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Abstract

The main objective of the project is to develop advanced, low cost conversions of small (< 25 hp) gasoline internal combustion engines (ICEs) to run on hydrogen fuel while maintaining the same performance and durability. This final technical report summarizes the results of i) the details of the conversion of several small gasoline ICEs to run on hydrogen, ii) the durability test of a converted hydrogen engine and iii) the demonstration of a prototype bundled canister solid hydrogen storage system. Peak power of the hydrogen engine achieves 60% of the power output of the gasoline counterpart. The efforts to boost the engine power with various options including installing the over-sized turbocharger, retrofit of custom-made pistons with high compression ratio, an advanced ignition system, and various types of fuel injection systems are not realized. A converted Honda GC160 engine with ACS system to run with hydrogen fuel is successful. Total accumulative runtime is 785 hours. A prototype bundled canister solid hydrogen storage system having nominal capacity of 1.2 kg is designed, constructed and demonstrated. It is capable of supporting a wide range of output load of a hydrogen generator.

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Executive Summary

This is the final technical report under the contract (DOE Award No: DE-FC26-06NT43026) from October 1 2006 to September 30, 2010. The report summarizes the research and development results of i) the conversion of small (< 25 hp) gasoline internal combustion engines (ICE) to run hydrogen fuel, ii) the durability test of a hydrogen engine and iii) the demonstration of a prototype bundled canister solid hydrogen storage system. A Briggs & Stratton V-twin engine is successfully converted to run on hydrogen. This is accomplished using an advanced carburetor system (ACS) that consists of a BEAM T-60C regulator and a throttle body. An optimized combination of regulator parameters, Venturi size and air-fuel ratio is reached via trial and error to achieve stable engine operation on hydrogen. Peak power of the hydrogen engine with ACS is 7.9 hp (5.9 kW) at 3,600 rpm, which is approximately 60% of the power output (13.5 hp or 10.2 kW) of the unmodified gasoline engine. The attempt to boost the engine power by installing a IHI RHF-3 turbocharger along with the ACS reduces the output peak power to 6.0 hp (4.5 kW) which is additional 16% power loss. The power loss may be attributed to an increased intake restriction by the compressor of the turbocharger, or the backfiring causing lean combustion. Another possible cause is the air flow capacity of engine is inadequate to drive the turbine to a high speed to achieve boost pressure. A solution is to replace it by a smaller version of turbocharger which is not available in the market at the present time. A Briggs & Stratton V-twin engine is also equipped with a retrofit of a pair of custom-made pistons (higher compression ratio of 14:1), a MDS MC-4 advanced ignition system and ACS to test run with hydrogen. The engine operated stably without knocking and backfiring is under the condition of very lean fuel/air lambda ratio, which limits peak power output at about 5.5 hp (4.1 kW). To boost the engine peak power, a great deal of effort is also devoted to the development of various fuel injection systems for a water cooled Honda GX350 engine. The results are disappointed that the engine governor and mass air flow are unsuccessful to work together to stabilize rpm and respond to load resulting in engine backfiring. Since the generator causes so many problems to the hydrogen injection, extra effort is made to tune the hydrogen engine by fuel injection control without a governor linked throttle. With the throttle-less setting and other fuel injection strategies, the Honda GX350 hydrogen engine operates smoothly and produces up to 4.7 hp (3.5 kW) electrical output which is 50% of the output peak power (9.4 hp or 7 kW) obtained from the

performance of this engine at 3,600 rpm using liquid propane (LPG). Pushing more power by moving into a territory of rich fuel burning, at lambda ratio of 1.5 or lower, the engine is experienced backfiring. The durability test is conducted on an in-house converted small single cylinder engine (Honda GC160 engine) using the commercial T-size hydrogen cylinders. The total runtime of the hydrogen engine is 785 hours. One hundred sixty-nine T-size cylinders are consumed during the durability test. Ninety-eight of them are able to continuously run the generator from full to empty at constant output load of 1kW. Runtime varied from 250 min. to 300 min. resulting to an average runtime of 267 min. per T-cylinder and an average hydrogen flow rate of 2.31 grams per min. (25.9 slpm). Although the engine durability does not reach to the originally planned 1000 hours, it is still a plausible accomplishment by considering the fact that Honda GC 160 engine is a very economic engine for backup generator. A prototype bundled canister solid hydrogen storage system with a nominal hydrogen capacity of 1.2 kg is designed, constructed and demonstrated. The bundled canister system consists of sixteen Ovonic 85G250B canisters mounted in a liquid coolant jacket. The result shows the prototype bundled canister system is capable to sustain 522 min. and 254 min. runtime for the average flow rates of 26 slpm and 52 slpm (required for the generator to deliver the output load of 1kW and 2kW), respectively at liquid coolant temperature of 40 °C.

Result Details

A. Hydrogen Engine Development

I. Engine selection and acquisition

The engine selection is based on the following two criteria: i) 95% of generator engines are typically overhead valve engines, and most transportation engines are of the overhead cam type and ii) the preference of having engine equipped with forced oil lubrication is allowed the use of turbocharger to boost the power. After extensive research and comprehensive market survey on the availability of different kinds and sizes of engines, it is found only twin cylinder engines for generators have forced lubrication. Two particular engines are selected for this development

program and the corresponding photos are shown in Figures. 1 (engine #1) and 2 (engine #2). Engine #1 is a 173 cc overhead cam automotive engine, used in light weight vehicles, marketed in 34 countries, including the United States and engine #2 is a Briggs and Stratton 480 c.c. V-twin engine, overhead valve, used for back-up power generator.

Figure 1: Photo of an overhead cam engine for light weight transportation (engine #1).

Figure 2: Photo of an overhead valve V-twin engine for stationary power (engine #2).

II. Baseline gasoline engine power and emission test

Since the attempts to obtain any data from the engine distributors or manufacturers are unsuccessful, the performance of the dynamometer and emission test for baseline gasoline of both the automotive engine #1 and generator engine #2 is carried out in-house. The engine dynamometer test is conducted using Land and Sea dyno cell. A water brake absorber is connected to the engine shaft. Torque and rpm sensors installed on the absorber are wired to DYNOMite hand held data acquisition controller. A personal computer is connected to the DYNOMite controller by a RS232 communication cable. Dynomax 2000 engine dynamometer software on the PC runs the data acquisition, records and processes the data of each engine test. By measuring the torque produce by engine and rpm, engine horsepower can be calculated in real time by Dynomax 2000.

Engine exhaust emission is monitored by using a TIF 5GA five gas automotive exhaust emissions analyzer. Its emissions probe is inserted to the very end of exhaust pipe to avoid contact with high temperature. The sample exhaust gas is first filtered, and then sucked into the analyzer by an internal pump. This analyzer uses non-dispersive infra-red absorption to measure and display the volume concentrations of hydrocarbon (HC), carbon monoxide (CO), and carbon dioxide (CO₂). The concentrations of oxygen (O₂) and nitric oxides (NO_x) are measured by electrochemical sensors. This detector can also calculate and display air/fuel ratio, or Lambda. Sample data at preset time intervals (from 1 to 10 seconds) can be captured and stored into its internal memory. After the test, the stored data can be read manually and keyed into a PC MS Excel data sheet for graphing.

The generator engine #2 is a V-twin Briggs and Stratton tri-fuel engine that can run on gasoline, CNG, and propane. The experimental setting for gasoline baseline performance test is shown in Figure 3. The dyno cell is on the left side of the engine, and a one gallon gas tank is shown suspended on a steel arm to the right of the engine. Engine oil pressure is also monitored by using a dial gauge that is installed right below the gasoline tank.

Figure 3: Engine #2 and dynamometer on the test bench for gasoline performance measurements.

Since the generator engine is controlled by a governor to run at a constant speed of 3600 rpm, the power curve at its peak value is plotted against time. Figure 4 shows a typical graph plotting engine peak power, engine torque and rpm versus time. The peak power of the baseline gasoline generator engine #2 is about 13.5 horsepower (hp) or 10.2 kW.

Figure 4: Baseline gasoline performance test for generator engine #2 - torque, rpm, and power curves vs. time. The plateau gives the peak power at about 13.5 hp (10.2 kW).

The emissions test is conducted by operating engine #2 at power outputs of 3.8, 7.0, 10, and 13.5 hp (or 2.9, 5.3, 7.5 and 10.2 kW, respectively), according to general EPA guidelines. The five gas concentrations and Lambda are plotted versus time in Figure 5. It indicates that the NO_x emission increases with engine output power up to 10 hp, and that it decreases slightly when the engine runs at peak power. The drop of NO_x in peak power may be attributed to richer fuel/air ratio. As indicated in Figure 5, the combustion changes from lean burn at low power range to rich at high power. CO concentration increases with engine power and as the fuel/air mixture gets richer toward wide opened throttle. Hydrocarbon (HC) concentration of the exhaust is found to be very steady at about 200 ppm.

Figure 5: Five gas emissions are plotted versus time (CO, CO_2 and O_2 in volume percentage, NO_x , and HC in ppm). The emissions step-wise change corresponds to engine output powers. The emission test was conducted with gasoline fuel.

III. Development of advanced carburetor system (ACS) and test

An advanced carburetor system (ACS) is built based on a commercially available throttle body for gaseous fuel, such as natural gas and propane gas. Modifications to the throttle body are made to accommodate different sizes of Venturi insert. Its arrangement is illustrated in Figure 6. This comprises a throttle, a Venturi, a hydrogen gaseous fuel port, and an adjustment screw for the air-fuel ratio. The throttle is linked to the generator governor to limit and maintain the engine rpm at 3,600. As noted above, the Venturi insert can be changed. The hydrogen fuel port is connected with a gaseous fuel regulator shown in Figure 7.

Figure 6: Low restriction ACS throttle body for the hydrogen engine.

Figure 7: Two-stage hydrogen gas regulator.

The particular regulator used along with the customized ACS is a Beam model T 60C, made by Cerritos, CA. It regulates incoming hydrogen gas, at pressure up to 250 psig (1.7 MPa), down to about 4 psig (0.03 MPa) by its primary stage. The primary pressure can be measured through the rest port, and adjusted by a primary spring retainer. Gaseous fuel at the primary pressure will be drawn further to the Venturi by a slight vacuum, where the hydrogen mixes with air during the intake stroke. The amount of hydrogen gas released from the regulator to the Venturi is proportional to the air velocity. The air-fuel ratio can also be controlled by an orifice located between the Venturi and the regulator whose size can be changed by the adjustment screw.

The actual air-fuel ratio of the combustion mixture is monitored by an ECM air-fuel ratio analyzer – a Recorder Model 1200. ECM Model 1200 uses a wide range oxygen detector (from 0 to 22 %) that samples the exhaust. It is capable of displaying the air-fuel ratio for hydrogen fuel.

Since hydrogen has a flame speed of 2.65 to 3.25 m/s, which is an order of magnitude faster than that of gasoline, the advanced spark timing can be retarded compared to that of the gasoline engine. The Briggs & Stratton V-twin OHV engine has a stocking ignition timing 23 degrees

before top dead center (BTDC). It uses MAGNETRON ignition, a self-contained transistor module. The armature produces the ignition when the magnet on the flywheel rotates to its proximity. To retard the ignition timing, a special flywheel key is made and installed to shift the magnet position on the flywheel relative to the armature. The modification for retarding ignition timing is illustrated in Figure 8. The final ignition timing for hydrogen is set at 12 degrees BTDC.

Figure 8: Schematic of a shifted flywheel key for retarding ignition timing.

IV. Emission test of engine with modified ACS

With retarded ignition timing and an optimized ACS installed to the engine, hydrogen engine performance and emissions are tested. Results show that the hydrogen engine runs smoothly at lean air-fuel mixtures. The lowest air-fuel ratio that enables the engine to run without back-firing is about 1.5. At that air-fuel ratio, peak power is about 7.9 hp (5.9 kW) – compared to 13.5 hp (10.2 kW) when the unmodified engine is operating on gasoline. NO_x levels are measured at 1500 ppm. Details of the hydrogen engine performance and emission test results are shown in Figures 9 and 10. The data show that the engine output power reduction by converting the engine using only the ACS is approximately 40%. During the tests, it is also observed that NO_x

emissions are correlated to the air-fuel ratio. The lower the air-fuel ratio is, the higher the NO_x emissions are. Due to the larger reduction in peak power, it is extremely challenging to fully restore the engine's peak power by applying only a higher compression ratio. However, it is believed that adding a turbocharger can potentially boost the engine's power to approach the output of the original gasoline engine.

Figure 9: RPM, torque, and horsepower profiles of the modified hydrogen engine with ACS.

Figure 10: NO_x emissions of the modified hydrogen engine with ACS.

V. Hydrogen engine with turbocharger and ACS

Two models of IHI turbochargers for small engines are acquired and used for power recovery effort of the hydrogen engine. They are IHI turbo models RHB-3 and RHF-3 respectively, as shown in Figure 11 (a & b). These models of turbochargers are typically used for motorcycle gasoline engine at power range up to about 50 hp (or 300 cc to 600 cc) gasoline engines. The model RHB-3 is the preceding version of RHF-3. The Model RHF-3 has water cooling capability, which is desirable for enhanced reliability and improved engine operation. The work is exclusively performed on model IHI RHF-3 with water cooling capability.

Figure 11: a) IHI RHB-3 turbocharger and b) IHI RHF-3 water cooled turbocharger.

Figure 12 is a schematic illustrating the basic structure of a hydrogen engine that combines a gaseous fuel carburetor system and turbocharger. As the engine exhaust blows through the turbine compartment of the turbocharger, the high speed, high temperature combustion gases and water vapor drives the turbine to an extremely high rotation rate, up to 200,000 rpm. The rotational power is transported to the compressor through a link shaft. The dense air and fuel mixture is forced into the engine combustion cylinder during the intake stroke. As described early in section III, ACS system is connected directly to the throttle body for the hydrogen engine converted without using turbocharger. As the air and fuel mixture between the intake valve and the turbo compressor was under boost pressure, the ACS system is removed from the

throttle body, and attached to the upstream of the turbo compressor. Engine oil is pumped and routed around the shaft area to provide lubrication and cooling. The pressurized oil passes through the turbo shaft and then feeds back to the engine crankcase. It is important to have a much larger outlet orifice than the inlet orifice to reduce the back pressure of the turbocharger oil. A high back pressure of turbocharger lubricant oil may cause oil seepage into the chambers of both exhaust turbine and air compressor along the link shaft. The initial failure of tested model IHI RHB-3 may be attributed to a high back pressure of lubricant oil. Additional water cooling is provided to prevent the air and fuel mixture from being heated. Since the engine power is direct proportional to boost pressure and mixture density, a cooler mixture is a very effective means for gaining engine power.

Figure 12: A schematic of a single cylinder turbocharged hydrogen engine.

Figure 13 shows the small V-twin Briggs & Stratton engine retrofitted with an IHI RHF-3 turbocharger and an ACS system. The IHI RHF-3 turbocharger is made for general application of any small gasoline engines, and comes with no fitting and adaptor in both air and exhaust sides. ECD has to design and make all the pipeline and pipe adaptors, as well as water cooling inlet and outlet connectors. The connection between the turbo compressor and the throttle body

is made by using three-layer silicone pipe. The silicone material absorbs vibration and also sustains expected boost pressure. Since the air and fuel mixture at the throttle body is under boost pressure, the gas mixture may leak out along the interface between the throttle shaft and throttle body. Therefore a shaft seal is installed around the throttle shaft to prevent gas leakage. The throttle shaft provides a linkage to the governor that regulates the engine speed at approximate 3600 rpm.

The hydrogen engine equipped with the IHI RHF-3 water cooled turbocharger is initially test-run on hydrogen at 3600 rpm and no load condition for about 10 minutes. It is observed that the IHI RHF-3 turbocharger can sustain the oil pressure without oil leakage, has sufficient cooling to keep the temperature in the compressor side close to ambient.

Figure 13: A small hydrogen engine equipped with a turbocharger and ACS system.

The performance tests are carried out at two states of the IHI RHF-3 turbocharger: the first with its exhaust waste gate open, the second with the waste gate closed. When the waste gate of the turbocharger is open, the exhaust bypasses the turbine. Therefore, the open state of waste gate does not produce boost pressure. Figure 14 shows the performance profiles of the hydrogen engine in open state. The engine peak power at about 3600 rpm is 5.6 hp (4.2 kW), which indicates that the peak power of the hydrogen engine reduces by 30% from what produced by an engine equipped with ACS alone. This power loss is attributed to an increased intake restriction by the compressor of the turbocharger.

Figure 14: Engine torque in foot pound, and horsepower tested with opened waste gate of turbocharger.

With the waste gate closed, the engine exhaust is forced to blast through the turbine of the turbocharger. The high speed, high temperature exhaust drives the turbine to an extremely high rotational speed. The higher the turbine speed is, the higher the boost pressure of the air/fuel mixture. Figure 15 shows the performance profiles of the turbocharged hydrogen engine with closed waste gate. The test result reveals that the peak power at about 3600 rpm is 6 hp (4.5 kW), which indicates that the turbocharger is still not fully functional. The lack of power increase from an under-driven turbocharger is also consistent with the negligent reading of boost pressure.

Figure 15: Engine torque in foot pound, and hp tested with closed waste gate of turbocharger.

During the performance test of the turbocharged hydrogen engine, it is observed that an air/fuel ratio richer than Lambda 1.8 induces backfiring. The fact that turbocharged hydrogen engine fails to gain power may also be related to lean combustion. Therefore, eliminating backfiring at richer mixture may allow the engine to gain more power. The technique to eliminate backfiring at rich mixture is to replace the ACS by a hydrogen fuel injection. Another reason for the fact that the turbocharger fails to function properly is the air flow capacity of engine is inadequate to drive the turbine to a high speed to achieve boost pressure. A solution to that problem is to replace it by a smaller version of turbocharger which is not available in the market after a comprehensive search. One other possible solution to the above problem is to increase the power output by using pistons of a higher compression ratio.

VI. Optimization of hydrogen engine with high compression ratio

With respect to the fact that the previous effort failed to recover the power of small hydrogen engine by retrofitting a turbocharger, the back-up approach to turbocharger is to increase the power output by using pistons of a higher compression ratio. Although the hydrogen engine with a higher compression ratio may not have the comparable potential for power boost to turbocharger, it is technically simpler and feasible to achieve a modest gain in power and fuel efficiency. More importantly, it is much lower cost. To gain power from an aspirated hydrogen engine with a higher compression ratio, it is important to achieve a stable burning at a fuel/air mixture close to a stoichiometric ratio. To obtain a steady burning of a rich mixture, it is critical to retard ignition timing and eliminate the waste spark.

Hydrogen has a higher auto-ignition temperature than gasoline and natural gas. By increasing the engine compression ratio from 9.0 to 14.5, the engine fuel economy increases. The gain in fuel efficiency directly contributes to a peak power increase of the hydrogen engine. To increase the engine compression ratio, the dead volume of the combustion chamber is reduced by installing new pistons. Figure 16 shows a pair of pistons for the original compression ratio and the higher ratio. The new piston for higher compression ratio is custom-made by Arias Pistons in California.

Figure 16: An original piston and a higher compression piston.

A second engine featuring a combination of higher compression pistons and ACS is converted to operate on hydrogen. This engine has also been integrated with a power generator, as shown in Figure 17. Although the hydrogen engine is not yet optimized and fine-tuned, the generator produces steady power of more than 5 kW. The electrical load is provided by three furnaces and a bank of light bulbs.

Figure 17: A hydrogen engine with higher compression pistons, integrated with a power generator supplying to 3 furnaces and 8 light bulbs.

As described in the above session, a V-twin Braggs and Stratton engine is retrofitted with a pair of custom-made pistons, increasing the compression ratio to 14.5 from the original 9. This engine is also modified with an ACS hydrogen fuel delivery system. The finished engine is integrated with an electric generator. A preliminary test then indicates that the engine can run smoothly, and supplied power to an electric load of about 5 kW. However, this engine featured with a higher compression ratio is not dyno-tested and optimized. Therefore, the exact power output is unknown.

i. Initial dynamometer test

The engine with a higher compression ratio is pulled out from the power generator, and mounted on the dynamometer test bench. The initial test of the engine produced about 5 peak hp at a 3600 rpm. Furthermore, the engine is observed to run smoothly only at an air/fuel Lambda ratio of 2.0 or higher, indicating an extreme lean burning under the condition of a higher compression ratio and the original ignition timing. Conversely, the engine modified to run on hydrogen with ACS and the original compression ratio, can run smoothly at an air/fuel Lambda ratio as low as 1.5. The analysis suggests that the stock ignition timing has to be retarded in order to improve the performance of the hydrogen engine with pistons of higher compression ratio.

ii. Stock ignition System

The stock engine is equipped with two magneto ignition devices, with each connected with a spark plug in individual cylinder head. The detailed can be seen in Figure 18. The magneto device has a ferro-magnetic coil mounted on the engine body along the peripheral of the flywheel, and a permanent magnet installed in the outer edge of the flywheel. The coil is energized as the magnet passes by. The high voltage pulse induced in the coil produces a spark between the electrode gaps of the ignition plug. The original coil position results in a spark timing of 23 crank-degrees before the TDC (top dead center). The stock ignition also produces a waste spark in the exhaust stroke for every engine cycle. The waste spark may be desirable to ignite fuel residual in exhaust for gasoline engine, but is highly suspicious of causing backfiring of hydrogen engine, when the valve timing overlap occurs at the tail of the exhaust and the onset

of the intake. Pre-ignition caused by excessively advanced ignition results in power loss, and even engine damages.

Figure 18: Original fixed spark timing, magneto ignition coils.

iii. Programmable ignition System

In order to optimize the performance of the hydrogen engine with a higher compression ratio, the original stock magneto ignition system is removed, and replaced by a MSD programmable ignition system. In addition, the ignition trigger to the camshaft wheel is also moved, which renders only one firing every other cycle, hence, eliminates the waste spark associated with crank ignition system. The ignition retard is implemented by using a programmable ignition digital control module MSD MC-4. The wiring configuration of the ignition control module is shown in Figure 19.

Figure 19: The wiring schematics of a programmable ignition system.

The Programmable MC-4 uses a high speed RISC microcontroller to control the ignition's output while constantly analyzing the various inputs such as supply voltage, trigger signals, and rpm. The MC-4 features a capacitive discharge (CD) ignition design. The majority of stock and aftermarket ignition systems are inductive ignitions. In an inductive ignition, the coil must store energy and step up the supplied voltage to maximum strength between each firing. At higher rpm, since there is less time to charge the coil to full capacity, the secondary voltage falls short of reaching its maximum energy level, which results in a loss of power or a top end miss. The MC-4 Ignition features a capacitor which is quickly charged to 490 - 505 volts and stores this energy until the ignition is triggered. With the CD design, the voltage sent to the coil is always at maximum power even at high rpm. The MC-4 produces full power multiple sparks for each firing of a plug. The number of multiple sparks that occur decreases as rpm increases, however the spark series always lasts for 20° of crankshaft rotation. Above 3,300 rpm there is simply not enough time to fire the spark plug more than once, so there is only one full power spark. The real spark timing for the converted hydrogen engine is determined by two parameters: one is cam wheel trigger position; another is run curve retard set by the MSD MC-4 programmable digital ignition control system.

iv. Ignition trigger and pickups installation

To get rid of the waste spark associated with the ignition based on the crank wheel triggers, two Hall Effect triggers are installed on the engine casing near the camshaft wheel. One permanent magnet made of a samarium cobalt alloy that has an operation temperature of 250 °C, is attached to the camshaft wheel by high temperature epoxy glue. The detail of the pickups and magnetic trigger can be seen in Figure 20. The alignment of the magnetic trigger and pickup 1 is positioned at 25 crank degrees before the TDC, which produces an ignition spark in the cylinder 1 at 25 crank degrees before TDC. The pickup 2 is installed in a position that is 225 degrees cam-angle with respect to pickup 1. The real sparking timing can be verified by reading an angular scale with timing lighter. The angular scale is amounted on the position of original magneto coil, as illustrated in Figure 21. Although the physical trigger/pickup alignment produces sparks at 25 crank degrees before TDC, the real timing can be further retarded by programming from the ignition system. As a result, the spark timing can be variable from 25 to 0 degrees before TDC, since the maximum retard by the MSD system is 25 degrees. The MSD ignition system also allows the use of variable timing at various engine rpm.

Figure 20: The pickups and magnet mounted on the camshaft wheel and casing.

Figure 21: Markup for verification of real spark timing.

v. Electronic design of Hall Effect pickups

The housing of Hall Effect pickups available from MSD and other vendors are made in different geometries and sizes. These pickup sensors are either electrically dysfunctional, or their footprint do not fit into the specific mounting position on the engine casing. Additionally, the cost of pickup sensors can be expensive relative to the prices of small engines. For these reasons, ECD has designed a functional Hall Effect pickup circuit by using low cost semiconductor components. The cost of these components is less than 5 dollars, versus 150 dollars for each pickup sensor from vendors. The detail of the pickup circuit is presented in Figure 22. It consists of an omni-polar Hall Effect IC sensor made by Allegro Microsystems Inc., and two NPN transistors, plus three current regulating resistors. The output of the pickup circuit is a square wave, with the output voltage signal normally at 12 volts, and changes to near zero as it is triggered by a passing magnet mounted on the camshaft wheel.

Instead of packing all the components in a threaded tubular housing, like most pickups purchased from vendor, the Hall Effect sensors shown in the left of Figure 22, which have small footprints (3x2x3 mm each), can be singled out and readily accommodated into a limited space on the engine casing, so that the circuit can be triggered by the magnet mounted on the camshaft wheel.

The rest of the electronics can be placed outside the housing of the Hall Effect sensor, and electronically connected.

Figure 22: Electronic layout of a Hall Effect pickup.

VII. Test the engine with higher compression ratio

After above mentioned optimization of the 460 cc V-twin Braggs and Stratton engine, it is equipped with a retrofit of a pair of custom-made pistons (compression ratio 14:1), a MDS MC-4 advanced ignition system and ACS. The MSD ignition system is programmable for various spark timings, and no waste spark is produced. During the dynamometer tests, spark timings from 5 to 23 degrees BTDC are experimented. It is found that highly advanced spark causes knocking, while aggressively retarded spark exacerbates backfiring. The optimal spark timing is at 15 degree BDTC. In addition to the variation of spark timing, air/fuel ratio is also adjusted by ACS from high lambda value (lean mixture) to low lambda value (rich mixture). The desired lambda ratio is to be as close as possible to 1 at stoichiometric air/fuel mixture. The experiment demonstrates that the lowest Lambda value achievable by the ACS is 1.5. Any richer mixture is prone to backfiring, regardless of spike timing. A typical performance test is shown in Figure 23. As it shows, engine could only be operated stably under a condition of very lean fuel/air lambda ratio, which limits peak power to about 5.5 hp (4.1 kW).

Figure 23: A run time profile of power, torque and rpm of the air-cooled engine equipped with pistons of higher compression ratio (14:1).

Although the mechanism for backfiring in the condition of no waste spark is not yet fully understood, poor oil screening by piston rings and a high cylinder temperature may be attributing factors. Without waste spike at the moment of overlap between exhaust and intake valve openings, the reason for persistent backfiring at richer fuel/air mixture may have something to do with glaring micro-particle of carbon soot. The soot dust can be produced by burning lubricant oil that leaks into the combustion chamber through piston rings. Additionally, a thermocouple probe fixed to the outside surface of engine head gives a reading at 220 °C. A higher cylinder, valve temperatures may exacerbate backfiring.

Not only is the engine performance greatly compromised, frequent backfiring can also cause engine damaging. Backfiring is a serious fire hazard for hydrogen fuel delivery system. Eliminating backfiring is crucial for engine performance, engine reliability, and engine durability. Combination of hydrogen fuel control limitation by carburetion delivery system and high engine temperature of air-cooled engine hold us in handicap for developing better performing hydrogen engines. For that reason, it is decide to replace the air-cooled Briggs Stratton engine by a water-cooled engine of comparable power, and substitute fuel injection system for carburetion.

VIII. Performance of water-cooled engine with ACS

The majority of small engines less than 20 hp (15 kW) is air-cooled. Water-cooled engine for residential backup and recreation vehicle generators is less-common. However, Honda Motor Co. is manufacturing a 5 kW generator that uses a water-cooled engine. Generator models EV6010 and EV4010 are marketed in the United States. The engine model GX360K1 is used for both generators and is water-cooled and also flexible fueling by either gasoline or gaseous fuel. The generator comes with fuel delivery system for liquid propane gas (LPG), or natural gas.

The Honda GX360K1 engine is a 4 stroke, overhead cam, 2 cylinder, and water-cooled engine. The displacement of the engine is 359 cc, with a compression ratio of 8.5:1. It uses a type of transistorized magneto ignition, with fixed spark timing at 24 degree BTDC. Unlike the air-cooled V-twin Briggs Stratton engine that has an odd firing cylinder configuration, the engine has a configuration of inline cylinders, which uses an even-firing ignition sequence. The Honda GX360K1 engine is dismounted from the electric generator, and installed on the bench of dynamometer test. Figure 24 shows the engine with its crankshaft coupled with a Land and Sea water brake dyno-cell. Since the engine comes with a gaseous fuel delivery system, the baseline performance test is carried out by using LPG. LPG is obtained readily from a local barbeque equipment store.

Figure 24: A water-cooled generator engine mounted on dynamometer test bench.

Figure 25 is a performance profile that plots the power and torque against running time at engine speeds around 3600 rpm. From the graph, this engine makes about 9.4 hp (7.1 kW) peak power with LPG as fuel.

Figure 25: The Baseline power and torque at around 3600 rpm, fueled by propane.

To test-run this engine on hydrogen, the original duel carburetor is connected with the Beam model T-60 regulator used previously for the air-cooled V-twin Briggs and Stratton engine. This setting can be readily put together and tested. With the stock ignition at 24 degree BDTC unchanged, maximum peak power is approximately 3 hp. With the duel fuel carburetor replaced by the low restriction ACS throttle body used for the V-twin engine, the peak power is increased to about 5 hp. Attempt for drawing more power out of the hydrogen engine by enriching fuel/air mixture beyond an air/fuel lambda ratio of 1.5 is failed in a similar fashion to the effort made on the air-cooled engine. This indicates that the ACS carburetion system has to be replaced by a more controllable hydrogen fuel injection system.

IX. Development of fuel injection system for water-cooled engine

Adopting a hydrogen fuel injection system is an alternative to improve the engine performance,

specifically increasing peak power at engine speed around 3600 rpm and eliminating backfiring. Compared with air-aspirated low restriction carburetion or advanced carburetor system (ACS), fuel injection is more complex but possesses greater flexibility for hydrogen delivery in an ICE engine. Previous effort using ACS finds the backfiring is persistent as the air/fuel Lambda ratio is reduced to 1.5. Therefore, the power output of the hydrogen engine is limited to a lean fuel mixture. In a fuel injected system, fuel can be introduced to the combustion chamber at various pressures and in precise timing. There are two different methods for hydrogen injection. The first is to stream it to the shared intake manifold or the intake port of individual cylinder. The second is to inject fuel directly into the combustion chamber from the cylinder head.

i. Installation of fuel injection system

An Electromotive TEC³ engine control system is used for hydrogen fuel injection. Figure 26 shows the TEC³ unit that is connected by a wiring harness to the Honda GX350 engine sensors and is interfaced with a computer. This electronic engine control system is widely used for automotive gasoline engines up to 12 cylinders. The closed loop control sequence of the unit is to acquire engine inputs, to process the inputs with pre-set instruction, and finally to generate outputs for engine control.

Figure 26: An Electromotive TEC³ fuel injection control unit.

Two essential sensor inputs for TEC³ are: engine speed/position sensors, and engine load sensors. Other optional sensors include coolant temperature sensor, manifold air temperature sensor, knock sensor, and exhaust oxygen sensor. The TEC³ ECU features a high-speed microprocessor for inputs sampling, conditioning, and processing. It has eight dedicated channels of low-side drivers for fuel injectors and four independent drivers for direct firing ignition control. Other features include four user-definable general purpose outputs for controlling devices like the turbocharger, intercooler, or idle motor. The computer software is a windows-based user friendly interface used to calibrate engine configuration, input sensors parameters, set engine ignition timing and fuel injection pulse-width. Basically, it provides a simple interface to quickly set-up engine configuration by inputting parameters such as engine cylinder number, estimated engine horsepower, sensor parameters, and injector sizes, etc. The TEC³ software also provides the capability of so-called “on-flying” adjustment for engine ignition and fuel pulse-width. “On-flying” adjustment makes it a powerful tool for performance optimization by adjusting control outputs as the engine is running.

ii. Installation of trigger wheel for engine position sensor

For the TEC³ engine control system to work, it depends on two primary analog inputs: one is engine speed and position; the other is engine load. The engine position sensor tells the control unit precisely the crank angle position of piston in cylinder number 1. Based on the engine position and speed, the control unit generates output signals to trigger the spark plugs and fuel injector in desired timing. The engine load sensor determines how much fuel to be introduced into the engine cylinders in terms of fuel injection pulse-width.

The electromotive TEC³ engine control system uses a 60 (-2) tooth crank trigger wheel to produce a high resolution engine position input. It basically tracks the engine cranking to an accuracy of 6 degrees. The high resolution engine position sensing is most critical for ignition timing as engine speed changes rapidly (high acceleration or deceleration). When the engine is used for power generation the engine speed stabilizes ideally at 3600 rpm to produce 60 Hz electricity. For this reason the original magneto coils of the Honda GX350 engine are retained in

our initial test, although a fixed ignition timing and waste spark resulted. Only the fuel injection control function of the TEC³ is utilized in this case.

An alternative to the 60(-2) crank trigger wheel is a 120(-4) tooth cam trigger wheel installed on the cam shaft. Since the cam trigger wheel is smaller in diameter, and is more conveniently modified and accommodated, it is used in all of our hydrogen injection work this quarter. Figure 27 shows the 120 (-4) cam trigger wheel (3.25 inches in diameter) mounted on the engine's cam wheel.

Figure 27: Cam trigger wheel mounted on the engine cam shaft.

The details of the modified trigger wheel and adapter are shown in Figure 28. The trigger wheel works together with an induction reluctance pick-up sensor (shown in the left side of Figure 27) that sends a voltage wave signal to the TEC³ control unit. Since the camshaft trigger wheel is small in diameter and has smaller teeth size a chisel pointed pick-up sensor is used to achieve adequate depth of wave signal. For reliable position sensing the gap between the chisel tip and tooth top must be less than 0.003 inch. For this reason, the tolerance of trigger wheel and adapter should be less than 0.001 inch.

Figure 28: Cam trigger wheel on the left, adapter on the right.

iii. Installation of engine load sensors

The second major engine input is engine load. The TEC³ control system uses the load input to determine the amount of fuel to be injected. The commonly used engine load sensors are MAP (manifold air pressure) and MAF (mass air flow). MAF sensors are made for a specific engine model, and only available for certain vehicle models as aftermarket automotive parts. It is hard to find a compatible MAF sensor for small engine. Therefore, a MAP sensor is initially used as the load sensor for the Honda GX350 engine. A port is opened on the intake air manifold to provide a connection for the MAP sensor, as shown in Figure 29. To alleviate the strong pulse effect, a small orifice may be added between the MAP sensor and the port.

Figure 29: MAP sensor connected to air intake manifold.

The operating principal of MAP load sensor is simple. As the engine idles with no load the engine throttle is virtually closed and a minimal amount of air is allowed to pass to the engine cylinder. Because of the air flow restriction, the combustion chamber and the air intake manifold up to the closed throttle are in a state of partial vacuum during the intake stroke. When a load is added, the engine speed reduces. The engine governor acts to open the throttle to let more air into the cylinder to stay at the set engine speed (3600 rpm). As soon as the throttle opens up the pressure of the intake manifold increases. The increase in pressure is due to more air drawn into the engine cylinder during the intake stroke. The MAP sensor measures absolute pressure of the air intake manifold and translates the pressure to a DC voltage signal from 0 to 5 volts in a linear scale. Therefore, the magnitude of the MAP signal is proportional to the mass air intake or engine load. The MAP signal is the basis for the control of the pulse-width of the hydrogen fuel injectors.

iv. Installation of fuel injectors

Electronically controlled fuel injection has the following advantages over carburetion for hydrogen fuel delivery: accurate and precise fuel metering, no air restriction, rapid and smooth response to engine acceleration, etc. These benefits from fuel injection lead to cleaner fuel combustion, higher fuel efficiency, higher power, and better response to engine load. While the carburetor is a mechanical and pneumatic Venturi device, the fuel injector is a solenoid controlled fuel nozzle. The fuel nozzle opens when an electric current pulse flows through the solenoid coil, which allows pressurized fuel to rush through.

Hydrogen is a lightweight gaseous fuel but is anticipated to have similar flow dynamics and controllability as gaseous hydrocarbons. Fuel injector widely used for natural gas or liquefied petroleum gas is suitable for hydrogen fuel injection. The fuel injectors used in the experiments are manufactured by Clean Air Power Inc (CAP). Four different injector sizes are available, giving different flow rates and working pressures. The largest size CAP injector has an operation pressure of about 8 bars (0.8 Mpa), and rated at an air flow rate of 432 slpm under a pressure drop of 6.5 bars (0.65 Mpa). The smallest size CAP injector has an operation pressure of 20 bars (2 Mpa) and an air flow rate of 115 slpm under a pressure of drop of 6.5 bars.

The CAP gaseous fuel injector falls within the low impedance category with its solenoid coil resistance at 1.6 ohms. With only one injector driven by a single output channel a 2 amps setting on the TEC³ control configuration is recommended.

Unlike the carburetion fuel delivery, fuel injection is a lot more flexible and can be implemented in many configurations. First, fuel injectors can be installed either to the air intake manifold or to the intake port for each cylinder. Second, fuel injection can be accomplished by either one single or multiple fuel injectors. If two injectors are used, one injector is called the primary and the other the staged. Generally speaking, the primary injector delivers fuel to engine for idling or maintaining constant speed, while the staged injector adds fuel for engine load and acceleration. It is found that the configuration of using two injectors in the air intake manifold achieved good controllability and engine performance. This configuration of hydrogen injection is shown in Figure 30 where the two ports made for port fuel injection are left plugged. The staged injector provides fuel to meet the load. In general, the staged injector is larger than the primary injector.

Figure 30: Manifold hydrogen fuel injection using one primary and one staged injectors.

v. Engine tests and performance

Since the generator engine requires near constant speed the initial projection is the engine governor and MAP sensor are aimed to work together to stabilize rpm and respond to load. However, extensive test indicates that the governor and MAP controlled hydrogen fueling do not cooperate; hence they do not render steady operation. Compared to the fuel injection control in a gasoline engine, it is far more challenging to implement hydrogen fuel injection in a small engine. This technical difficulty may derive from the characteristics of hydrogen fuel. The fundamental difference between petroleum and hydrogen fuels is their volumetric energy densities. A stoichiometric combustion of hydrogen requires a mixture of 28 volumetric percent of hydrogen and 72 percent of air, which means that a significant portion of engine displacement is taken by gaseous hydrogen fuel. As the governor reacts to engine load, the governor motion opens the throttle to admit more air to maintain the set speed. Because the overshoot motion of the governor opens the throttle widely leading to a sudden MAP pressure swing to atmospheric pressure. This sudden MAP voltage increase triggers a burst of hydrogen injected into the air intake manifold. The burst of hydrogen leads to a momentary rich mixture that is prone to backfiring. For this reason the TEC³ fuel injection system responds very poorly to the true fuel need in a governor equipped small hydrogen engine.

Little improvement is observed when a throttle position sensor (TPS) is used to offset the effect on the MAP signal by an overreacting governor and hydrogen burst pressure. In fact, the governor linked throttle is just not working well with the hydrogen fuel injection control system. This technical problem is likely more severe with a small engine since the added load can reduce the engine speed every rapidly. Rich hydrogen mixture at low engine speed tends to cause backfiring.

Since the governor causes so many problems to the hydrogen injection, extra effort is made to tune the hydrogen engine by fuel injection control without a governor linked throttle. Essentially, a hydrogen injection control is implemented to a throttle-less engine. This strategy has achieved greater success since hydrogen is capable of extreme lean burning.

Without a throttle that restricts the amount of air drawn into the cylinder the MAP sensor is no longer effective in sensing the engine load. To replace the load sensing role played by the MAP sensor a current sensor is installed at the generator power output circuit. The setup for the current sensor as a load sensor is shown in Figure 31. The current sensor produces a compatible DC voltage signal from 0 to 5 volts in response to the output AC current from 0 to 50 amps. As the power demand increases higher current passes through the detector of the current sensor and produces a signal for higher input to the TEC³ control unit. By connecting the current detector signal to either MAP or TPS sensor harness connectors it effectively controls the fuel injection's pulse-width according to electric load the generator.

Figure 31: Current detector used to replace MAP sensor for engine load input.

Examples of fuel map programmed via TEC3 windows based software is shown in Figure 32 (a,b,c). It demonstrates that the fuel injection pulse-widths can be programmed in table format, where x and y coordinates corresponds respectively to the load inputs and engine rpms. The volumetric efficiency correction shown in Figure 32(a) is originally designed to compensate engine air flow deviation from linear thermodynamic behavior. In the current application, it serves as an effective means for adjusting the amount of fuel injected into the engine to stabilize the rpm around 3600. As it can be seen from the cell data, the fuel is set to reduce as engine rpm

is higher than 3600 and to increase as the rpm drops below 3600.

Figure 32(a): A VE correction table set to stabilize the engine speed.

Figure 32(b) is a staged injector pulse-width table expressed as the percentages of full pulse-width set for the primary injector. The staged injector, set in such manner, provides the majority of fuel to meet the engine load.

Figure 32(b): A staged injector pulse-width table set to provide fuel to meet engine load.

Figure 32(c) is a typical fuel map in full pulse-width in milliseconds for the bench-tested Honda GX350 hydrogen engine when hydrogen inlet pressure is set at 40 psig.

Figure 32(c): A typical fuel map in full pulse-width for the Honda GX350 hydrogen engine.

Since the throttle-less hydrogen engine has no governor for its speed control, the speed of engine must be limited by the TEC³ control unit. By using decreasing injector pulse-width at a speed higher than 3600 rpm, as described in the previous paragraph, the TEC³ unit can set the engine speed limit for fuel cut. In this study, 4000 rpm is set to be the speed limit. With the throttle-less setting and other described fuel injection strategies, the Honda GX350 hydrogen engine operates smoothly and produces up to 3.5 kW (4.7 hp) electrical output. As we pushed for more power by moving into a territory of rich fuel burning, at a lambda ratio of 1.5 or lower, the hydrogen engine is observed to incur back firing.

X. Development of direct fuel injection system for air-cooled engine

The performance of the liquid-cooled Honda GX350 engine equipped with either port or throttle body fuel injection indicates hydrogen delivery system based on aspiration is prone to backfiring. The problem of back-firing is aggravated as the air/fuel Lambda value crosses over a threshold of 1.5 toward the stoichiometric ratio 1. A lean burning hydrogen engine reduces its power by as much as 50%, compared to its gasoline counterpart. This leaves direct fuel injection to be an ultimate approach to boost power. There are two methods to implement direct fuel injection: the first one is to use a so-called spark-injector; another is to use a standalone injector. The spark-

injector can be installed conveniently by replacing the spark plug. No additional modification to the engine cylinder head is needed. Since the spark-injector combines the functions of a fuel injector and a spark plug, the technical design of the spark-injector is very sophisticated and hard to build. With the advantage of technical simplicity and short development time, the approach to use a standalone injector is more practical and viable. Furthermore, a single cylinder air-cooled Honda GX160 engine is chosen for the test of the prototype direct fuel injector. Honda GX160 engine is much smaller compared to the Honda water-cooled twin cylinder engine tested for port and throttle body hydrogen injection with the following specifications: overhead cam, 160 cc displacement, 9:1 compression ratio, air cooled. The overlap of valve timings for intake and exhaust is found to be about 25 crank-degrees. With the success of direct hydrogen injection in a single cylinder engine, the technology may be applied to twin cylinder engines.

i. Direct fuel injection system

The first version of the prototype direct hydrogen fuel injector is shown in Figure 33. It has a threaded end for connecting to the cylinder head. The injector body is made up of aluminum alloy 2024, which is strong and has good machinability. Inside the injector is a Lee automotive check valve. The other end is a tapped NPT female connector, to which a CAP gaseous fuel injector is attached. The distance from the female connector to the CAP injector is kept to minimum, since the pressurized hydrogen trapped in the cavity between the Lee check valve and the CAP injector leaks from the check valve into the exhaust.

Figure 33: Cut-away view of the first version of direct hydrogen injector.

The above described direct fuel injector is tested and found capable of holding the cylinder pressure when hydrogen is introduced from the intake port. This indicates that the Lee check valve is able to stop the back-streaming of the high pressure combustion mixture. However, the Lee check valve is found to malfunction after a few minutes of testing, when hydrogen is introduced to the chamber through the direct injector. By disassembling the Lee check valve and examining its components the cause of failure is attributed to the spring. The spring is tempered after being exposed to the heated combustion mixture.

The second version of the prototype direct hydrogen fuel injector is shown in Figure 34. It consists of a lower body piece and an upper body piece. The lower body has the portion of injector housing which connects to the engine cylinder head. The core of the injector housing is a poppet valve that is normally close. The poppet valve is closed by an expansion spring engaged with the upper portion of the long stem. The upper body of the injector includes a hydrogen inlet port and an electromagnetically actuated push-pin. When the electromagnetic coil is powered by the engine injector controller, an electromotive force is induced in the ferrous steel block connected to the push-pin. This force overcomes the spring tension and opens the poppet valve to allow hydrogen to be directly injected into the combustion chamber during the intake and compression strokes.

Figure 34: Cut-away view of the second version direct hydrogen injector.

This design combines two functions into a single injector. One function is the checking of high cylinder pressure during the power stroke. The other is to allow hydrogen fuel to flow into the cylinder at accurate time and in controlled amount. The poppet valve at the nozzle prevents back-streaming of the high pressure (likely as high as 3000 psig or 20.4 MPa), and high temperature combusted gas. The tension on the stem of the poppet valve exerted by the spring keeps the poppet valve normally closed. When the electromagnetic coil is excited, the poppet valve is pushed open. The amount of hydrogen is determined by the pulse width of injector controller. While the failure of the first version solves the problem by moving the spring's exposure to extreme temperature, the second version solves the problem by moving the spring out from the hot combustion zone to an area where the temperature is much lower due to heat dissipation and additional cooling of hydrogen flow. Another benefit of the improved design is that it prevents hydrogen from being vented into the exhaust during the exhaust stroke. This is accomplished by adjusting the tension of the spring to offset the hydrogen inlet pressure. As long as the spring tension for the poppet valve is higher than the force exerted by inlet pressure, the poppet valve stays closed.

ii. Modification of cylinder head

Honda GX160 engine is an air-cooled single cylinder engine with the following specifications: overhead cam, 160 cc displacement, and 9:1 compression ratio. The overlap of valve timings for intake and exhaust is found to be about 25 crank-degrees, which is much smaller compared to the Honda water-cooled twin cylinder engine tested for port and throttle body hydrogen injection. Valve overlap is a critical parameter for reducing back-firing. Generally, wider valve timing overlap causes more severe problem of backfiring. The occurrence of back-firing in a lean fuel-air mixture is the major hurdle to overcome when improving power output air aspirated hydrogen engines. Direct hydrogen injection introduces fuel after the intake valve is shut, which eliminates back-firing. For this reason, direct inject hydrogen engines have the potential to achieve engine power comparable to the gasoline fueled counterpart.

Figure 35 shows the cylinder head of the Honda GX160 engine. The space between the spark plug and intake valve allows an opening to accommodate the direct hydrogen injector. In the

experiment, a 3/16-20 straight threaded hole is made. The carbon deposit shown on the wall of cylinder head and wall may have been caused by previous runs using carbon based fuel or more likely when lubricant leaked through the piston rings.

Figure 3: Engine cylinder head modified for hydrogen injector.

iii. Installation of engine control sensors

One cannot simply install and use direct hydrogen fuel injector without the precise control of the fuel injection. Hydrogen fuel must be introduced into the cylinder at precise timing and in accurate amounts. More specifically, fuel injection must start right after the intake valve is fully closed, and ends before the ignition spark. If the engine runs at 3600 rpm, the maximum pulse-width for direct fuel injection is about 8 ms.

To realize the required precision of hydrogen fuel injection, synchronized, full sequential injection control is needed. Unlike the phased, sequential engine control, the full sequential injection requires the use of a cam trigger in addition to the crank trigger. The cam trigger tells the control system which stroke the engine is on. With a cam reference for the electronic control

system, the injector fires only once per engine cycle.

With two triggers to be installed on the small Honda GX160 engine, it is difficult to accommodate trigger wheel and sensors. To find a solution, minor modifications to the engine's flywheel and cam cover were made. Figure 36 shows a crank trigger wheel of 60-2 tooth tightly fitted between the engine block and flywheel. The crank trigger magnetic sensor is mounted on a bracket attached to the engine block. In order to adjust the injection timing, arc slots are machined on the trigger wheel, as shown in Figure 37.

Figure 36: Crank trigger for injection control system.

Figure 37: Crank trigger wheel with machined arc slots.

Figure 38 shows the position of the cam trigger sensor fixed on the cam cover. It is an omni-polar Hall Effect sensor, which produces a square wave pulse when triggered by a magnet. The magnet for triggering the cam sensor is mounted on the cam wheel, which is invisible in the figure.

Figure 38: Cam trigger for synchronized full sequential injection.

It is found that second version of direct hydrogen injection adaptor still cannot sustain the combustion heat causing failed in a short order. Failure analysis indicates the intense heat of hydrogen combustion would either temper the spring, or oxidize the poppet valve and seat. Damaged spring and seal result in a faulty check valve. The faulty adaptor allows high pressure hot gas to leak out from the engine combustion chamber. The engine stalled due to lost compression.

iv. The version of direct hydrogen injection adaptor

To improve the functionality of the injection adaptor, a third prototype direct hydrogen fuel adaptor has been designed. Its configuration is illustrated in Figure 39.

Figure 39: An illustrative design of the third version of a direct injection adaptor.

The major difference from the two previous versions is two check valves are included in the newly designed one. The first check valve is placed next to the nozzle, and is made of a ceramic ball and stainless steel seat. The ceramic ball is also confined between the nozzle and the seat. The heat resistive ceramic ball moves toward the seat to shut, and back down the nozzle to open. The main function of the first check valve is to prevent the back-streaming of hot combustion gas during the power stroke.

Since the ceramic ball is made of materials such as alumina and silicon nitride making it lightweight. The new ceramic ball has been shown to shut and open the check valve reliably. Although a spring may enhance the shut-off, no spring can sustain the heat. Without using a spring, the first check valve exhibits some leakage when shut off. Therefore it does not function well by itself. However, it does block the intense heat and reduce the pressure significantly.

The second check valve, an embedded Lee check valve, is located at the center of the adapter's upper body. The Lee check valve used in the adaptor has a Lohm number of 250. For larger engines a lower Lohm number is recommended to achieve higher hydrogen fueling capacity. Since the first ceramic check valve blocks the heat, the Lee check valve maintains a lower temperature during operation. The Lee check valve works in favorable condition of temperature and pressure. Therefore, it can reliably shut the high pressure combustion gas to maintain the

engine's compression.

v. Installation of the third version direct injection adaptor

The above described direct injection adaptor is built, and tested to be leak-free before installation. A threaded adaptor head is then mounted between the intake port and spark plug on the cylinder head of the Honda GX160 generator engine. The adaptor's hydrogen inlet is connected to the outlet of the CAP (made by Clean Air Power) gaseous fuel injector. The adaptor and the CAP injector should be attached to each other as close as possible so the channel cavity volume from the Lee check valve to outlet opening of the CAP injector can be minimized. This cavity volume must be minimized, because the pressurized hydrogen fuel trapped in the end of the injection stroke vents and is wasted during the exhaust stroke. The installation configuration of the adaptor and CAP fuel injector is shown in Figure 40.

Figure 40: Direct injection adaptor and CAP injector installed on GC160 engine.

vi. Strategy for third version direct injection timing control

The Honda GC160 engine equipped with direct hydrogen fuel injection must have separate systems for ignition and fuel injection timing controls. For that reason, the ignition is provided by its original transistorized magneto ignition. The original spark timing is fixed at about 24

crank degrees before the top dead center (TDC). Since the ignition coil is energized by a magnet installed on the crank flywheel, there exists a waste spark during the exhaust stroke. To retard the ignition timing, an arc-slotted bracket is constructed and mounted to displace the ignition coil installed on the engine body. The detail of ignition retardation is shown in Figure 41.

Figure 41: Spark timing retarded by using a bracket with arc slots.

As opposed to the port hydrogen injection which occurs during the intake stroke, the direct hydrogen injection takes place during the compression stroke when both the intake and exhaust valves are closed. Since the air in the combustion chamber is compressed, the hydrogen pressure for direct injection must be higher than the pressure of the compressed air. For the Honda GC160 engine, whose compression ratio is 8.5, the maximum air pressure at top dead center position of compression is 110 psig (0.75 MPa). Therefore the hydrogen pressure must be higher than 110 psig, if the hydrogen is to be injected into the chamber at the later stage of compression stroke.

The direct hydrogen injection timing is controlled by the Electromotive TEC³ unit. A control mode is set at single engine and full sequential. The full sequential control produces only one injection pulse every engine cycle. The correlation between the ignition, cam synchronization trigger, and direct injection timings is a little complex, hence confusing for those not familiar with the process. To facilitate understanding, their relationship is illustrated in Figure 42. The

spark ignition occurs at the end of compression, marked on the cam wheel. The timing for fuel injection must happen in any position during the compression stroke but before ignition. The cam trigger must be installed at a spot between 180 to 6 crank degrees before the start point of injection. Therefore, the best spot for the cam trigger is at the end of the intake stroke.

Figure 42: A schematic illustrating the correlation between cam trigger position, ignition and direct injection timing on camshaft wheel.

vii. The test of third version direct hydrogen injection

Test-run with the third version direct injection adaptor is conducted on the Honda GC160 engine. Hydrogen fuel is injected at the beginning of compression stroke, under a pressure of 200 psig. Engine operation is observed and found to run unstably and prone to stalling by backfiring. The backfiring may be caused by hydrogen in the intake port. Hydrogen in the intake port is likely leakage through the loosely shut intake valve in the early phase of compression stroke. By retarding injection timing by about 30 crank degrees, the engine is observed to run much smoothly, but still stalls when a load is added. It is possible to smooth out the stalls by further retarding injection and ignition timings.

B. Durability Test of Hydrogen Engine

Among the technical approaches to convert small hydrogen ICE engines, direct hydrogen injection is the most effective in boosting power output, but the approach is at its exploratory stage and the cost is much high. Due to its technical complexity, more research work is needed in terms of performance and durability of hydrogen injector. Both the hydrogen fuel injection adaptor and its electronic control system are not reliable, and far from being optimized. It is difficult to carry out the durability test by using the small hydrogen engine equipped with direct fuel injection. In contrast, ACS is very economical and a very simple conversion method. Its operation is very reliable, and its maintenance is minimal. In addition, the in-house resource has significantly cut back after several restructuring at ECD. Great effort has been made on enhancing the small hydrogen ICE engine's performance and the development of solid hydrogen storage canisters, the durability test of small hydrogen ICE engine is delayed. For all the above reasons and considering the limited remaining funding, it is decided to conduct the durability test by small single cylinder hydrogen engine converted by ACS at ECD's test facility. Since the engine size is small, engine heat and exhaust can be readily handled and convenient of carrying the test in-house. Additionally, hydrogen fuel cost incurred during durability test is expected to be substantially lower when compared to the larger, multiple-cylinder engines.

I. Convert Honda GC160 engine with ACS system for durability test

It is a relatively simple task to convert the Honda GC160 engine by using advanced carburetion system (ACS). The following steps describe the conversion:

1. Remove gasoline tank, and disconnect the gasoline fuel line and filter from the carburetor.
2. Discard the choke and its linkage on the engine.
3. Plug the gasoline fuel port on the carburetor.
4. Add a Garretson carburetor adaptor between the air filter and carburetor.
5. Assemble the carburetor, the Garretson carburetor adaptor, and the air filter together, and bolt them to the engine intake port using a gasket between them.

6. Complete the linkage between the throttle and governor leverages.
7. Install a load block between the outlet of a Beam Model two stage dry gas regulator and the Garretson carburetor adaptor for the adjustment of fuel/air ratio.
8. Install the Beam Model regulator vertically to accept a hydrogen inlet pressure between 35 to 250 psig (0.24 to 1.7 MPa.).
9. Connect the hydrogen fuel line to the inlet of the Beam Model regulator.
10. Add Mobile-One™ fully synthetic oil to the engine for lubrication.
11. Start the engine without load.
12. Add load to the generator and adjust the fuel/air ratio for smooth engine operation.
13. Adjust the tension of the governor to set the engine speed at 3600 rpm.
14. Check the electric output to verify the voltage is at 110 Volts.

The Honda EN 2500 generator has a rated steady power output of 2.0 kW electric. The converted hydrogen generator produces peak power of 1.5 kW. A load of 1 kW is added to the generator for conducting durability test.

II. Fail-safe system for hydrogen engine durability test

Generally engine durability test is carried out in a dynamometer laboratory for the development of new engines. Construction of dynamometer laboratory must comply with safety codes for hazardous materials building, to be blast rated, sound/noise rated, fire rated, and adequately ventilated. Computerized engine test equipment is also required for carrying out durability test. The equipment includes dynamometer control hardware, data acquisition sensors and data display software. The dynamometer measures engine output torque and power, and associated sensors are used to measure air flow, fuel flow, engine cylinder pressure, engine speed, and engine emissions.

Since the Honda GC160 is a commercial engine used for back-up power generation, the durability test of the converted hydrogen engine can be carried out by a real world operation of the hydrogen generator for extended hours with an electrical load.

Lab152 at ECD's Rochester Hills facility is constructed as a dynamometer laboratory for the small hydrogen engine. It has a ventilation system built specifically for small engine exhaust. The laboratory is also equipped with a hydrogen gas detector and UV flame detector. These sensors are constantly monitored by a state of arts building alarm system that is directly tied to the fire department of the city of Rochester Hills.

The experiment setup for small hydrogen engine durability test in Lab. 152 is shown in a P&I diagram marked as Figure 43.

Figure 43: Instrument and pipe diagram of hydrogen ICE durability test.

The test system is comprised of four subsystems: i) hydrogen fuel, ii) hydrogen ICE engine and exhaust ventilation, iii) electrical generator and load and iv) sensors and feed-back protection.

Hydrogen in compressed gas cylinder (maximum pressure at 2400 psig or 16.3 MPa) in size Ø9 x 60 is supplied by Air Gas Inc. Industrial purity is adequate for the ICE engine. Hydrogen usage for the durability test is estimated about 1 kg for every 8 hours. Two tanks of compressed hydrogen are consumed in a single work day.

As seen in Figure 43, hydrogen from the high pressure cylinder passes through a regulator to reduce its pressure to ~ 50 psig (0.34 MPa). Down from the regulator, the low pressure hydrogen flows through a quarter inch diameter stainless steel tube in a length of about 10 feet, and reaches to a pneumatic valve. After the valve, the hydrogen line is then connected with a Beam Model T60 two-stage regulator. The hydrogen outlet of the two-stage regulator is attached to a load block for fuel/air ratio adjustment, before being connected to a gaseous fuel carburetor adaptor. Hydrogen pulses into the engine intake manifold by Venturi vacuum pressure during the intake stroke.

The engine exhaust is fed into a ventilation pipe, mixed with ambient air, and drawn out the building by a blower at the end of the vent. A temperature sensor is mounted at the entrance of the ventilation pipe. If either the vent blower stops, or is not turned on, the temperature at the entrance will increase rapidly by the engine exhaust. The temperature controller will detect a higher temperature than normal (> 80 °F or 26.7 °C) and shut down the engine.

The load of the hydrogen generator EN2500 is provided by plugging in an array of light bulbs. A constant load of 1 kW is used for the durability test.

Hydrogen is a highly flammable gas, and its leakage into a confined space can cause a fire and/or explosion. For that reason, fail-safe measures must be implemented for an extended period of durability test. Four fail-safe (or feedback) loops were added to the test system. Under any of the following circumstances, these safety feedback loops will trigger the pneumatic valve on the hydrogen line to close: i) Engine failure or faulty generator, ii) Building fire alarm system is triggered to level one (warning), iii) Exhaust ventilation malfunction and a blazing flame and iv) Hydrogen leakage.

As it is illustrated in Figure 43, a solenoid valve controls air pressure to activate the pneumatic valve in the hydrogen line. It is powered up by the electricity from the engine generator. The solenoid's power circuit includes three relay switches in series. These relays are tied to hydrogen detector, temperature sensor and building fire-alarm. The building fire alarm system has a second layer of hydrogen detection, besides flame detection.

The durability test system for the hydrogen engine is shown in Figure 44. A cage is made to cover the engine mainly for hydrogen detection. Any hydrogen leakage from the engine body or hydrogen delivery components arises to the top of the cage and accumulates under the sealed aluminum roof. A hydrogen detector is mounted right under the ridge beam of the cage. The hydrogen detector is preset at 8000 ppm (about 20% of LEL) to trip. The relay switch connected to the hydrogen detector will cut off power to the solenoid at a preset hydrogen level. Loss of power to the solenoid turns off the hydrogen fuel line and the hydrogen engine stops.

Figure 44: Front view of the durability test system for hydrogen ICE generator.

The durability test is accomplished by simply accumulating engine run-time at constant load condition. The proposed target is 1000 hours. The actual run-time of the hydrogen engine may depend on engine condition and other critical components. After vigorous reviews and hazop analysis of the test system, ECD's safety staffs approve the safety features and recommend the durability test be conducted during work hours.

The goals of small hydrogen ICE engine durability test are to investigate the impact of hydrogen combustion on engine components, such as piston, piston rings, aluminum cylinder wall, engine valves, lubricant and dry gas regulator for hydrogen fuel delivery and metering. During the test

period the conditions of the engine cylinder wall, piston, spark plug and other engine parts is designed to be periodically examined. Inductively coupled plasma (ICP) technique is used to analyze the engine oil to study the worn off of the aluminum cylinder wall.

III. Procedures of operation

The routine performance for durability test follows the following steps:

1. Check the oil level of the engine. If it is low, add oil. If it is dirty, change it.
2. Check for any loosen engine part. If so, fix it before running the engine.
3. Turn on compressed air, and check air pressure to be 60 psig.
4. Make sure the load and solenoid plugs are firmly attached to the generator.
5. Open the hydrogen tank and leak-detect the entire hydrogen line.
6. Push the green button switch to turn on the exhaust ventilation fan.
7. Open the valve of the hydrogen canister.
8. Push a spring loaded primer on the Beam Model regulator to release some fuel into the engine intake manifold.
9. Pull the recoil to start the engine.
10. Log the following data: mark up the check list, load on/off, start-up time, bottle pressure, shut off time, hours run, and accumulated (meter) hours, bottle pressure at shut off time, oil check/oil sample/oil change, hydrogen tank change, ventilation fan off.
11. Stop the engine by either turning off the engine ignition switch or shut the hydrogen tank valve.
12. Log the following data: shut off time, hours run, accumulated (meter) hours, bottle pressure at shut off time, oil check/oil sample/oil change, hydrogen tank change, ventilation fan off.
13. Close the hydrogen tank and compressed air.
14. Allow a few minutes to cool before switching off the ventilation fan.
15. If the hydrogen tank is emptied, change the tank and leak-detected the connection.

IV. Engine components examination

The converted Honda Economy EN 2500 generator ICE engine operated under a constant electrical load of 1 kW, which is about 70% of the maximum output when it is fueled by hydrogen. In general, the durability test of the hydrogen engine progresses smoothly as a daily routine without any major glitches. In addition to routine hydrogen gas cylinder replacement and scheduled lubricant changes - regular engine oil changes every 50 hours, the wear conditions of engine components and especially the valve recession, boroscopic examination is conducted approximately every 200 hours.

i. Boroscopic examination after 200 hours runtime

According to the operation manual, it is recommended that the spark plug be replaced in every 200 hours of run time. At a runtime of 203 hours, the initial spark plug (model NGK BPR6ES) is unplugged from the cylinder head, leaving an opening into the combustion chamber. A boroscopy is employed to scrutinize the engine wear. Three major engine parts are particularly examined: intake and exhaust valves and their seats; cylinder wall, and piston head.

Figure 45 shows the appearance of the engine intake valve and its seat after 200 hours of runtime. The reddish coating on the valve and the vicinity of its seat area is mainly iron oxide (rust) caused by high temperature steam and excess oxygen in lean burning combustion. Based on the appearance of well-defined edges, and their cleanness, the valve and its seat are still in a very good condition. A similar look of the engine exhaust valve after a runtime of 200 hours is shown in Figure 46. The reddish iron oxide coating may be somewhat thicker comparing to that in the area of intake valve, due to higher temperature in the port of exhaust. The condition is still in good shape. However, some degree of recession for both intake and exhaust valves are sustained.

Figure 45: A photograph of the engine intake valve after a runtime of 200 hours.

Figure 46: A photograph of the engine exhaust valve after a runtime of 200 hours.

A typical appearance of the engine cylinder wall after a runtime of 200 hours is shown in Figure 47. It shows only a few, rare vertical lines (from the bottom to the top) in the direction of longitudinal axis of the cylinder. These are probably scratches caused by hard particles trapped

between the piston rings and cylinder wall. The crosshatches (from the left to the right) on the cylinder wall are the original polishing or brushing marks left by cylinder honing. They are intentionally left to retain oil for lubrication between the piston rings and cylinder wall. Deep vertical scratch line is rarely observed on the cylinder wall, which indicates that wear is minimal.

Figure 47: A photograph of the engine aluminum cylinder wall after a runtime of 200 hours.

Figure 48 shows the rough surface of the aluminum alloy piston top. This surface feature is typical of a cast metal. Beyond the surface's toughness, it seems that oxidation layer or coating deposit is formed on the piston head. More detail about the nature of the surface deposit will be examined when the engine is overhauled at the end of the durability test.

Figure 48: A typical texture of the piston head after a runtime of 200 hours.

In addition to the above described boroscopic features of the major engine parts, Figure 49 shows a photograph of the spark plug replaced at a runtime of 200 hours. The overall condition of the spark plug is very good. The same type of reddish oxide coating as that on the engine valves occurs to the electrode area exposed to combustion chamber. No substantial electrochemical corrosion is observed on both electrode areas.

Figure 49: A photograph of the replaced spark plug after a runtime of 200 hours.

Based on the above results, the overall hydrogen engine condition is very good after a runtime of 200 hours. It showed a minimal amount of wear on major components.

ii. Boroscopic examination after 703 hours runtime

Figure 50 shows the appearance of the engine intake valve and its seat after 703 hours of runtime. Compared to their conditions after 200 hours of runtime, they showed more severe wear and deeper indentation.

Figure 50: A photograph of the engine intake valve after a runtime of 703 hours.

A similar view of the engine exhaust valve after a runtime of 703 hours is shown in Figure 51. Compared to the condition of the intake valve, the seat indentation of the exhaust valve seems much deeper due to harsh conditions caused by engine exhaust. The exhaust valve also shows more severe wears. The images of both intake and exhaust valves illustrate a dry condition surrounding the valve/seal interfaces, due to the lack of lubricant.

Figure 51: A photograph of the engine exhaust valve after a runtime of 703 hours.

A photograph of the engine cylinder wall after a runtime of 703 hours is shown in Figure 52. It shows much more vertical lines (from the bottom to the top) caused by piston displacement. Some of the crosshatches (from the left to the right) on the cylinder wall are still visible; indicating the wear to the cylinder wall may be not severe. Since vertical scratches increase, hydrogen blow-by through those lines may also increase, which may reduce the engine's fuel economy.

Figure 52: A photograph of the engine's aluminum cylinder wall after a runtime of 703 hours.

Figure 53 shows the morphology of the aluminum alloy piston top. It appears that an oxidation

layer or coating of different nature formed on the piston head grows coarser as the engine runtime increases. Its detailed structure of this top layer is addressed at the end of the durability test when the engine is taken apart.

Figure 53: A typical texture of the piston head after a runtime of 200 hours.

After 703 hours service runtime, the boroscopic examination of the engine combustion chamber shows that engine valves and seats sustained substantial wear and erosion. Furthermore, the engine's output power seems to reduce, due to intake valve recession and increased flow resistance.

iii. Examination of engine components after 785 hours runtime

Durability test conducted on a small single cylinder hydrogen engine is stopped when a significant power loss in engine performance is recorded due to the engine stalling out. Total cumulative runtime of the hydrogen engine for the entire durability test is 785 hours. Although the engine durability did not reach to the originally planned 1000 hours, it is still a plausible accomplishment by considering the fact that Honda GC 160 engine is a very economic engine for backup generator.

As described in the previous sessions, valve recession is a cumulative wear during the durability test. Excessive wear on both the valve and the seat decreases or eliminates the clearance

between the valve stem and the cam rock arm. The diminished valve clearance leads to decompression, power loss, backfiring, lower fuel economy, and leakage through the valve stems into the cam case. Therefore, valve clearance must be repeatedly adjusted as the engine loses compression. The interval of valve clearance adjustment is typically at 200 hour runtime.

Over the course of durability test, the valve clearance is adjusted three times, at 350, 500, and 678 hour runtime respectively. As the valve recession accumulates, the valve poppet is increasingly sunk into the valve seat, which narrows the pathway for the intake air and fuel mixture. A narrower intake pathway increases the resistance of gas flow, which reduces the volumetric efficiency of the hydrogen engine. Our examination indicates that the hydrogen engine started to experience gradual power loss at a runtime of 500 hours. To regain the power output, adjustment has to be made to enrich the fuel/air mixture.

Figure 54 shows the appearance of the engine intake and exhaust valves after a runtime of 785 hours. The sealing surfaces of both valves sustained severe wear. The valve is made of stainless steel which is relatively soft. The impact area of the valve is abraded and the valve poppet becomes a sharp edge. Both the intake and exhaust valves suffer comparable amounts of wear. This indicates that the main cause of failure is due to the mechanical wear between valve and valve seat, instead of temperature erosion.

Figure 54: A photograph of the engine intake and exhaust valve after a runtime of 785 hours.

Figure 55 shows the appearance of the engine's exhaust and intake valve seats, the contact area of both seats are indicated by arrows. The valve seats appear to suffer deep indentation from the mechanical wear of the valve body, after the 785 hours of operation.

Figure 55: The engine exhaust and intake valve seats.

Figure 56 is a photograph showing the cam wheel with the cam lobe at the center. The cam wheel and cam lobe are both made of plastic as a single piece. The markers on the picture point to the original lobe height and worn-off area respectively. As it can be seen, the lobe is worn off by as much as 1/16 inches in the area contacting with the rocker arm. This large amount of reduction in cam lobe height translates to a decreasing opening of both valves. The restricted openings of intake valve severely decrease flow rate, hence reduces the volumetric efficiency of the intake stroke. This results in significant power loss of the hydrogen engine.

Figure 56: A photograph of cam wheel with its lobe marked to indicate wear after a runtime of 785 hours.

Figure 57 shows the piston of the engine that has a runtime of 785 hours. The piston is still in good condition. No excessive wear is found on the piston rings. There is a thick layer of deposit on the piston head, which may result from the combustion of engine lubricant that is either seeped through from the piston rings or escaped from the crankcase with the blown-by hydrogen/air mixture.

Figure 57: Appearance of the piston after a runtime of 785 hours.

Figure 58 shows the appearance of the cylinder wall of the hydrogen engine at the end of durability test. The inspection the cylinder wall does not reveal excessive wear. The wall surface does not show deep vertical scratches, which indicates the engine is well lubricated during the course of durability test.

Figure 58: A photo showing the cylinder wall of the hydrogen engine with 785 hour runtime.

In conclusion, noticeable wear occurred on the engine valves and valve seats is determined to be the major factor for engine failure instead of temperature erosion.

V. Engine oil Analysis for hydrogen engine

Selection of proper engine oil is very crucial for extending engine service life and improving fuel economy. The main function of engine oil is lubrication of moving parts. It cleans these parts, and inhibits corrosion. It acts a coolant to move the heat away from the moving parts such as crankshaft bearings, connecting rods, cam-shafts and piston rings. Engine oil also enhances fuel economy by forming seal between cylinder wall and piston rings to reduce blow-by, and to sustain compression.

There are two major types of motor oils: one is conventional motor oil; another is fully synthetic motor oil. The conventional one is made up of mineral oil and additives. Mineral oil is refined from crude oil pumped from ground as petroleum. The base of the conventional motor oil is a mixture of complex hydrocarbons such as polyalphaolefins (PAO), polyinternal olefins (PIO), Esters, and other inorganic compounds in low contents. The base stock of synthetic oil is made up of artificially synthetic compounds, such as polyalphaolefins and synthetic esters. Synthetic oil has therefore more consistent molecular composition than mineral oil, while mineral oil is a mixture of organic compounds of various molecules sizes, also includes wax and other impurity in small quantity. Added to the synthetic base oil are various additives to enhance oil performance, such as wear reduction, cleaning, anti-foaming, corrosion and oxidation inhibition, and viscosity stabilization.

For the durability test of the small hydrogen engine, 10W-30 Mobil One fully synthetic motor oil is selected, since synthetic oil is more stable in higher temperature that is typical of air cooled engine. Extensive tests of automobile engines indicate that synthetic oil has longer service life than mineral based motor oil. In spite of its relatively higher cost, synthetic oil performs exceptionally better in reducing engine wear, oil burning, oil volatility, and exhibits chemical stability in the environment of higher engine temperature.

The first oil change is performed after a runtime of 5 hours as a typical new engine incurs excessive wear and contaminants like dusts and metallic debris left in engine assembly. The second oil change is done at a service time of 30 hours. After that, for the entire course of durability test, engine oil is routinely changed at 50 hour service time. Additional oil changes take place at 70 and 80 hours service time. An outside oil analysis service company (Polaris Laboratories, headquartered in Indianapolis) performs the tests for all oil samples including the baseline Mobil one 10W-30 fully synthetic oil.

Engine oil analysis and its implication to engine operation can be challenging, due to the complexity of motor oil, and various functionalities of modifiers and additives. Established engine oil analysis involves tests of chemical and physical properties of base oil. Because motor oil is in constant contact with engine moving parts, small or microscopic sized wear particles and contaminants accumulates as suspension in oil. It is crucial that oil analysis characterizes the

suspension and contaminants. Oil tests on a regular basis monitors engine operation, and helps identify abnormal wear of engine parts. A typical oil test looks into the following chemical and physical properties:

1. Basestock viscosity
2. Wear metal, additive, or contaminants concentration
3. Water content
4. Particle sizes distribution
5. Total base and acid number

Figure 59 represents a typical report of engine oil analysis performed by Polaris Laboratories. It is an analysis of the engine oil after 50 hours service time including concentrations of 20 metals, viscosity, water content, total acid number, and particle distribution.

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Figure 59: An analysis report for engine oil with a service time of 50 hours.

The oil analysis is performed on a series of samples taken from the changed oil at various engine runtimes, namely at 66, 274, 376, 477, 582, 682, and 785 hours. The plot of the worn-off metal concentration versus engine runtimes is shown in Figure 60. It shows that the engine wear in the end of the durability is severe. The engine oil at the end of the engine life (785 hours) also exhibits higher viscosity than that of new oil and extremely high particle counts (ISO Code 27/27/27) compared to new oil's 21/19/15. The reason for the accelerated engine oil deterioration may relate to the relative high temperature caused by rich fuel/air ratio, and to more oil escaped into combustion chamber due to higher vacuum created during the intake stroke. The higher degree of vacuum may be caused by reduced flow rate of air/fuel mixture, since the intake valve opening is much smaller due to the severe wear of the cam lobe and valve recession. It is difficult to stipulate the cause behind the smaller bump at 376 hours on Figure 60.

Figure 60: Wear metal concentrations plotted against oil service time.

The major reaction gas of hydrogen combustion is steam. Therefore, it is likely that the steam condenses and dissolves in oil. Water content in oil greatly deteriorates the performance of motor oil. However, the oil analysis indicates that the water content in all oil samples is of an insignificant amount, less than 0.1%. One explanation for the low water content may have something to do with the high engine temperature that is typical of air cooled engine.

Viscosity is a very important parameter of the base-stock of engine oil. The technical implication of viscosity of oil is a measure of oil's resistance to shear, or a measure of resistance to flow. Higher viscosity means higher resistance to flow (thick or heavy oil), while low viscosity indicates low resistance to flow (thin or light oil). The reported viscosity by Polaris is kinematic viscosity in temperature 100 degrees Celsius. Kinematic viscosity is determined from absolute (dynamic) viscosity divided by the density of oil. The absolute (dynamic) viscosity of oil can be determined by directly measuring shear stress and shear rate. The unit of kinematic viscosity of oil is centistokes (cSt or mm^2/s). Dynamic viscosity is measured by viscometer. The unit for dynamic viscosity is mPa s or centipoise (cP).

To rate engine oil, SAE (Society of Automotive Engineers) establishes eleven viscosity grades for winter grade oils and non-winter grade oils: 0W, 5W, 10W, 15W, 20W, 25W, and 20, 30, 40, 50, 60. The number with a W stands for the pour point of oil expressed Celsius, whereas the number without a W is rating according to the kinematic viscosity measured at 100 °C. For multi-grade engine oil, two-number label, for example 10W-30 is used. The first number stands for the pour point rating in cold temperature and second number stands for viscosity rating in high engine temperature. Multi grade engine oil contains additives in base-stock, viscosity modifier, to meet the viscosity range required by both cold and hot engine conditions.

The oil analysis indicates that the viscosity of the engine oil is slightly reduced for all the samples except the one used in the end of the durability test. However, the reduction is not significant. For example: the viscosities of Mobil One fully synthetic oils having runtimes of 0, 50, 70, and 80 hours are: 10.8, 10.2, 10.2, and 10.1 cSt respectively. For reference, water at 20 °C is 1.0038 cSt. This demonstrated that the synthetic engine oil shows superior performance by retaining a stable viscosity during long service life.

Acid number reported by oil analysis is a measure of the amount of alkaline metal hydroxide added into the lubricant. Since acids are formed as oil oxides and ages, total acid number is indicative of age of oil. It can be used to determine when oil must be changed.

The analysis shows the baseline Mobile One has a total acidity number at 2.59. As oil service

times increases, the total acid number seems to increase but not conclusive. For example: the acid number for oil samples with runtimes of 50 and 70 hours are 3.07, and 4.70. However, the oil sample with 80 hour runtime shows a much lower total acid number, at 2.69. This inconsistency may be resulted from test error.

As oil service time increases, the size and concentration of wear or contamination particles in suspension become greater. Particle count test measures particle concentration per milliliter at a given size (microns), and its results are reported as an ISO Cleanliness Code along with particle distribution. The ISO 4406 standard for oil cleanliness uses a three number system, for example, 12/15/10. The first number represents the concentration of particles greater than 4 microns; the second number those greater than 6 microns; and third those greater than 14 microns. By using the ISO particle size range code system, the oil's ISO Cleanliness code can be determined. The higher the number in each category is, the higher the concentration of wear particles.

The ISO Cleanliness Codes for oil samples with runtimes of 0, 50, 70, and 80 hours are 20/17/14, 18/17/14, 18/17/15, and 20/19/16 respectively. It indicates that the concentration of particles greater than 6 and 14 microns is much higher in oil with longer service time. As larger particles cause deep cut, and fast wear, this is consistent with the results of wear metal concentration described earlier.

C. Solid Hydrogen Storage System Development

Hydrogen storage is very critical component for hydrogen utilization. Solid hydrogen storage is the preferred method over compressed high pressure and liquid hydrogen for various applications including motive, stationary and portable power. Solid hydrogen storage systems offer several advantages over both compressed and liquid hydrogen storage systems. Some of the advantages of solid hydrogen storage systems include:

- Safety
- Low pressure operation
- Compactness

- No cryogenic temperatures or “boil-off”
- Scalability to virtually any size
- Tailorability of delivery pressure
- Utilization of waste heat
- High volumetric energy storage density
- Long cycle life

ECD has developed proprietary metal hydride hydrogen storage technology that is compact and operates at low pressure. For this project, an innovative approach of modular solid hydrogen storage system configuration is proposed. The major advantages of modular structure are its low cost in the development phase, flexibility of system sizing, and marketability.

I. Conceptual designs of hydrogen storage system

Two conceptual designs of solid hydrogen storage system for stationary power generation are proposed, particularly targeting applications of output power at about ≤ 5 kW. As illustrated in Figures 61 and 62, the solid hydrogen storage systems are comprised of four major components: i) hydrogen storage materials, ii) pressure vessel, iii) hydrogen gas manifold and iv) containment of heat exchange fluid. For the current design, no active internal heat exchange elements are incorporated inside the individual vessels.

Figure 61: Exploded view of a hydrogen storage system using aluminum canisters.

Figure 62: Exploded view of a hydrogen storage system using stainless steel tubes.

II. Hydrogen storage materials

The hydrogen storage system uses a variety of reversible metal hydrides. These hydrogen storage materials typically absorb hydrogen atoms into the interstitial sites of a crystal lattice at room temperature and under low pressure, typically less than 500 psig (34 MPa). As hydrogen is absorbed, heat will be released from the medium as a result of a reaction between hydrogen atoms and the metallic alloy. At the end of absorption, the metallic alloy is fully transformed to a new phase of hydride. During desorption, hydrogen atoms transport out of the storage medium by diffusion. Hydrogen gas forms at the surfaces of storage medium particles, and escapes as outlet pressure is below desorption pressure. At the same time, heat is needed to release hydrogen from the hydrogen storage medium. The chemical reaction for reversible hydrogen storage materials can be expressed as:

The kinetics of such hydriding reaction is determined by how fast the exothermic heat is removed from the storage medium. Therefore, a metal hydride hydrogen storage system must incorporate an efficient heat exchanging system.

Many metal alloys show the above-described attributes of reversible hydrogen storage including TiMn₂ based alloys, commonly called AB₂ type, TiFe based, AB type, La(Ce, Misch metal)Ni₅, AB₅, etc. ECD has engaged in the R&D of hydrogen storage materials for more than thirty years and developed a family of AB₂ type low temperature metal hydride (LTMH) alloys. They demonstrate to have high hydrogen capacity and excellent cycle stability up to thousands of absorption/desorption cycles.

One of the favorable features of LTMH alloys is the tailorability of the equilibrium plateau pressure. Figure 63 shows an example of the pressure-temperature relationship of three Ovonic LTMH alloys. These alloys have a heat of formation in the range of 24-30 kJ/mole H₂. The criteria that are used to select an appropriate alloy for a particular application are i) type of waste heat available, ii) required delivery pressure, and iii) allowed maximum charging pressure.

Figure 63: Pressure-temperature relationship of three LTMH alloys.

In general, the canisters designed for the uninterruptable power system (UPS) application contain only a passive heat exchange system. Canisters are required to deliver at least 10 psig pressure at 5°C and the maximum pressure must be less than 250 psig at 30°C. Among three alloys shown above, OV694 is ruled out because it cannot provide the required deliverable pressure and flow rate at 5°C due to its low desorption plateau pressure. OV610 has a relatively high plateau

pressure that is required a charging pressure higher than 400 psig at 25°C. OV679 alloy can meet all the requirements for the portable power canister application.

III. Portable canisters

ECD has developed a family of portable canisters for various stationary applications with the hydrogen capacity ranging from 7 to 80 g. Figure 64 shows a schematic drawing of an Ovonic model 85G250B canister. A fully charged 85G250B canister provides more than eleven hours (663 minutes) of fuel cell run time at a stack power of 140W.

- Size: 3.5" diameter x 16.6" length
- Reversible capacity: 74-80 g depending on air flow rate/temperature
- Weight: 6.5 kg (14.5 lbs)
- Staubli quick-connect interface

Figure 64: Ovonic portable power canister model 85G250B.

The canister design shown in figure 64 and other canister designs have been tested both internally and by independent outside agencies and shown to be in compliance with applicable codes and standards, resulting in the first US DOT competent authority of this kind for transport of metal hydrides in portable canisters. Tests performed at ECD laboratories demonstrate that canisters remain safe after absorption/desorption cycling in excess of 500 cycles. Figure 65 compares strain data for respective canisters at 5 and 503 cycles. Charging conditions included 250 psig H₂ pressure in a 25°C liquid bath for 3 hours while hoop strain was measured on the canister surface in the maximum absorbed condition. Following absorption, the canisters were desorbed at 50°C for a minimum of one hour. The data shows that the highest strain values

reached at the end of absorption are 1300 $\mu\epsilon$ and 1800 $\mu\epsilon$ for the canisters at 5 and 503 cycles, respectively. The maximum value measured for canister cycled more than 500 cycles is below the maximum allowable (1854 $\mu\epsilon$) according to DOT limits of for a pressure vessel with a minimum wall thickness of 0.145 inches constructed of 6061-T6 aluminum alloy. Furthermore, the test result from a DOT recognized independent agency confirmed that there was no change of burst pressure (>4500 psig) indicating no reduction of canister strength.

Figure 65: Microstrain data during absorptions for canisters on cycles #5 and #503.

Figure 66 shows the hydrogen refilling profile of the 85G250B canister by 250 psig hydrogen pressure and i) 25 °C water bath , ii) room temperature moving air and iii) 20 °C quiescent air conditions. At 250 psig, the canister can be refilled to up to 90% of its capacity in about 75 minutes in 25 °C water bath. By lowering the temperature of a water bath, or by increasing the hydrogen refilling pressure, the refilling duration can be shortened. More than 6 hours are needed to refill the canister under the moving air condition.

Figure 66: Hydrogen refilling profile for the 85G250B canister.

IV. A prototype Ovonic bundled canister storage system

A prototype bundled canister storage system consists of sixteen Ovonic 85G250B canisters equipped with a stainless steel thermal jacket. A manifold of eight canisters having two four-canister arrays inserted into the thermal jacket is shown in Fig. 67(a). Each canister is incorporated with a TPRD safety device on the valve assembly (yellow arrows in Fig. 67(a)). A second set of the canister manifold is inserted at the opposite side of the thermal jacket. Figure 67(b) illustrates the ECD engineer to perform the leak check around the canister manifold prior to seal the system with the end plates. The canister manifold is designed to hold the helium pressure up to 900 psig.

Figure 67: (a) A manifold of eight canisters inserted into the thermal jacket and (b) the ECD engineer performs the helium leak check around the canister manifold.

The assembled buddle canister storage system is then ready for testing. Figure 68 shows the storage system connected to the test stand in Lab 151.

Figure 68: The bundled canister solid hydrogen storage system in Lab 151.

V. ECD hydrogen testing facility and capability

ECD hydrogen facility includes a specially designed hydrogen-rated section that houses several testing laboratories equipped with a centralized hydrogen delivery system, UV flame and hydrogen sensors, alarm systems and remotely controlled in operation. A hydrogen tube trailer managed by Praxair serves as a central hydrogen delivery system to supply the hydrogen through the hydrogen pipeline to each individual lab with the pressure up to 1500 psig. In-house cooling and heating systems provide the water/glycol (50/50) mixture operated in the temperature range from 5 to 80 °C. Lab 151 shown in Fig. 69 is used to conduct the performance test on the storage system. A high purity 2200 psig helium cylinder installed in the test stand system is used to leak check the storage vessel system prior to the hydrogen absorption/desorption experiments. Figure 70 is a PI diagram of the test stand in Lab 151 including three flow meters and controller covering the ranges of hydrogen flow rate of 100 and 1000 slpm and 1000 gpm, respectively and a hydrogen delivery pressure up to 1200 psig.

Figure 69: The test stand system in Lab 151.

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Figure 70: PI diagram of Lab 151.

VI. Performance test of a prototype Ovonic bundled canister system

i. Average hydrogen flow rate derived from the T-size cylinder

The durability test conducted on a small single cylinder hydrogen engine (Honda GC160 engine) is using the commercial T-size hydrogen cylinders. One hundred sixty-nine T-size cylinders are consumed during the durability test. The total recorded runtime of the hydrogen engine for the entire durability test is 785 hours. Ninety-eight hydrogen cylinders are able to continuous run the generator from full to empty to deliver a constant 1kW load. Runtime of the T-size cylinders varied from 250 to 300 min. as summarized in Fig. 71. A typical T-size cylinder at the pressure of 2200 psig contains 617g - 625g of hydrogen. The red line in the graph represents an average runtime of 267 min. per T-cylinder. An average hydrogen flow rate is 2.31 grams per min. (25.9 slpm) which is required to run the engine in order to power a constant 1kW load.

Figure 71: Runtime of T-size hydrogen cylinders.

ii. Results of a prototype Ovonic bundled canister hydrogen storage system

A prototype Ovonic bundled canister hydrogen storage system consists of sixteen 85G250B canisters with a nominal hydrogen capacity of ~1.2 kg (13,500 std. liter). Figure 72 displays the first desorption profile of the system after the activation in Lab 151. The absorption parameters prior to the desorption are 500 psig H_2 pressure and the coolant temperature set at 6 °C for overnight charging. For the desorption, the following parameters are used, a constant H_2 flow rate of 135 slpm (blue line in Fig. 6) and the coolant temperature (inlet: orange line and outlet: red line) set at 50 °C for the beginning 10 min. then by increasing to 60 °C for the next 60 min. followed by further increasing to 80°C for the remaining of desorption. The H_2 desorption capacity is 1236 g in agreement with the nominal system capacity resulting from a fully activated storage system. The storage system is able to deliver a constant flow rate of 135 slpm for 93 min. Overall, 1121 g or 91% of the total capacity manages to supply the required flow rate.

Figure 72: Desorption data (first desorption) of the bundled canister system.

In general, a solid hydrogen storage vessel having H₂ capacity in the range of 0.5 – 1.0 kg works well with the incorporation of a build-in heat exchanger unit to handle the thermal management requirement. Typically, this kind of vessel enjoys better volumetric storage density and attractive absorption/desorption kinetics than the ones without an internal heat exchanger; i.e. such as in the case of the canister. However, the drawback is the vessel is too rigid in shape and size and is very difficult to change in design. On the other hand, the advantage of the bundled canister system can tailor the design to fit into any irregular space and still retain a reasonable good volumetric storage density than the compressed H₂ storage system counterpart. Previous experience indicates charging pressure, temperature and time are three major variables to determine the H₂ capacity and the charging kinetics of any hydrogen storage system. In this investigation, four experiments were conducted under the charging pressures of 500 psig and 750 psig, the coolant temperature of 15°C and 25°C and a constant charging time of 60 min. The data are summarized in Fig. 73. The result indicates the higher charging pressure and/or the lower charging temperature correlate to the better charging kinetics and the higher H₂ capacity. The charging kinetics and the H₂ capacity follow the descending sequence of 750 psig/15 °C, 750 psig/25 °C, 500 psig/15 °C and 500 psig/25 °C. The data derived from the 500 psig/25 °C experiment is the only one having the H₂ capacity below the nominal 1.2 kg. The charging

parameter of 500 psig/15 °C for 60 min. is chosen as the standard charging condition in the remaining study.

Figure 73: Charging kinetics conducted from the bundled canister system.

Figure 74 represents a set of absorption/desorption experiment under the following conditions: absorption - 500 psig, 15 °C and 60 min. and desorption – a constant flow rate of 25.8 slpm with the coolant temperature gradually increased to 40 °C during the first 20 min in desorption. The H₂ capacity was 1231 g in which 97% (1199 g) of the capacity could supply 25.8 slpm for near 9 hrs (522 min) that is more than double the runtime from a T-size cylinder.

Figure 74: A set of abs./des. experiment conducted from the bundled canister system.

Runtimes recorded from various desorption flow rates at 40°C are shown in Fig. 75. The higher H₂ desorption flow rate required for delivering the higher engine output power correlates to the shorter runtime; from 522 min. at 26 slpm to 254 min. at 52 slpm to 69 min. at 135 slpm. The solid blue line represents the power curve fit to the data points to guide reader's eyes.

Figure 75: Runtime vs. desorption flow rate at 40 °C from the bundled canister system.

Reproducibility of the bundle canister system as well as the reliability of the test stand unit in Lab 151 are demonstrated in Fig. 76. Two consecutive absorption/desorption experiments are carried out using the same absorption parameter as the one shown in Fig. 74 followed by the desorption using a constant flow rate of 26 slpm and coolant temperature of 20 °C. The entire experiment lasts for more than 21 hrs. After the completion of 500 psig charging, the pressure quickly falls as soon as the hydrogen begins to release from the bundled canister storage system and reaches to 240 psig, near the equilibrium plateau pressure of the alloy at 20 °C, within 30 min. Both H₂ capacity (1196 g vs. 1198 g) and the runtime required to deliver a constant flow rate of 26 slpm (both at 504 min.) are essentially identical between the two experiments.

Table 1 summarizes the runtimes recorded from the experiments taken from the bundled canister system at four different desorption flow rates and four coolant temperatures. The corresponding data are also plotted in Fig. 77.

Figure 76: Two consecutive abs./des. experiments to demonstrate the reproducibility of the bundled canister system.

In general, the runtime increases linearly with the increase in the coolant temperature for a given constant flow rate. In the case of 26 slpm, the runtimes are within 6% difference between 6 °C and 40 °C. There is only ~11% difference in runtimes between 6 °C and 40 °C for the 52 slpm flow rate condition. On the other hand, for 135 slpm, the runtime at 40 °C is near five times longer than the one at 6 °C; 80 °C can further increase the runtime for more than 6.6 times longer than the one at 6 °C. Desorption at the higher flow rate would benefit the most for the higher available temperature.

Table 1: Runtime vs. Temperature at various flow rates

	26	52	90	135
T(°C)	slpm	slpm	slpm	slpm
6	489	227		14
20	504	242		43
40	522	254	140	69
80				93

Figure 77: Runtime vs. coolant temperature at various desorption rates.

D. Conclusions

Based on the consideration that a large portion of small ICE engines are used for back-up power generation, a Briggs & Stratton 480 cc V-twin overhead valve engine is the first one selected for the development of hydrogen conversion. The baseline gasoline performance and emission tests indicate the Briggs & Stratton V-twin engine delivers an actual output power of 13.5 hp (10.2 kW) at 3,600 rpm. Without using a catalytic converter, the gasoline operation exhibits high concentrations of CO, unburned HC and NO_x emissions. A Briggs & Stratton V-twin engine is successfully converted to run on hydrogen using an ACS that consists of a BEAM T-60C regulator and a throttle body. An optimized combination of regulator parameters, Venturi size and air-fuel ratio is reached via trial and error to achieve stable engine operation on hydrogen. Peak power reaches to 7.9 hp (5.9 kW) at 3,600 rpm, which is approximately 60% of the power output of the unmodified gasoline engine.

Since the air-aspirated hydrogen engine with only ACS reduces its peak power as much as 40% compared to its unmodified gasoline operation. The most effective option to restore the engine

peak power is to add a turbocharger system. IHI RHF-3 water cooled turbocharger is the smallest one made for general application and is typically used for motorcycle gasoline engine at power range up to 50 hp (37.6 kW). The attempt to boost the engine power by installing a IHI RHF-3 turbocharger along with the ACS is un-successful. The output peak power is unexpectedly reduced to 5.6 – 6.0 hp (4.2 – 4.5 kW) no matter the waste gate of the turbocharger is closed or not. This is another 25 % drop from what produced by an engine equipped with ACS alone. The power lose may be attributed to an increased intake restriction by the compressor of the turbocharger, or the backfiring causing lean combustion. Another possible reason is the air flow capacity of engine is inadequate to drive the turbine to a high speed to achieve boost pressure. A solution is to replace it by a smaller version of turbocharger. Unfortunately, there is no such smaller size available in the market after the diligent search.

One of the Braggs & Stratton V-twin engines is equipped with a retrofit of a pair of custom-made pistons (higher compression ratio of 14:1), a MDS MC-4 advanced ignition system and ACS. The engine operated stably without knocking and backfiring is under the condition of vey lean fuel/air lambda ratio, which produces peak power output at about 5.5 hp (4.1 kW).

The Honda GX360K1 engine (359 cc, with a compression ratio of 8.5:1) is a 4 stroke, overhead cam, 2 cylinder and water-cooled engine and is selected to convert running with hydrogen. The baseline performance of this engine delivers 9.4 hp (7 kW) output peak power at 3,600 rpm with liquid propane as fuel. Test run this engine with hydrogen equipped with the low restriction ACS throttle body reaches to the peak power of 5 hp (3.8 kW). Attempt for drawing more power out of the hydrogen engine by enriching fuel/air mixture beyond an air/fuel lambda ratio of 1.5 is failed in a similar fashion to the effort made on the air-cooled engine. The alternative way of restoring the lost peak power is to substitute with fuel injection system for carburetion. A great deal of effort is devoted to the development of various fuel injection systems for a water cooled Honda GX350 engine. It is disappointed the learn that the engine governor and mass air flow are un-successful to work together to stabilize rpm and respond to load resulting in engine backfiring. Since the generator causes so many problems to the hydrogen injection, extra effort is made to tune the hydrogen engine by fuel injection control without a governor linked throttle. With the throttle-less setting and other fuel injection strategies, the Honda GX350 hydrogen

engine operates smoothly and produces up to 4.7 hp (3.5 kW) electrical output. Pushing more power by moving into a territory of rich fuel burning, at lambda ratio of 1.5 or lower, the engine is experienced back firing.

High engine durability test is conducted on the converted Honda GC160 engine with ACS system. The task is carried out in Lab 152, constructed as a dynamometer lab for the small hydrogen engine. The load of the hydrogen generator Honda EN2500 is provided by plugging in an array of light bulbs. A constant load of 1 kW is used for the test. Total cumulative runtime of the hydrogen engine is 785 hours. Although the engine durability did not reach to the originally planned 1000 hours, it is still a plausible accomplishment by considering the fact that Honda GC 160 engine is a very economic engine for backup generator.

A prototype Ovonic bundled canister solid hydrogen storage system consisting of sixteen Ovonic 85G250B canisters having 1.2 kg H₂ nominal capacity is designed, constructed and assembled. The prototype system is characterized under various conditions to demonstrate its impressive absorption and desorption features. The results show the system is capable of supporting a wide range of output load of a hydrogen generator and are summarized below:

- The bundled canister storage system has demonstrated the capacity of 1.2 kg under the charging condition of 500 psig or higher pressure and the coolant temperature of 15 °C for 60 min.
- Higher pressure and/or lower coolant temperature in charging correlate to the higher H₂ capacity.
- The system is mostly operated at the pressure of 150 - 250 psig at 20 °C instead of 2200 psig for a T-size cylinder.
- The bundled canister storage system is capable of delivering near 9 hrs at a constant flow rate of 26 slpm, required to deliver a constant 1kW load from a small single cylinder hydrogen engine that is more than doubled runtime from a commercial T-size hydrogen cylinder.
- Increasing desorption flow rate results in shorter runtime.
- Increasing coolant temperature at any given desorption flow rate would increase in

runtime. It tends to benefit the most for the higher desorption flow rate.