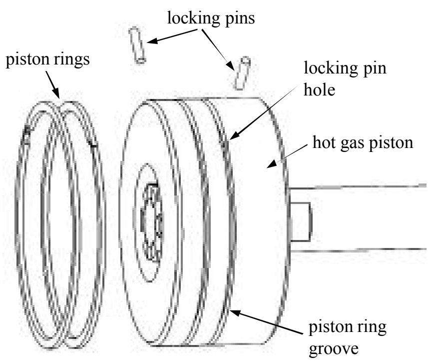

Figure 4: Exploded view of hot gas piston with locking pins.

4.4 COMBUSTION OF HYDRAULIC FLUID

Hydraulic fluid is very flammable under certain conditions. The hydraulic fluid used in this prototype had a flash point of 200°C, at which point it could produce fumes capable of ignition. Since the hydrogen peroxide decomposed at 740°C, the hydraulic fluid needed to be isolated from the hot gases to avoid possible combustion. A dead volume of insulating air behind the hot gas piston at the limit of its stroke, shown in Figure 2, prevented the hot gas from coming into close proximity with the hydraulic fluid. The depth of the dead volume was 1.3 cm (0.50 in.), which was estimated to be adequate to ensure that there was never direct contact between hot gas and hydraulic fluid, nor any bulkhead directly heated by the hot gas on one side and exposed to the hydraulic fluid on the other. Some heat conduction occurred from the hot gas cylinder to the hydraulic cylinder, but this conduction did not result in excessively high hydraulic fluid temperatures.

4.5 HIGH SPEED HYDRAULICS

The hydraulic piston required a seal to minimize fluid leakage between the piston and hydraulic cylinder bore. Traditional hydraulic piston applications include large, slow moving devices such as backhoes and dump trucks. The seals used in these machines rarely experience piston speeds exceeding 1.0 m/s. Since the FPHP needed a seal capable of a maximum piston speed near 8.0 m/s, bronze-impregnated Teflon seals manufactured by Claron were chosen to perform at this level. These seals could withstand temperatures of 200°C and hydraulic piston speeds up to 15 m/s.

There was a significant pressure drop across the check valves that facilitated the pumping of the hydraulic fluid. The flow of an incompressible fluid through a restricting orifice is proportional to the square root of the pressure drop across the orifice divided by the specific gravity of the fluid [14]:

$$Q = c_v \sqrt{\frac{\Delta P}{G}} \tag{4}$$

Solving for the pressure drop ΔP,

$$\Delta P = \frac{GQ^2}{{c_v}^2} \tag{5}$$

where G is the specific gravity of the fluid, Q is the volumetric flow rate, and c_v is the valve geometry dependant flow constant. Peak flow occured at the maximum free piston speed. Cavitation could occur during the intake of the fluid if the pressure drop across the check valve reduced the pressure of the fluid to its vapor pressure. Such a condition would be very undesirable since it would result in the formation of bubbles in the fluid, causing rapid volume changes and pressure fluctuations. A significant drop in pressure would also bring dissolved air out of solution in the hydraulic fluid, producing more bubbles. The hydraulic fluid would become much more compressible if it contained many bubbles in suspension. On the intake stroke, the pressure forcing the fluid through the check valve was only atmospheric pressure, so a pressure drop of approximately 1.0 atm (15 psi) produced cavitation. To remedy this situation, a wide range of commercial sources was consulted for a compact check valve with a high c_v value. The check valves with sufficiently high c_v values to avoid cavitation at the theoretical maximum flow rate of 130 liters/min [11] were very large, usually near 15 cm in length with large pipe thread connectors on each end. These valves would have been very bulky and cumbersome to integrate into the design of the FPHP. The solution to this problem was to pressurize the reservoir above atmospheric pressure to prevent the hydraulic fluid from dropping to below its vapor pressure at the inlet of the FPHP. This technique enabled the use of a more compact check valve while avoiding cavitation. The check valves used on the FPHP were made of brass and had a moderate c_v factor of 5.5 gallons/min/(psi)$^{1/2}$. Evaluating Eq. (5) at the maximum flow of 130 liters/min using hydraulic fluid with a specific gravity of 0.88, the pressure drop across the check valves is approximately 230 kPa (34 psi). Pressurizing the reservoir to 310 kPa (45 psi) eliminated cavitation under all circumstances.

4.6 SAFETY CALCULATIONS

The most critical safety aspect of the design of the FPHP was ensuring adequate wall thickness of the cylinders to avoid rupturing due to hoop stress levels. The minimum wall thickness for a uniform cylinder is given by Eq. (6):

$$t_{\min} = \frac{f_s PR}{s_Y} \tag{6}$$

where P is the pressure in the cylinder, R is the inner radius, σ_Y is the yield strength of the material, and f_s is the safety factor [15]. The minimum wall thickness of the FPHP hot gas and hydraulic cylinders are 5.1 mm (0.20 in.) and 4.5 mm (0.177 in), respectively. The hydraulic and hot gas cylinders walls were designed with a minimum safety factor of five under maximum pressure conditions. This high safety factor accounted for exhaust and fluid ports and other irregularities machined into the walls of the cylinders, cyclic loading effects, and possible weakening due to corrosion from the hot steam.

4.7 EXPERIMENTAL HARDWARE

The following figures depict various aspects of the prototype FPHP system. Figure 5 shows a photograph of the main components of the FPHP before assembly. Figure 6 is a photograph of the FPHP system with the main components labeled.

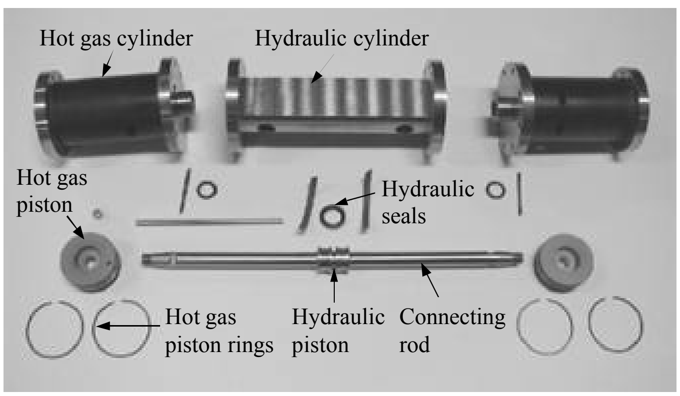

Figure 5: FPHP hardware.

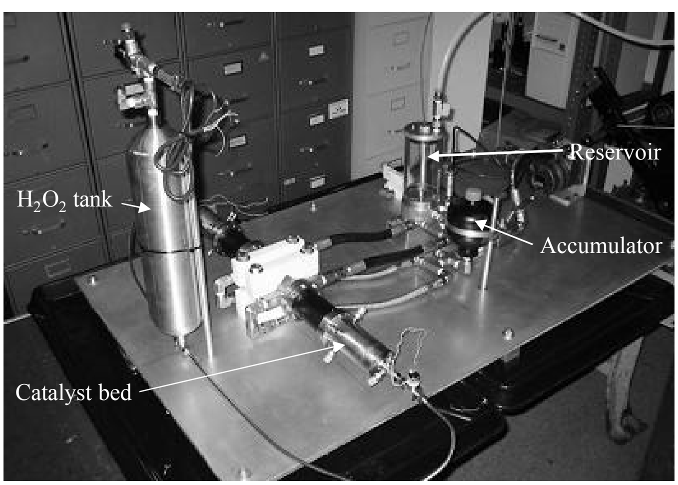

Figure 6: FPHP system.

5 EXPERIMENTAL EVALUATION AND RESULTS

A LabVIEW data acquisition (DAQ) system allowed the FPHP to run at a steady speed under open-loop control. The DAQ system controlled the solenoid valves using two parameters: injection time and delay time. During injection, the solenoid valve opened, allowing hydrogen peroxide to flow under high pressure from the hydrogen peroxide tank into the catalyst bed. The DAQ system also recorded the performance of the FPHP, sampling the pressure and temperature data at 100 Hz. Ashcroft pressure transducers measured the pressure values and K-type thermocouples measured the temperatures. In the following plots, the solenoid valve signals are represented as square waves for reference alongside the pressure data. The transient accumulator pressure curve, shown in Figure 7, was taken from a manually controlled run. Figures 8-10 show steady-state data taken from an experimental run with a 0.50 second injection time and a 4.5 second delay time. The steady state temperature of the hot gas cylinders was measured at approximately 270°C. This operating temperature was well below the adiabatic decomposition temperature of the hydrogen peroxide because the hot decomposition gas mixed with cooler gas in the cylinder at the start of each cycle. Heat loss to the surrounding environment further dropped the temperature.

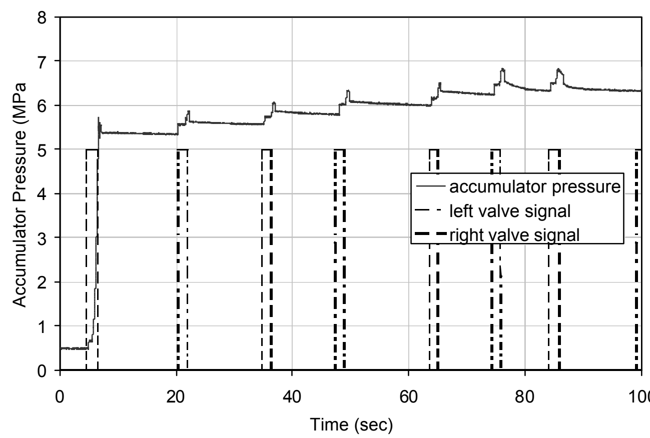

Figure 7: Start-up transient accumulator pressure.

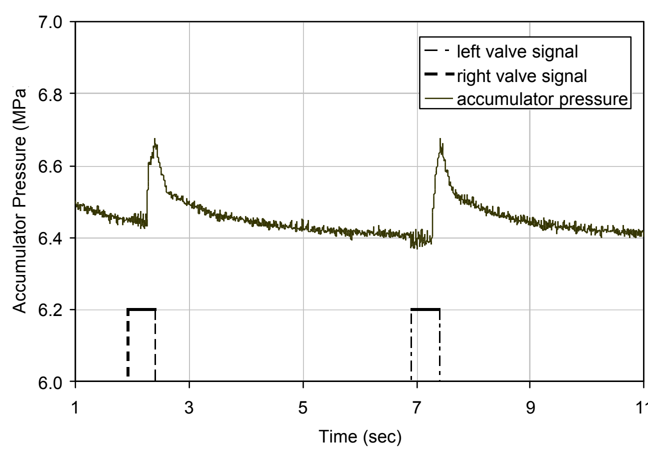

Figure 8: Steady-state accumulator pressure profile.

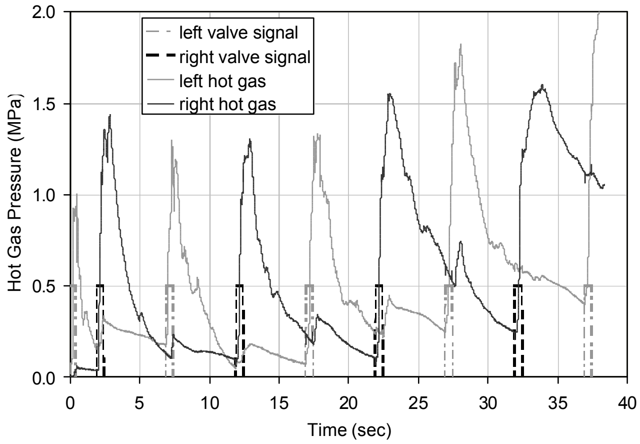

Figure 9: Hot gas pressure.

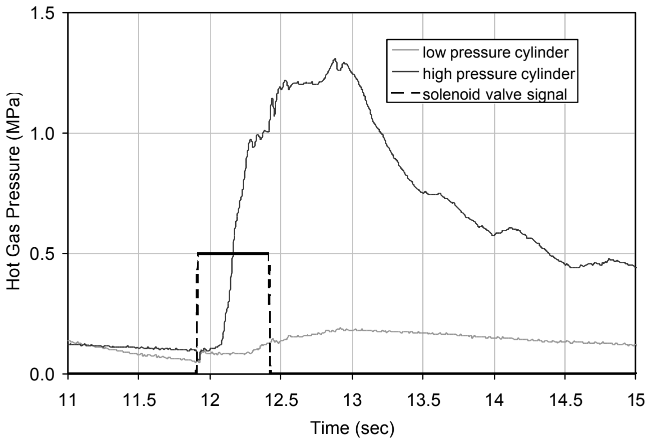

Figure 10: Detail view of hot gas pressure rise

6 DISCUSSION

6.1 PERFORMANCE OVERVIEW

It is evident that the measured performance of the FPHP was far below the expected theoretical performance. The plots of the accumulator pressure (Figs. 7,8) demonstrate the ability of the FPHP to successfully pump hydraulic fluid at approximately 6.5 MPa (940 psi), but only at a low flow rate of 0.48 liters/min (0.13 gallons/min). The hydraulic flow rate was not measured directly, but estimated by multiplying the displaced volume of the hydraulic cylinder by the operating frequency of the free piston assembly. It was assumed that leakage across the hydraulic seals was not significant. The achieved hydraulic pressure was very near the desired pressure of 6.9 MPa (1000 psi) and only fell below the mark because the pressure relief valve was adjusted by manually turning a large screw, thus making it difficult to achieve an accurate pressure relief point. The maximum power was approximately 50 W, taken from a run with a 0.50 second injection time and 1.5 second delay time. The maximum conversion efficiency was a low 1.2%, a result of slow hydrogen peroxide decomposition and poor harnessing of the hot decomposition gases, detailed in the following subsections. The hydrogen peroxide tank was estimated to supply enough monopropellant for approximately 15 minutes of operation at an output of 50 W, sufficient operating time for experimental purposes.

Although not measured directly, the peak free piston speed was estimated to be less than 1.0 m/s, much lower than the theoretical 8.0 m/s initially predicted by the simulations, resulting in the low flow rate and low power output of 50 W. The low free piston speed prevented the free piston from carrying sufficient momentum to drive past the exhaust port, allowing the hot gas in the cylinder to leak out slowly as soon as the exhaust port was partially uncovered. The back pressure of the hydraulic fluid stopped the free piston after the pressure in the hot gas cylinder dropped below approximately 1.0 MPa (150 psi). This situation prevented a complete exhaust of the expanded hot gas to the atmosphere at the end of each cycle, since the gas would bleed out gradually rather than explosively exiting the cylinder. Figure 10 shows clearly the relatively shallow pressure drop after the exhaust port was uncovered. A significant amount of hot gas remained trapped in the cylinder, accumulating over several cycles. The general trend of rising pressure, even in the low-pressure hot gas cylinder of the FPHP, is evident in Fig. 9. As a result, the cycle times of the free piston under full load were much slower than predicted. Due to the low free piston speeds, hot gas leakage around the hot gas piston rings became more significant, resulting in additional lost work. There were two primary reasons for the low piston speeds and the resulting poor performance: slow hydrogen peroxide decomposition and a large effective clearance volume.

6.2 SLOW HYDROGEN PEROXIDE DECOMPOSITION

The FPHP required intermittent low-flow hydrogen peroxide injection, resulting in poor performance of the catalyst beds. The catalyst beds used in the FPHP were commercially available components designed to operate with a high nominal hydrogen peroxide flow rate of 0.11 kg/s under steady flow conditions. The maximum flow rate through the solenoid valves, estimated using Eq. (4), was an order of magnitude below the nominal flow of the catalyst bed. As the hot gas cylinder pressure increased, the hydrogen peroxide flow rate was further reduced.

The internal geometry of the catalyst bed was a series of fine silver screens that decomposed the hydrogen peroxide on contact. The decomposition process was conjectured to proceed as follows:

1) Liquid hydrogen peroxide was injected at high velocity when the solenoid valve opened.
2) Some hydrogen peroxide contacted the catalyst directly, decomposing rapidly. The rest trickled out of the inlet tube at low velocity when the solenoid valve shut.
3) A significant amount of hydrogen peroxide pooled in the long inlet tube, slowly decomposing at low pressure when it came into contact with the catalyst screens.

The above process resulted in an effective mass flow rate of hot gas that was much lower and lasted longer than the initial mass flow rate and injection time of the liquid hydrogen peroxide. This explains the necessity of allowing a large delay time after each injection cycle to let the injected hydrogen peroxide decompose completely. There is evidence that even with very large delay times of five seconds, there was still a residual amount of hydrogen peroxide decomposing in the catalyst bed that never fully vented to the atmosphere. The hot gas pressure, shown in Fig. 10, dropped after the hot gas piston uncovered the exhaust port, yet there were actually small pressure *increases*, visible at 14 and 14.5 seconds, in this portion of the pressure curve. This residual hydrogen peroxide could build up over several cycles and exacerbate the observed increasing trend in the pressure profiles, as documented in Fig. 9. These pressure increases eventually resulted in extremely high cylinder pressures on both sides of the free piston assembly, producing incomplete free piston stroke lengths.

6.3 LARGE CLEARANCE VOLUME

The internal volume of the catalyst bed contributed to the clearance volume in the hot gas cylinder when the free piston assembly was in the full left or full right position. There was a large visible cavity at the hot gas exit side of the catalyst bed, and there was likely a significant amount of space within the catalyst mesh portion. Short steel tubes leading to the two pressure transducers on each catalyst bed also contributed to this volume. This resulted in added free expansion of the hot gas and less work extracted from each stroke. The free piston assembly also moved more slowly since the peak pressure was reduced.

These two detrimental effects, a reduced decomposition rate and increased clearance volume, can be modeled in the dynamic simulation developed by McGee [11]. When the aforementioned changes were made in the simulation, the theoretical pressure curve closely matched the experimental data.

7 CONCLUSIONS

The Free Piston Hydraulic Pump (FPHP) represents a new concept for a power supply for mobile robotic applications, integrating a monopropellant-based system with a free piston pump. The FPHP was conceived, designed, constructed, and tested to prove its functionality. In testing with 90% hydrogen peroxide, the prototype FPHP achieved a hydraulic power output of 50 W and a conversion efficiency of 1.2%. While these performance ratings are too low for a viable mobile robotic power supply, the FPHP demonstrated the concept of a monopropellant driven hydraulic power supply. A more effective catalyst bed with a greatly reduced internal volume and a lesser distance between the solenoid valve and catalyst would result in a more rapid rise in hot gas pressure when hydrogen peroxide is injected, producing a more viable device with a greater power density and higher efficiency. By thus improving the performance of the FPHP, it could be used for field applications in anaerobic environments.

ACKNOWLEDGEMENTS

This project received funding from the Office of Naval Research grant number N000-14-98-1-0669.

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