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Development of an Inexpensive Free Piston Linear Motor Compressor for Home Refueling

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Abstract

In the U.S., there has been steadily increasing utilization of natural gas in place of gasoline and diesel as a transportation fuel. The waste collection industry has seen as many as 50% of new trucks being manufactured to run on dedicated compressed natural gas (CNG). This recent movement has been driven by the low cost and large reserves of U.S. natural gas. While there has been substantial growth in the medium- and heavy-duty truck markets, there has been very little impact in the light duty vehicle sector which constitutes about 70% of the energy used for transportation in the U.S.

The limiting factors in the light-duty market in the U.S. have been vehicle capital cost and fueling infrastructure availability. Vehicle costs have already dropped substantially over the past several years, and are expected to continue their downward trend with the increasing involvement of vehicle manufacturers and increased sales volume. With the cost premium of CNG vehicles coming down, fueling infrastructure remains the primary hurdle for the light-duty market to see significant growth. Currently there are about 1400 CNG stations in the U.S. with a little more than half of those being open to the public. This is more than two orders of magnitude less than the number of public gasoline stations in existence in the U.S.. With the high cost of a full-scale NGV fuel station, home fueling may serve as an attractive alternative pathway for CNG infrastructure and light-duty natural gas vehicle (NGV) expansion.

Home refueling is seen as a stepping stone towards substantial adoption of CNG throughout the light-duty market because it reduces the need for public stations by allowing commuters a convenient method of fueling their vehicles at home. The issue with current home refueling appliances (HRA) is that they increase the cost of adoption by $4000-$6000 including installation. Supported by Advanced Research Projects Agency-Energy (ARPA-E), the project team of Gas Technology Institute (GTI), University of Texas Center for Electromechanics, and Argonne National Lab are developing a novel free piston linear motor compressor (FPLMC) that is expected to have a substantially reduced cost and increased reliability over the current state of the art. This is accomplished by driving a four stage reciprocating compressor using a single piston and linear motor, eliminating all but a single moving part. The design is also completely free of any lubricants, eliminating the risk of gas contamination due to oil carryover. The team has modeled the performance, tested individual components, developed the solid model, and is proceeding with fabrication and commissioning which will be completed in 2014. This paper will describe the modeling results, design and performance of the prototype FPLMC.

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Introduction

Gas Technology Institute (GTI), University of Texas Center for Electromechanics (CEM), and Argonne National Lab (ANL) were awarded $4.2 million in 2012 by the Advanced Research Projects Agency – Energy (ARPA-E) of the U.S. Department of Energy under their program titled Methane Opportunities for Vehicular Energy (MOVE). The goal of the project is to develop a Free Piston Linear Motor Compressor (FPLMC) that will enable natural gas vehicles (NGV) to be inexpensively filled to 3600 psig (250 bar) from 0.25 psig (<1 bar), which is a common residential gas supply pressure. The project objectives are closely tied to the performance and expected manufacturing cost of the device; therefore, these were the driving metrics for many of the design decisions that were made over the last 24 months.

The goal of the project is to develop a low cost ~$500 compressor, capable of compressing gas from a residential natural gas supply directly into an NGV at a rate of 2 standard cubic feet per minute (SCFM) (56 liters per minute) (approximately 1gasoline gallon equivalent (GGE) per hour), while using less than 1.7 kWh/GGE. In addition, the compressor must be extremely durable, with a target lifespan of 15,000 hours or better. Lastly is should be designed to be ~100 lbs (45 kg), in order for a single person to be able to install it in, or next to, a garage.

Background

In the U.S., there is presently an abundance of natural gas which has led to a revived interest in NGVs. Currently, transportation constitutes about 28% (27 Quadrillion BTUs) of the national energy consumption, as shown in Figure 1(EIA, 2011). Petroleum provides 93% of the transportation energy, while natural gas (3%) and various renewables (4%) make up the remaining 7% of the energy consumption. However, digging deeper into the natural gas consumption data reveals that 94% of the natural gas used for transportation is for pipeline transportation, while only 6% is actually used in vehicles (ORNL, 2012). Therefore, NGVs make up less than 0.2% of the total transportation energy consumption in the U.S.

This is low, but not entirely unimpressive considering the number of NGVs on the road. Natural Gas Vehicles of America (NGVA) currently estimates that there are about 150,000 NGVs operating in the U.S. while there are about 250 million vehicles total (NGVA, 2014). Therefore, NGVs make up about 0.06% of the total vehicles used in the U.S., yet consume 0.2% of the energy. This is surprising until the driving factors are understood. Natural gas is an attractive transportation fuel because it is widely available, clean, and much less expensive than gasoline; however, the conversion of a vehicle to run on natural gas is expensive.

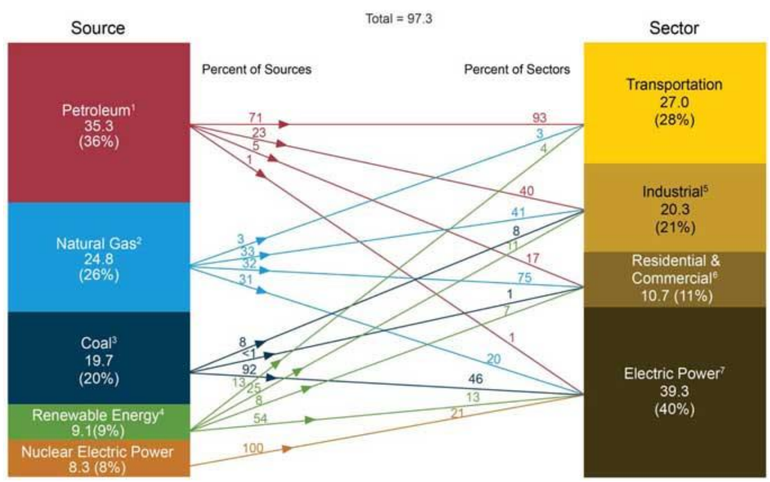

Figure 1. U.S. energy consumption by source and sector.

A NGV conversion in the U.S. costs between $5000 to $10,000 for a light-duty car or truck, depending heavily on the vehicle type and volume of compressed natural gas (CNG) storage. As the

NGV market has matured these costs have dropped, yet no emissions certified conversion has dropped below the $5000 mark.

Therefore, in the absence of state and government incentives, the primary way to take advantage of the fuel cost savings is to consume a lot of fuel. According to the Energy information Administration (EIA), the average cost of regular gasoline in the U.S. in 2014 is $3.51, while the average cost of a gasoline gallon equivalent (GGE) of natural gas purchased at a public station is $2.11, $1.40 less per gallon (EIA, 2014)(CNGnow, 2014). $2.11 per gallon is a very attractive price, but with the current cost of vehicle conversions a driver would still have to use over 3500 gallons just to break even. Depending on the vehicle's fuel economy, that translates to between 50 and 150 thousand miles. Therefore, it's not surprising that the average NGV uses much more fuel than the average U.S. vehicle, and adoption by the average driver has been slow.

Now that the problem is understood, what is the solution? The best way to reduce NGV cost is to increase the rate of adoption, and increase sales volumes. This can be accomplished by increasing the availability and benefits of owning a natural gas vehicle by making an affordable home refueling appliance (HRA) available to the public, which highlights the goal of this project. HRAs provide a couple additional benefits to owning a NGV. The first is that the vehicle owner can fill at home or at work without having to drive to a CNG station that may be miles away and possible have to wait in line to fuel. This should help to increase the adoption rate because it makes owning an NGV possible anywhere there is natural gas, while adding convenience because gas and diesel vehicles generally can't be filled at home or work either. Second, they help to reduce the cost of the fuel per GGE. Using an average residential gas rate of $10.33 per thousand cubic feet (MCF), and an electric rate of $0.12 per kWh, a GGE only costs about $1.50 (EIA, 2014) to dispense into a vehicle at home. The cost reduces further if the owner already uses a lot of gas, such as a large house, or even a small business, as the cost per cubic foot of gas decreases substantially as gas consumption increases. At about $6 per MCF the cost per GGE is already under $1. However, this second benefit has to be balanced with the additional cost of the HRA itself such that the HRA doesn't further increase the payback time on the total investment. Using the 3500 gallon payback number from above, multiplied by the additional savings of having a HRA ($0.60) versus filling at a public station, the installed price of the HRA needs to be less than $2100 otherwise it will increase the payback period. This value will obviously change depending on the specific area, but provides a reasonable target for the price of the HRA.

Realistically, the development of a cost effective HRA is not going to immediately have a significant impact on the market because the opportunity for a quick payback is still limited to high mileage drivers, even with the availability of an HRA. The potential that the HRA holds is that it will increase the availability of compressed natural gas to small businesses and home owners that are not within a convenient number of miles of an existing public station and can't afford to build their own. This will help the market grow, which will help to further reduce the cost of vehicle conversions, continuously improving the payback of switching to natural gas.

Free Piston Linear Motor Compressor (FPLMC) Design

The 36 month design project was divided into several tasks in order to manage the design and development of the FPLMC. These tasks included the preliminary design, detailed design, component validation, fabrication, and testing. The team is currently 24 months into the project, and has completed the preliminary design, detailed design, component validation, and has nearly completed the fabrication task.

During the preliminary design phase, the team conducted an extensive literature review to better understand how various compressors were designed including the sealing systems, valves, controls, assembly, etc. This included disassembly and evaluation of two existing, state of the art HRAs. It was concluded that while current compressor are well designed, they suffer from the requirement of many moving parts that must be precisely machined and balanced. This obviously adds cost, and increases the chance for components to malfunction, wear, or break.

Once the design review was complete the team began developing models for the motor and compressor. They started as separate models; however, it quickly became apparent that the models would need to be combined. A traditional compressor can be designed as a stand-alone system that simply has an appropriately sized motor attached to the crankshaft in order to provide the power. In this case however, a compressor driven by a linear motor is a highly interdependent system that requires precise controls and feedback in order to operate efficiently. Therefore, a dynamic compressor simulation was designed in Matlab Simulink to interface with various electric motor models also designed using the same tool. The compressor model was designed using a first law analysis of the individual sub-components which make up the compressor system. This approach allowed for changes to the stroke length, frequency, piston area, etc. to be studied to determine the average flow, temperatures, and mechanical power requirements. Losses such as friction and dead space were also incorporated into the model in order to calculate a realistic power requirement. A representative schematic of the model operation is shown in Figure 2.

The compressor stages were treated as adiabatic, which is a simplified assumption; however, the team was unable to find a reliable alternative. As a result, the team accounted for all the known losses such as friction and pressure drop through the valves, and then oversized the motor by about 20% to compensate for any additional losses during compression. Heat transfer from the interstage tubing was calculated using thermal resistance which was determined from an in depth study of the interstage tubing design using computational fluid dynamics and experimentation to validate the model. Check valves were also simulated by calculating the isentropic flow through a variable orifice that was defined by the size of the valve and lift height. The lift height was determined by the effective area of the valve being acted on by the pressure, and the opposing downstream back pressure and spring force.

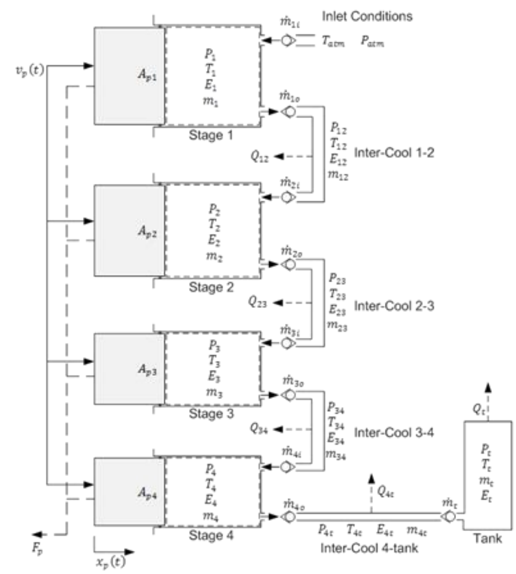

Figure 2. Compressor model schematic.

The motor simulations included the wall power, power electronics to run the motor (rectifier, link capacitor, and inverter), the detailed motor controller and motor model with interconnections and parameters defined using 3-D electromagnetic finite element analysis. A full transient simulation that runs in the Matlab/Simulink environment was used, and system states such as current, voltage, position, velocity, temperature, and pressure were available throughout the model. Control of the compressor was accomplished with a state control of position and velocity necessary to achieve the desired flow rate. The actual position and velocity were compared to referenced signals to produce a force command which was used by the motor controller to produce the appropriate voltage at the motor terminals. This tool was valuable for establishing specifications for the system components, and closed the final loop on the simulation, allowing it to predict end to end performance of the linear motor compressor.

Early graphical outputs of the simulated compressor temperature and pressure curves can be seen in Figure 3. The pressure and temperature of every stage can be seen with respect to the piston position. In addition, the net piston force versus displacement is shown in Figure 4. These are older plots, and not representative of the current design parameters, but effectively highlight the basic performance characterized by the simulation. Finally, Figure 5 represents an overview the transient dynamics of the system. The motor force versus time curve highlights the challenge of controlling the motor.

The team modeled, simulated, and downselected six different motor topologies in order to identify the best possible design for the FPLMC. The motor topologies were all modeled and simulated as described above in order to determine if the performance characteristics met the design targets for the project. From there, commercial and manufacturability characteristics such as size, weight, ease of fabrication, etc. were evaluated and scored for each design. Next, the overall risk in terms of controllability and safety was evaluated and scored. Lastly, an in depth cost analysis was conducted for each design using several financial models found in the literature. The total scores were added and the design with the highest score, or chance of success, was selected. This completed the preliminary design study and allowed the team to move forward with the detail design of the downselected system.

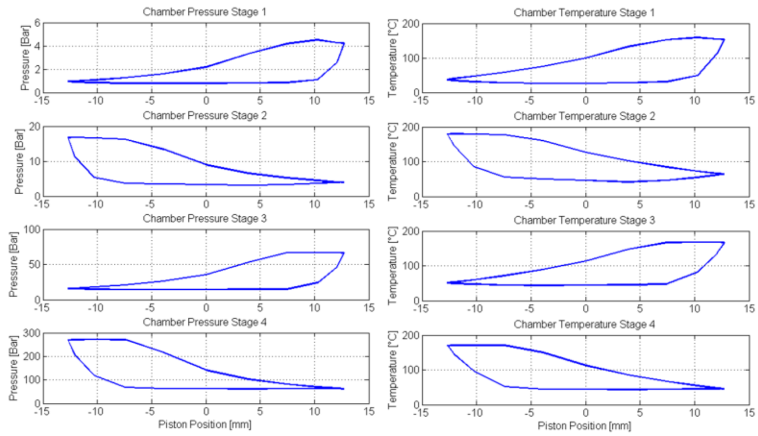

Figure 3. Pressure (left) and temperature (right) cycles of each stage from the dynamic model.

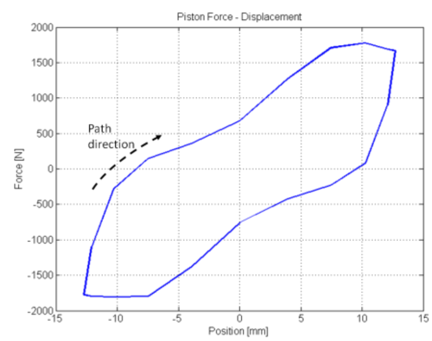

Figure 4. Net force-displacement curve for free piston compressor design.

During the detailed design task, the motor and compressor solid models were refined to more accurately represent the component details. In addition, individual component tests were conducted on the seals, valves, springs, interstage tubing, and motor controller. These tests were used to more accurately define the performance of the components, and then input the realistic performance into the motor/compressor dynamic simulation in order to better represent the expected true system performance. The simulation was also continuously refined to better account for thermodynamic losses, heat removal and overall power consumption.

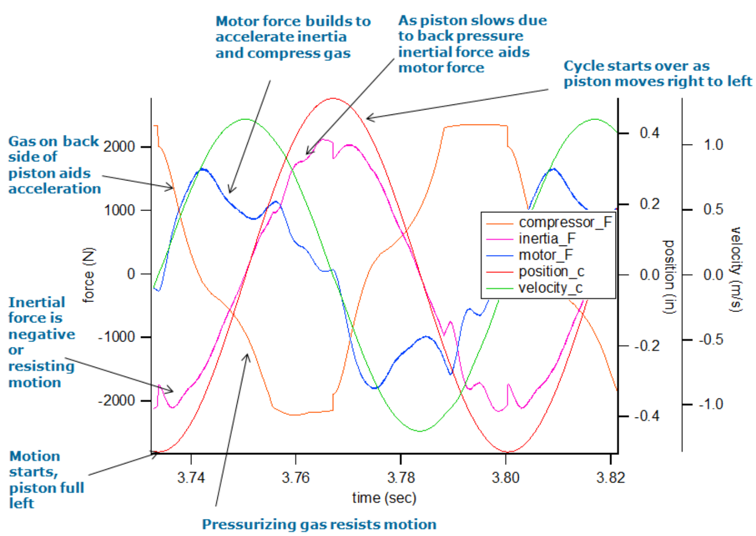

Figure 5. Compressor transient dynamics from simulation.

The seals were initially tested using a pin on disk wear test. Seal and coating combinations were tested at the peak velocity and temperature in a natural gas environment for the equivalent of 10,000 meters of travel, as shown in Figure

6. A total of nearly 30 combinations were tested using this apparatus, and then ranked based on the friction coefficient, and wear of the coating and seal. The most promising candidates were then tested in a high pressure test rig that pressurized the seals to full pressure using natural gas, and reciprocated a small piston at the expected operating frequency, shown in Figure 7. To date, the best seal and coating combination has run for over 3500 hours at peak pressure, and is still holding pressure. The team is optimistic about this combination being used in the prototype compressor since the compressor will not run at full pressure continuously. Therefore, it is believed the current testing is accelerated, although exactly how accelerated is difficult to determine.

The compressor valves were originally going to be purchased from a well-known manufacturer; however, due to the low flow and small physical size, no commercial valves were found to be available. As a result, the team decided to develop a novel valve design that can be injection molded from a PEEK material. The design was refined using CFD and FEA in order to determine the flow performance as well as strength and toughness to withstand the rapid pressure cycles for the life of the compressor. Spring options, including magnets, were also evaluated in depth to provide the closing force for the valves. The magnets seemed ideal from a durability standpoint; however, high temperature cyclic testing proved otherwise as the force profile of the magnets was substantially degraded, including supposedly high temperature resistant materials that were investigated. As a result, the team identified a suitable mechanical spring that should work well. The overall lift of the valves is very small, ~0.1-0.2 mm, therefore the cyclic stress seen by the spring is low, and should result in a long life. The valve spring system was ultimately tested using compressed air and high speed solenoid valves in order to map the actual performance. The valves were also filmed using a high speed camera in order to verify that they remained open throughout a pressure cycle without chattering, and closed before the pressure in the system reversed and forced them shut.

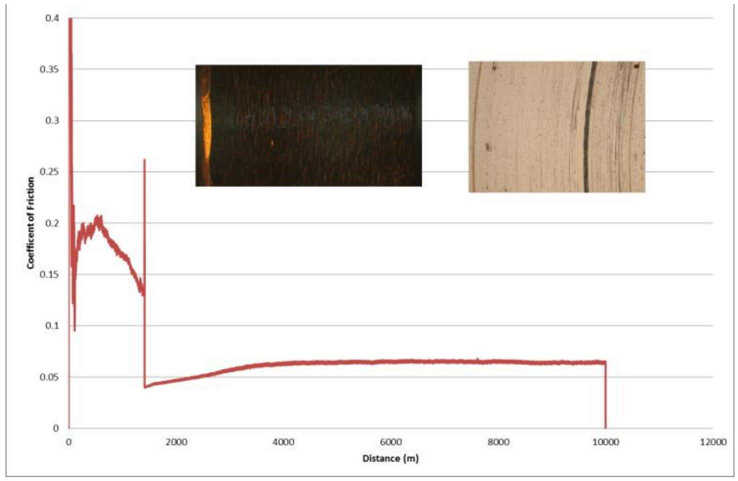

Figure 6. Example of tribo. testing and measured friction coefficient of a promising seal material.

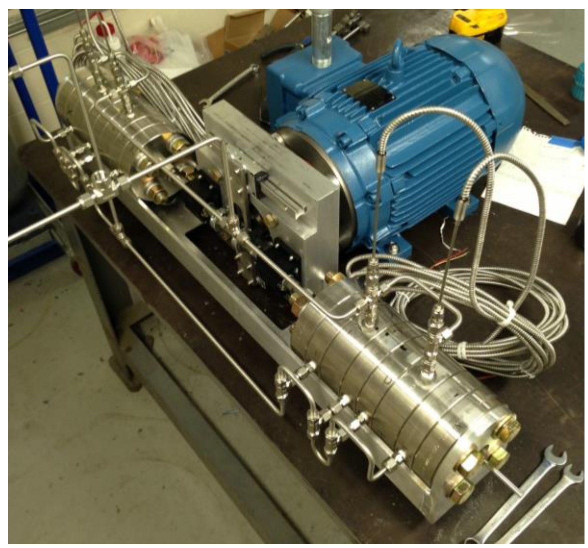

Figure 7. High pressure seal test rig.

A detailed analysis of the interstage cooling was also conducted in order to select the highest performing option for the lowest cost. Various designs were considered including plain tubing, radially and longitudinally finned tubing, and expanded metal foams. The various options were simulated and then tested in order to validate the performance around the actual compressor using a fan in the lab setting. The results showed that the plain tubing and radial fins were the best options in terms of performance and cost; however, plain tubing was ultimately selected due to the slightly reduced risk of fouling, as well as ease of fabrication.

Lastly, the motor controller was simulated using small scale off the shelf components and springs to represent the compressor stages. This approach is not a perfect simulation because the linear force profile of the springs doesn't exactly match the force profile of the compressor; however, it still allowed the model to be refined, and the feedback loop to be evaluated. Multiple techniques were investigated to provide feedback to the controller including inductance, a potentiometer and linear encoder. The linear encoder was ultimately selected because of the efficiency, reduced signal noise and accuracy. The bench scale test was able to repeatedly position the reciprocator to within 0.5 mm of the end of the stroke. This isn't as accurate as a mechanical system, or even some commercially available linear motors, but for the selected motor topology, cost, and target performance goals, it is adequate.

With the individual components validated, and the simulation showing promising performance, the team chose to move forward with the current components, and finalize the system design before beginning fabrication. The motor and compressor solid models were refined using Solidworks, and ultimately integrated into a single model. The interface between the motor and compressor was somewhat challenging because the piston is machined from a single piece of steel and then rigidly attached to the motor. This acted to simplify the piston design as it can be turned out of one piece of stock, but complicates the design of the other compressor components because they must be very accurately aligned from end to end of the compressor. The team finally decided to assemble the compressor components within sleeves at either end of the compressor. In this manner, as long as the sleeves were kept concentric, then the compressor stages would be concentric as well. The team also modeled the layout of the interstage tubing, as well as all the additional components for prototype testing such as pressure transducers and thermocouples for the motor and gas stream performance measurements.

Once the modeling was complete, the team began fabricating the components. The majority of the components are simple enough that they could be machined at GTI and CEM using the equipment available. However, several components such as the housing, piston and motor laminations had to be sent out of house in order to be machined accurately. Currently, fabrication is still ongoing, but expected to be completed by the end of September to early October, 2014.

Once the fabrication is complete, the motor and compressor will be tested independently at CEM and GTI respectively. The full scale motor will be tested against large springs that simulate the compressor stages in order to further refine the control algorithm and feedback loop before testing the system with gas. Separately, GTI will test the compressor using a more traditional scotch yolk drive mechanism that is being fabricated to accurately represent the performance of the linear motor. The scotch yolk will be powered using a traditional variable frequency drive electric motor in order to accurately control the speed of the compressor. This way the team can start slow before ramping up to full speed and pressure.

Once the individual components are tested, the components will be integrated into the full FPLMC. The fully assembled unit will initially be tested using nitrogen while the controls and feedback loop are refined. Once the unit is working, the whole unit will be shipped to GTI where a test loop has been developed to allow for continuous, unattended testing for 1000+ hours. The test loop contains a small pressure vessel (~1.5 cubic feet water volume) so that full fills can be demonstrated, but also is able to maintain a fixed outlet pressure so that the compressor can be tested extensively at a variety of fixed pressures. The test plan is to begin at lower pressures (500-1000 psig) and slowly work up to 3600 psig in order to ensure everything is working as expected. Once the compressor is running consistently at 3600 psig, the system will be allowed to run continuously by regulating pressure out of the pressure vessel and feeding it back to the inlet natural gas supply. In this manner the team can continuously test the compressor without releasing gas directly to atmosphere.

In addition to the static pressure durability test, the team is fabricating a second compressor that will be extensively tested within an existing environmental chamber at GTI. The chamber has the capability of reaching temperatures of -40-120 F, making it possible to test the FPLMC under nearly any anticipated real world operating condition. During testing, the FPLMC will continuously fill vessels as it would during normal operation. This will allow the team to not only test the operation under actual conditions, but also test that the compressor is capable of temperature compensation during temperature extremes. Between the durability and environmental testing, the team is confident that many of the issues can be worked out of the system in preparation for the beta prototype.

The final step of the existing project will be to fill actual vehicles in order to demonstrate that the compressor can work under real world conditions. This will be a straightforward test after the extensive durability testing is complete; however, it is considered important in order to show the real world performance of the compressor.

At the conclusion of the project next year, the team will have proven the FPLMC to be a reliable, durable compressor capable of refueling vehicles under a range of operating conditions. However, the compressor will still not be ready for commercialization, and will require the design, controls, aesthetics, etc. to be developed to a point that it can be placed as a standalone device at a home or business. The beta unit will also require additional testing, including field testing in order to get customer feedback before making any final modifications. The unit will then be ready for certification under the existing listing standard, CSA 12.6, or under the newly developed ANSI HRA code NGV 5.1 depending on when the new code is complete. Once certified, the FPLMC can be sold anywhere in the US, and begin to revolutionize the industry.

Respectfully Submitted,

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