WGCPARIS2015
WORLD GAS CONFERENCE
"GROWING TOGETHER TOWARDS A FRIENDLY PLANET"

26th World Gas Conference | 1-5 June 2015 | Paris, France

# Development of an Inexpensive Free Piston Linear Motor Compressor

Jason Stair
Engineering Project Manager
Gas Technology Institute
USA

## Table of Contents

## 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.

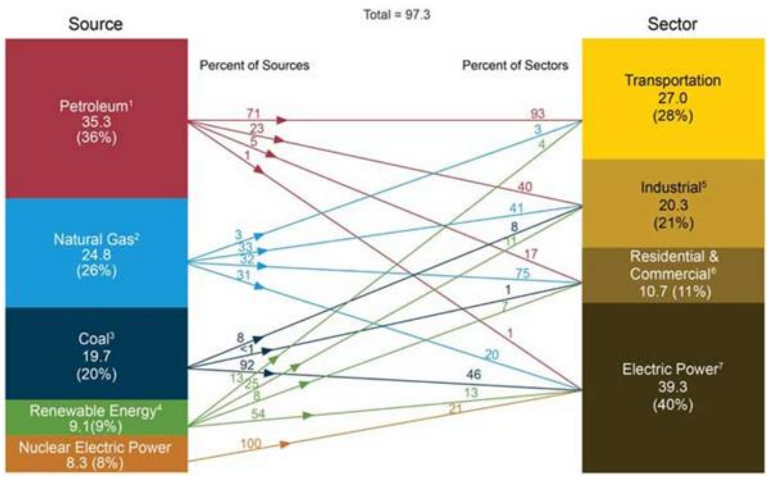


Figure 1. U.S. energy consumption by source and sector.

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.

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 fastest way to breakeven on the added cost of the vehicle 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 was $3.51, while the average cost of a gasoline gallon equivalent (GGE) of natural gas purchased at a public station was $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. 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.

Recently, oil prices have dropped substantially, significantly reducing the advantage of choosing natural gas over gasoline or diesel. Some might think this price drop will kill the NGV market; however, reality has shown that this is not the case. CNG still maintains a price advantage over gasoline and diesel, meaning that the lowest oil prices seen in recent history are still nowhere near the energy cost of natural gas in the US. In addition, the rapid decrease in oil price shows that oil can be very volatile over a short timeframe, whereas the natural gas prices are expected to remain relatively flat over the next 10 to 20 years. Vehicles are a relatively long term investment, and many fleet operators understand this, and are still choosing to convert to natural gas.

## Aim

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, making those 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 1 gasoline 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 between maintenance. 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.

## Methods

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 30 months into the project, and has completed the preliminary design, detailed design, component validation, fabrication, and is currently testing the FPLMC.

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.

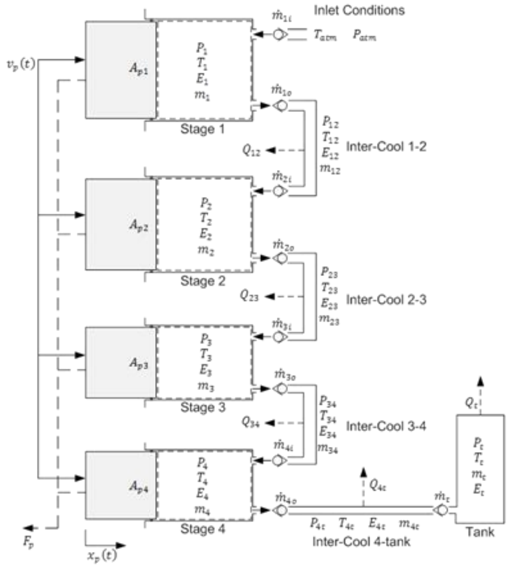


Figure 2. Compressor model schematic.

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

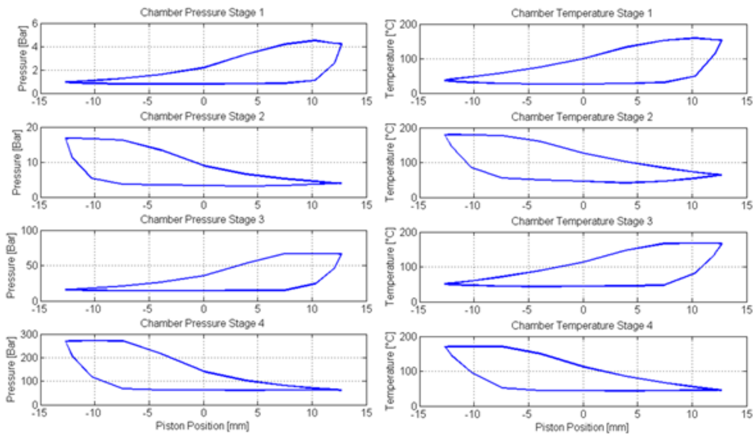


Figure 3. Pressure (left) and temperature (right) cycles of each stage from the dynamic model.

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

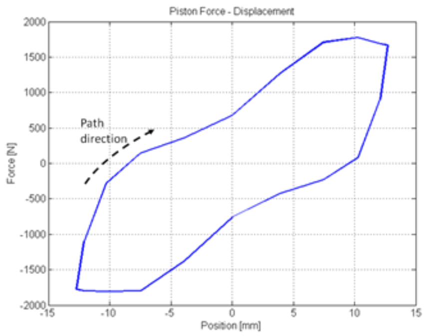


Figure 4. Net force-displacement curve for free piston compressor design.

effective area of the valve being acted on by the pressure, and the opposing downstream back pressure and spring force.

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.

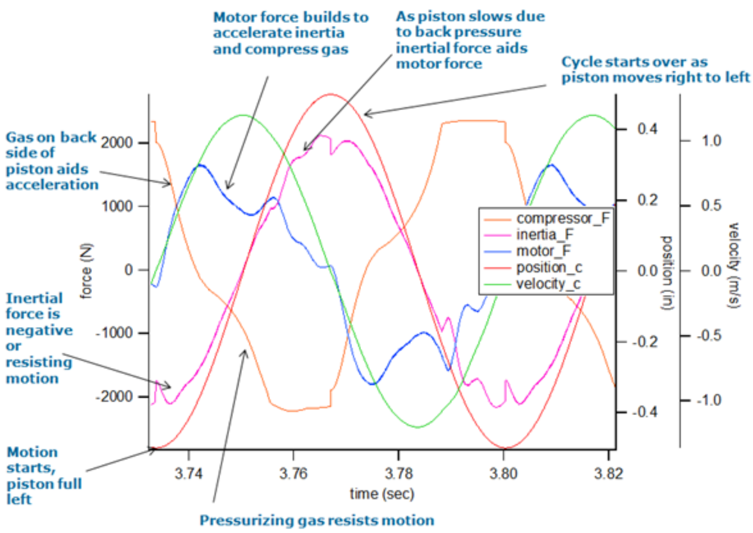


Figure 5. Compressor transient dynamics from simulation.

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 of the transient dynamics of the system. The motor force versus time curve highlights the challenge of controlling the motor.

The team modelled, simulated, and downselected six different motor topologies in order to identify the best possible design for the FPLMC. The motor topologies were all modelled 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.

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.

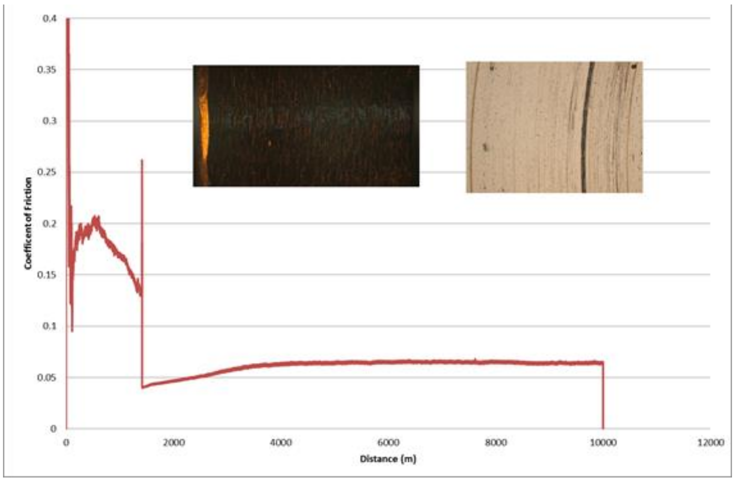


Figure 6. Example of tribo. testing and measured friction coefficient of a promising seal material.

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 each pressure cycle without chattering, and closed before the pressure in the system reversed and forced them shut.

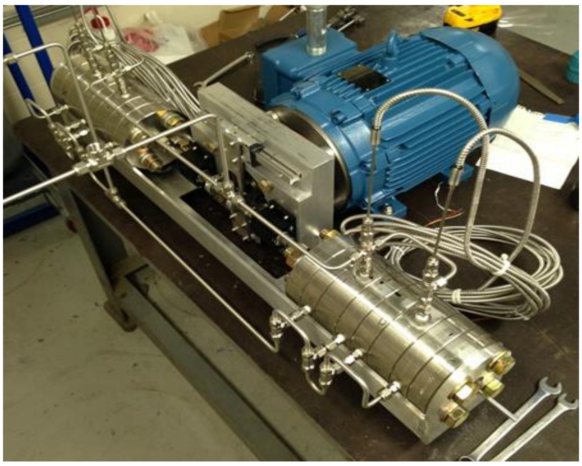

Figure 7. High pressure seal test rig.

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 modelled 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 modelling 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. Fabrication was completed towards the end of 2014, and the linear motor and compressor were tested independently to validate their performance.

## Results

The linear motor and compressor were each tested independently in order to verify their performance. The linear motor was tested against springs that simulated the forces that would be seen in the compressor. These tests uncovered some electrical noise issues, as well as some design issues with the motor itself, but ultimately the team was able to successfully test the motor and controls at full stroke and frequency. In parallel, the compressor was being tested using a scotch yoke that simulated the sinusoidal path the piston would take in the actual device. The compressor was tested using first nitrogen, then natural gas. The compressor was first tested using a very small volume attached to a back pressure regulator. These tests proved that the compressor worked, and was able to reach full pressure. After the initial validation, the compressor was tested by successfully filling a 4 GGE cylinder from empty to 3600 psi. With this final test it was apparent that the independent systems worked, and could be coupled together.

The compressor and linear motor components were finally mated together in December of 2014. The full system was tested using nitrogen and an adjustable back pressure regulator that was used to build up to a target pressure. The testing was done slow, starting with a partial stroke and frequency, and only compressing up to a few hundred psi. Over about two days, these parameters were increased until the compressor was operating at the full 1 inch stroke at 15 Hz, and compressed gas up to 3600 psi. With this test the linear compressor was finally deemed successful.

The successfully tested free piston linear compressor is currently being integrated into a fully instrumented test loop that will allow it to run continuously in order to test the performance and durability of the whole system. The test loop will record the flow and power, as well as the temperature and pressure in and out of every compression stage. This data will allow the team to fully characterize the performance of the system while filling a vessel, or running at a constant pressure. This testing should also help the team to identify any design issues that will need to be improved for the beta unit.

## Conclusions

The free piston linear compressor was successfully tested up to full pressure with nitrogen. While the system still needs to be tested using natural gas, the nitrogen tests prove that the design works as intended, and simply needs to be refined into a commercial product. The testing to date also verifies that the compressor can effectively be controlled with a reduced stroke and/or frequency, resulting in a reduced flow and power consumption. This could be very useful for alternative applications such as variable flow refrigeration that is used to adjust the cooling capacity with demand. The compressor has also been successfully started at full pressure, meaning that a blowdown tank, which is frequently required for reciprocating compressors, is not required.

The team is excited about the prospects and applications of this technology, and is currently looking for commercial partners to help develop a commercial product.

## References

"CNG National Average." Average Prices Across the United States. 2014. CNGnow. Web. 29 August 2014. < http://www.cngnow.com/average-cng-prices/pages/default.aspx>

"Domestic Consumption of Transportation Energy by Mode and Fuel Type." Transportation Energy Data Book. 2012. Oak Ridge National Laboratory. Web. 29 August 2014. <http://cta.ornl.gov/data/chapter2.shtml>

"Natural Gas Prices." Natural Gas. 2014. U.S. Energy Information Administation. Web. 29 August 2014. < http://www.eia.gov/dnav/ng/ng_pri_sum_dcu_nus_a.htm>

"Press Kit." 2014. Natural Gas Vehicle for America. Web. 29 August 2014. <https://www.ngvamerica.org/media-center/presskit/>

"Primary Energy Consumption by Source and Sector." Annual Energy Review. 2011. U.S. Energy Information Administration. Web. 29 August 2014. <http://www.eia.gov/totalenergy/data/annual/pdf/sec2_3.pdf>

"Weekly Retail Gasoline and Diesel Prices." Petroleum and Other Liquids. 2014. U.S. Energy Information Administration. Web. 29 August 2014. <http://www.eia.gov/dnav/pet/pet_pri_gnd_dcus_nus_a.htm>