

THESIS

DESIGN CONSIDERATIONS FOR AN ENGINE-INTEGRATED RECIPROCATING  
NATURAL GAS COMPRESSOR

Submitted by

Mohammad Malakoutirad

Department of Mechanical Engineering

In partial fulfillment of the requirements

For the Degree of Master of Science

Colorado State University

Fort Collins, Colorado

Fall 2014

Master's Committee:

Advisor: Thomas H. Bradley

Peter Young  
Daniel Olsen

Copyright by Mohammad Malakoutirad 2014

All Rights Reserved

## ABSTRACT

### DESIGN CONSIDERATIONS FOR AN ENGINE-INTEGRATED RECIPROCATING NATURAL GAS COMPRESSOR

This thesis presents the development of an engine retrofit concept to turn a ICE vehicle's engine into a compressor for convenient natural gas refueling, as opposed to building a smaller secondary standalone unit. More specifically, this project seeks to outfit an internal combustion engine (ICE) to serve the dual purposes of providing vehicle propulsion and compression for natural gas refueling with minimal hardware substitution.

The principal objective of this thesis is to describe and analyze the dynamic and thermal design considerations for an automotive engine-integrated reciprocating natural gas (NG) compressor. The purpose of this compressor is to pressurize storage tanks in NG vehicles from a low-pressure NG source by using one of the cylinders in an engine as the compressor. The engine-integrated compressor is developed by making minor changes to a 5.9 liter displacement diesel-cycle automotive engine. In this design, a small tank and its requisite valving are added to the engine as an intermediate storage tank to enable a single compressor cylinder to perform two-stage compression. The resulting pressure in the compressor cylinder and storage tank is 25 MPa, equivalent to the storage and delivery pressure of conventional compressed NG delivery systems. The dynamic simulation results show that the high cylinder pressures required for the compression process create reaction torques on the crankshaft, but do not generate abnormal rotational speed oscillations. The thermal simulation results show that the temperature of the storage tank and engine increases over the safety temperature of the NG unless an active thermal management system is developed to cool the NG before it is admitted to the storage tanks.

Results are then translated into vehicle-level operating costs and petroleum consumption for a dual-fuel NG-diesel vehicle.

## ACKNOWLEDGMENT

The presented work in this thesis would be done without Dr. Thomas Bradley support during the path of my research and education. I would like to express my gratitude to him for his help and support during my education.

I wish to express my appreciation to my family, Yasaman, Alireza, Zohreh, Hengameh, Hasti, and Ali who have always been supportive of me during my life and education.

Finally, I wish to express my gratitude to my colleagues and friends, especially Austin Leischer, Justin Wagner, Jake Bucher, Shawn Salisbury and Benjamin Geller, who have been helpful during my education and my study.

Thank you,

Mohammad Malakoutirad

## TABLE OF CONTENTS

ABSTRACT.....	ii
ACKNOWLEDGMENT.....	iv
TABLE OF CONTENTS.....	v
LIST OF TABLES.....	vii
LIST OF FIGURES.....	viii
1 INTRODUCTION.....	1
1.1 Natural Gas Overview.....	1
1.2 Transportation Emissions Overview.....	2
1.3 Compressed Natural Gas Vehicle Overview.....	2
1.4 Importance of Localized CNG Fuel Infrastructure.....	3
1.5 Thesis Objective.....	4
1.6 Research Questions.....	5
1.6.1 Research Question 1.....	5
1.6.2 Research Question 2.....	5
1.6.3 Research Question 3.....	6
1 STRUCTURAL AND FUNCTIONAL DESCRIPTION AND ANALYSIS.....	7
2.1 Structural Description.....	7
2.2 In-cylinder Natural Gas Compression Process.....	9
2.3 Surge Tank Pressurization Process.....	10
2.4 Storage Tank Pressurization Process.....	12
2 EXPLICIT DYNAMICS MODEL DESCRIPTION AND RESULTS.....	15
3.1 Model Description.....	15
3.2 Pressurizing Surge Tank.....	16
3.3 Crank Shaft Torque Results.....	19
3 DETAIL DESIGN CONSIDERATIONS.....	22
4.1 Model Description.....	22
4.1 Dynamic Torques and Rotational Energy Requirements.....	22
4.2 Thermal Considerations.....	26
4.2.1 Adiabatic Compression Model.....	26

4.2.2 Convective Heat Transfer Model.....	29
4.3 Summary of Results.....	32
4 NATURAL GAS UTILITY FACTOR.....	34
5.1 Natural Gas Utility Factor Overview.....	34
5.2 Utility Factor Definition for Dual Fuel Vehicles.....	35
5.3 Distance Utility factor Results.....	38
5.4 Yearly Fuel Cost.....	41
6 DISCUSSION.....	44
6.1 Research Question 1.....	44
6.2 Research Question 2.....	45
6.3 Research Question 3.....	45
7 CONCLUSION.....	47
8 REFERENCES.....	49
9 APPENDIX.....	51
9.1 Controller.....	51
9.2 Model Development.....	52

## LIST OF TABLES

Table 1. Typical Composition of Natural Gas on Terms of Mole and Mass Fraction [2].....	1
Table 2. Summary of the Survey Data.....	39

## LIST OF FIGURES

Figure 1. Schematic of Engine-integrated Reciprocating Natural Gas Compressor .....	9
Figure 2. Pressure inside the Compression Cylinder During the Compression Process. Figure A Shows the Compression Cylinder Pressure During the First Few Seconds of the Compression Process. Figure B Shows the Compression Cylinder Pressure During the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa. ....	10
Figure 3 . Pressure inside the Surge Tank During the Compression Process. Figure A Shows the Surge Tank Pressure During the First Few Seconds of the Compression Process. Figure B Shows the Surge Tank Pressure During the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa.....	12
Figure 4 Pressure inside the Surge Tank During the Compression Process. Figure A Shows the Storage Tank Pressure During the First Few Seconds Of the Compression Process. Figure B Shows the Storage Tank Pressure during the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa.....	13
Figure 5. Pressure of the Storage Tank during the Entire Pressurizing Process.....	14
Figure 6. Pressure inside Surge Tank during the Compression Process.....	17
Figure 7. Compression Cylinder Pressure during Pressurizing Process of Surge Tank and Storage Tank .....	18
Figure 8. Pressure inside the Cylinder, Surge Tank, and Storage Tank during the Compression Process. ....	19
Figure 9. Crankshaft Torque and Piston Pressure in Explicit Model (Positive Torque Represents Compression and Negative Torque Represents Decompression in Cylinder).....	20
Figure 10. Developed Simulink Model for the Detail Design Consideration.....	22
Figure 11 Crankshaft Torque and Piston Pressure (Positive Torque Represents Compression and Negative Torque Represents Decompression in Cylinder).....	24
Figure 12. Crankshaft Velocity Fluctuation during the Compression Process .....	26
Figure 13. Temperature within the Compression Cylinder under the Adiabatic Compression Conditions .....	27
Figure 14. NG Temperature within the Surge Tank under Adiabatic Compression Conditions ..	28
Figure 15. NG Temperature within the Storage Tank under the Adiabatic Compression Conditions .....	29
Figure 16. NG Temperature within the Compression Cylinder under Compression Conditions with Heat Transfer to Engine Cooling Jacket .....	30
Figure 17. NG Temperature within the Storage Tank under Compression Conditions With Heat Transfer Of Tank-To-Air Cooling .....	31
Figure 18. NG Temperature within the Surge Tank Cylinder under Compression Conditions with Heat Transfer from Surge Tank-to-Air Cooling .....	32
Figure 19. J2841 Distance Utility Factor Based on the NREL Research Fleet DNA .....	40
Figure 20. Yearly Fuel Cost of Each Type of Vehicle .....	42
Figure 21. Weighted Daily Diesel Gallon Equivalent Fuel Consumption for Diesel.....	43
Figure 22. Compressor Controller and Input and Output Signals into the controller.....	51
Figure 23 .Controller Operation.....	52
Figure 24. Heat and Mass Flow Measurements in the Model .....	53
Figure 25. Pressure and Temperature Measurements in Model .....	53

# 1 INTRODUCTION

## 1.1 Natural Gas Overview

Natural gas (NG) is produced from a gas well or it is produced as a by-product of crude oil production. NG is made mostly by methane, however; it contains other components such as ethane, propane, nitrogen, and some other components [1], which are presented in Table 1.

**Table 1. Typical Composition of Natural Gas on Terms of Mole and Mass Fraction [2]**

<b>Natural Gas Composition</b>	<b>Mole Fraction</b>	<b>Mass Fraction</b>
Methane (CH <sub>4</sub> )	0.9229	0.8437
Ethane (C <sub>2</sub> H <sub>6</sub> )	0.0360	0.0623
Propane (C <sub>3</sub> H <sub>8</sub> )	0.0080	0.0206
Butane (C <sub>4</sub> H <sub>10</sub> )	0.0029	0.0099
Pentane (C <sub>5</sub> H <sub>12</sub> )	0.0013	0.0053
Hexane (C <sub>6</sub> H <sub>14</sub> )	0.0008	0.0039
Carbon dioxide (CO <sub>2</sub> )	0.0100	0.0252
Nitrogen (2)	0.0180	0.0289
Water (H <sub>2</sub> O)	0.0001	0.0001
<b>Total</b>	<b>1</b>	<b>1</b>

According to Semin, R.A. Bakar and A.R. Ismail, NG has five advantages over petroleum fuels: 1. It is cheaper than gasoline and diesel fuel 2. The combustion of NG produces low air emissions 3. Its combustion produces less greenhouse gas emissions 4. Its use extends petroleum supplies 5. It is available in large quantities in North America [1]

NG can be stored and compressed at high pressures and be used as primary fuel in vehicles. However, unlike gasoline and diesel fuel, NG needs to be stored and compressed in large volume storage tanks, which reduces the cargo space. The pressure of the NG storage tank ranges between 20 MPa and 25 MPa, depending on the size of the tank and driving range expectations.

### **1.2 Transportation Emissions Overview**

Transportation is a major contributor to the productivity of the US economy, but the air pollution costs and fuel import costs of a petroleum-fueled transportation system are high. Transportation produces 79 % of total CO emission, 50 % of total nitrogen oxide emissions, 26% of greenhouse gas (GHG) emissions, and 42% of the total volatile organic compound emissions in the U.S [3]. Although regional air quality, and specific GHG emissions of transportation have been improving with advancements in transportation technologies, many regions of the US do not meet federal requirements for air quality and total GHG emissions continue to increase [3]. Transportation fueled by NG engines has been demonstrated to produce fewer CO<sub>2</sub>, CO, and HC emission than gasoline vehicles [4], and it is a near-term feasible technology for improving the sustainability of the transportation sector [5].

### **1.3 Compressed Natural Gas Vehicle Overview**

Compressed NG powered vehicles (CNGVs) have not become mass-market vehicles in the US for a variety of reasons. Consumer and fleet adoption of CNGVs are hampered by sparse fueling infrastructure, limited driving range, and the incremental cost of NG vehicles relative to conventional petroleum vehicles. NG fueling infrastructure primarily consists of a sparse network of non-public fueling stations. There are approximately 1300 NG fueling stations in the US [6] as compared to over 150,000 gasoline stations. The scarcity of NG fueling infrastructure is exacerbated by the limited driving range available from conventional CNGVs. For example,

the driving range of a Chevrolet Cavalier dual-fuel compressed natural gas (CNG) vehicle is 110 miles, approximately 1/3 of the driving range of a conventional vehicle. Finally, NG powered vehicles have a several thousand dollar<sup>1</sup> higher purchase price than conventional petroleum powered vehicles. CNGVs are therefore subject to the conundrum in which inadequate infrastructure limits the market for the CNGVs, and the limited market for the vehicles discourages investment in a costly CNGV fueling infrastructure.

#### **1.4 Importance of Localized CNG Fuel Infrastructure**

Many researchers believe that the solution to the NG infrastructure and cost conundrum is the development of localized (as opposed to centralized) business or home NG compressor and refueling systems. This localized refueling system would compress the low pressure NG that is delivered to many businesses and homes through the preexisting, low-pressure NG infrastructure and would transfer the compressed NG to the vehicle. This concept has several advantages over the more conventional centralized CNG infrastructure solutions. First, the allocation of fueling infrastructure costs to a single vehicle means that the cost and availability of the fueling infrastructure is independent of the state of CNG vehicle market penetration. Second, by placing fueling infrastructure at the locations where the vehicle regularly parks, refueling can be performed daily or more often without inconvenience to the driver. The challenge that must be overcome to improve the feasibility of localized refueling infrastructure is its cost. There exist a variety of localized (home) NG compression systems that will fuel a CNG vehicle by locally compressing the NG available from low-pressure residential-type infrastructure. These systems

---

<sup>1</sup> The Natural-gas alternative. The pros and cons of buying a CNG Powered car.  
( <http://www.consumerreports.org/cro/2012/03/the-natural-gas-alternative/index.htm> )

have a high incremental cost because they are not subject to mass-production, they have tight tolerances, high parts count, and they require high-power, high-torque power supplies.<sup>2</sup>

### **1.5 Thesis Objective**

Conventionally, compressed NG (CNG) vehicles are refueled using a high-cost, centralized, and sparse network of CNG fueling stations designed for the use of fleet customers. An engine-integrated reciprocating NG compressor has the capability to disrupt the incumbent CNG market by enabling the use of NG for personal transportation, fueled at home, from the preexisting low-pressure NG infrastructure, at low parts count, using conventional components, and therefore at low incremental costs.

This thesis proposes the development of an engine-integrated, low-parts count, reconfigurable NG compressor as an innovative technology to address the cost barrier to localized NG compression and fueling. This concept allows the engine controller to reconfigure one of the engines' combustion cylinders to function as a reciprocating multi-stage compressor that directly refuels the vehicle's CNG tank using the low-pressure residential NG infrastructure. Such a system allows for the advantages of home CNG refueling while minimizing the incremental cost of materials, components, and systems. To characterize this technology, this thesis describes the structural and functional description which includes the mechanical structure of the compressor system and NG compression process, and provides a set of thermodynamic and mechanics models and results to characterize the design requirements of the system which includes dynamic torque and crankshaft velocity evaluation. Finally, in this thesis, heavy duty commercial natural gas-diesel dual-fuel vehicles are evaluated to better understand the effects of localized NG refueling infrastructure on fuel cost and fuel economy in these type of vehicles.

---

<sup>2</sup> BRC FuelMaker, <http://www.brcfuelmaker.it/eng/casa/phil.asp>

## **1.6 Research Questions**

During the concept development and proposal tasks of this project, a modeling statement of work was constructed with the objective of understanding the design requirements of the engine/compressor system. Two main research questions were proposed to help to understand two main potential failure mechanisms for the system.

### **1.6.1 Research Question 1**

The first potential failure mechanism involves the rotational dynamics of the engine crankshaft. During the compression cycle, the engine will experience cylinder pressures of up to 25 MPa (significantly higher than normal combustion pressures). The effects of this dynamic loading on the engine crankshaft dynamics are unknown, leading to the following research question.

If high torque is applied to the crankshaft due to the high pressure in the compression cylinder, does the flywheel of the selected engine need to be resized to avoid velocity fluctuations on the crankshaft?

### **1.6.2 Research Question 2**

The second potential failure mechanism involves the thermal behavior of the engine and compressed gas during the NG compression cycle. In this case, the in-cylinder temperatures, and gas storage temperatures are constrained by the material properties of the compression system components. The thermal behavior of the system was unknown, leading to the following research question.

Does compressing NG inside the compression cylinder raise the temperature over the regulation limit temperature? If yes, what solution can be provided to avoid this temperature rise?

### **1.6.3 Research Question 3**

Finally, the conditions of use of the CNG vehicle were unknown in its heavy-duty vehicle application. The dual-fuel concept has been proposed as a means to get maximum advantage from the home refueling concept explored in this study, but the framework for understanding the cost of operation, fuel economy and conditions of use of a dual-fuel home refueled vehicle has not been developed. This leads to the final research question.

How sensitive is the fuel consumption and fueling cost of a dual-fuel commercial vehicle to NG powered range?

## 2.1 Structural Description

This section describes the vehicle components and mechanical systems that make up the engine-integrated compressor system. The primary components of the engine-integrated compressor system are the CNG tanks, the CNG internal combustion engine (ICE), the Surge Tank, and the system of control valves, as shown in Figure 1.

CNGVs store NG in high-pressure on-board tanks that have rated pressures of 20-25 MPa [7] conventionally; these tanks are filled from a high-pressure connection to a high-pressure CNG refueling system. According to Skycng and Cleanvehicle, there are four main types of the CNG storage tanks. These tanks are designed in different sizes with various materials, and they vary in price and size. Aluminum or steel are most common, as they are inexpensive, but heavy. In hoop-wrapped composite tanks, a metal liner is strengthened by a composite wrapping of glass or carbon fiber. In fully wrapped composite tanks, the metal liner takes very little stress and the majority of loads are taken by the composite material. Fully wrapped non-metallic tanks are made of a plastic liner, fully wrapped by carbon fiber<sup>3</sup>. In this thesis, the steel storage tank has been assumed.

One aspect of this thesis investigates the potential of filling the on-board CNG tank by compressing NG from the fuel supply line pressure (~1.02 bar absolute) to tank pressure using an engine-integrated compressor. Fueling the on-board tank using low-pressure NG obviates the need for high-pressure CNG infrastructure.

---

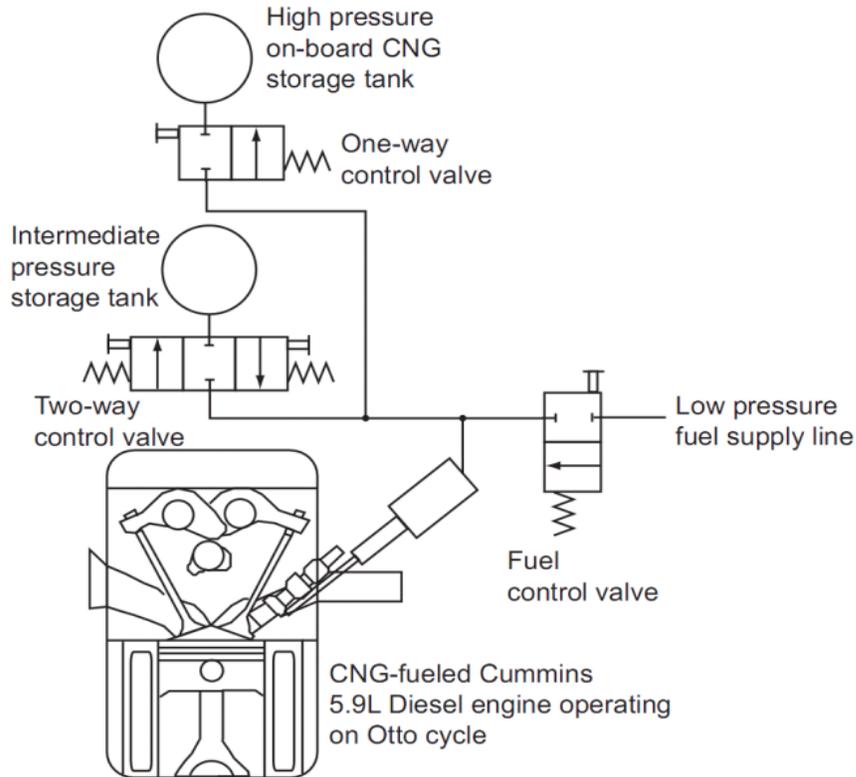
<sup>3</sup> <http://skycng.com/cngcylinders.php>" & "<http://www.cleanvehicle.org/technology/CNGCylinderDesignandSafety.pdf>

CNG-powered engines are similar in structure and control to conventional ICEs. There are three primary ways to fuel engines using NG. The first is a dedicated NG engine; these engines run on only NG. These engines can be designed and made for NGVs, or they can be derived from existing diesel engines [1]. Second is a dual fuel engine, these engines mix 50-75 percent NG with diesel, and they can run on either this mix of diesel and NG, or on diesel fuel only [1]. The third type of the NG engines are bi-fuel which engine can run on either diesel or NG.

For this design a 6 cylinder, inline, 5.9 liter heavy-duty diesel Cummins engine is selected to be converted to NG powered engine. Modified NG injectors and emissions control equipment is assumed to be used for converting conventional ICEs to NG-powered ICEs. In this thesis, the modification of the engine is not evaluated in detail. A modified diesel-cycle engine is chosen for this concept description to allow the mechanical components of the engine to tolerate repeated in-cylinder pressures of up to 25 MPa [8]. In this concept, by using the internal components of the ICE as a reciprocating compressor, we can reduce the parts count and therefore manufacturing costs of the integrated compressor and engine systems. A reciprocating compression cylinder is powered mechanically from the engine crankshaft as the other 5 cylinders perform conventional ICE operations.

Conventional diesel engines have compression ratios (CR) between 14 and 22 [9], primarily limited by clearance volume requirements and diminishing returns on engine efficiency with increasing CR. Because of this CR limitation, the pressurization of the NG from line pressure to tank pressure must be performed in multiple stages. In this concept we use “Surge Tanks” to store intermediate pressure NG between the stages of compression to allow for multi-stage compression of the NG to storage pressure using a single compression piston. The surge

tanks are envisioned as rigid, metallic cylinders that are connected to the cylinder with a system of control valves. Under each stage of compression, the cylinder can achieve a compression ratio of between 14 and 22, which implies that between 2 and 3 stages of compression (and between 1 and 2 surge tanks) are required to generate a tank pressure of 25 MPa from the near ambient supply pressure.

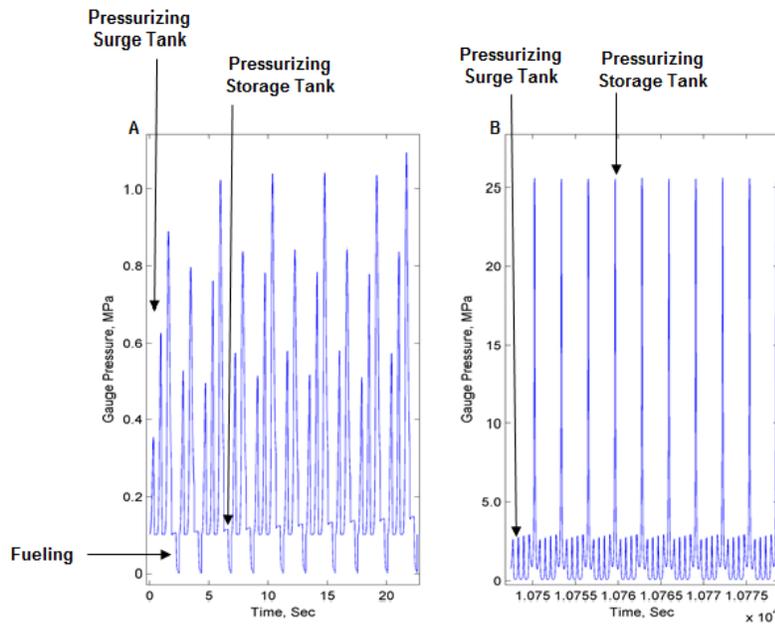


**Figure 1. Schematic of Engine-integrated Reciprocating Natural Gas Compressor**

## 2.2 In-cylinder Natural Gas Compression Process

Upon connection of the vehicle to the home NG fueling line, one of the cylinders of the ICE begins to operate as a compressor powered by the other 5 cylinders of the ICE. Low pressure gas is admitted into this compression cylinder by way of the inlet check valve. As the piston moves upwards in the cylinder, the NG is compressed. The action of the one-way valves shown in Figure 1 are such that the cylinder will only transfer pressurized NG from the

compression cylinder to Surge Tank when the in-cylinder NG pressure is greater than the pressure in the Surge Tank . As the cylinder moves past its top dead center (TDC) position, the in-cylinder pressure will decrease below the pressure in the Surge Tank, closing the one-way valve between the compression cylinder and the Surge Tank. The compression cylinder then expands the remaining in-cylinder NG until the cylinder reaches near ambient pressure, at which point low pressure gas is admitted into the compression cylinder from the home NG fueling line. The cycle repeats to provide compression of the NG from the fueling line, and to and from the surge tank, in turn. The compression cylinder pressure during the pressuring process is shown in Figure 2.



**Figure 2. Pressure inside the Compression Cylinder during the Compression Process. Figure A Shows the Compression Cylinder Pressure During the First Few Seconds of the Compression Process. Figure B Shows the Compression Cylinder Pressure During the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa.**

### 2.3 Surge Tank Pressurization Process

As described in the preceding paragraph, the NG in the compression cylinder increases in pressure as the cylinder approaches TDC, and the one-way valve between the compression

cylinder and Surge Tank moves NG from the cylinder into the Surge Tank. Details of the Surge Tank pressuring process are shown in Figure 3. For this conceptual design study, the compression of NG is simulated using a Polytropic process with a Polytropic exponent (k) of 1.3 as in Equation 1, whereas the ratio of specific heats for isentropic process is 1.4 [10].

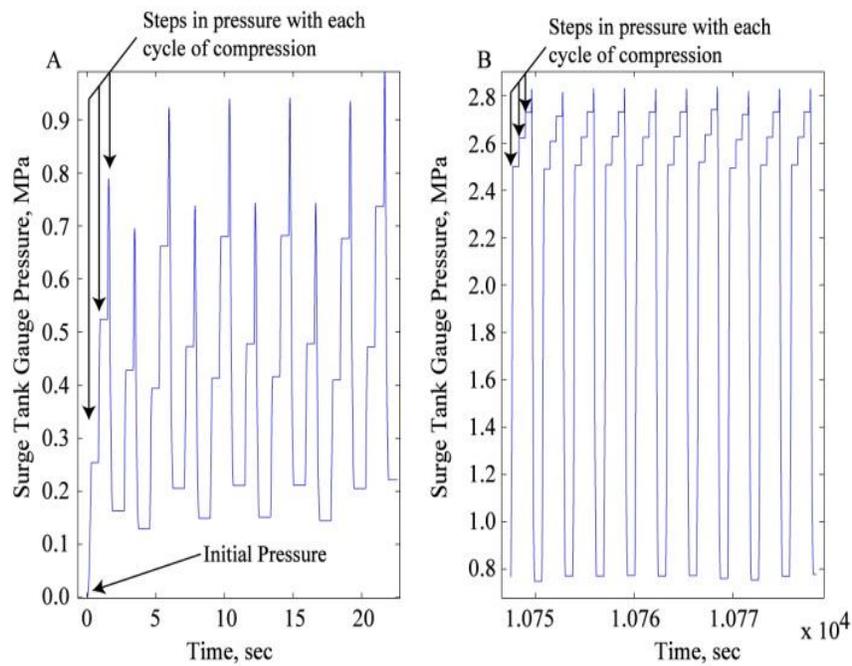
**Equation 1**

$$\frac{P_{\text{initial}}}{P_{\text{final}}} = \left(\frac{V_{\text{initial}}}{V_{\text{final}}}\right)^k = \left(\frac{1}{r_{\text{compression ratio}}}\right)^k$$

As the compression cylinder passes through TDC, the pressure in the compression cylinder begins to decrease and the one-way valve between the compression cylinder and Surge Tank closes. The cylinder expands the remaining NG until new NG is admitted to the cylinder at near-atmospheric pressure near BDC. This process repeats itself to until the pressure in the Surge Tank reaches its initial pressure setpoint (which here is assumed to be 0.7 MPa as an example). Each cycle of compression moves incrementally less mass to the Surge Tank, and increases the pressure within the Surge Tank incrementally less until the pressure in Surge Tank reaches its pressure setpoint.

As the pressure in the Surge Tank rises above its setpoint, the compression system begins to implement its second stage of compression. As the cylinder nears BDC, the control valve between the surge tank and the compression cylinder opens and the high pressure NG is transferred from the Surge Tank back to the compression cylinder. The control valve closes again when the compression cylinder reaches BDC. At this point in the example, the state of the compression cylinder is that it is at intermediate pressure and temperature and is at BDC. The compression cylinder therefore compresses from intermediate pressure as it moves from BDC and generates high pressure NG to fill the vehicle storage tank. The setpoint at which the

compression system performs this second stage compression varies between 0.7 MPa and 2.8MPa as a function of storage tank pressure. The behavior of this setpoint is visible in the simulated surge tank pressure traces shown in Figure 3. During the first seconds of compression, the surge tank will reach the setpoint pressure of 0.7MPa before it pressurizes the storage tank, thereby lowering its pressure. During the last seconds of the compression cycle, the surge tank will reach a setpoint pressure of 2.8MPa before it pressurizes the storage tank.

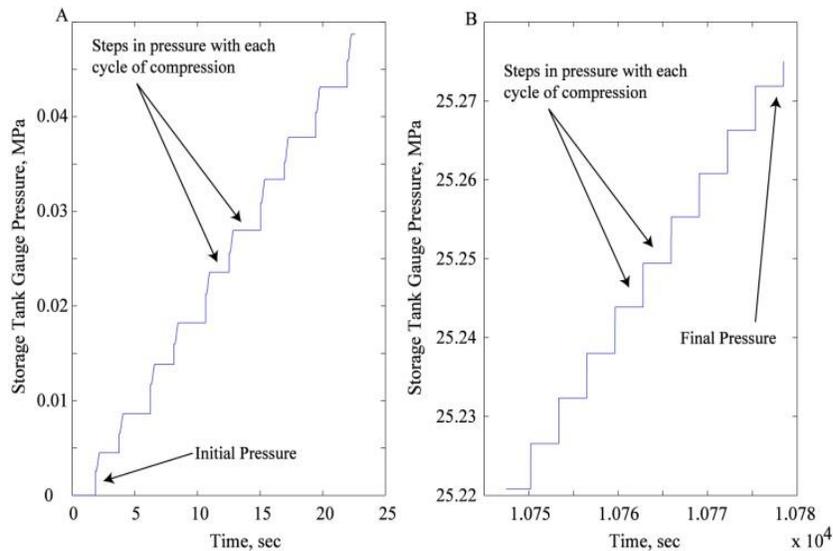


**Figure 3 . Pressure inside the Surge Tank during the Compression Process. Figure A Shows the Surge Tank Pressure During the First Few Seconds of the Compression Process. Figure B Shows the Surge Tank Pressure During the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa.**

#### 2.4 Storage Tank Pressurization Process

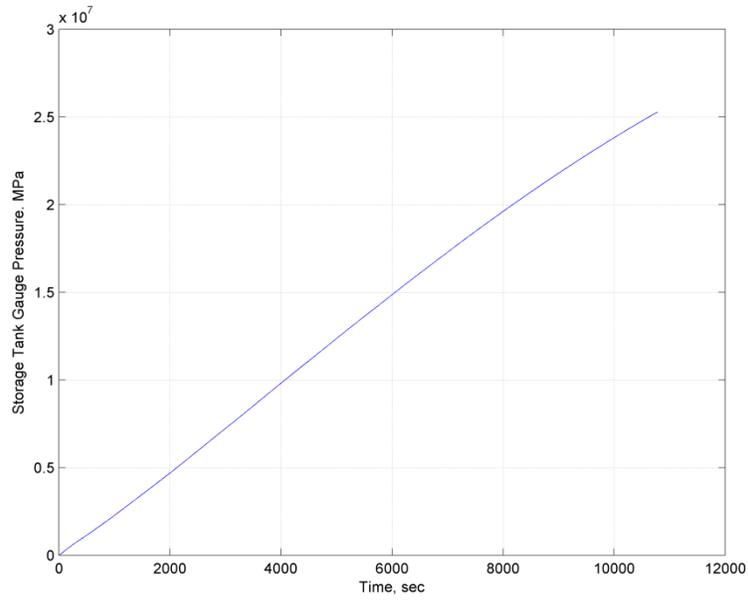
The vehicle's NG storage tank is a simple storage vessel connected to the compression cylinder using a one-way valve. This valve transfers NG to the storage tank when the pressure of the compression cylinder is greater than the pressure in the storage tank. The volume of the storage tank is much greater than the volume of the cylinder or surge tanks, so that the pressure of the storage tank builds slowly with each cycle of the multi-stage compression process towards

a 25 MPa rated pressure. The pressure of the storage tank is shown in various levels of detail in Figure 4. Early in the compression process, the storage tank is at low pressure and each cycle of pressurization increases the tank pressure by approximately 0.05 MPa. The tank continues to receive compressed NG and increase its pressure until it reaches a final pressure of 25 MPa. The pressure between the compression cylinder, the Surge Tank and the storage tank drops significantly at beginning the pressuring process; however, the pressure drop reduces by building more pressure in the Surge Tank and the storage tank



**Figure 4 Pressure inside the Surge Tank During the Compression Process. Figure A Shows the Storage Tank Pressure During the First Few Seconds Of the Compression Process. Figure B Shows the Storage Tank Pressure during the Final Seconds of the Compression Process as the Storage Tank Pressure Reaches 25 MPa.**

Figure 4 shows the pressure in the storage tank at the beginning and end of the pressuring process. In addition, in Figure 5 shows the pressure of the storage tank during the entire process of pressuring.



**Figure 5. Pressure of the Storage Tank during the Entire Pressurizing Process**

### 3.1 Model Description

The modeling of the engine integrated NG compressor was developed in two stages. The first stage refers to the explicit model which was developed to demonstrate the operation of the designed system and obtain a better understanding of the problem. This model is not as accurate as the implicit model (DAE model) developed in the second stage; however, the development of the explicit model played a significant role in improving the DAE model by revealing the weaknesses and inaccuracies in the assumptions during the design process. These assumptions were then revised for the advanced model in the second stage. In this chapter the results from the explicit model are presented.

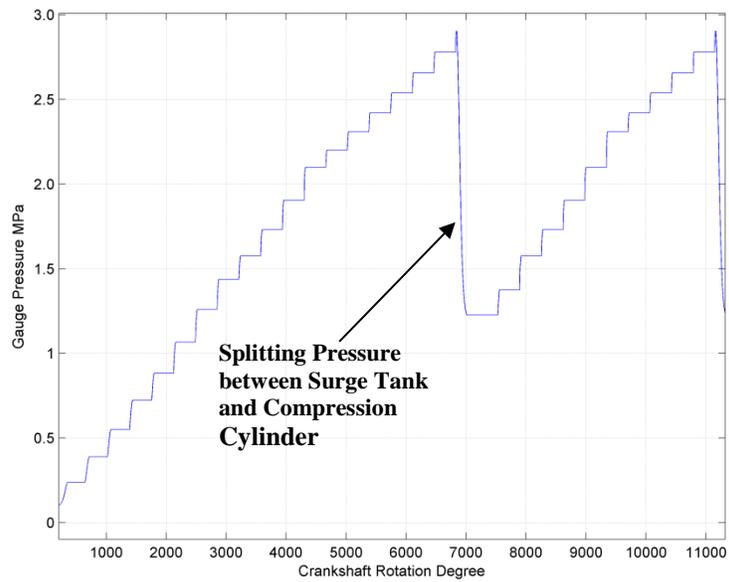
The explicit model was developed with degree-by-degree simulation of the crankshaft rotation. Euler integration was used to derive an explicit difference equation for the pressure and cylinder volume as a function of crank angle. In this model the pressurized gas is transferred from the compression cylinder to the Surge Tank and storage tank when the pressure of the NG inside the compression cylinder becomes greater than the pressure of the NG in the Surge Tank and storage tank. Using the explicit model, the piston pressure can be used to estimate the crankshaft torque. The valves between the Surge Tank, storage tank, and the compression cylinder are not simulated in this model because of the complexity of the valves' operations and their modeling. Thus, it is assumed that the valves do not affect the pressures or flow rates in the compression cylinder, Surge Tank, and storage tank. However, the omission of the valves in the investigated systems is the main source of inaccuracy in this model. Therefore, while the compression cylinder pressurizes the gas in the Surge Tank and the storage tank, the NG pressure

in both tanks are equal to NG pressure in the compression cylinder, and it is not feasible to see pressure drops across the control valves.

Because of its simple formulation, the explicit model is a fast model to run because there are no differential equations in this model, and valves are not simulated. Moreover, the other cylinders operations and the environment around the Surge Tank, storage tank, and the compression cylinder are not simulated. Additionally, heat transfer is ignored between the gas and environment in this model.

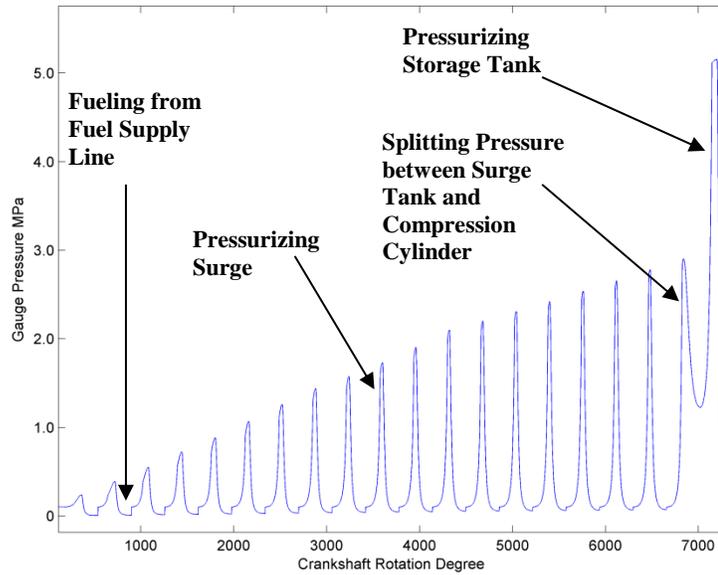
### **3.2 Pressurizing Surge Tank**

The pressure of NG in Surge Tank is shown in Figure 6. NG is pressurized in compression cylinder by compression of gas from bottom dead center to top dead center. While the gas pressure in the compression cylinder is greater than the gas pressure in the Surge Tank, the Surge Tank can be pressurized by the compression cylinder. This process continues for each revolution of the crankshaft as long as the pressure of the compression cylinder is greater than the pressure in the Surge Tank ( $P_{cyl} > P_{surge1}$ ). When the NG pressure in the compression cylinder is less than the NG pressure in the Surge Tank, the pressure of the gas in the Surge Tank stays constant until the next revolution. At each revolution, if the NG pressure in the cylinder at the bottom dead center is less than 0.1 MPa, the compression cylinder will receive NG with 0.1 MPa pressure from the low pressure fuel supply line. This process will continue until the pressure in the Surge Tank reaches a setpoint of 2.8 MPa. At this time, the Surge Tank pressure is split between the compression cylinder and the Surge Tank.



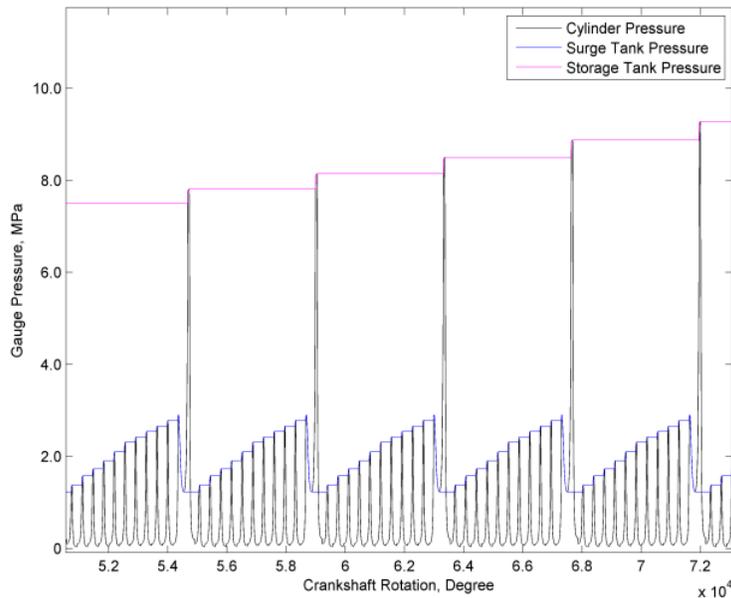
**Figure 6. Pressure inside Surge Tank during the Compression Process**

Figure 7 shows the compression cylinder pressure trace as a function of crank angle. In this picture, four different stages of pressuring are shown which are 1) fueling from the low pressure supply pipe line, 2) pressurizing the surge tank, 3) splitting pressure between compression cylinder and Surge Tank, and 4) pressurizing the storage tank.



**Figure 7. Compression Cylinder Pressure during Pressurizing Process of Surge Tank and Storage Tank**

As shown in Figure 6, after the surge tank reaches its setpoint of 2.8 MPa, the surge tank fills the compression cylinder to a pressure of 1.5 MPa (at BDC) in the example above. The compression cylinder then fills the storage tank to high pressure as the second stage of the compression process. Figure 8 shows the compression cylinder pressure trace during both stages of the compression process. Having higher pressure at the top dead center facilitates pressurizing the storage tank. This process continues at each revolution of the crankshaft rotation as long as the pressure of the compression cylinder is greater than the pressure of the storage tank ( $P_{cyl} > P_{storage}$ ). At each crankshaft revolution, the compression cylinder loses mass and pressure; therefore, it is necessary to build high pressure in the compression cylinder again. Thus, the Surge Tank has to be pressurized repeatedly until the pressure in the storage tank reaches 25 MPa.



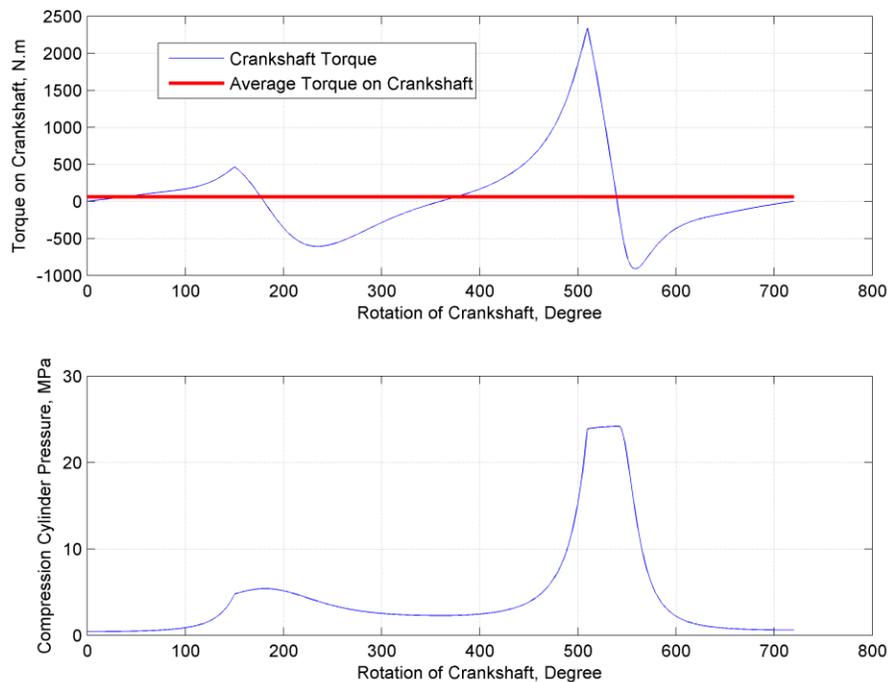
**Figure 8. Pressure inside the Cylinder, Surge Tank, and Storage Tank during the Compression Process.**

### 3.3 Crank Shaft Torque Results

The high pressure in the compression cylinder is the result of the high torque on the crankshaft, which may consequently leads to high noise and vibrations in the engine. Therefore, it is essential to determine the crankshaft torque. The explicit model is used to calculate the maximum torque on crankshaft during the pressuring process of the storage tank. Figure 9 shows the torque results at maximum pressure in the compression cylinder. In this simple model of crankshaft dynamics, the crankshaft speed is held constant. Therefore all of the torque to overcome the torque required for the compression cycle is assumed to be provided by the engine flywheel.

As it is shown in Figure 9, positive torque on the crankshaft occurs at the same time as the compression process in the compression cylinder. In other words, energy transfers from the flywheel to the crankshaft; on the other hand, negative torque represents the energy that transfers

from the crankshaft to the flywheel [11]. Moreover, the average torque is the output torque from the flywheel which is the average of the torque over 720 degree of crankshaft rotation. The maximum torque on the crankshaft in this model is 2400 N·m during the final compression of the NG in the compression cylinder, and the maximum range of the torque fluctuation is 3300 N·m, which occurs during the compression of the NG to the storage cylinder, and the expansion of the NG after the one way control valve closes.



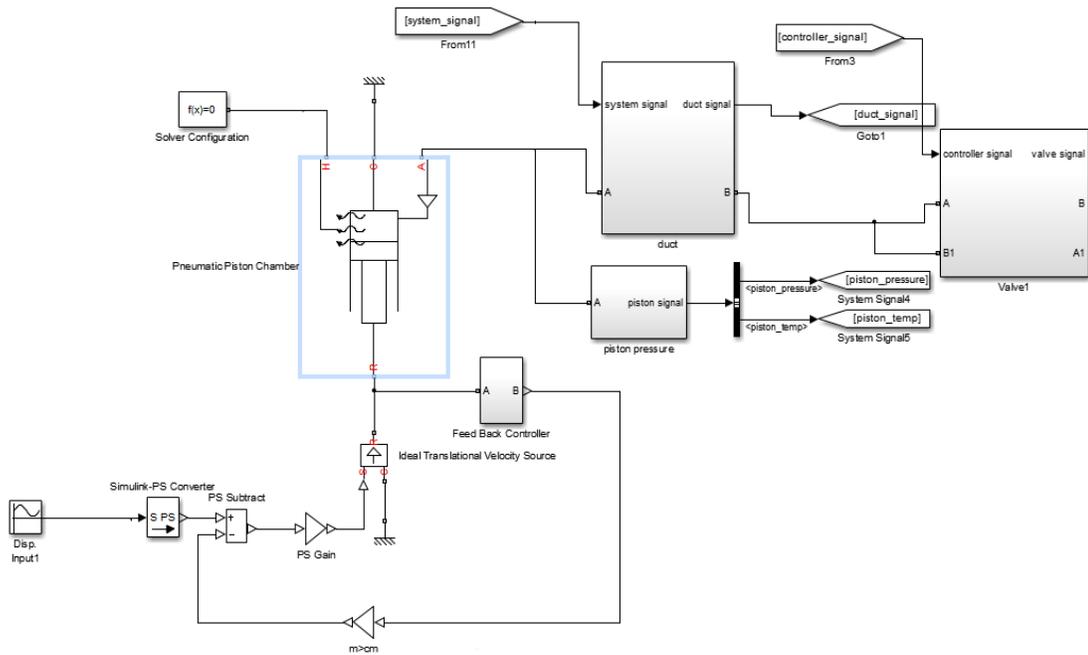
**Figure 9. Crankshaft Torque and Piston Pressure in Explicit Model (Positive Torque Represents Compression and Negative Torque Represents Decompression in Cylinder)**

Both of the torque and compression cylinder pressure figures display sharp edges because the pressure in the compression cylinder is built as a function of crankshaft rotational degree creating unsmooth pressure data and consequently unsmooth torque data. Therefore, each pressurizing process can take up to a degree of crankshaft rotation; however, each stage can also take less than a degree to occur, and this creates a source of inaccuracy in the explicit model

leading to unsmooth edges in the shape of the figures of compression cylinder pressure and crankshaft torque.

**4.1 Model Description**

This section represents the results of detail design analysis which is developed in MATLAB & Simulink and also describes the variable time domain model which solves sets of DAE differential equations to model the rotational and thermal state of the engine/compressor system. This modeling format allows for a more detailed investigation for the characteristics, design tradeoffs and performance of the engine/compressor system



**Figure 10. Developed Simulink Model for the Detail Design Consideration**

**4.1 Dynamic Torques and Rotational Energy Requirements**

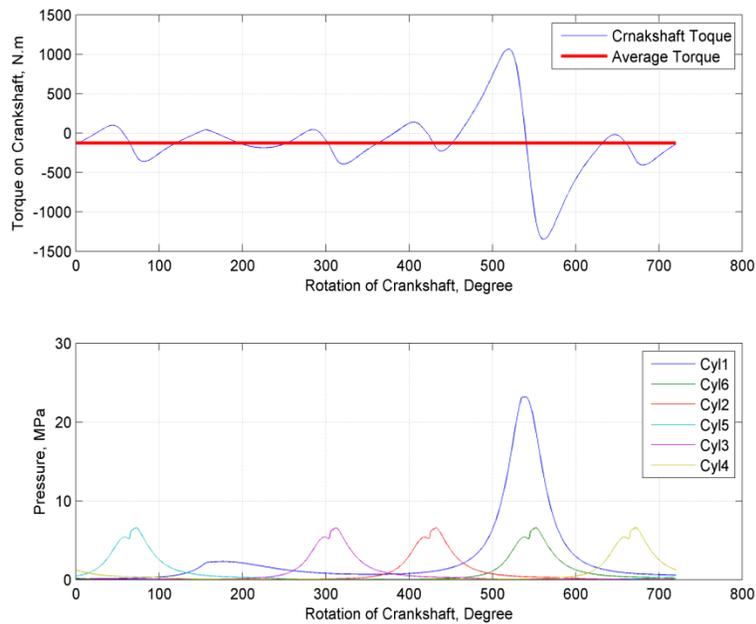
In practice, the function of this complicated system will be constrained by the detailed design considerations surrounding a few key tradeoffs among the system’s components and capabilities. In this section of the thesis, the detail of these key tradeoffs within two problem

domains that were identified through peer review, commercialization workshops, and discussion with ARP Ae project managers: dynamic torque loads and thermal considerations.

Generating high pressures in the compression cylinder requires that high torques be applied and received by the engine crankshaft which was represented in the explicit model in chapter three.

In this DAE model, the model is developed over the variable time domain which allows to open and close the valves in less than the time for one degree rotation of crankshaft, and this creates the smoother pressure fluctuation and torque calculation results. The example engine which is used in this thesis has 6 inline cylinders. There are 120 degrees phase differences between the cylinders in this example engine. Due to standard four-stroke ICE orientation, cylinders 1 and 6, cylinders 2 and 5, and cylinders 3 and 4 are respectively at the same positions, but there is 360 degrees of crank angle phase difference between pairs; however, in this study the worst case scenario is considered when combustion of cylinder 6 occurs at the same time that high pressure (25 MPa) in the compression cylinder (cylinder 1) occurs. The dynamics of applying large torques to the compression cylinder's piston, connection rod, and crankshaft journals will have important implications for engine noise, vibration and component lifetime.

One means to analyze these considerations is to model the dynamics of the engine-integrated compressor with the objective of quantifying the magnitude of any increase in engine speed fluctuations. The selected engine for this design has a stock flywheel with inertia of 2.1 kg m<sup>2</sup>, and the instantaneous velocity fluctuation at standard engine operation at 1000 rpm is  $\pm 10$  rpm. If the engine speed fluctuations are significantly increased by the dynamics of the simultaneous compression and Otto combustion cycles, then the behavior will have to be considered to mitigate durability, noise, vibration, and harshness problems.



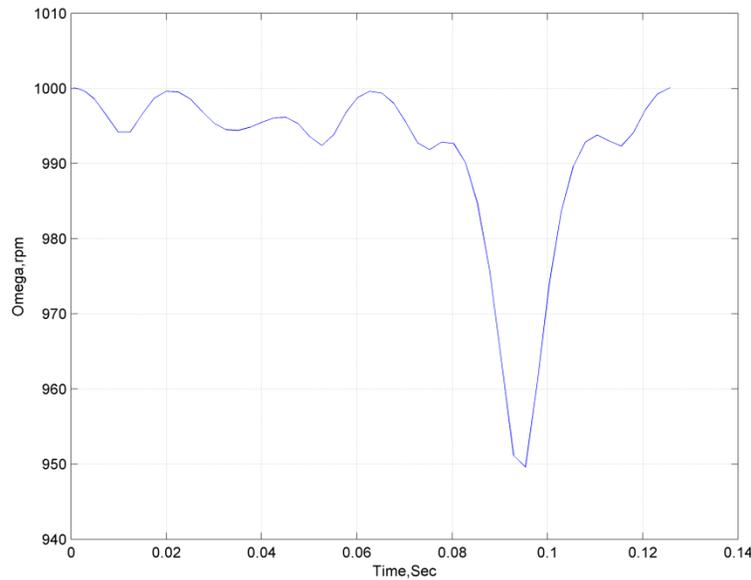
**Figure 11 Crankshaft Torque and Piston Pressure (Positive Torque Represents Compression and Negative Torque Represents Decompression in Cylinder)**

As illustrated in Figure 11, five of the cylinders of this engine operate at the normal pressures associated with an Otto cycle internal combustion engine. Relative to the other cylinders, the compression cylinder develops a large torque pulse during its second-stage compression and subsequent expansion processes. As all cylinders share a common, conventional crankshaft, this high torque pulse makes the engine operate with higher speed fluctuation during the NG compression cycle. For the example engine considered in this study, the engine torque pulses due to the compression and expansion of NG inside the engine compression cylinder will cause instantaneous torque spikes up to 1100 Nm. This torque magnitude is less than the magnitude which was obtained in the explicit model; however, it is more accurate due to the structure of the DAE model. Figure 11 shows the torque and pressure results in the compression cylinder and other cylinders in the example engine.

To translate these forces into the magnitude and direction of speed fluctuation, we can model the rotational dynamics of the example engine as cylinder 1 undergoes the compression cycle, and as cylinders 2 through 6 undergo Otto combustion cycles. Again, this calculation assumes that the cylinder 1 (the compressor cylinder) and 6 operate out of phase to demonstrate the speed fluctuation under the worst case scenario in which combustion torque in cylinder 6 and expansion torque in cylinder 1 both accelerate the crankshaft. As shown in Figure 11, the peak pressure in cylinder 1 (compression cylinder) occurs at 540 degrees of crankshaft rotation (TDC), as is expected for a polytropic compression cycle. The peak pressure in cylinder 6 occurs at 550 degrees of crankshaft rotation, as is expected for a pure combustion cycle with conventional ignition delay.

The 10 degree misalignment in peak pressure between Cylinder 1 and Cylinder 6 means that for cylinders 1 and 6, the sum of the integrated torque in expansion is larger than the sum of their integrated torque in compression, as can be seen in Figure 11. Figure 12 shows the velocity fluctuation of the crankshaft. As it is shown in Figure 12, operating the engine in a combined combustion and compression mode generates a crankshaft velocity fluctuation between 950 rpm and 1000 rpm. Although this represents an increase in engine velocity fluctuations relative to the stock engine model, a 100 rpm speed fluctuation is not abnormal for automotive applications [12].

These results demonstrate that although there may be some durability and NVH mitigating systems required to achieve OEM-level requirements, the high inertia and I-6 architecture of the conventional engine mean that the engine will be able to run while performing compression cycles.



**Figure 12. Crankshaft Velocity Fluctuation during the Compression Process**

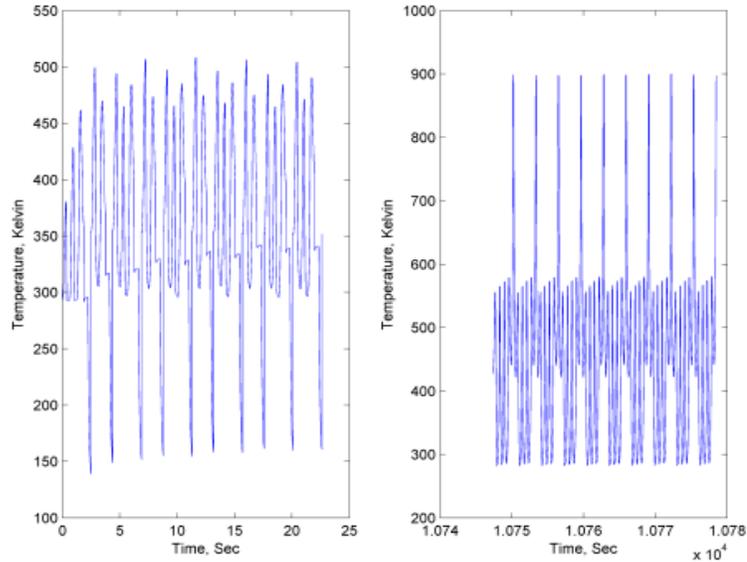
#### **4.2 Thermal Considerations**

The CNG will increase in temperature as it is compressed, and must therefore be cooled to enable it to be stored in compliance with regulations [13]. In order to follow the regulation, it is required that NG be stored in the storage tank at local ambient temperature. In practice, the rapid compression of NG that this engine-integrated compression system makes available would require the active cooling of the NG before it can be placed into the NG storage cylinder. To develop an understanding of the thermal requirements of this cooling system we can model the thermodynamics of the NG compression and heat transfer process.

##### **4.2.1 Adiabatic Compression Model**

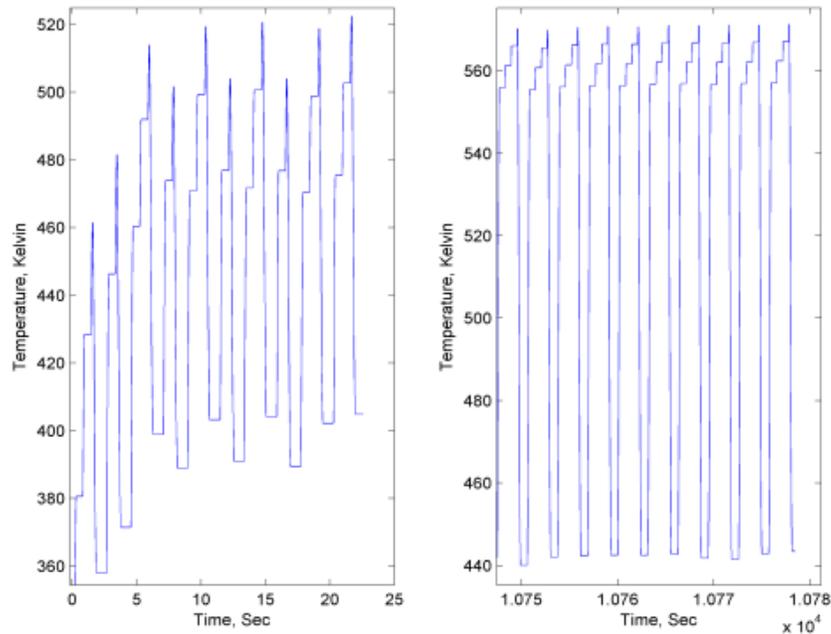
In this section, we describe the adiabatic compression of NG in the engine/compressor. The adiabatic temperature results help to understand the cooling power required in this application. During the adiabatic compression process, the CNG temperature in the compression cylinder increases from 300 K at near-atmospheric pressure to more than 900 K at 25 MPa. The compression cylinder has small volume and high pressure; therefore, its temperature increases

significantly. The temperature results for the compression cylinder under adiabatic compression conditions are shown in Figure 13.



**Figure 13. Temperature within the Compression Cylinder under the Adiabatic Compression Conditions**

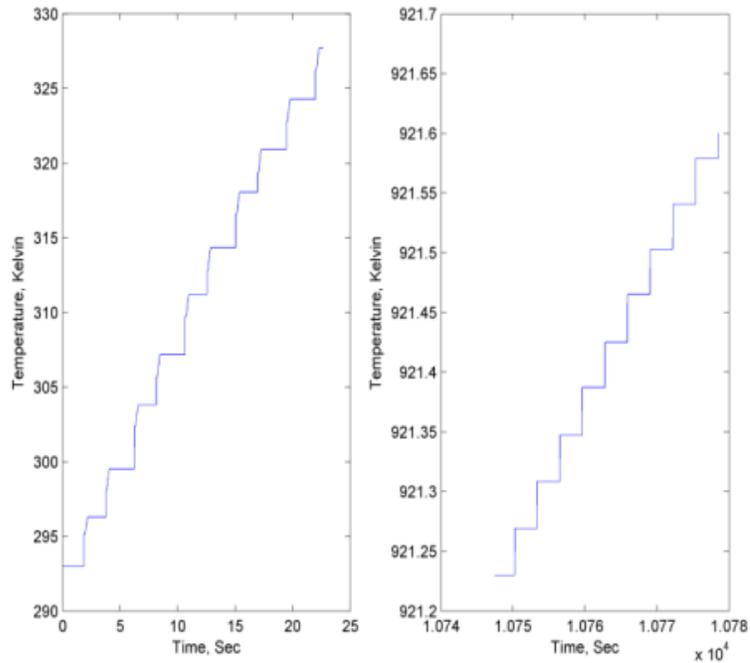
The Surge Tank temperature under adiabatic compression conditions is shown in Figure 14. The maximum temperature of the surge tank reaches 570K, which is lower than the maximum temperature in the storage tank and compression cylinder. The Surge Tank has lower pressure than the storage tank and compression cylinder; therefore, its temperature does not increase as much as the compression cylinder and storage tank temperature.



**Figure 14. NG Temperature within the Surge Tank under Adiabatic Compression Conditions**

In addition, CNG temperature in the storage tank increases to more than 900 K. The temperature result for the storage tank under the adiabatic compression condition is shown in Figure 15. The high pressure of the storage tank makes the NG temperature increase significantly inside the storage tank.

The high temperature result of the NG in the storage tank and compression cylinder shows that transferring heat from the NG inside the compression cylinder with the existing engine cooling system does not cool the NG enough; therefore, the transferred NG from the compression cylinder into the storage tank has higher temperature than local ambient temperature. Consequently, we must design a system to reduce the temperature of the NG in the storage tank and Surge Tank during the compression process in order to make the compressed NG inside the storage tank at local ambient temperature.



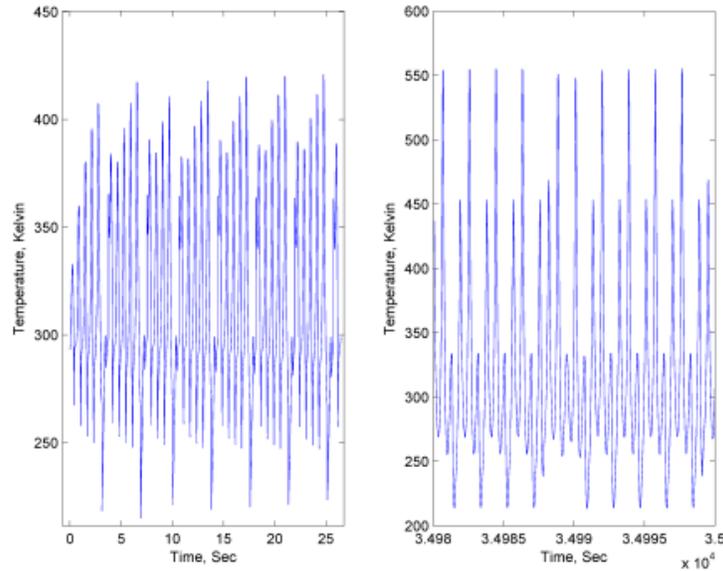
**Figure 15. NG Temperature within the Storage Tank under the Adiabatic Compression Conditions**

#### 4.2.2 Convective Heat Transfer Model

The first thermal system that the compressed NG comes in contact with is the engine itself including cylinder walls, head, and exhaust valves. These components are thermally regulated during the engine normal operation by the engine cooling jacket and heat exchanger systems which have a heat exchange capacity of tens of kW of thermal power. The convective heat transfer model seeks to show the thermal behavior of the NG inclusive of the effects of the engine cooling system.

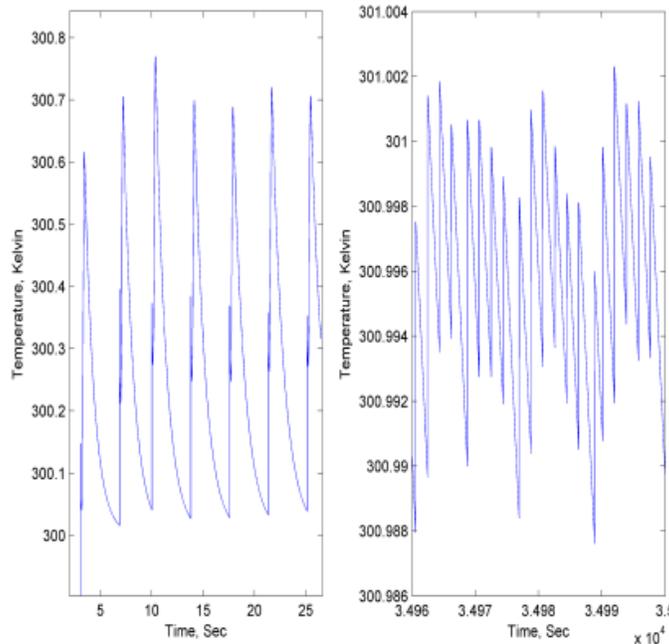
This cooling system includes a water jacket which circulates water around the compression cylinder to decrease the temperature of compressed NG. It is assumed that cylinder wall and NG are in thermal equilibrium in the compression cylinder during the compression

process. The temperature results for compression cylinder with heat transfer to engine water cooling jacket have been shown in Figure 16.



**Figure 16. NG Temperature within the Compression Cylinder under Compression Conditions with Heat Transfer to Engine Cooling Jacket**

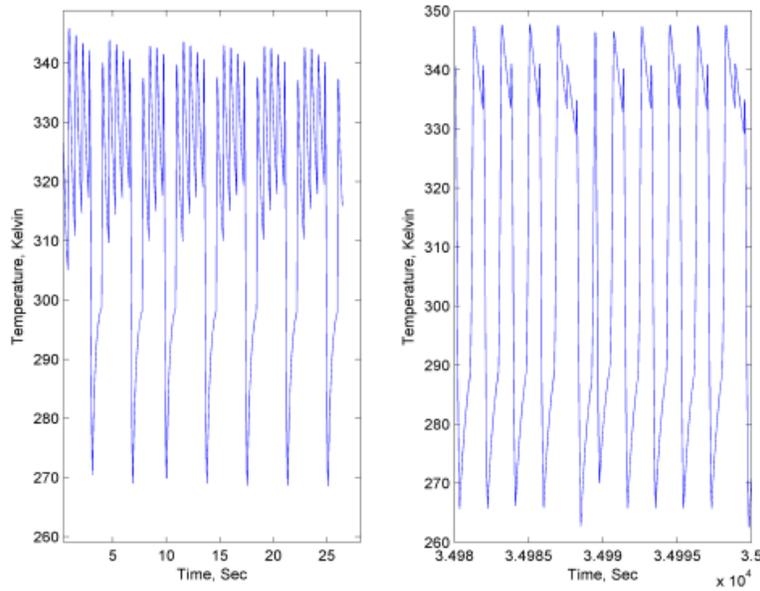
The cooling of the storage tank and Surge Tank can be achieved by developing a NG-to-air cooling system because the storage tank has high volume and surface area, and the Surge Tank has smaller pressure in comparison to the storage tank and compression cylinder. Therefore, transferring heat from the storage tank wall and Surge Tank can be performed by using air cooling. A model has been developed to show the results of the NG-to-air cooling system. In this model, two small fans are used to cool the storage tank and Surge Tank; one positioned near each tank to enable forced convection. This cooling system allows the CNG which has cooled already inside the compression cylinder to become cooler and reach room temperature, which is presumed approximately 300 K. The temperature results of the storage tank with heat transfer of gas to air is shown in Figure 17.



**Figure 17. NG Temperature within the Storage Tank under Compression Conditions with Heat Transfer of Tank-To-Air Cooling**

The small size of the compression cylinder and high pressure gas inside it generates more heat per volume than the storage tank and Surge Tank. Consequently, it is essential to use water which has a higher heat capacity and heat convection coefficient than air to cool the NG inside the compression cylinder. On the other hand, the Surge Tank has lower pressure which does not allow the gas temperature to increase as much as the compression cylinder and storage tank temperature. Therefore, the Surge Tank can be cooled by air and it facilitates cooling NG in the storage tank.

The Surge Tank temperature with heat transfer of gas to air is shown in Figure 18. The temperature of the Surge Tank after cooling fluctuates from 350 K to 270 K with heat transfer from the gas to the air which has 300 K temperature in this model. Its low temperature is still higher than NG liquid temperature (120 K); therefore, compressed NG will not be liquefied in the Surge Tank by cooling the gas inside the Surge Tank.



**Figure 18. NG Temperature within the Surge Tank Cylinder under Compression Conditions with Heat Transfer from Surge Tank-to-Air Cooling**

### 4.3 Summary of Results

This chapter presents the details of the thermal and dynamic torque models which were developed as part of the DAE model. The torque results show that the maximum torque which is applied to the crankshaft at the maximum NG pressure in the compression cylinder is three times more than the torque which applied by other 5 cylinder operating at Otto cycle combustion pressures. The crankshaft velocity fluctuation increases when the final stage of compression of the NG occurs inside the compression cylinder from 10 rpm to 50 rpm of fluctuation. The heat transfer results show that the temperature of the NG inside the storage tank and compression cylinder increases from 300 K at atmosphere pressure to more than 900K under an adiabatic compression process at 25MPa. The adiabatic temperature results help to understand the required methods and materials for cooling the NG inside the compression system. Based on the adiabatic results, the engine water jacket and air is proposed to be used to cool the NG to near-ambient

temperature in the storage tank. Air is delivered by two fans which has to be added to the cooling system of the engine.

### 5.1 Natural Gas Utility Factor Overview

NG has many advantages over petroleum in transportation applications. For instance, NG is currently cheaper than petroleum and U.S has immense NG resources. NG is also a safer fuel in comparison to gasoline and diesel fuel. The ignition temperature of NG ( $822^{\circ}$  K) is higher than gasoline and diesel fuel therefore, it is safe source of fuel in the terms of ignition temperature. Moreover, petroleum engines produce more pollutants such as  $\text{CO}_2$ , CO, and HC than NG engines [3]. Therefore, using NG as a vehicle fuel can be a possible solution to the emission problems and increasing fuel costs of the transportation sector. On the other hand, using NG as vehicle fuel has several disadvantages. NG vehicles have lower driving ranges than petroleum vehicles and NG storage tanks can take up the entire space of the vehicles' trunk. If the advantages of the NG and petroleum vehicles are combined, the resulting dual-fuel and bi-fuel vehicle technology could provide the next step in transportation innovation.

Beside the dual-fuel technology discussed in previous sections, there are bi-fuel vehicles which are a unique class of automobile systems which use two different kinds of fuel separately. Bi-fuel vehicles can run on either fuel independently. A common example of these vehicles are plug-in hybrid electric vehicles (PHEVs) which can run independently on either electricity or petroleum. Another category of bi-fuel vehicles is NG and diesel vehicles, and these vehicles can use either diesel or NG separately.

There are two operation modes in the NG-diesel dual-fuel vehicles. First, when the engine operates 100 percent on diesel fuel, second when the engine operates on NG and diesel simultaneously under a dual-fuel operation mode. In the second operation mode, NG is used as

main source of fuel, and diesel is to assist in igniting the NG fuel. NG has higher ignition temperature than diesel; therefore, diesel fuel is used to facilitate the ignition of NG without requiring changes to the ignition system of the original diesel engine.

Current NG or electric vehicles (EV) have a lower driving range than petroleum vehicles. For instance, the electric vehicle Nissan Leaf, has a driving range of 84 miles on a single charge<sup>4</sup> and the CNG Honda civic has 240 miles of combined highway and city driving range<sup>5</sup>. For comparison, the Honda Civic has 430 miles of driving range. The lower driving range of NG vehicles or EVs can be solved by combining these power sources with petroleum. Then, the fundamental question that arises is how to properly evaluate the effects (measure the sensitivity in improvement of the fuel cost and fuel consumption while using dual-fuel and bi-fuel) of using dual-fuel and bi-fuel on the driving ranges of these vehicles. The Society of Automotive Engineering (SAE) has proposed a utility factor (UF) standard for evaluating hybrid electric vehicles (HEVs). This standard has been used widely for evaluating HEVs and PHEVs and is useful for objectively comparing their fuel usage.

In this section, the concept of UF is used to evaluate dual-fuel vehicles for metrics of fuel consumption, cost and driving range. The UF of dual-fuels NG-diesel heavy-duty commercial vehicles is calculated by using data obtained from National Research Energy Laboratory's (NREL's) research on heavy-duty commercial vehicles.

## **5.2 Utility Factor Definition for Dual Fuel Vehicles**

UF is defined for PHEVs in SAE J2841 as shown in Equation 2.

---

<sup>4</sup> <http://www.nissanusa.com/electric-cars/leaf/charging-range/range/>

<sup>5</sup> <http://automobiles.honda.com/2013/civic-sedan/civic-natural-gas.aspx>

## Equation 2

$$UF_{\text{distance}(R_{CD})} = \frac{\sum_{k=1}^N \min(d(k), R_{CD})}{\sum_{k=1}^N d(k)}$$

In this definition, charge depleting is an operating mode in HEVs and PHEVs when in the car only runs on battery power and it depletes the initial battery charge.  $R_{CD}$  represents the distance driven powered by the primary fuel source. In the case of a PHEV,  $R_{CD}$  is the charge depleting range of the vehicle. Recharging a PHEV can be done during the night at home or during the work time at a charging station close to workplace. Therefore, a useful definition for evaluating driving ranges is the daily distance UF which is defined as the statistical probability that an average vehicle will be driven less than or equal to  $R_{CD}$  during a particular driving day [14].  $R_{CD}$  in charge depleting mode varies based on both vehicle load and battery size. For instance, a vehicle with a large battery size can have longer  $R_{CD}$  on a particular drive cycle than the same vehicle with a smaller battery size in the same drive cycle. On the other hand, if a battery size is used for two vehicles with different loads, the one which has higher load can have a lower EV driving range than the vehicle with lighter load. In evaluating UF, the battery size is not considered, and it is assumed the vehicles can be characterized by a deterministic and constant  $R_{CD}$ .

To construct an analogy between PHEVs and dual fuel vehicles, we can consider that the batteries in PHEVs resemble the NG in DFNDVs (the NG acts as the range limiting energy source in this case). Therefore,  $R_{CD}$  can be considered as the driving range which vehicles operate on the dual-fuel operation mode, and the symbol  $R_{CD}$  is replaced with  $R_{NG}$  in this definition to express the range available under a dual-fuel operation mode. In the formula for daily distance UF (Equation 3),  $d(k)$  represents the daily distance that a single vehicle ( $k$ ) drives.

Daily distance UF for the DFNDVs is the ratio of the  $R_{NG}$  to the total distance traveled during a day if  $R_{NG}$  is smaller than  $d(k)$ , otherwise this value is equal to 1. Over the course of  $N$  travel days, an aggregate UF can be calculated as a function of  $R_{NG}$ . In this concept similar to PHEVs, it is assumed vehicles' NG tank is filled by fueling during night time or at work. Therefore, the engine integrated compressor concept is a useful tool to achieve this goal.

### Equation 3

$$UF_{\text{distance}(R_{NG})} = \frac{\sum_{k=1}^N \min(d(k), R_{NG})}{\sum_{k=1}^N d(k)}$$

As expressed in the formula above, the daily distance utility factor for DFNDVs is the ratio of the vehicle's daily driving distance while operating under dual fuel mode, to the total daily driving distance of the vehicles. This definition shows the fraction of the daily distance dual fuel vehicles can be driven using dual fuel during the day.

Based on the assumptions listed above, UF can now be used as a tool to estimate the weighted fuel cost to drive the DFNDVs. This concept can specify the total amount that a driver can save over the year by dual-fuel NG-diesel car, as opposed to relying solely on diesel fuel.

To calculate the weighted fuel cost of the DFNDVs over a year, Equation 4 is used. In this equation,  $d_{\text{year}}$  is the total distance that a vehicle drives per year (km/year).  $FC_{\text{dual mode}}$  is the fuel consumption of DFNVs while NG and diesel is used as the fuel source simultaneously. The fuel economy of these vehicles increases about 20 percent on average at different loads when they operate in dual fuel mode [15]. In this concept, the fuel economy of the NG is expressed in terms of diesel gallon equivalence in 25 MPa (DGE) which is converted to liter per

km.  $C_{NG}$ , and  $C_{diesel}$  are the cost of the CNG and diesel respectively.  $FC_{diesel}$  is diesel fuel consumptions in units of liter per km.

The ratio of NG consumption to diesel fuel consumption differs based on the engine load [15]. The average portion of the NG inside the engine is 57 % of total fuel [15] which is designated by the symbol  $r$  in the Equation 4.

#### Equation 4

$$C_{UF\ weighted} = d_{year}(UF_{distance} \cdot FC_{dual\ mode} [C_{NG} * r + C_{diesel} \cdot (1 - r)] + (1 - UF_{distance}) \cdot FC_{diesel} \cdot C_{Diesel})$$

Using a similar formulation, UF can be used to estimate the weighted fuel consumption over the year. This idea supports a better understanding of the fuel economy of DFNDVs in the modern market. Moreover, it can also be utilized to directly compare the costs of NG-diesel bi-fuel vehicles to diesel and dual-fuel vehicles. In this concept, the diesel gallon equivalent (DGE) is used to convert CNG fuel economy to equivalent diesel fuel economy. The following equations calculate the weighted UF fuel consumption over a trip for diesel fuel and NG respectively in terms of  $Lkm^{-1}$ :

#### Equation 5

$$DFC_{UF\ weighted} = UF * FC_{Dual\ mode}(1 - r) + (1 - UF)FC_{Diesel}$$

#### Equation 6

$$NGFC_{UF\ weighted} = UF * FC_{Dual\ mode} * r$$

### 5.3 Distance Utility factor Results

In this study, the UF for 5 different types of heavy-duty vehicles are calculated based on data collected from a survey conducted by NREL. Data sets for each type of vehicle differ in

terms of the number of vehicles and the number of days each vehicle is driven. The summary of these data is shown in Table 2.

**Table 2. Summary of the Survey Data**

<b>Type of Vehicles</b>	<b>Number of Vehicles</b>	<b>Number of Days</b>	<b>Average Speed (Miles/Hour)</b>	<b>Average Distance per day (miles/day)</b>
Delivery Vans	92	1282	19.78	46.54
Delivery Trucks	9	128	24.57	73.13
School Buses	210	922	23.02	57.00
Bucket Trucks	19	207	22.61	24.67
Service Vans	7	81	21.20	22.75

The daily distance UF for each vehicle is then calculated and shown in Figure 19. This figure shows total fraction of the daily distance that each type of commercial vehicles is driven in dual fuel mode. .

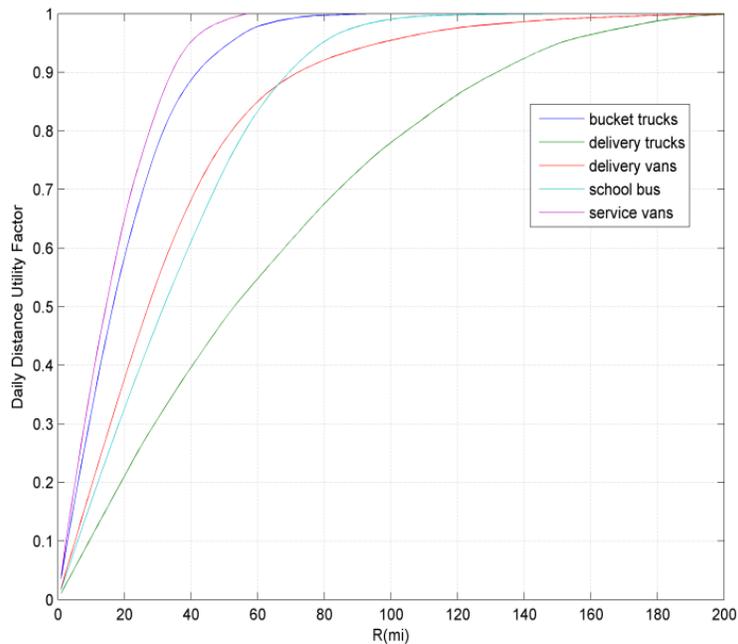
Figure 18 shows that the surveyed service vans have the highest daily distance UF, and these vehicles can achieve 95 percent of their daily driving distance operating in dual-fuel mode when they have enough NG fuel source to operate for 40 miles on dual-fuel operation mode. This result is consistency with the average driving distance of service vans, NREL’s surveyed service van fleet has which have a lowest average distance per day of any of the surveyed vehicle types.

The bucket truck has the second highest utility factor, and can achieve 90 percent of their daily driving distance in dual-fuel mode when they have 40 miles dual-fuel range. School buses have a lower utility factor than delivery vans for dual fuel ranges under 63 miles. After this point, the UF for the school buses slightly increases above the UF for the delivery vans. This is

due to the presence of a few high daily distance delivery vans, whereas there are no school buses that drive more than 100 miles in a day.

The lowest distance UF in this category belongs to the delivery trucks. The average daily driving distance for the delivery trucks is higher than the other four vehicle types in this survey. Delivery trucks can operate on dual-fuel mode for 40 percent of their daily distance when their dual mode driving range is 40 miles.

As is represented in Figure 19, a dual fuel range of approximately 40 mi leads to between 50% and 100% displacement of the diesel-fueled driving mode. This suggests that a small NG storage tank (to provide approximately 40 mi of dual fuel range) would significantly reduce the diesel fuel consumption, emissions and costs of these heavy duty vehicles without requiring a full function NGV conversion.



**Figure 19. J2841 Distance Utility Factor Based on the NREL Research Fleet DNA**

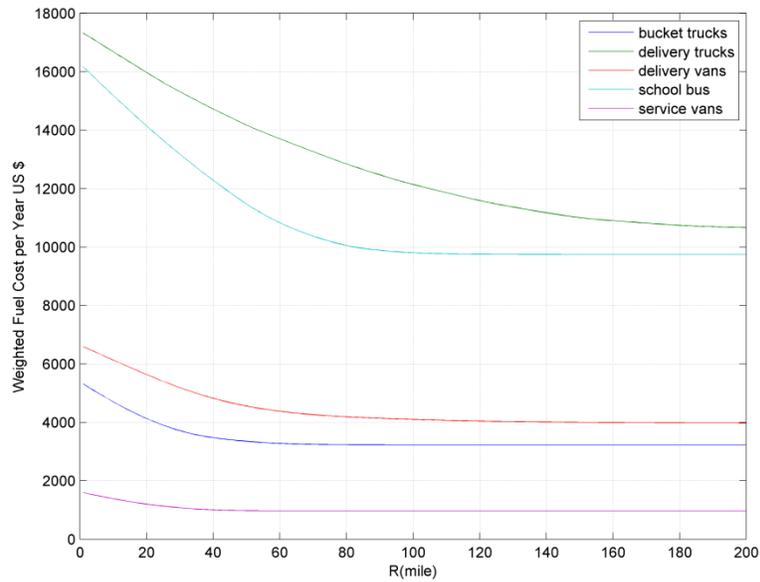
#### 5.4 Yearly Fuel Cost

The fuel costs of the surveyed vehicles are a function of their UF, as shown in Equation 4. To calculate the magnitude of the costs savings available with increasing dual fuel range, we will assume a cost of NG of \$2.09/DGE while diesel costs \$3.89/gallon<sup>6</sup>. By increasing the dual-fuel operating range of these vehicles, the fuel cost decreases (as shown in Figure 20).

These results show that school buses can save \$6000 every year by having 80 miles of daily dual-fuel operation mode range. Delivery trucks can save \$4500 by having 80 miles of dual-fuel operation mode range. Delivery vans, bucket trucks and service vans save \$2400, \$2100, and \$652 per year, respectively. School buses and delivery trucks have the lowest daily distance UF, meaning they are driven more than the other vehicles in this category, and their fuel costs are more sensitive to changes in the UF. In other words, by increasing the dual-fuel operation range (which is equivalent to increasing  $R_{NG}$  in the Equation 3), the vehicles owner can save more money during a year. In general, increasing the NG operation range decreases the yearly fuel cost of dual-fuel NG-diesel heavy-duty commercial vehicles based on the data in this survey.

---

<sup>6</sup> <http://www.afdc.energy.gov/fuels/prices.html>

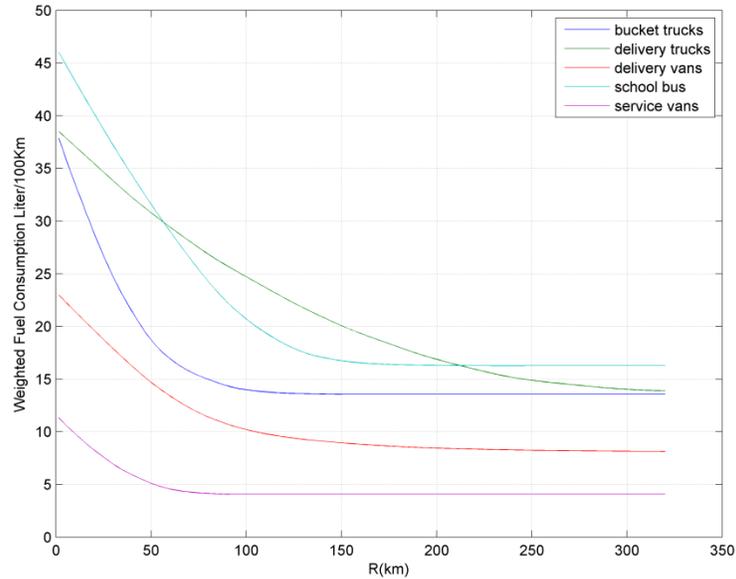


**Figure 20. Yearly Fuel Cost of Each Type of Vehicle**

The daily diesel fuel consumption of each surveyed vehicle as a function of  $R_{NG}$  is shown in Figure 21. Increasing the  $R_{NG}$  has the effect of decreasing the diesel fuel consumed by the dual-fuel vehicles. The daily diesel fuel consumption of these dual-fuel vehicles does not reach 0 when the UF reaches to one due to the usage of the diesel fuel as an ignition source for NG in dual-fuel operation mode.

Decreasing diesel fuel consumption by increasing the dual fuel range of these vehicles has numerous benefits. For example, for the service vans surveyed for this study, a dual fuel range of 50 miles lead to a 50% reduction in diesel fuel consumption. This corresponds to a 50% reduction in petroleum consumption (as the lifecycle petroleum content of NG is negligible), and a 22% reduction in GHG emissions. The criteria pollutant emissions of these vehicles have the potential to be decreased by similar amounts; however, the emission of the DFNDVs should be

evaluated more carefully because their emission does not necessarily decrease while these vehicles operate in low load due to the incomplete combustion of NG in the engine.



**Figure 21. Weighted Daily Diesel Gallon Equivalent Fuel Consumption for Diesel**

As is discussed in this section, UF concept is used to show the sensitivity of the dual-fuel vehicles metrics of performance to using the NG as a source of fuel. To meet the assumptions implicit in the UF calculation, the heavy duty vehicles in this survey have to be fully refueled at the start of each driving day. Therefore, using the home engine integrated NG compressor can be a useful tool to accomplish the goal of having the full fuel tank before start of the driving day.

## 6 DISCUSSION

In this section, I summarize the results associated with each of the research questions.

### 6.1 Research Question 1

Research Question 1 looks at the torque applied on the crankshaft and the velocity fluctuations of the crankshaft and investigates the necessity of resizing the flywheel to avoid velocity fluctuations.

This question is answered through the development of a dynamic simulation of the engine's rotation as it undergoes the combined combustion and ignition cycles, as described in Chapter 4. I find that the torque on crankshaft fluctuates from -1300 N·m to +1100 N·m at the highest pressure in the compression cylinder. This high torque variation occurs at the final stage of pressurizing the storage tank. Therefore, it represents the worst case scenario of the torque spike on the crankshaft. This results in a crankshaft velocity trace with a maximum of  $\pm 50$  rpm fluctuation at the same time which crankshaft receives the highest torque spike. These results are obtained based on the flywheel size of a 5.9 liters Cummins engine which has an inertia of  $2.1 \text{ kg m}^2$ . During the final pressurizing stage of the storage tank, torque on the crankshaft and pressure in the compression cylinder has a lower value; therefore, the velocity fluctuation is less than 50 rpm before the final stage of the storage tank pressurizing.

The velocity fluctuation of the crankshaft in automotive applications is more than  $\pm 100$  rpm [12] which in the modeled engine is  $\pm 50$  rpm during the worst case scenario of the combustion/compression cycle. Therefore, a new flywheel is not required for the selected 5.9 liters Cummins engine to avoid the high velocity fluctuations.

Even though the 5.9 liter Cummins engine does not need the resized flywheel, other engines with smaller flywheels may require a resized flywheel with higher inertia to reduce the velocity fluctuation in the engine at high pressure.

### **6.2 Research Question 2**

Research question 2 investigates the temperature of the gas in compression cylinder, storage tank and Surge Tank, and also it looks for the way to cool the NG if its temperature exceeds the required regulation temperature limit for the storage of NG.

This question is answered by through the development of a thermal model of the NG compression cycle. This effort is described in Chapter 4 Section 4.1, in which the thermal results are shown. A synopsis is outlined below.

NG should be stored in the storage tank at the local ambient temperature to meet the regulation requirements [13]. Under adiabatic conditions, the NG temperature in compression cylinder reaches to over 900 K, the NG temperature in the Surge Tank is ~ 600 K, and the NG temperature in the storage tank is over 900 K. A cooling system was designed using a liquid cooling jacket for the compression cylinder, and an air cooling system for the surge tank and storage tank. The resulting system is able to cool these components and the NG to near-ambient conditions over the course of the compression operation.

### **6.3 Research Question 3**

This research questions seeks to find the sensitivity of fuel cost and diesel fuel consumption of dual-fuel heavy duty commercial vehicles to use of NG in dual-fuel operation mode.

This question is answered in Chapter 5 through the development of a utility factor (UF) to characterize the statistical driving conditions under which medium duty vehicles are operating.

The NG dual fuel UF definition is proven to be a helpful tool used to approach this question and was integrated with the NREL DNA fleet surveyed data.

As it was shown in Figure 20 the yearly cost of the fuel in the delivery trucks and the school buses is more sensitive to dual fuel range than the other vehicles in this survey because school buses and delivery trucks have higher fuel consumption and higher average driving range in this survey. Therefore, their fuel cost will reduce significantly by increasing the dual-fuel NG-diesel driving range of these vehicles.

The diesel fuel consumption of the surveyed vehicles decreases by increasing the dual-fuel NG-diesel driving range Figure 21. Bucket trucks and schools buses fuel consumption are more sensitive to dual-fuel NG-diesel driving range because the UF of these vehicles is sensitive to the dual-fuel operating mode driving range and also because they have high diesel fuel consumption.

Generally, the diesel fuel consumption and yearly cost of the dual-fuel NG-diesel commercial heavy duty vehicles depend on the utility factor and the fuel consumption of these vehicles. Increasing the utility factor by expanding dual-fuel NG-diesel power range can decrease the diesel fuel consumption and the yearly fuel cost.

## 7 CONCLUSION

This study has analyzed the design considerations required to enable the development and commercialization of an engine-integrated reciprocating NG compressor. The structural and functional architecture of the concept show that the engine-integrated reciprocating NG compressor can be constructed with minimal additional parts-count and control system development from conventional diesel ICEs. The rotational dynamics, and heat transfer of the NG compression system have been identified as design considerations of note and this study has addressed these potential barriers to the realization of the concept. The rotational dynamics considerations can be solved by using a Surge Tank to split the compression forces into two stages, a large-scale I-6 engine where opposing cylinders are undergoing compression and combustion cycles simultaneously, and where engine inertia is high enough to overcome any induced rotational dynamics problems. The thermal considerations can be solved by removing only a fraction of the high-pressure and high-temperature NG from each compression cycle, by using the water jacket around the compression cylinder of conventional engine system to perform thermal regulation of the cylinder and piston, and by enabling simple forced convection around the storage tanks.

This design can be a disruptive technology which includes those that can reach customers that are not served by the mainstream product. They are generally technologically straightforward, consisting of off-the-shelf components put together in a new product architecture that can realize different benefits than the incumbent technologies.

At present, compressed NG (CNG) vehicles are refueled using a high-cost, centralized, and sparse network of CNG fueling stations. As these systems were designed for the use of fleet

customers (an incumbent and high profit margin customer), they are often not available for public refueling<sup>7</sup>. At the same time that the cost of NG to fuel CNGVs has gone down in recent years, a consumer demands for low-cost, utilitarian, and environmentally preferable personal transportation has emerged with the production, and sales of electrified and hybridized vehicles. As such, an engine-integrated reciprocating NG compressor has the capability to disrupt the incumbent CNG market by enabling the use of NG for personal transportation, fueled at home, from the preexisting low-pressure NG infrastructure, at low parts count, using conventional components, and therefore at low incremental costs. By enabling low-cost NG fuel to enter the personal transportation market, the potential for this or another technology to disrupt the NG fueling energy sector is high.

---

<sup>7</sup> US Department of Energy, Alternative Fueling Station Locator, 2013, available at <http://www.afdc.energy.gov/locator/stations/>

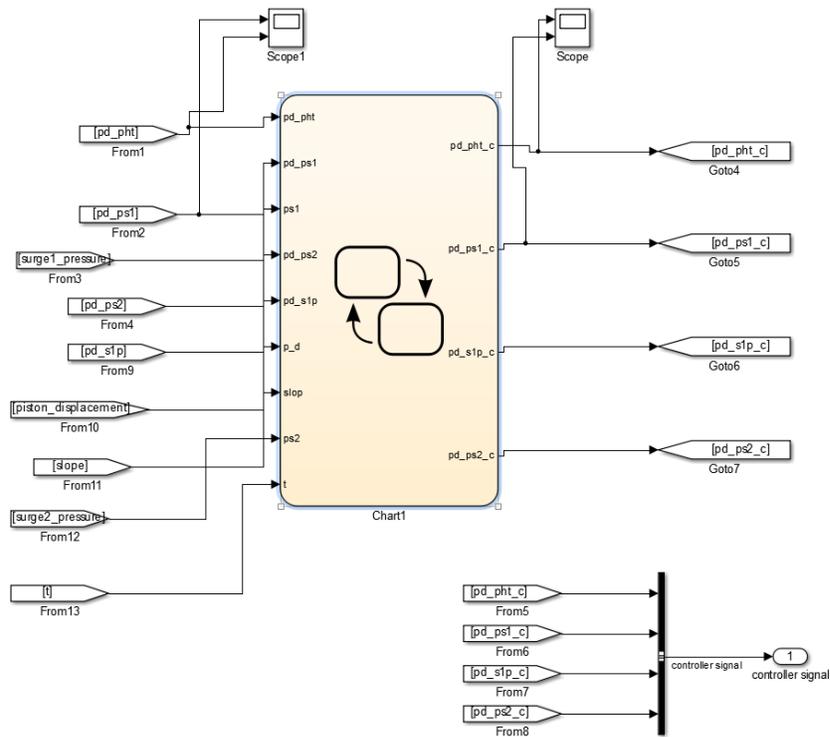
## 8 REFERENCES

- [1]. Semin, and Rosli Abu Bakar. "A Technical Review of Compressed Natural Gas as an Alternative Fuel for Internal Combustion Engines." *American Journal of Engineering and Applied Sciences* 1.4 (2008): 302-311. Print.
- [2]. Chan, S.h., and H.m. Wang. "Effect of natural gas composition on autothermal fuel reforming products." *Fuel Processing Technology* 64.1-3 (2000): 221-239. Print.
- [3]. Wang, M, and H Huang. "A Full-Fuel-Cycle Analysis of Energy and Emissions Impacts of Transportation Fuels Produced from Natural Gas." *Transportation Technology R&D Center* 1 (1999): 134.
- [4]. Aslam, M.U., H.H. Masjuki, M.A. Kalam, H. Abdesselam, T.M.I. Mahlia, and M.A. Amalina. "An Experimental Investigation of CNG as an Alternative Fuel for a Retrofitted Gasoline Vehicle." *Cleaner Production* 1 (2006): 8. Print.
- [5]. Hekkert, Marko P., Franka H.J.F. Hendriks, Andre P.C. Faaij, and Maarten L. Neelis. "Natural Gas as an Alternative to Crude Oil in Automotive Fuel Chains Well-to-Wheel Analysis and Transition Strategy Development." *Energy Policy* 1 (2005): 16. Print.
- [6]. Xuping, Li, Ogden Joan M, and Kurani Kenneth S. "An Overview of Automotive Home and Neighborhood Refueling." *Institute of Transportation Studies* 1 (2009).  
<http://escholarship.org/uc/item/75m7618w> (accessed June 12, 2013).
- [7]. Wegrzyn, J. , and M. Gurevich. "Adsorbent Storage of Natural Gas." *Elsevier Science Ltd* 55 (1996): 13.  
[http://econpapers.repec.org/article/eeeappene/v\\_3a55\\_3ay\\_3a1996\\_3ai\\_3a2\\_3ap\\_3a71-83.htm](http://econpapers.repec.org/article/eeeappene/v_3a55_3ay_3a1996_3ai_3a2_3ap_3a71-83.htm) (accessed May 30, 2013).
- [8]. Megel, Marc, Barry Westmoreland,, Guy Jones, Ford Phillips, Douglas Eberle, and Mark Tussing. "Development of a Structurally Optimized Heavy Duty Diesel Cylinder Head Design Capable of 250 Bar Peak Cylinder Pressure Operation." *SAE International* 1 (2011): 20.
- [9]. Kirkpatrick, Allan T, and Colin R. Ferguson. *Internal Combustion Engines: Applied Thermosciences*. Fort Collins: Greek edition, 2008.
- [10]. Mokhatab, Saeid, and William A. Poe. *Handbook of natural gas transmission and processing*. 2nd ed. Waltham [Mass.: Gulf Professional Pub., 2012. Print.
- [11]. Norton, Robert L.. *Design of machinery: an introduction to the synthesis and analysis of mechanisms and machines*. New York: McGraw-Hill, 1992. Print.
- [12]. Davis, Roy I., and Robert D. Lorenz. "Engine Torque Ripple Cancellation With an Integrated Starter Alternator in a Hybrid Electric Vehicle: Implementation and Control." *IEEE Transactions on Industry Applications* 39 (2003): 10. Print.
- [13]. Menon, V.C., and S. Komarneni. "Porous Adsorbents for Vehicular Natural Gas Storage: A Review." *Journal of Porous Materials* 1 (1998): 16. Print.
- [14]. Bradley, Thomas H., and Casey W. Quinn. "Analysis of plug-in hybrid electric vehicle utility factors." *Journal of Power Sources* 195.16 (2010): 5399-5408. Print.

- [15]. Wan Mansor, Wan Nurdiyana, Jennifer S. Vaughn, and Daniel B. Olsen. "Emissions and Efficiency Evaluations of a 6.8 Liter Diesel Derivative Dual Fuel Engine." Canadian Society for Mechanical Engineering International Congress 1 (2014): 6. Print.

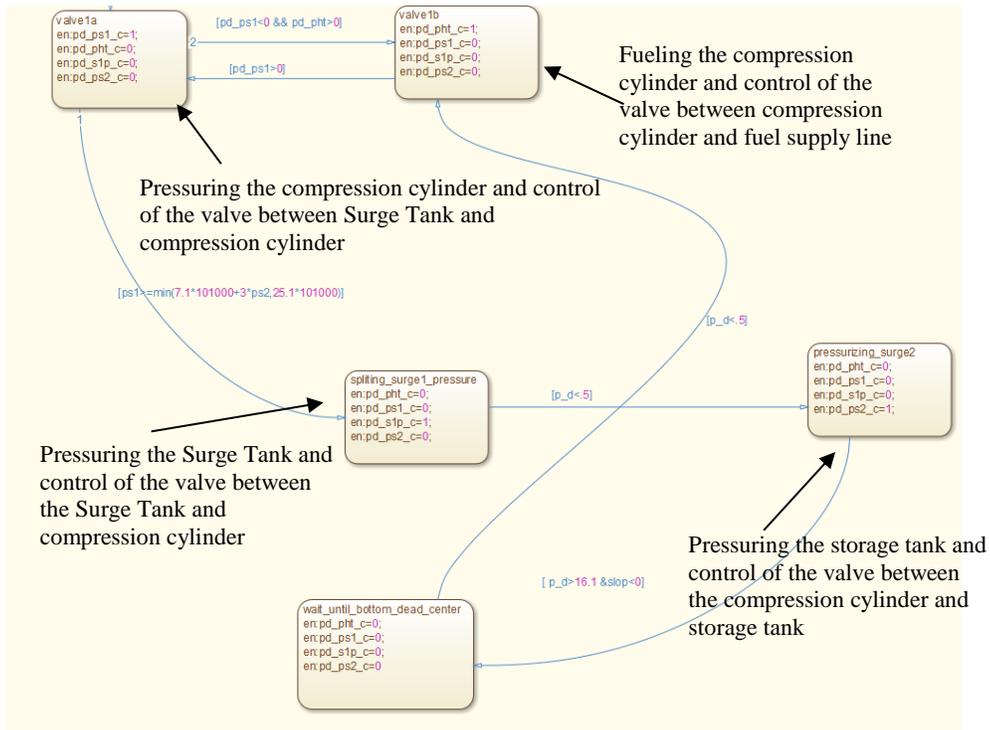
**9.1 Controller**

Figure 22 shows the controller of the compressor system and the input and output signals into the controller. The main input signals are the pressure difference between the compression cylinder, the Surge Tank and the storage tank which enable controller to open and close the valves and define the output signals to the valves.



**Figure 22. Compressor Controller and Input and Output Signals into the Controller**

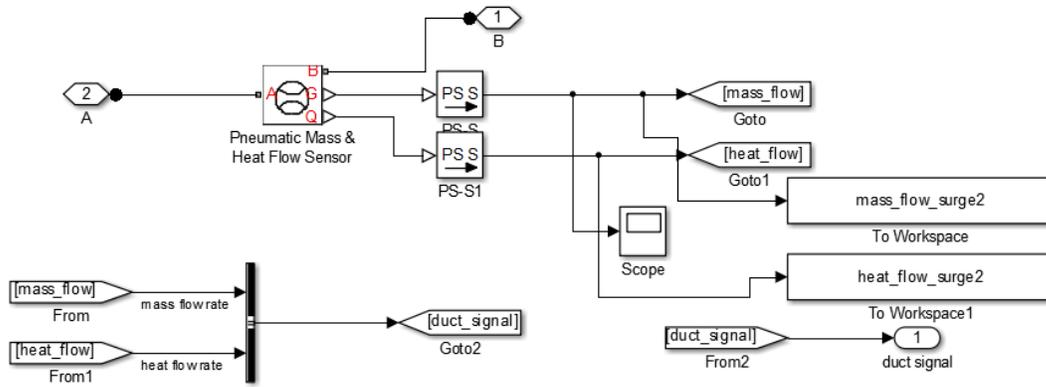
Figure 23 show the controller stages which are 5 stages in this controller. The pressurizing and fueling compression cylinder are two main stages for compression cylinder. In addition pressurizing the Surge Tank and storage tank are other two main stages which are shown in Figure 23



**Figure 23 .Controller Operation**

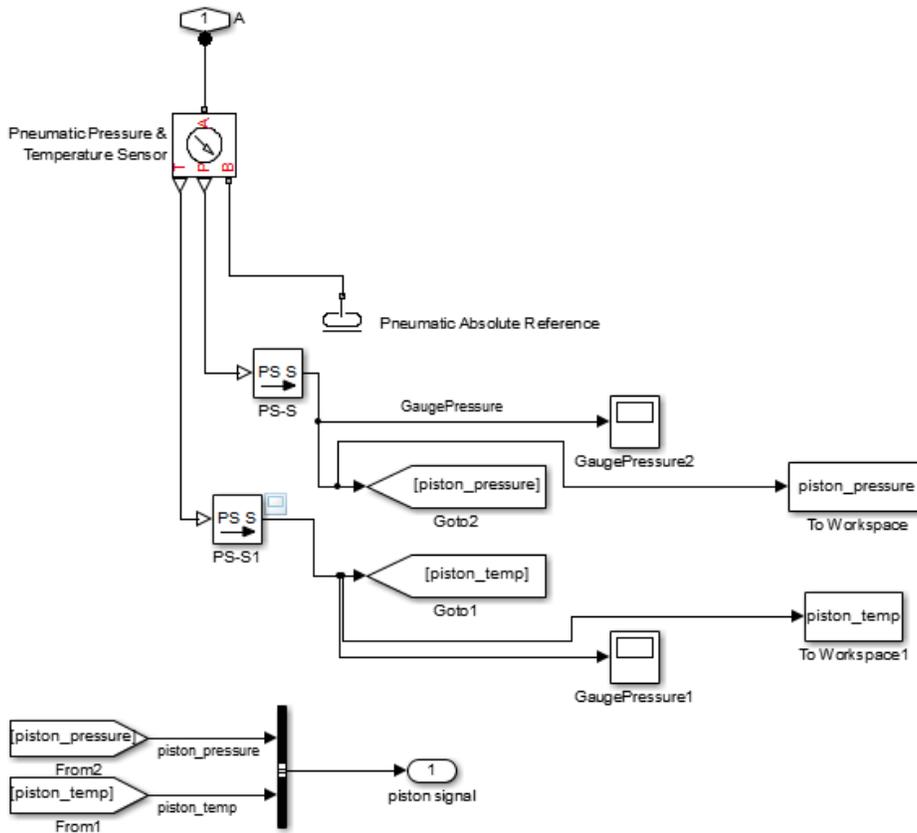
## 9.2 Model Development

The model of the compressor system is developed in the Simscape and Simulink in MATLAB and all the parts is designed and modeled in detail in order to measure the essential parameters f or this design Figure 24 shows the approach which is used to measure the heat and mass flow in this design.



**Figure 24. Heat and Mass Flow Measurements in the Model**

Figure 25 shows the blocks and methods are used to measure the compression cylinder temperature and pressure



**Figure 25. Pressure and Temperature Measurements in Model**