Development and Operation of a Self-Refueling Compressed Natural Gas Vehicle

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Abstract

A dual-mode engine has been developed where in one mode all engine cylinders fire normally, providing locomotion for the vehicle. In the other mode, one cylinder of the engine is used to compress low pressure residential natural gas (NG), in multiple stages, to a standard U.S. compressed natural gas (CNG) vehicle storage tank pressure of 248 bar [3,600 psig]. This allows a natural gas vehicle (NGV) to be refueled anywhere there is access to the natural gas distribution network. Experimental studies were conducted on this prototype engine and are reported here. Knowledge gained from applying numerical models combined with empirical results led to the realization of a self-refueling natural gas vehicle. On the test stand, air (a surrogate for NG) was compressed by the engine to over 138 bar [2,000 psig] filling an 11.8 GGE [140 L] tank in 150 minutes. On the vehicle platform, the integral compressor had an approximate refueling efficiency of 70%, with an electrical-equivalent parasitic load of 12.6%. Idling of non-compression cylinders and the distance between the compressor and the three-way valves used to control the compression staging were noteworthy sources of inefficiency within the system. At the conclusion of the project the vehicle powered by the dual mode engine was driven over 161 km [100 miles] using self-compressed natural gas which originated from the local utility.

Keywords:

Natural gas, transportation, CNG, NGV, compressor, home refueling

Highlights

- A mode switching natural gas SI engine was developed and then installed in a vehicle.
- The engine can compress NG and combust in the same cylinder at different times.
- The vehicle was driven on public streets using its own CNG for 161 km.
- The system can increase the adoption rate of NG fuel in transportation.

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Preprint submitted to *Applied Energy*

June 5, 2015

34 **Nomenclature**

35 γ

36 η

37 Ψ

38 bar

39 kPa

40

ratio of specific heats, -

efficiency, %

parasitic load, %

pressure, $10^5 \frac{N}{m^2}$

pressure, $10^3 \frac{N}{m^2}$

41 1. Introduction

42 Compared to traditional fuels like gasoline and diesel, combustion of natural
 43 gas (NG) releases much less CO_2 into the atmosphere, a potent greenhouse gas
 44 (GHG) [1]. In fact, NG emits less CO_2 per unit of energy than any fossil fuel
 45 [2]. It has been shown that the use of CNG in a retrofitted gasoline engine can
 46 result in lower emissions of carbon-monoxide, carbon-dioxide, and unburned
 47 hydrocarbons [3]. The adoption of NG as a transportation fuel can act as
 48 a transition step towards a low carbon future where transportation relies on
 49 renewable sources of energy [4][5]. For further discussion of the impact of NG
 50 on GHG emissions, please consult Moniz et. al., Peterson et. al., or Curran et.
 51 al. [2][6][7]. Despite these benefits, only 0.03% of the 26.13 trillion cubic feet
 52 (Tcf) of NG consumed in the U.S. during 2013 was used for vehicle fuel [8].

53 Over 60 million homes and businesses in the United States (U.S.) are con-
 54 nected to the NG supply network [9]. The proliferation of horizontal drilling
 55 and shale gas made accessible by hydraulic fracturing has unlocked huge supplies
 56 of NG in the U.S.. According to the U.S. Energy Information Administration,
 57 shale gas drilling accounted for 9.7 Tcf of NG production in the U.S. during 2012
 58 and is forecasted to grow to 19.8 Tcf by 2040 [10]. This increase in natural gas
 59 reserves has led to the decoupling of domestic NG and global petroleum prices,
 60 with the price of NG falling substantially [11]. It was stated by Peterson et. al.
 61 that natural gas vehicle (NGV) adoption is influenced by the required payback
 62 period and infrastructure growth, but primarily driven by low NG prices and
 63 correspondingly high oil prices [6]. NG must be compressed in order to achieve
 64 an acceptable vehicle driving range. Typically the gas is stored inside a com-
 65 pressed natural gas (CNG) tank pressurized to 248 bar [3,600 psig], which has
 66 about a quarter the energy density of gasoline [7].

67 In many locations, the retail price difference between gasoline or diesel and
 68 CNG can be over \$2.00 per a gallon of gasoline equivalent (GGE) on an energy
 69 basis. This price difference is expected to continue for several decades, owing
 70 primarily to the abundant domestic supply of NG [12]. This presents an oppor-
 71 tunity for individuals and fleet owners to save a large amount of money on fuel
 72 cost for vehicles, with greater savings occurring the more miles driven. NG usage
 73 for transportation provides benefits beyond the GGE cost savings over conven-
 74 tional fuels. An increase in U.S. NG-fueled vehicles will reduce regulated air
 75 emissions, increase revenues spent on domestic NG exploration and production,
 76 and decrease dependence on imported petroleum [13][14].

77 As of October 2014, there are 770 public CNG refueling stations in the U.S.,
 78 compared to over 100,000 gasoline stations (as of 4th quarter 2012) [15][16].
 79 There are very few passenger cars available which use NG as a fuel, partly
 80 because of the lack of convenient refueling stations. Often referred to as fast-fill
 81 stations, refueling times at these locations are similar to conventional gasoline
 82 stations. One reason for the current absence of CNG refueling stations is capital
 83 cost. A CNG station costs around \$1.7M to construct compared to around
 84 \$150,000 for the average gasoline station (assuming three underground storage
 85 tanks) [17][18]. Interestingly, the lack of refueling infrastructure can also be

attributed to minimal consumer demand due to the limited number, and higher price, of CNG vehicles available from automakers [6]. In the work described here, the combustion and compression functions are combined into a single cylinder, creating an on demand integral compressor for vehicular applications. This allows a NGV to be refueled anywhere there is access to NG, even if the NG is at very low pressure, thereby removing the fueling infrastructure barrier inhibiting widespread adoption of CNG for U.S. transportation.

This internal combustion engine (ICE) operates in two distinct modes. In one mode, all engine cylinders fire normally providing locomotion for the NGV. In the other mode, one cylinder of the engine is used to compress low-pressure residential NG (0.017 bar [0.25 psig]), in multiple stages, to a standard U.S. CNG vehicle storage tank pressure of 248 bar [3600 psig]. When in this mode, the remaining engine cylinders idle to provide the necessary compression energy. Experiments were conducted to evaluate the performance of the compressor first using air as the working fluid and subsequently NG. This paper contributes to existing literature by: 1) Establishing a new, alternative method to produce CNG fuel for a NGV to further facilitate the adoption of NG in transportation, 2) presenting a novel dual-purpose engine cylinder for use with NG, 3) experimentally characterizing a proof-of-concept self-refueling vehicle, and 4) comparing the new concept to existing CNG fueling technology.

The ensuing parts of this paper present and discuss the development and operation of the self-refueling NGV. In the next section entitled Methodology, a more detailed description of the bimodal engine is presented. Then, the various iterations of laboratory experimental setups and the data acquisition system, including the instruments used, are discussed. Section 3 presents the logic created to automate the refueling process. Next, section 4 presents pertinent results from experimental testing and discusses the implications of these results. Finally, section 5 presents observations and conclusions gleaned from the experimental work.

2. Background

It was shown by Robinson and Beaty that reciprocating compressors are effective for compressing gases to high pressure ratios [19]. Furthermore, integrating a compressor into an internal combustion engine is not novel. For example, in the oil and gas industry wellhead boosters are used to pressurize NG at the drilling site for low pressure wells. In the 1970's GrimmerSchmidt began selling their MonoBlock air compressors and NG well boosters. This compressor used four cylinders of a V-8 engine for compression of air or NG while the other four cylinders supplied the mechanical work [20]. More recently (circa 2000), and led by economics, wellhead gas boosters such as the GasJack became increasingly popular to increase production from shallow, near-depleted well sites. The addition of a compression booster to a well site is capable of doubling or tripling daily production on gas wells producing less than 500 Mcf per day [21]. The GasJack booster uses a modified 460 Ford V-8 engine where one bank of cylinders, unchanged from the stock engine, burns NG from the well

(assuming it is suitable for burning). The second bank of cylinders is outfitted to pump natural gas.

Functional at-home NG refueling has a precedent, namely a line of small scale (1-5 kW [1.3-6.7 HP]) NG compressors that have been available to U.S. consumers for some time. A prime example of a home refueling NG compressor is the Fuelmaker Phill [22]. The Phill is a home refueling compressor that can be installed indoors or outdoors to refuel a CNG vehicle using residential pressure NG. It is a multi-stage reciprocating compressor, with four compression cylinders of successively smaller volumes coupled to a common crankshaft. The total cost including installation is typically over \$6,000 [23]. The following will briefly outline the performance of this compressor such that a comparison can be drawn with the new compressor presented in this paper.

The unit will compress gas to 248 bar [3,600 psig] at a flow rate of 0.5 Gasoline Gallon Equivalent (GGE) per hour using 0.85 kWh of energy on average, according to the manufacturer [22]. At this refueling rate, in one hour, the gas compressed would represent 60,138 kJ [57,000 BTU] of chemical energy. Assuming the electricity used came from a thermal power plant with a heat rate of 9496 kJ/kWh [9,000 BTU/kWh] (i.e., 37.9% BTE) and transmission and distribution losses are 8%, then the compression penalty (parasitic load) associated with the Phill compressor is approximately 14.3%. This device is considered state-of-the-art for residential CNG vehicle refueling. The primary disadvantage of the device and home refueling in general is the fill rate. If a vehicle had an 11.8 GGE CNG tank, it would take nearly 24 hours to completely fill up. With this disadvantage in mind, the current project aimed to use the engine itself for compression to improve fueling times.

3. Methods

In the following, after the bimodal engine concept is more thoroughly explained, the laboratory experimental setup is presented in sections 3.2 and 3.3. The majority of testing occurred in the laboratory and primarily used air as the working fluid. After the prototype engine was thoroughly tested with air, a new experimental setup was constructed in the laboratory space to conduct compression tests with NG. Shortly after those tests, the system was moved into the prototype vehicle. Compression with NG is presented in section 3.4 and experiments conducted with the vehicle in order to approximate the refueling efficiency are outlined in section 3.5.

3.1. Bimodal Engine Description

The following is a description of a prototype bimodal NG engine and integral compressor. As mentioned in the introduction the engine has two modes, driving and refueling. In refueling mode, one of the engine cylinders (cylinder 1) acts as a compressor. This particular gas compressor can be classified as a three-stage reciprocating positive displacement compressor. In order to create this engine, a custom cylinder head was designed and manufactured out of billet aluminum.

172 A high resolution model was constructed in the MATLAB Simulink environ-
 173 nment to simulate the compressor functionality of the engine. This 1-D lumped
 174 parameter model used differential equations for pressure and temperature within
 175 control volumes originating from Guzzella and compressible flow through ori-
 176 fices to determine mass flow rates [25]. The model provided a means to conduct
 177 detailed analysis of individual components in the refueling system and evaluate
 178 the overall performance of the design. While the majority of the hardware used
 179 was produced by the engine OEM or aftermarket suppliers, the rest had to be
 180 custom designed for this particular application. Additional details concerning
 181 the model and its use as a design aid for the compression cylinder can be found
 182 in Echter et al. [26]. The custom cylinder head created for this bimodal engine
 183 can be seen in Figure 1.

Figure 1: On the left, a CAD image of the bimodal engine cylinder head. On the right, a sectional view of the cylinder head highlighting the compressor valves (A), intermediate volume (B), and the inlet and outlet check valves housed in the valve block (C).

184 When behaving as a compressor, while the single cylinder is compressing
 185 NG, it is powered by spark-ignited (SI) NG combustion in the remaining engine
 186 cylinders. In this mode, the intake and exhaust valves in the compression cylin-
 187 der are deactivated hydraulically by utilizing collapsible lifters. The spark plug
 188 in that cylinder is also deactivated. The compression cylinder was modified to
 189 include two additional valves which, when opened, allow the flow of NG in and
 190 out of the compression cylinder. Thus, there are four valves in cylinder 1, while
 191 cylinders 2-6 each contain two valves. To aid the following discussion, the image
 192 on the right side of Figure 1 displays the compressor functionality unique to this
 193 bimodal engine. Highlighted by callouts, the refueling (compressor) valves (A),
 194 intermediate port volume (B), inlet check valves, and outlet check valve ports
 195 (C) can be seen.

196 The intermediate port volume acts as a dead volume, increasing the clear-
 197 ance volume of the compression cylinder and thereby reducing the compression
 198 ratio (CR) from 13:1 in combustion mode to 4.7:1 for compressor operation.
 199 Two small refueling valves were added rather than one large valve to maximize
 200 the effective flow area considering the size and shape of the selected location.

201 These compression valves remain open during the upward and downward strokes
 202 of the piston, thus, gas is compressed with each revolution of the crankshaft.
 203 The prototype bimodal engine has three distinct stages of compression. Stage
 204 1 compression occurs when low pressure residential natural gas enters the com-
 205 pression cylinder and then flows into Tank 1, the first of two staging tanks.
 206 Stage 2 compression occurs when the pressurized gas within Tank 1 is routed
 207 into the compression cylinder and then flows into Tank 2, the second staging
 208 tank. Finally, Stage 3 compression occurs when the gas in Tank 2 is directed
 209 into the compression cylinder one last time, where it then flows into Tank 3, the
 210 main high pressure storage tank of the system.

211 Heavy-duty diesel engines are some of the most robust ICEs in production
 212 today and they are designed to withstand peak in-cylinder firing pressures of
 213 greater than 200 bar [27]. The goal of this project was to compress NG in the
 214 ICE cylinder to 250 bar. Considering this fact, a Cummins model 6BT 12-
 215 valve 6-cylinder 5.9L turbo diesel engine was chosen for the prototype due to its
 216 reputation as a robust ICE and the availability of high performance aftermarket
 217 parts which were used to increase the stock peak cylinder pressure capability.
 218 The engine was converted to spark ignited NG operation by way of a second
 219 electronic control unit (ECU), replacing diesel injectors with spark plugs and
 220 by lowering the firing compression ratio from 17:1 to 13:1 [28].

221 *3.2. Laboratory Details*

222 The two key thermodynamic properties to be measured when conducting
 223 compression experiments are pressure and temperature. For experiments with
 224 the bimodal engine, in-cylinder pressure was monitored using a Kistler 6056AU20
 225 pressure transducer. Pressure was recorded outside the compression cylinder in
 226 the valve block using a Kulite HEM-375-5000A pressure transducer.

227 Temperature was recorded using Omega type-K thermocouples. Each of
 228 the three storage tanks had a thermocouple and a pressure transducer to mon-
 229 itor tank temperature and pressure. Tank 1 had an Omega PX309-1KG5V
 230 transducer and Tank 2 had an Omega PX309-2KG5V. Tank 3 had a Kulite
 231 HEM-375-5000A transducer, the same model and accuracy of the transducer
 232 used with the valve block.

233 Data acquisition and control was achieved using a National Instruments
 234 CompactRio (cRIO) model 9074. In addition to the actuation of valves and the
 235 monitoring of vital engine ECU information such as current speed and engine oil
 236 pressure, data was recorded using LabVIEW. For the majority of experiments
 237 conducted in the lab, a sampling rate of 10 Hz was used, but some experiments
 238 had a sampling rate of 20 kHz in order to adequately resolve the pressure trace
 239 inside the compressor valve block.

240 The engine dynamometer was a 125 HP 230V TECO-Westinghouse Max-
 241 E1, 3-phase AC motor controlled by a Yaskawa A1000 variable frequency drive
 242 (VFD). The VFD is connected to two Yaskawa CBDR-2110B braking units
 243 which are each connected to two resistor banks.

244 3.3. Laboratory Experimental Setup

245 Earlier experiments were conducted on the test stand using multiple heat
246 exchangers [29]. Here the discussion is focused on a streamlined laboratory
247 approach where a single heat exchanger was employed.

248 The schematic for the simplified single heat exchanger experimental setup
249 can be seen in Figure 2. The system functions by cycling three-way valves
250 and therefore access to tanks. A number of different filling experiments were
251 conducted. Initially, the three-way valves were actuated manually (using the
252 computer to flip a relay switch) and the filling times required to reach varying
253 pressures goals such as 34.5, 69, and 138 bar [500, 1,000, and 2,000 psig] were
254 recorded. Later, actuation of the valves was automated using logic to determine
255 the minimum and maximum cutoff pressures for Tanks 1 and 2. The scheme
256 used to switch between staging tanks was varied in an effort to optimize the
257 refueling time.

OSU Lab Testing Schematic – 3 stages v5/16/14

Figure 2: Schematic of the single heat exchanger experimental setup used for air compression experiments. Not to scale.

3.4. Laboratory and In-Vehicle Natural Gas Compression Setup

The following will discuss NG compression experiments conducted in the lab and on the prototype vehicle en route to on-road operation, as well as the estimation of the overall refueling efficiency. When the transition in the laboratory from compressing air (as a NG surrogate) to compressing NG occurred, the experimental setup was simplified to a two-stage compressor system. This was done by removing one of the two three-way valves used to control compressor staging. Due to the removal of the third compression stage, the maximum pressure attainable by the compressor was significantly reduced. The absolute maximum theoretical pressure that could be reached with NG was in the ballpark of 34.5 bar [500 psig]. The reduced compression capability would ultimately limit the vehicle range, but also allow for lower pressure rated copper

270 tubing to be used. Figure 3 is a schematic of the NG compression setup used
 271 in the laboratory and on the vehicle.

Figure 3: Schematic of the 2-Stage system used for compression experiments using natural gas. The same setup was used both in the laboratory and in the vehicle.

272 The advantage of copper tubing is a remarkable improvement in thermal
 273 conductivity over stainless steel (roughly 20 times). No heat exchanger was used
 274 with this setup; instead, all heat rejection was handled by the copper tubing
 275 itself. In the vehicle, two approximately seven-foot lengths of copper tubing
 276 (serving as the heat rejection apparatus) were run along the frame, connecting
 277 the storage tanks with the compressor. The two-stage compression process
 278 would not generate nearly as much thermal energy as three-stage, 138 bar [2,000
 279 psig] experiments. However, at the time, it was not known empirically if the
 280 tubing would dissipate the required thermal energy. Additionally, the three
 281 Optimum Composite Technologies model C19A45L10Y20 CNG fuel tanks used
 282 have an inner aluminum shell (that is then wrapped in carbon fiber) which
 283 is capable of storing excess heat. There is approximately 33.1 kg [73 lbs] of

aluminum in each tank, a substantial thermal reservoir.

In the interest of safety, only a few NG compression tests were run in the laboratory, successfully reaching tank pressures of 24 bar [350 psig], and later 25.5 bar [370 psig]. The testing in the lab was done primarily to shake down the system and troubleshoot any problems before moving to the vehicle. For example, the logic implemented to switch from driving to refueling mode and vice versa was tested first in the lab.

3.5. Filling Efficiency Experiments

In order to empirically determine the efficiency for this refueling process, two quantities had to be determined: the chemical energy content present in the vehicle storage tanks at the end of the refueling period and the chemical energy consumed by the ICE in order to fill the storage tanks with compressed gas. The first of these two quantities is relatively easy to determine. Pressure transducers and thermocouples measured the pressure and internal temperature of the tanks, allowing the mass of NG present to be calculated using the ideal gas equation of state.

In order to quantify the NG fuel consumed by the ICE to idle and power the compression process, an external source of high pressure CNG (six-pack of 49.8 L, 165.5 bar [2,400 psig] bottles) was connected to the fuel regulator on the vehicle. The regulator throttles the pressure to approximately 5.5 bar [80 psig] before being injected upstream of the intake manifold. The pressure in the external source of CNG used to fuel the ICE was monitored in two ways: 1) using an analog pressure gauge connected to the six-pack, and 2) logging the fuel storage pressure reading from a transducer located inside the previously mentioned vehicle fuel regulator using the aftermarket NG conversion engine control unit (ECU).

Low pressure (0.017 bar [1/4 psig]) NG from the lab building was connected to the refueling port of the vehicle. A bellows-type commercial gas meter (Elster AL-1000) was installed at the low pressure source and the quantity of gas used (in cubic feet) was recorded. A simple diagram of the experimental setup is in Figure 4.

The volume flow was later converted to mass of NG using the ideal gas equation of state with a pressure of 0.017 bar [1/4 psig] and the average source gas temperature reading of 36°C. The engine speed in RPM was logged by the ECU. The goal/result of this particular test was quantifying the mass (energy) of NG that was required to fuel the engine to both idle and power the compression process at the same time, as well as the quantity of NG pumped from the low pressure source into the storage tanks during that time.

A second experiment was designed to determine the fuel consumed by the ICE when idling on five cylinders, i.e., to determine the amount of energy required to run only the engine without compressing. To do so, the compressor cylinder was operated as an air-spring by turning off the intake and exhaust valves, while keeping the compressor valves closed. Air would be compressed and then expanded repeatedly in this cylinder. The six-pack of CNG previously

Figure 4: Experimental setup for the in-vehicle NG compression experiment.

mentioned was again used to fuel the ICE and the pressure in a single bottle was monitored in the same manner. A simple diagram of this experimental setup can be seen in Figure 5. In the diagrams seen in Figures 4 and 5, the compressor cylinder is placed outside of the engine so it may be easier to understand the experiments. The compressor is integrated into the ICE in the physical system.

Figure 5: Diagram of the experimental setup used to estimate the idle fuel consumption.

The result of this experiment was a function for estimating the instantaneous fuel consumption rate (with respect to engine speed) required to only idle the ICE. Additionally, the approximate amount of energy that was solely used to compress NG—the useful energy—was determined by taking the difference between the gross energy consumed and the estimated idle energy.

338 4. Calculation of Compression Stage Cutoff Pressures

339 As mentioned previously, the prototype bimodal engine utilizes three distinct
 340 stages of compression. For a moment, consider a two-tank system with only two
 341 stages of compression. When in refueling mode, the compression cylinder in this
 342 bimodal engine has a fixed compression ratio, which results in a fixed pressure
 343 ratio, given by Equation 1 assuming isentropic compression, where gamma is
 344 the ratio of specific heats ($\gamma=1.3$ for methane and $\gamma=1.4$ for air) [30]. For this
 345 particular engine, the compressor CR was 4.7:1.

$$P_{ratio} = \left(\frac{V_{BDC}}{V_{TDC}} \right)^\gamma = (CR)^\gamma \quad (1)$$

346 Therefore, when Tank 2 is being filled using the pressurized gas from Tank
 347 1 as the source, it is impossible to gain any additional pressure within Tank 2
 348 when the pressure in Tank 1 falls to:

$$P_{Tank1} = \left(\frac{P_{Tank2}}{P_{ratio}} \right) \quad (2)$$

349 In other words, Equation 2 says that when Tank 1 reaches this minimum cutoff
 350 pressure, no more mass can be moved from Tank 1 into Tank 2 (this same logic
 351 would apply with three tanks in the system when filling from Tank 2 to Tank
 352 3). With the minimum cutoff pressure known, now the optimal maximum cutoff
 353 pressure needs to be determined; in other words, how much pressure should be
 354 in Tank 1 before the system switches to filling Tank 2, using the compressed
 355 gas in Tank 1.

356 Figure 6 shows a plot of pressure versus time for the filling of a single tank.
 357 It is evident that the slope of the pressure curve is at its greatest value (i.e,
 358 high dP/dt) initially and as time goes on the slope diminishes (i.e. low dP/dt)
 359 until eventually reaching an asymptote (zero dP/dt). In other words, there are
 360 diminishing returns the longer the compressor is used to fill the tank.

361 Based on this information, it is clear that while one can pump more mass
 362 into Tank 2 in a single filling cycle if you fill Tank 1 to a higher pressure, it takes
 363 a lot of time to gain those final few units of pressure. It is more advantageous to
 364 fill Tank 1 to a lower pressure and switch to filling Tank 2 (with Tank 1 as the
 365 source) more frequently, taking advantage of the high dP/dt . To clarify, this
 366 keeps the operation on the part of the pressure response curve with the greatest
 367 slope, resulting in faster filling times (higher overall mass flow rate). However,
 368 since the pressure ratio given by Equation 1 is fixed, setting the maximum cutoff
 369 pressure of Tank 1 to low values does not allow high pressures to be reached in
 370 Tank 2. In order to reach the highest possible pressure in Tank 2, the maximum
 371 cutoff pressure for Tank 1 must be gradually increased.

372 The best way to do this is to take the calculated value for the minimum
 373 cutoff pressure from Equation 2 and simply add an offset to that value, such as
 374 69 or 138 kPa [10 or 20 psig], to create the maximum cutoff pressure for Tank
 375 1, as presented in Equation 3.

Figure 6: Plot of the tank pressure over time for a single tank fill using the single heat exchanger experimental setup and air as the working fluid.

$$P_{Tank1,max,cutoff} = \frac{P_{Tank2}}{P_{ratio}} + P_{offset} = P_{Tank1,min,cutoff} + P_{offset} \quad (3)$$

Since the minimum cutoff pressure always increases as the pressure in Tank 2 increases, this offset will allow Tank 2 to gradually fill to higher and higher pressures. Using a simple, ideal gas, isothermal (for the gas in the tanks) compressor model, Figure 7 shows the effect of varying the offset from 69 kPa [10 psig] up to 345 kPa [50 psig].

This plot suggests that keeping the offset small allows higher pressures in Tank 2 to be reached in shorter amounts of time. As the offset grows larger, it takes longer for Tank 1 to reach the maximum cutoff pressure until eventually it becomes impossible to reach. The fixed pressure ratio limits the maximum pressure that can be reached within Tank 1; for example, assume this limit is 621 kPa [90 psig]. If the offset used to determine the maximum cutoff pressure is sufficiently large, such as 345 kPa [50 psig], there comes a point when Tank 1 is incapable of reaching the desired pressure. In this case, this occurs when the minimum cutoff pressure is greater than 276 kPa [40 psig]. Ultimately, this suggests that switching more often will lead to faster fill times for the NG storage tanks on this self-refueling NGV.

Figure 7: Tank 2 pressure over time for different Tank 1 offset values.

5. Results and Discussion

5.1. Air Compression in the Laboratory

In most of the compression experiments the engine was motored by the dynamometer because it was easy to hold a constant RPM and safer than burning NG. In later compression experiments, NG was burned in five cylinders while the engine was coupled to the AC motor in order to determine the effect of combustion on the temperature of compressed gas leaving cylinder 1 (the compression cylinder).

Afterward, the engine was decoupled from the dynamometer and allowed to spin freely under its own power while air was compressed. Without the aid of the dynamometer, the speed of the engine is a function of the load torque and the throttle position. Depending upon the pressure of the source gas and the back pressure of the destination tank, each filling stage loads the engine a different amount. For example, when switching to Stage 2 filling (that is, filling Tank 2 using the pressurized gas in Tank 1) from Stage 1, the load on the engine is increased and the RPM will decrease if the throttle is not opened a sufficient amount. When the Stage 2 fill is complete and the refueling system returns to Stage 1, the engine load is decreased and unless the throttle is also decreased the engine speed will increase rapidly. The engine load was primarily a function of the inlet pressure on the compression cylinder, but the back pressure in the tank was also a factor.

In Figure 8 and Figure 9, the pressure within each of the three tanks in the system is shown for a single refueling experiment. Air was the working fluid and

415 the engine was coupled to the dynamometer. Until around the two minute mark,
 416 all three tanks are being filled in the first stage of compression. After this mark,
 417 stage two filling occurs and the pressure in Tank 2 and Tank 3 increases while
 418 the pressure in Tank 1 correspondingly decreases. Around the seven and a half
 419 minute mark, the first stage three fill occurs. The pressure in Tank 3 increases
 420 sharply as gas inside Tank 2 is routed into the compressor. Tank 2 pressure
 421 decreases and Tank 1 pressure remains constant during this time. After this
 422 initial stage three fill, the cycle of multiple stage one and stage two fills followed
 423 by a single stage three fill continues for the entirety of the experiment. The
 424 decrease in Tank 3 pressure after each Stage three filling event, seen in Figure
 425 9, is associated with cooling within the tank volume.

Figure 8: Plot of system tank pressures versus time for a refueling experiment with air as the working fluid. The engine was coupled to the dynamometer. Units are in gauge.

Figure 9: View of the initial behavior of the system for the same experiment shown in Figure 8.

426 The reader may notice that the highest pressure reached in the main storage
 427 tank (Tank 3) was approximately 139 bar [2,020 psig] and was achieved in 150
 428 minutes. Recall that all three tanks were the same volume (11.8 GGE [140 L])
 429 and the cylinder displacement volume was 0.98 L. The time it took to reach
 430 34.5 and 69 bar [500 and 1,000 psig] was just over 30 minutes and just under
 431 70 minutes, respectively. The highest pressure reached in the laboratory for the
 432 compression of air was 144.8 bar [2,100 psig]. This is below the standard U.S.
 433 NGV CNG storage tank pressure of 248 bar [3,600 psig]. While the prototype
 434 bimodal engine was designed to handle in-cylinder pressures of greater than 250
 435 bar, it was decided that doing so in the laboratory posed an unnecessary risk. If
 436 a catastrophic failure occurred, the project would be critically delayed before the
 437 engine could be put into a vehicle and driven. The 138 bar [2,000 psig] mark was
 438 reached five times and 248 bar [3,600 psig] was achieved in the laboratory using
 439 the bimodal engine and an external pressure amplifier (Haskel AGD-32) driven
 440 by engine-compressed air. As an aside, if sorbent tank technology continues to
 441 progress, reaching extremely high pressures may no longer be necessary [31].

442 5.2. Natural Gas Compression In-Vehicle

443 The following plot in Figure 10 shows the pressure within the vehicle storage
 444 tanks during a NG compression test. Recall that only two stages existed on the
 445 vehicle (stage two was comprised of two 140 L tanks) and therefore the maximum
 446 possible pressure was limited. The experimental setup is that of Figure 4.

447 Over the course of 40 minutes, the pressure in the main storage tank was
 448 increased from approximately 0.138 bar [2 psig] to 18.1 bar [263 psig], with a

Figure 10: Pressure versus time for a NG refueling experiment conducted on the prototype vehicle. Units are in gauge.

449 corresponding temperature rise of 10°C. The staging tank was taken from an
 450 initial pressure of approximately 0.276 bar [4 psig] to 3.17 bar [46 psig]. The
 451 amount of energy added to the system during this time was equivalent to about
 452 190.6 MJ, assuming a lower heating value (LHV) of 50 MJ/kg for methane gas
 453 [30].

454 The compressibility factor was significant enough to warrant inclusion in the
 455 ideal gas equation of state calculations for mass and was found using a Refprop
 456 MATLAB function available from NIST assuming methane as the fluid. The
 457 result of this experiment was a value for the energy content consumed by the
 458 engine in order to compress residential NG and store this CNG on the vehicle,
 459 ~80 MJ. The ratio of gross energy content required to compress NG into the
 460 storage tanks to the energy content present in the tanks at the end of a refueling
 461 cycle provides a gross compression penalty/cost (also referred to as a parasitic
 462 load) of ~42%, seen in Equation 4.

$$\Psi = \frac{E_{Gross, Compression}}{E_{NG Tank}} = \frac{80 MJ}{190.4 MJ} = 42\% \quad (4)$$

463 However, this efficiency cannot be compared directly to the project goal of a
 464 parasitic load for an at-home refueling station goal of <5% [11]. This is because
 465 that number is defined as a ratio of electrical energy consumed to the energy
 466 content in the vehicle storage tanks, as defined by Equation 5.

$$\Psi = \frac{E_{compression}}{E_{NG Tank}} < 5\% = \frac{1.78 kWh}{GGE} \quad (5)$$

$$\Psi = \frac{E_{Gross, Compression}(\eta_{BTE})}{E_{NG Tank}} = \frac{80MJ(0.3)}{190.4MJ} = 12.6\% \quad (6)$$

467 The parasitic value seen (Ψ) in Equation 6 is directly comparable to the
 468 parasitic goal mentioned above. The brake thermal efficiency (BTE) of the
 469 engine is assumed to be 0.3 [30]. However, this is an estimate and if the BTE
 470 is less than 0.3 then less energy was needed for compression and the parasitic
 471 load becomes even smaller (8.2% at a BTE of 0.2). For this ratio, consider
 472 the numerator to represent the energy required at the shaft to operate the
 473 compressor and the denominator to represent the energy acquired in the tank
 474 over the same time span. Note that this parasitic load is lower than that of the
 475 Phill unit previously discussed (14.3%).

476 After combustion, the shaft energy that was consumed by the bimodal ICE
 477 during refueling was split between overcoming friction to idle the engine and
 478 powering the compressor cylinder. The energy needed to simply overcome fric-
 479 tion and idle the engine can be considered an unavoidable penalty, one which
 480 has already been paid by electrical energy at the thermal power plant location
 481 (efficiency of the gas and/or steam turbines) and during transmission and dis-
 482 tribution to the consumer (T&D efficiency). In order to quantify how much
 483 energy was used to simply idle the engine, another experiment was conducted,
 484 shown in Figure 5.

485 After the engine was warmed, it was idled at four different engine speeds
 486 (ranging from 750 to 1600 RPM) for fifteen minutes at each point. The mass
 487 of fuel consumed at each engine speed was calculated using the ideal gas law
 488 equation of state and a compressibility factor. A second-order polynomial was
 489 fit to the data, as seen in Figure 11, to give a reasonable estimate of the instan-
 490 taneous fuel consumption rate in g/s versus engine RPM, with an adjusted R2
 491 value of 0.992.

Figure 11: Plot of the fuel consumption rate versus engine idle speed. The four idle speeds tested are plotted along with a line of best fit. The vertical lines show the upper and lower RPM bounds during the compression experiment conducted prior to this test.

The energy required to both idle the engine and compress gas is known from the first NG compression experiment. This fuel consumption rate relationship was then used with the engine RPM data logged during the NG compression experiment in order to find an approximate value for the energy that was used solely to idle the engine and overcome friction and other engine parasitic losses during the compression experiment. Additionally, the energy that was only used for NG compression was determined.

During the NG compression test (the pressure plot seen in Figure 10) an equivalent of approximately 271.9 MJ of total energy (i.e., 100%) was input to the vehicle, assuming a LHV of 50 MJ/kg. Approximately 190.4 MJ of this energy (70%) consisted of compressed gas present in the vehicles fuel storage tanks at the end of the experiment. Additionally, the equivalent of 56.8 MJ of energy (20.9%) was consumed by the engine to idle and overcome internal losses such as friction, while approximately 24.7 MJ of energy (9.1%) was needed to drive the compressor used to fill the fuel tanks. As one can see, for this prototype, more than twice the energy needed to compress NG was spent to idle the engine. In future engine incarnations, the engine will be more heavily loaded by compression, for example, by a higher compression ratio and/or by using multiple cylinders for compression such that the friction load will become a small percentage of the gross power.

To determine the NG compression efficiency of this prototype, the same compression energy efficiency formula used by the Argonne National Laboratory GREET model, seen in Equation 7, was used [32]. This efficiency equation originates from a General Motors study [33].

$$\eta_{compression} = \frac{LHV_{CH4}}{LHV_{CH4} + \left(\frac{w_{adiabatic, theoretical}}{(\eta_{compressor, adiabatic})(\eta_{engine, bte})} \right)} \quad (7)$$

516 This equation is the ratio of the LHV of methane (in kJ/kg) divided by the
 517 LHV of methane plus the actual amount of work it took to compress the low-
 518 pressure methane gas. Dividing the theoretical adiabatic compression work (in
 519 kJ/kg) by the adiabatic efficiency of the compressor gives the amount of energy
 520 required at the shaft. Dividing that number by the efficiency of the ICE yields
 521 the specific methane energy required for compression.

522 For the prototype bimodal engine the compression efficiency formula is sim-
 523 ilar, except, since the mass/energy of NG used is known (using the LHV) and
 524 the compressor and engine efficiencies are not, the units will be kJ rather than
 525 kJ/kg. The form used is given by Equation 8. It is the ratio of the energy in
 526 the storage tanks divided by the sum of the energy in the storage tanks and the
 527 energy consumed by the ICE to compress the NG [33].

$$\eta_{compression, gross} = \frac{E_{Tanks}}{E_{NG Tanks} + E_{ICE}} \quad (8)$$

528 Two compression efficiencies are presented below, the gross and net. The
 529 gross efficiency is the actual efficiency of this prototype during a favorable
 530 portion of a refueling cycle and is approximately 70%¹. The net compression effi-
 531 ciency represents the estimated energy the ICE required to compress NG with-
 532 out the idle penalty. Both efficiencies are calculated in Equations 9 and 10.
 533 Note that the percentage of fuel energy consumed to compress the NG is the
 534 difference between η and 100%.

$$\eta_{compression, gross} = \frac{E_{Tanks}}{E_{NG Tanks} + E_{ICE}} = \frac{190.4 MJ}{(190.4 + 81.5) MJ} = 70\% \quad (9)$$

$$\eta_{compression, net} = \frac{E_{Tanks}}{E_{NG Tanks} + E_{compressor}} = \frac{190.4 MJ}{(190.4 + 24.7) MJ} = 88.5\% \quad (10)$$

535 The net efficiency suggests an upper bound on the theoretical efficiency
 536 that could be attained if the energy required to idle was negligible compared
 537 to the energy used to power the compression process—a situation that could
 538 be approached by designing the engine and compression process for optimum
 539 efficiency, not high pressure operation as was done here. It also shows the

¹The refueling started with low tank pressures, which allowed the greatest mass flow rates to occur. As tank pressure builds, the mass flow rate declines. Also note that the vehicle used two-stage compression, while being designed to use three stages. The third stage would have boosted overall compression efficiency, but was not included due to safety and time constraints.

540 impact of overcoming engine friction and other parasitic losses on the overall
 541 compression efficiency.

542 One of the largest factors influencing efficiency is the CR of the compres-
 543 sor. With every ICE there is going to be energy used to overcome friction and
 544 pumping losses during idle operation—this is unavoidable. These losses are a
 545 function of engine speed, increasing as RPM increases. The energy used by the
 546 compressor is in addition to this idle energy. If the CR of the compressor were
 547 higher, then more of the total energy used by the ICE would go towards the
 548 compression of gas rather than idle the engine. A higher CR would thus allow
 549 the fuel storage tanks to reach higher pressures in less time and improve the
 550 overall compression efficiency. This idea is one reason why a three-stage system
 551 is more efficient than a two-stage one. The third stage uses a larger proportion
 552 of the consumed fuel energy for compression than the first or second stages.

553 A Sankey diagram highlighting the magnitude of different energy flows for
 554 the refueling process is shown below in Figure 12. As one can see, the engine is
 555 using fuel from its own storage tank to idle and compress NG. Losses primarily
 556 occur from thermal energy leaving the system and are shown on the bottom
 557

Figure 12: Sankey energy flow diagram which shows the relative energy flow magnitudes for the self-refueling NGV.

558 Figure 13 presents the finished product, a working NGV fueled by residential
 559 NG compressed using the vehicle's own engine. This vehicle was successfully
 560 driven over 161 km [100 miles], averaging 5.1 km/L [12 MPGGE].

Figure 13: Photo of the self-refueling natural gas vehicle right before going on a drive.

6. Conclusions

A mode switching natural gas SI engine was operated on a dynamometer test stand and then installed in a vehicle. The key points from experimental studies are given below.

- The engine can compress air and combust in the same cylinder, but at different times. Air was compressed, using three stages, to >138 bar [2,000 psig] in an 11.8 GGE [140 L] tank in 150 minutes.
- NG was compressed by the same engine on the test stand using a simplified two stage system. In this case, the pressure reached was 24.8 bar [360 psig] in 40 minutes.
- The engine was installed in a roadworthy vehicle and driven on public streets using its own compressed natural gas for 161 km [100 miles] averaging 5.1 km/L [12 MPGGE]. On the vehicle, the integral compressor had an approximate refueling efficiency of 70%, with an electrical-equivalent parasitic load of 12.6%.

Much of the work done in the laboratory was focused on simply proving the concept of a bimodal engine and developing the prototype. Safety was the number one concern, not minimizing sources of pressure loss or maximizing the compressor efficiency. This is why only two stages were used for NG compression experiments. One way that the refueling rate could be vastly improved is to increase CR of the compressor cylinder by reducing the dead volume between the outlet check valves of the compressor and the entrance to the cylinder (callout

583 B in Figure 1). This would greatly increase the mass flow rate into the tanks
584 and allow higher pressures to be reached in Tank 1. The maximum pressure
585 ever achieved in Tank 1 was about 6.55 bar [95 psig] compressing air. If this
586 limit was doubled or tripled, then the main high pressure storage tank would
587 be capable of reaching the U.S. standard NGV CNG tank pressure of 248 bar
588 [3600 psig] in less time.

589 Residential NG is delivered to the consumer at pressures of less than 0.069
590 bar [1 psig] (typically 0.017 bar [1/4 psig]). Commercial consumers may receive
591 gas at 0.345 bar [5 psig] or higher. The low pressure of the gas source is one
592 of the most limiting factors in the refueling system since the highest possible
593 pressure that can be achieved at TDC of a compression stroke is largely dictated
594 by the inlet pressure. Thus, the greater the NG source pressure, the higher the
595 pressure that can be reached from one stage of compression. If higher pressure
596 NG from the utility is not available, one thought to improve the source gas
597 pressure is to incorporate some kind of blower or supercharger to take the low-
598 pressure residential NG and boost the pressure to 1.38 or 2.07 bar [20 or 30
599 psig] before feeding the gas into the ICE reciprocating compressor. This will
600 practically eliminate the need for one stage of compression and improve the
601 refueling rate.

602 Another way that refueling performance could be significantly improved is
603 by minimizing the intermediate tubing volume between the engine, the three-
604 way valves (which control staging), and the tanks, while maintaining a large
605 flow area. Ideally, the tanks and three-way valves would have been right next
606 to the compressor cylinder, but that was not feasible. One possibility is to use
607 two or more engine cylinders for compression, thereby eliminating the need for
608 three separate stages. Using staging tanks introduces significant flow losses,
609 so removing the two intermediate staging tanks would significantly improve
610 compression performance. The next design iteration of this engine will address
611 these inefficiencies. Ultimately, further development of this technology, and
612 NGV infrastructure in general, is highly dependent upon future NG and oil
613 prices.

614 Acknowledgments

615 This work was supported by the Advanced Research Projects Agency-Energy
616 (ARPA-E), U.S. Department of Energy, under Award Number DE-AR0000259
617 and was also made possible with support of the Oregon State University Venture
618 Development Fund. Czero, Inc. contributed significantly to the preparation of
619 this experimental testing.

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