CONTROL OF FUGITIVE METHANE EMISSIONS THROUGH COMBUSTION OF COMPRESSOR VENT AND ENGINE CRANKCASE EMISSIONS

A Thesis

by

TREVOR DALE MURRAY

Submitted to the Office of Graduate and Professional Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

Chair of Committee,	Timothy Jacobs
Committee Members,	Jerald Caton
	Sergio Capareda
Head of Department,	Andreas A. Polycarpou

August 2017

Major Subject: Mechanical Engineering

ABSTRACT

At compressor stations, fugitive methane emissions from compressor piston rod packing and engine crankcases are vented directly into the atmosphere. In an effort to reduce compression station carbon footprint, this study evaluates the feasibility of combusting the methane emissions into carbon dioxide and thus reduce the global warming potential. This study focuses on running simulations to determine the methane reduction from rebreathing engine crankcase and compressor vent gases into the air intake of a large bore, natural gas, 2-stroke engine. The methane reduction percentage is observed over a range of rebreathed gas mass flow rates, and rebreathed gas composition.

It is extremely difficult to determine the composition of the engine blow-by gases in the crankcase, since the composition depends on a large variety of parameters. For this study, the emissions from the compressor was modeled as methane, and the emissions from the engine crankcase was modeled as products of combustion with a varying amount of methane concentrations. A sensitivity analysis was performed, and the observed pressure traces show that the engine performance is not affected by the addition of rebreathed gases. This insensitivity mainly results from the very small rebreathed flow rates compared to the air intake, and the adjustments made on engine parameters, boost pressure and fuel injection rate, to keep TER and the energy delivery rate the same.

The simulations also showed that the net methane reduction percentage was approximately 52%, no matter the study test conditions. It was discovered that the methane reduction depends on the trapping ratio of the engine; since these engines generally have trapping ratios around 50%, the actual methane reduction tends to be small. A 52% reduction rate is not desirable; the hope is to increase the reduction rate closer to 100%. More importantly to note, a substantial amount of complexity would need to be added to a typical compressor station just to reduce methane emissions by 1 kg/hr. Thus, at present, the idea of rebreathing compressor and engine crank case gases for methane emission reduction is not feasible. Future studies should focus on routing the emissions to a 4-stroke engine, a waste heat recovery system, or other combustion devices with higher trapping ratios.

DEDICATION

To my family, fiancée, and mentors.

ACKNOWLEDGEMENTS

There are special people I would like to thank for their efforts in affording me my wonderful education. First, I acknowledge my parents, Robert and Kelli Murray, who have been with me every step of the way. Their unconditional love, patience, support, and guidance throughout my life has helped me achieve my dreams and goals. I would also like to acknowledge my fiancée, Megan Story, and my brother, Zachary Murray, for their love and confidence in me. They were there for me through the good times as well as when life became rough. Thank you to the rest of my family for their love for and belief in me. I am fortunate to have these people in my life and they have helped make me the person I am today.

When attending graduate school at Texas A&M, I knew that I wanted to work in AERL. Little did I realize that I would be working with the best team imaginable. The group is welcoming and inclusive, and expressed interest in my research. Because of their motivation, I became a better researcher and could not have completed my work without this team. I hope that I am fortunate enough to work with a group like the AERL team in my future career.

I also acknowledge the PRCI team. It was a privilege to have the direction and guidance from Greg Beshouri, Gary Choquette, Jim McCarthy, Chris Nowak, Mark Patterson, and Dan Rem.

I am grateful for the support of my committee members, Dr. Jerald Caton and Dr. Sergio Capareda. These educators were unselfish with their time and were enthusiastic to learn about my work. They were also helpful in making sure I could reach all my deadlines. I am thankful and appreciative to have both distinguished professors on my thesis committee and reviewing my research.

I end my acknowledgements by recognizing a great mentor, Dr. Tim Jacobs, who has been with me on the duration of this journey, keeping me on tract while providing his knowledge and guidance. Dr. Jacobs was very interested in getting to know me as an individual and was genuinely excited to hear about my life events. His uplifting, and positive personality has made my time at Texas A&M a fun and memorable adventure. It was an honor to have the opportunity to work and learn under Dr. Jacobs.

CONTRIBUTORS AND FUNDING SOURCES

Contributors

Part 1, faculty committee recognition

This work was supervised by a thesis committee consisting of Professor Timothy Jacobs [advisor] and Professor Jerald Caton of the Department of Mechanical Engineering and Professor Sergio Capareda of the Department of Biological & Agricultural Engineering.

Part 2, student/collaborator contributions

All work for the thesis was completed independently by the student.

Funding Sources

This work was made possible by Pipeline Research Council, Interntational.

NOMENCLATURE

AERL	Advanced Engine Research Laboratory
AFR_{stoich}	Stoichiometric Air-to-Fuel Ratio
AMP	Air Manifold Pressure
AMT	Air Manifold Temperature
Baro	Barometric Pressure
BDC	Bottom Dead Center
CCV	Closed Crankcase Ventilation
CH4	Methane
CSU	Colorado State University
EGR	Exhaust Gas Recirculation
EM	Exhaust (Unburned) Methane
FM	Fumigated Methane
GT	Gamma Technologies
IMEP	Indicated Mean Effective Pressure
m	Mass Flow Rate
PCV	Positive Crankcase Ventilation
ppm	Parts Per Million
PRCI	Pipeline Research Council International
RS	Scavenging Ratio
SC	Short Circuit

SCM	Short Circuit Methane
SE	Scavenging Efficiency
TDC	Top Dead Center
TER	Trapped Equivalence Ratio
V_{trap}	Trapped Volume
X	Mole Fraction
ρ	Density
φ	Trapping Ratio
φ_{corr}	Corrected Trapping Ratio

TABLE OF CONTENTS

Page

LIST OF FIGURES

Page

LIST OF TABLES

Page

1 INTRODUCTION

It is desired to reduce fugitive emissions from pipelines and compressor stations. One of these fugitive emissions is methane, which has a global warming potential larger than carbon dioxide. Carbon dioxide has a global warming potential of 1.0 compared to methane which has a global warming potential of 3.7, almost 4 times larger than carbon dioxide [1]. Based on this, it will be more beneficial for the environment to convert methane into carbon dioxide rather than venting the methane into the atmosphere which is the current method of crankcase gas disposal. At compressor stations, there are two main sources of fugitive methane emissions, compressor vents attached to rod packing leaks, and engine crankcases. One possible solution to convert methane into carbon dioxide is to breathe the fugitive emissions into the air intakes of the large bore, 2-stroke, natural gas engines, which are driving the compressors, and combust the methane into carbon dioxide. Besides reducing the fugitive methane emissions from the compressor station, another expected benefit is the decrease in the volume of gas required to operate the natural gas engine which will lead to an increase in the pipeline efficiency. This solution raises a few questions that need to be answered before contributing further resources into research and design of a fumigated methane rebreathing system. One question is what is the nature or composition of the vent and crankcase gases. The gases from the compressor vent and engine crankcase will be breathed into the engine crankcase, therefore it is important to understand what concentration of gases or other species, such as oil, make up the breathed gases to protect the engine. A second question is how does the combustion reaction change if the intake air contains small concentrations of the breathed gases; is the possibility of pre-ignition and auto-ignition increased. Adding in different species into the engine cylinder can cause the combustion process to change, and it needs to be determined if these changes are non-existent, small, or large enough to damage the engine. A final question regarding this solution is how much of the fugitive methane emissions can be reduced by re-breathing the methane into the engine air intake. If this solution can only provide a small percentage reduction, it may not be worth the time and effort to use resources on this solution, so it is necessary to determine how effective this solution can be.

A detailed literature review over crankcase ventilation is to follow with the goal of answering the posed questions above. Then the simulation set up will be explained followed by the results and discussion section. And the report will be completed with conclusions and future work.

1.1 Literature Review

1.1.1 Sources of Fugitive Methane Emissions

A new study published in the journal *Nature*, has shown that global methane emissions in oil and gas production are 60-110% higher than current estimates. The study also revealed that methane leak in oil and gas production are 20-60% higher than previously estimated [2]. Fugitive methane leaks cause a few major negative effects. One being a loss of product, and therefore a loss of money. The other being negative environmental, health, and safety impacts. This report will focus on methane leaks and emissions from engines and compressors, specifically engine crankcases and compressor packing rod vents.

Compressors have six areas that can allow methane to leak into the environment: gas piping connections, compressor cylinder valve caps, compressor cylinder heads, unloading devices, piston rod pressure packing, and the collection for recovery or disposal [3].

Depending on the application, the sealing elements and the connections used at gas piping connections vary in size and pressure rating. Spiral wound metallic gaskets are a commonly used gasket today. The sealing effectiveness of these gaskets greatly depend on proper flange to flange alignment. Another commonly used connection is tubing with compression fittings. Again, proper fitting practices must be followed. If proper practices are followed when installing these connections, leakage can be eliminated. It is strongly recommended to perform a thermal pipe growth analysis, to provide allowance for pipe growth to help reduce alignment problems that may occur [4].

In the past, valve caps with a paper gasket were used to seal gases from leaking. This was not an efficient method, and the O-ring valve cap has replaced the older design. With proper installation, gas leakage at the valve caps can be eliminated. The O-rings must be replaced on a proper maintenance schedule, and workers need to be careful not to cut the O-rings. Explosive decompression is also a problem with O-rings [3]. To avoid this, it is important to check the compatibility of the O-ring material with the gas type in concern.

Compressor cylinder heads can have gas leakage if they are not properly installed. Today, metallic ring gaskets are used, which are gas tight and allow no leakage. If compressor cylinder is designed properly, and the correct installation procedures are followed, gas leakage can be eliminated [3].

Unloading devices include valve unloaders, valve pockets, head spacers, and head end clearance pockets which are mainly manually or pneumatically operated. These devices introduce a path for leakage at the actuation stem. Over the years, seal designs for the stem have improved and now offer better and longer sealing capabilities and life. If these devices and seals are installed and maintained properly, the gas leakage can be eliminated [3].

The main source of concern for compressors are leaks through the piston rod pressure packing [3][4]. The pressure packing creates a seal around the piston rod, which moves in and out of the cylinder at high speeds. These seals are designed to minimize gas leakage and it routes leaks to a collection point. These are dynamic seals, so wear is expected which will increase leakage. Seal designs have improved the stationary seals, but there has been little improvement for the seal between the packing rings and piston rod, which is the main leakage source. Leakage rates for traditional segmented packing rings are approximately 0.1 to 0.17 scfm when the packing seals are new. The leakage rate will increase as the wear of the seals increase. An “alarm” point to replace the packing seals is a leakage rate normally around 1.7 to 3.4 scfm.

The main source of fugitive methane emissions for engines is from the engine crankcase, where blow-by gases escape. Blow-by gases mainly escape around the piston rings [5]. Methane can also be emitted through the exhaust or through any other small leaks that may be present in the engine, but this project is not focused on those areas.

1.1.2 Methane Leak Audit and Leak Rates

Johnson, and Covington performed a methane leak and lost audit on a compressor and the engine running the compressor at five different compressor stations. Meaning, for this study there were five different compressors and engines being studied. Leak was defined as unintentional methane emissions, and lost was defined as designed methane emissions [6]. Data will be included for methane loses, but this project is concerned with the methane leaks as defined by Johnson and Covington. The audit focused on emissions from the exhaust of the engine, the engine crankcase, leaks on the engine, and the packing of the compressor. Figure 1-1 below show the results of the audit from the five different sites.

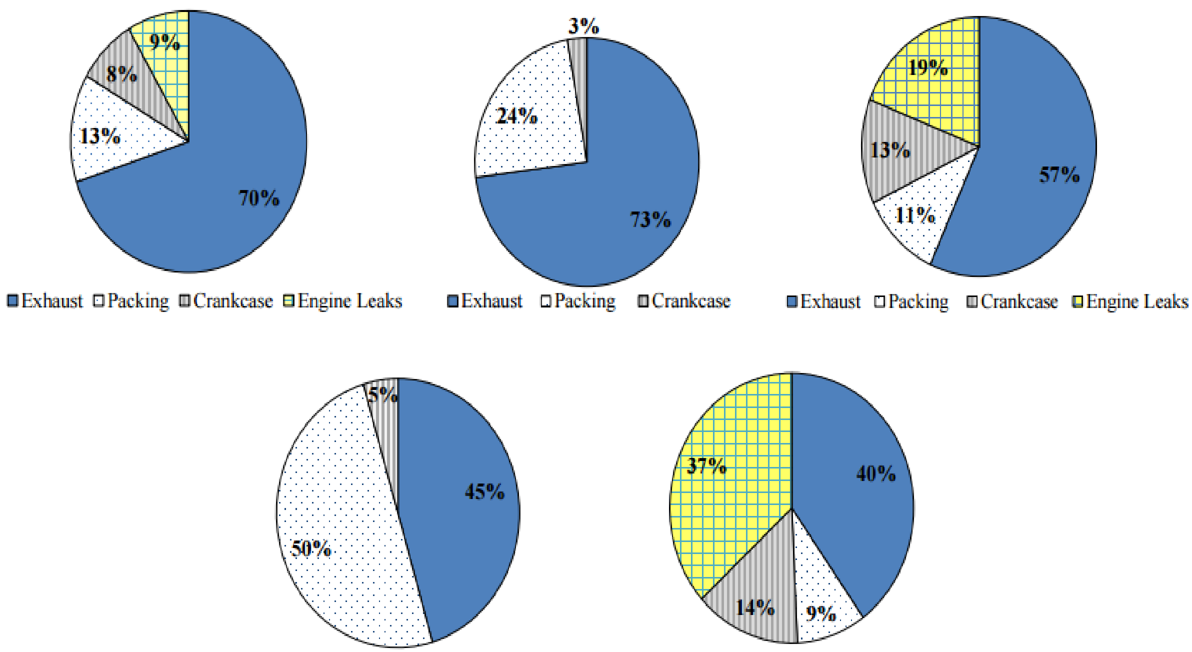

Figure 1-1: Audit Results from Five Sites. Reprinted from [6]

On the top row from left to right the pie charts illustrate the results from site one, two, and three, and the bottom row shows the results from left to right from sites four and five. It is noted that the largest source of methane leaks is from the exhaust of the engine largely due to incomplete combustion and short circuiting. This project is not concerned with this source of emissions, so the exhaust percentages from each site were taken out, and the percentages were recalculated. The adjusted results can be seen in Tables 1-1 to 1-5 below.

Table 1-1: Adjusted Results for Site 1

Source	Emissions %
Engine Crankcase	26.7
Engine Leaks	30
Compressor Packing	43.3

Table 1-2: Adjusted Results for Site 2

Source	Emissions %
Engine Crankcase	11.1
Engine Leaks	0
Compressor Packing	88.9

Table 1-3: Adjusted Results for Site 3

Source	Emissions %
Engine Crankcase	30.2
Engine Leaks	44.2
Compressor Packing	25.6

Table 1-4: Adjusted Results for Site 4

Source	Emissions %
Engine Crankcase	9.1
Engine Leaks	0
Compressor Packing	90.9

Table 1-5: Adjusted Results for Site 5

Source	Emissions %
Engine Crankcase	23.3
Engine Leaks	61.7
Compressor Packing	15

After little inspection, the compressor packing is the largest source of methane leaks. Engine leaks percentage also ranged from 0% to 61.7% of the methane leaks. This is a broad range that can skew the results dramatically. Proper maintenance and equipment installation procedures need to be followed to keep engine leaks down to zero percent, because the results show that it is possible. The data in Tables 1-1 through 1-5 were combined to calculate a total methane emissions from all five sites, with engine exhaust data still excluded. For engine crankcase, engine leaks, and compressor packing, the total methane emissions were 20%, 30%, and 50% respectively. All of these sources play a major role in the total methane emissions, but as noted earlier, compressor packing is the largest source of emissions based on the results of this audit.

Johnson and Covington also measured and reported select flow rates for the leaks at the sites. At site 1, two packing vent leak rates were measured to be 0.3 kg/hr and 0.8 kg/hour. At site 2, two packing vent leak rates were measured to be 0.022 kg/hr and 0.0016 kg/hr. At site 3, the engine crankcase leak rate was measured to be 2.2 kg/hr. At site 4, a compressor packing vent rate was measured to be 11.6 kg/hr. At site 5, a compressor packing vent leak rate was measured to be 0.35 kg/hr, and the crankcase leak rate was 1.1 kg/hr [6]. These numbers vary drastically between each site, possibly meaning that leak rates will mainly depend on the specific equipment, and it may be hard to approximate or assume an accurate and representative leak rate for simulation purposes.

1.1.3 Crankcase and Compressor Vent Gas Compositions

No article found through the literature review provided details about the composition of the vent gases drawn from the piston rod packing emissions. The compressors in focus are located on a natural gas pipeline, so it will be assumed that the composition of the gases emitting through the piston rod packing is entirely natural gas.

The composition of the gases in the engine crankcase is extremely hard to determine, or even approximate. Pav provides the pie chart in Figure 1-2 below, which illustrates the typical raw blow-by gas composition of port injection gasoline SI engines.

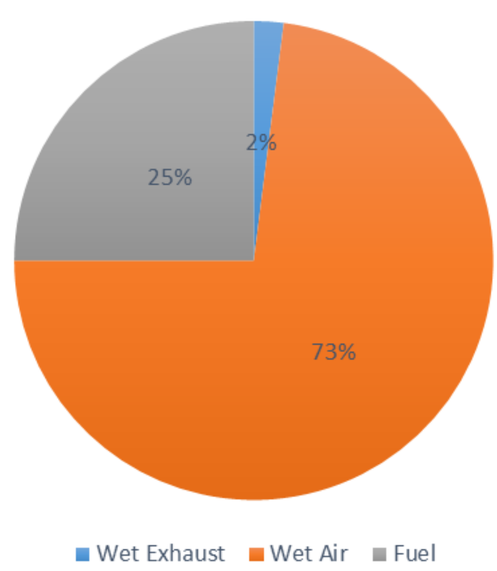

Figure 1-2: Typical Blow-by Gas Composition. Adapted from [6]

However, Pav does not define what is meant by “Wet Exhaust” and “Wet Air”. The water is also split between both of those terms and no percentage is provided for how much water is in “Wet Exhaust” nor “Wet Air”. This project also deals with a gasoline engine, not a methane engine. This composition is not ideal, but at the very least, it can provide a ground for comparison, and it can also be a useful source for initial conditions when running simulations. Pav also claims that exhaust gas and water steam fraction in the raw blow-by gas varies as a function of engine speed [7].

The Caterpillar Crankcase Ventilation Application and Installation Guide offered similar statements. Caterpillar states that the components expected to be found in the engine crankcase are wear particles, oil, fuel, gas, and air [8]. The term gas represents the products of combustion for the particular fuel used. Although the components found in blow-by are known, their specific composition varies on multiple parameters. The type of engine, age of the engine, fuel type, engine speed, load, and the previous maintenance history of the engine are all parameters that affect the specific composition, and volume of the blow-by gas. Cylinder pressure, piston ring pressure, and component wear will also change the volume of blow-by gases [9]. There will need to be additional controls added due to the fact the blow-by gases will cause the intake to be richer than before, so less fuel will need to be injected to maintain the desired air-to-fuel ratio [9].

Further experimentation needs to be conducted to help relate all the different parameters to their effects on blow-by gas composition. Without knowing what the composition of the blow-by gas is, it is difficult to determine how combustion will be affected with the introduction of these gases into the air intake system of an engine. A sensitivity analysis should be run with varying inlet gas compositions, to see how CCV can affect the combustion process.

1.1.4 Oil Damage to Engines

The main problematic component in the blow-by gas is the oil that is used in the crankcase to help lubricate the engine. If the blow-by gas is used in closed crankcase ventilation, it can help contribute to oil consumption [5][10][11]. When the oil is introduced into the engine, it causes air intake system fouling, and it helps poison the exhaust catalyst. Oil consumption can also lead to hydrocarbon deposits on several different pieces of equipment within the equipment such as valves, pistons, piston walls, and the intercooler. The image in Figure 1-3 below shows two sets of inlet and outlet valves [12]. The hydrocarbon deposits can easily be seen on the inlet valves compared to the outlet valves. This is going to negatively affect the intake flow, and volumetric efficiency which will decrease engine performance.

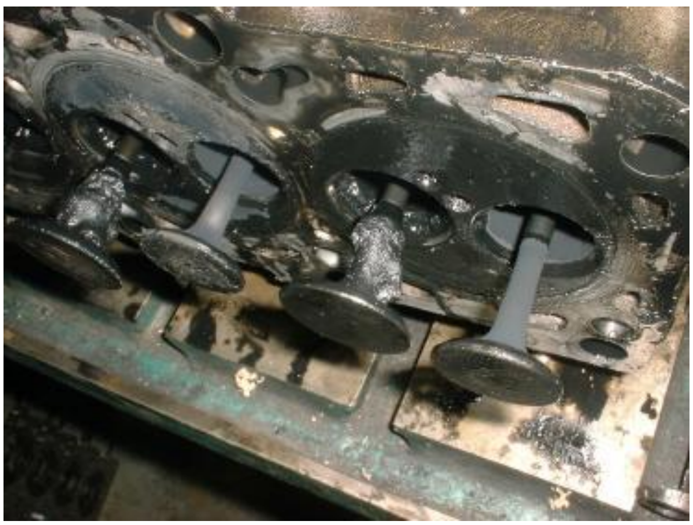

Figure 1-3: Hydrocarbon Deposits on Inlet Valves. Reprinted from [10]

Similar to Figure 1-3, this kind of deposit on the piston and piston walls will negatively affect combustion, and it will further increase blow-by leading to more oil consumption. The deposits can also heavily damage the piston and piston walls. When it occurs on intercooler walls, heat transfer will be effected and the engine will not be performing properly. Oil consumption can cause a large variety of engine problems, so there is a need for a system that can separate the oil contained in blow-by gases.

1.1.5 Oil Separation Methods

There are several different methods of separating oil from the blow-by gases. Candy and Guerbe discuss a couple of this methods.

Starting in the early 2000's the main method to separate the oil was the use of a cyclone separation system, which is represented in Figure 1-4 below.

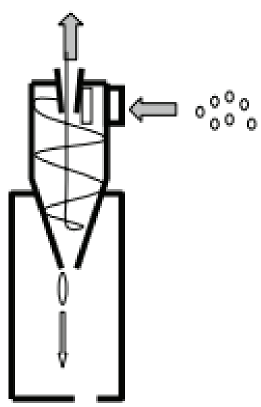

Figure 1-4: Diagram of a Cyclone Separation System. Reprinted from [5]

The inlet of the system accelerates the gas stream, and the stream rotates into the cylindrical body of the oil separator. The centrifugal force cause the oil droplets to impact the walls and slide down toward the oil collection tank. The clean gases leave the system through the opening in the top. One single system is normally not efficient enough, so they are designed with several cells set in parallel. One main advantage of the cyclone separate method is that there is little gas flow restriction. The main problem though is the system is not efficient when the oil droplets are 0.4μm or smaller in size [5].

A growing method today is the use of coalescing separators, which uses a fiber medium to catch the oil droplets. When the gas is introduced through the fiber medium, the oil droplets are collected along the fibers and collect together forming larger oil droplets. The air flow in the oil's mass pushes the oil droplets towards the lower part of the system, and the oil is evacuated by the drain and returned to the oil sump. When compared to the cyclone method, the coalescing separator system is more efficient, but the resulting air restriction can be high [5].

There are other methods that will be listed, but not discussed in detail as they can be expensive, consume considerable amounts of energy, or are not commonly used. One method being electrostatic separators. An electric field deflects the oil mist into the walls of the system, and then slides to the bottom and is drained out. A second method is using centrifugal oil separators. In this method, an oil pan rotates at a high speed which makes the gas stream rotate. This results in the oil droplets impacting the housing walls and is drained from the system. The last system is the rotating coalescing separator. This system is similar to the coalescing separators, but the system rotates with engine rotation speed. The oil droplets are caught in the fiber medium, and are removed from the separator due to centrifugal force [5].

Today, the most commonly used method is the cyclone separation system. This method is not efficient enough to meet the growing needs of a closed crankcase ventilation system. Coalescing separators are much more efficient, but they introduce a large pressure drop in the air flow. Low pressure coalescing separators do exist, and is recommended for use in a CCV system if low air flow restriction is desired. Krause, Spies, Bell, and Ebert ran an experiment to compare cyclone separators and separators with staple-fiber nonwovens that utilize a coalescing method. The results from the experiment demonstrated that using staple-fiber nonwovens allows the separation efficiency to be tripled compared to simple case-type separators [10].

1.1.6 Experiments & Conclusions Made

Xiao, Sohrabi, and Karim ran an experiment to determine the successfulness of introducing methane into the intake of a swirl chamber diesel engine [13]. They modified an existing diesel engine facility to make it suitable for their study. A diagram of the experimental apparatus can be seen in Figure 1-5 below.

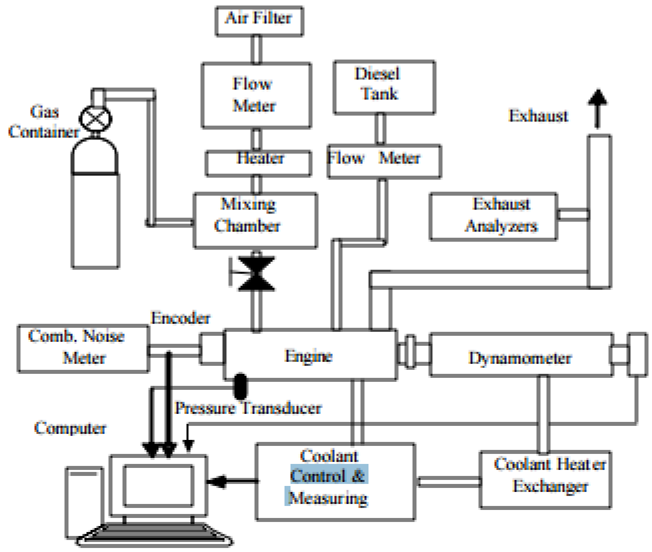

Figure 1-5: Schematic of Diesel Engine Testing Apparatus. Reprinted from [13]

The team introduced small amounts of methane into the engine intake system under various operating conditions. The diesel equivalence ratio, engine speed, and inlet air and coolant water temperature were each varied separately while holding the other parameters constant. Results from the experiment showed that 53% to 80% of the methane introduced into the system was converted into CO_2. The addition of the methane can increase power output, increase the exhaust temperature, and reduce specific diesel fuel consumption. And there was an increase of CO emissions which means that part of the methane introduced into the system did not fully oxidize [13]. This experiment does not exactly represent a CCV system that this project is investigation, but the results indicate what can be expected from this type of system.

Parker conducted an experiment that rerouted dry gas seal vent gas from a compressor into the intake of a gas turbine. A schematic of the experimental apparatus can be seen below in Figure 1-6.

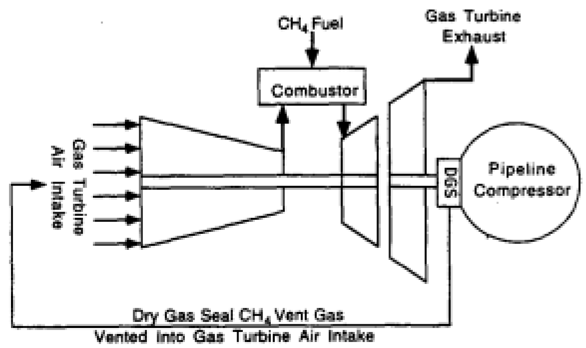

Figure 1-6: Schematic of the Gas Turbine Apparatus. Reprinted from [14]

The methane destruction from this experiment is comparable to the greenhouse gas destruction efficiency of flaring or incineration (86.9%). This approach is a cost-effective solution. Portions of the methane introduced into the gas turbine was emitted as CO. Again, this system is not the same as the system this project is concerned with, but the results can apply. The study shows that a significant percentage of the methane can be converted into CO_2 through combustion, instead of released into the atmosphere [14]. Neither of these experiments breathe the methane into a large 2-stroke engine, so the percentages and results can easily vary. What these experiments have in common is that the methane was injected into a system where it was combusted into carbon dioxide with large conversion rates. These studies show that rebreathing does work in reducing methane emissions, but how effective will it be on a 2-stroke engine still needs to be determined.

1.1.7 Pros of Closed Crankcase Ventilation

Experimentation has shown the different methods of converting fugitive methane emissions into carbon dioxide are successful. When run through a diesel engine, methane was converted at a rate of 80% when added in very small amounts [12]. When the methane emissions were routed to a gas turbine, the methane destruction efficiency was 86.9%. This high percentages indicate that a CCV system on a natural gas pipeline has the opportunity to be extremely successful. With the goal of reducing methane emissions, fumigating the blow-by gases also means that there are zero blow-by emissions, which contain several other combustion process species. By eliminating this source of emissions, it is expected to help reduce smog [15].

There are also other unintentional benefits with using a closed crankcase ventilation system. A CCV system will maintain a vacuum on the engine crankcase to draw away the blow-by gases. The vacuum will minimize the volume of gases in the crankcase which will lower the possibility of a crankcase explosion [16]. With the recovery of methane gases that would otherwise be vented to the atmosphere, there is an increase in volumetric efficiency, reduce specific fuel consumption, and increase power production [11][13][14]. There are other means of converting methane into carbon dioxide, such as using a flare that has a high conversion rate, but this is a wasteful method as it loses the energy in the methane. A CCV system will allow this lost energy, either through emissions or other wasteful conversion methods, to be utilized in the engine [17].

1.1.8 Cons of Closed Crankcase Ventilation

One of the main problems with CCV is the oil that is trapped in the engine blow-by gases. No matter how efficient the oil separation system is, oil will end up in the engine. This leads to air intake system fouling, exhaust catalyst poisoning, increased emissions, and resinous deposits on the intercooler, inlet valves, and piston ring grooves which can lead to decreased heat transfer, decreased volumetric efficiency, and further increased blow-by and oil consumption [5][10][18]. Oil separation systems that have a large oil separation efficiency typically have some sort of draw back. When it comes to coalescing separators, when they reach high efficiencies, they introduce a high air flow restriction into the system [5]. Not only will the oil harm the engine, and can also have negative effects on the CCV system itself. For example, the oil can be deposited onto a flow regulator valve and restrict the flow of the blow-by gases [18]. If a CCV is put into place, more rigorous and frequent maintenance plans must be followed to ensure proper engine operation.

When methane is introduced into the air intake, the goal is to convert 100% of the methane into CO_2, but this is an unrealistic goal to achieve. While studies have shown that a sizeable percentage can be converted into CO_2, the remaining percentage either partially oxidizes into CO, or remains methane. The carbon monoxide and methane is then released into the environment. While CO is not a greenhouse gas, it is a pollutant and has adverse effects in the upper atmosphere which can result in respiratory problems for animals [14].

Adding in a CCV system will add in more equipment, and complexity to the engine and compressor. There will be a loss of money due to operation of the new equipment which can include but is not limited to pumps, oil separators, and safety equipment. There will also be a need for upgrades to the controls that are currently in place. The two flows from the compressor and engine crankcase will need to be controlled to keep the air to fuel ratio the same. The boost pressure, and the fuel injection rate will also need to be injected to keep the trapped equivalence ratio, and the energy delivery rate the same.

1.1.9 Current CCV Technology

Two patents were found that detailed the entirety of a CCV system. The first one is titled "Engine crankcase ventilation". This design utilizes a pressure actuated regulating valve downstream of a fixed orifice to control the flow of the engine crankcase gases into the induction system. The device also has a controlled vacuum during normal conditions, but will allow pressure build-up during moments of excessive blow-by to allow other pressure actuated shutdown devices to operate [16]. A drawing of the system can be seen below in Figure 1-7.

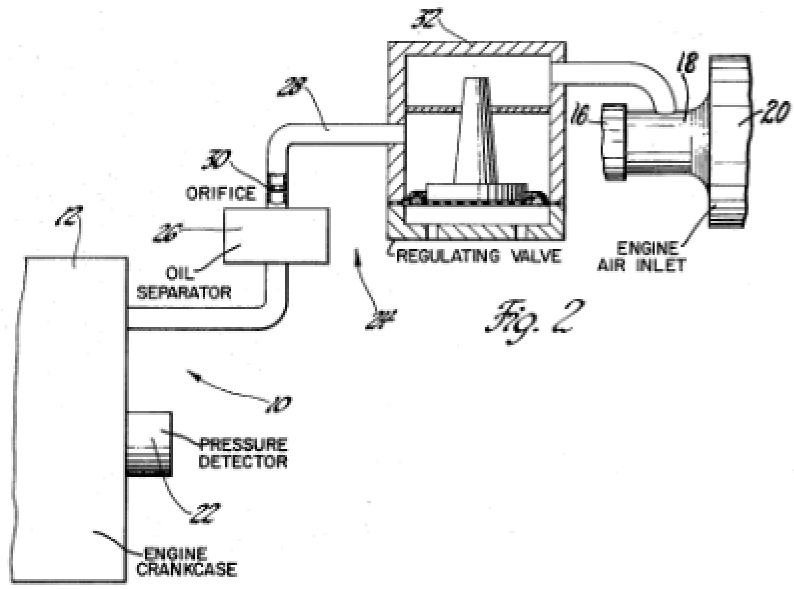

Figure 1-7: Drawing of Engine crankcase ventilation system. Reprinted from [17]

The second patent is titled "Greenhouse gas capture system and method", and it is extremely similar to the first patent design. Where the second patent differs is that the fluid drawn from the engine is diluted with a non-combustible fluid before the stream is sent to the engine for combustion [17].

There are several crankcase ventilation systems out in the market, but a large majority of these systems or for application in the automotive industry. There are very few CCV systems that are designed to operate on larger compressors and 2-stroke engines.

Caterpillar offers an ingestive, low pressure, positive crankcase ventilation system on their natural gas G3520C engine [8]. When using this system, all operation and maintenance procedures need to be strictly followed, and it should be expected that maintenance costs will be high using this device. Caterpillar also offers several recommendations to ensure the proper function of a positive crankcase ventilation system. A cleanable aftercooler should be used and maintained regularly. All blow-by gases must be filtered before being sent to the turbocharger, or engine intake. The system cannot freeze when it is operating in low ambient temperatures. The system must be able to handle two times the engine blow-by flow rates to be prepared for engine wear. Oil must be removed at a rate of 99.97%. The overall system must have a bypass for if the crankcase over pressurizes due to clogged filters [8]. These recommendations show that having a crankcase ventilation system adds complexity to the system and maintenance routine at the site.

REM Technology offers various models for its SlipStream® system. The SlipStream® utilizes vented hydrocarbons and uses them as a supplementary fuel source for natural gas engines. The system also has monitors and controls to ensure safe and reliable engine operation [19]. The SlipStream® can be used on a wide range of engines with an operating limit of 100 to 4,000 horsepower. There are three models for this system, the SS3, SS10, and SS50. The SS3 has a maximum flow rate of 3 kg/hr, the SS10 has a maximum engine fuel supplementation of 10%, and the SS50 has a maximum engine fuel supplementation of 50% [19].

2 SIMULATION SET UP

2.1 Real Engine Set Up

The engine being modeled in the simulations is a large bore, natural gas, 4-cylinder, Cooper-Bessemer GMV-4 2-stroke engine. This study is focused on using a large bore, 2-stroke engine to provide the combustion process for the conversion of methane into carbon dioxide, because that is the equipment that is convenient for use at compressor stations. The engine can be seen below in Figure 2-1. The engine runs at a slow speed of 300 rpm with a 14” (35.6 cm) bore, and a 14” (35.6 cm) stroke. The engine runs lean, and the fuel is directly injected into the cylinder. The fuel mass flow rate is 190 lb/hr, based on data received from CSU which can be found in Appendix A [19]. Each cylinder has a pre-combustion chamber, and it should be noted now that the combustion chamber was not modeled in GT-Power. The air supply for the engine is provided via a supercharger that is simulating a turbocharger. The air mass flow rate per cylinder is approximately 1960 lb/hr. The engine does not have a system for closed crankcase ventilation, this source of gases will need to be added in to the simulation model of the engine. It is noted that this engine is located in Colorado, so the atmospheric conditions are different there compared to sea level conditions, and the test space and model will be built incorporating those ambient conditions.

Figure 2-1: Cooper-Bessemer GMV-4 at CSU. Reprinted from [20]

2.2 Test Space Set-Up

From the literature review, it was determined that a sensitivity analysis needed to be performed to check engine performance for the various possibilities from rebreathing fumigated emissions. The analysis will adjust the rebreathed gas concentrations and the volume of rebreathed gases. Various test spaces were designed to encompass all the different conditions an engine may operate at. The results of the analysis will be observed to determine if the engine performance will be greatly affected by the addition of the rebreathed gases.

The parameters that are being varied in this study are mass flow percent of rebreathed gases introduced into the engine, composition of the rebreathed gases, boost pressure, and fuel flow rate. Each of these will be explained in more detail as the section continues. It is desired to keep the amount of gases rebreathed into the engine intake manifold small compared to the total volume of gases in the cylinder. To simplify calculations, the percentages will be based on mass flow rates for the air intake. The rebreathed mass flow rates were decided to be 1%, 3%, and 5% of the engine air intake for the engine crankcase gases, and 1%, 3%, and 5% of the fuel injection flow rate for the compressor gases.

The next parameter varied was the composition of the rebreathed gases. This was simply done by running different simulations with the focus completely on the engine crankcase gases or the compressor gases. For example, during the case for 5% of rebreathed gases, one test will have the 5% be composed of gases from the engine crankcase, and a different test will have the 5% be composed entirely of gases from the compressor. This also allowed for easier post analysis calculations to determine the percentage of methane reduction from each source. Based on the findings from the literature review, the engine crankcase gases will be modeled as products of combustion, including air and water. It is assumed that 100% of the oil particles are filtered out, and that there are no other participates in the gases. In blow-by gases, a typical measurement will show that methane is 1500-3000 ppm. Another parameter varied was having methane in the blow-by gases vary from 1500 ppm to 3000 ppm. The compressor gases will be modeled as pure methane.

When designing the test space there were two engine parameters that needed to be constant throughout the different cases, trapped equivalence ratio and energy delivery rate. True TER control accounts for the differences between air and residual products by including a scavenging model [21]. Equations 2-1 and 2-2 below are used to first calculate the uncorrected trapped equivalence ratio and the scavenging ratio, respectively. Equation 2-3 is then used to determine the scavenging efficiency, which is then used in Equation 2-4 along with the original TER to finally get the corrected TER [21].

$$\varphi=\frac{AFR_{stoich}*\text{Fuel Flow}*\rho_f}{\text{Speed}*V_{trap}*\frac{(AMP+Baro)}{29.92}*\frac{520}{(460+AMT)}*\rho_{air}} \tag{2-1}$$

$$RS=\frac{\text{Airflow}}{\text{Speed}*V_{trap}*\frac{(AMP+Baro)}{29.92}*\frac{520}{(460+AMT)}} \tag{2-2}$$

$$SE=1-e^{-RS} \tag{2-3}$$

$$\varphi_{corr}=\frac{\varphi}{SE} \tag{2-4}$$

Since gases, which include methane, are being introduced into the engine air intake, the airflow and the fuel flow will be changing for each case. The TER was kept constant by adjusting the boost pressure. The energy delivery rate was kept constant by adjusting the fuel flow rate into the engine from direct injection. Rebreathing the gases will add varying amounts of fuel into the air intake, which will end up in the cylinder. To keep the energy delivery rate constant, the fuel injection needed to be reduced as to not create a richer burn.

Varying all five of the parameters listed earlier, the test space for the study was developed. The test space can be viewed in Tables 2-1 through 2-9 below.

Table 2-1: Test space for 1% engine crankcase gases with 1500 ppm methane

	Gas Location	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
1% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 1500 ppm Methane	0.00	3.90	7.85	11.76	15.71	19.61
	Intake Air (lb/hr)	1960.00	1956.10	1952.15	1948.24	1944.29	1940.39
	Fuel Flow (lb/hr)	48.000	47.990	47.980	47.969	47.959	47.949
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-2: Test space for 3% engine crankcase gases with 1500 ppm methane

	Gas Location	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
3% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 1500 ppm Methane	0.00	11.76	23.51	35.32	47.03	58.83
	Intake Air (lb/hr)	1960.00	1948.24	1936.49	1924.68	1912.97	1901.17
	Fuel Flow (lb/hr)	48.00	47.97	47.94	47.91	47.88	47.85
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-3: Test space for 5% engine crankcase gases with 1500 ppm methane

	Gas Location	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
5% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 1500 ppm Methane	0.00	19.61	39.22	58.83	78.44	98.05
	Intake Air (lb/hr)	1960.00	1940.39	1920.78	1901.17	1881.56	1861.95
	Fuel Flow (lb/hr)	48.00	47.95	47.90	47.85	47.80	47.75
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-4: Test space for 1% engine crankcase gases with 3000 ppm methane

	Gas Location	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
1% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 3000 ppm Methane	0.00	3.90	7.85	11.76	15.71	19.61
	Intake Air (lb/hr)	1960.00	1956.10	1952.15	1948.24	1944.29	1940.39
	Fuel Flow (lb/hr)	48.000	47.980	47.959	47.939	47.918	47.898
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-5: Test space for 3% engine crankcase gases with 3000 ppm methane

	Gas Location	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
3% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 3000 ppm Methane	0.00	11.76	23.51	35.32	47.03	58.83
	Intake Air (lb/hr)	1960.00	1948.24	1936.49	1924.68	1912.97	1901.17
	Fuel Flow (lb/hr)	48.00	47.94	47.88	47.82	47.76	47.69
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-6: Test space for 5% engine crankcase gases with 3000 ppm methane

	Gas Location	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
5% Rebreathed Gases in Cylinder	Engine Crankcase (lb/hr) 3000 ppm Methane	0.00	19.61	39.22	58.83	78.44	98.05
	Intake Air (lb/hr)	1960.00	1940.39	1920.78	1901.17	1881.56	1861.95
	Fuel Flow (lb/hr)	48.00	47.90	47.80	47.69	47.59	47.49
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-7: Test space for 1% compressor vented gases

	Gas Location	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
1% Rebreathed Gases in Cylinder	Compressor Vent Methane (lb/hr)	0.00	0.09	0.19	0.29	0.38	0.48
	Intake Air (lb/hr)	1960.00	1959.91	1959.81	1959.71	1959.62	1959.52
	Fuel Flow (lb/hr)	48.000	47.910	47.810	47.710	47.620	47.520
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-8: Test space for 3% compressor vented gases

	Gas Location	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
3% Rebreathed Gases in Cylinder	Compressor Vent Methane (lb/hr)	0.00	0.29	0.58	0.86	1.15	1.44
	Intake Air (lb/hr)	1960.00	1959.71	1959.42	1959.14	1958.85	1958.56
	Fuel Flow (lb/hr)	48.000	47.712	47.424	47.140	46.850	46.560
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

Table 2-9: Test space for 5% compressor vented gases

	Gas Location	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
5% Rebreathed Gases in Cylinder	Compressor Vent Methane (lb/hr)	0.00	0.48	0.96	1.44	1.92	2.40
	Intake Air (lb/hr)	1960.00	1959.52	1959.04	1958.56	1958.08	1957.60
	Fuel Flow (lb/hr)	48.000	47.520	47.040	46.560	46.080	45.600
	Boost Pressure (inHg)	41	41.05	41.1	41.15	41.2	41.25

For all cases, the intake air flow decreased by the same value as rebreathed flow. The fuel flow also decreased by the same value as rebreathed fuel addition from the methane concentration in the gases. The beginning case for each table always starts with zero rebreathed gases to build a baseline to compare the other 5 cases.

2.3 Simulation Model

The engine was modeled using the computer software GT-Power. For model simplification, and to make troubleshooting easier, only one cylinder was modeled. Realistically, all four cylinders will not be operating under the same exact conditions, and the results will vary slightly, but can be representative of all four cylinders for the scope of this project. The 1-cylinder GT-Power model for the rebreathed engine crankcase gases with 3000 ppm methane can be seen below in Figure 2-2. The top left boxed labeled “inlet” provides the states, temperature, composition, and pressure, of the air flow into the engine. Downstream of the air intake is a control system that will measure and control the flow of air to the desired value listed in the test space. On the bottom left of the model, the box labeled “EngineCrankcase(3000)” provides the rebreathed gas flow. This box contains the fuel composition, temperature, and pressure. Downstream of this box, is a control system to measure and control the flow to the desired value listed in the test space. These two streams are mixed together and enter and exit the cylinder through the inlet and exhaust valves. The product composition is then exits the model to the box labeled “exhaust”.

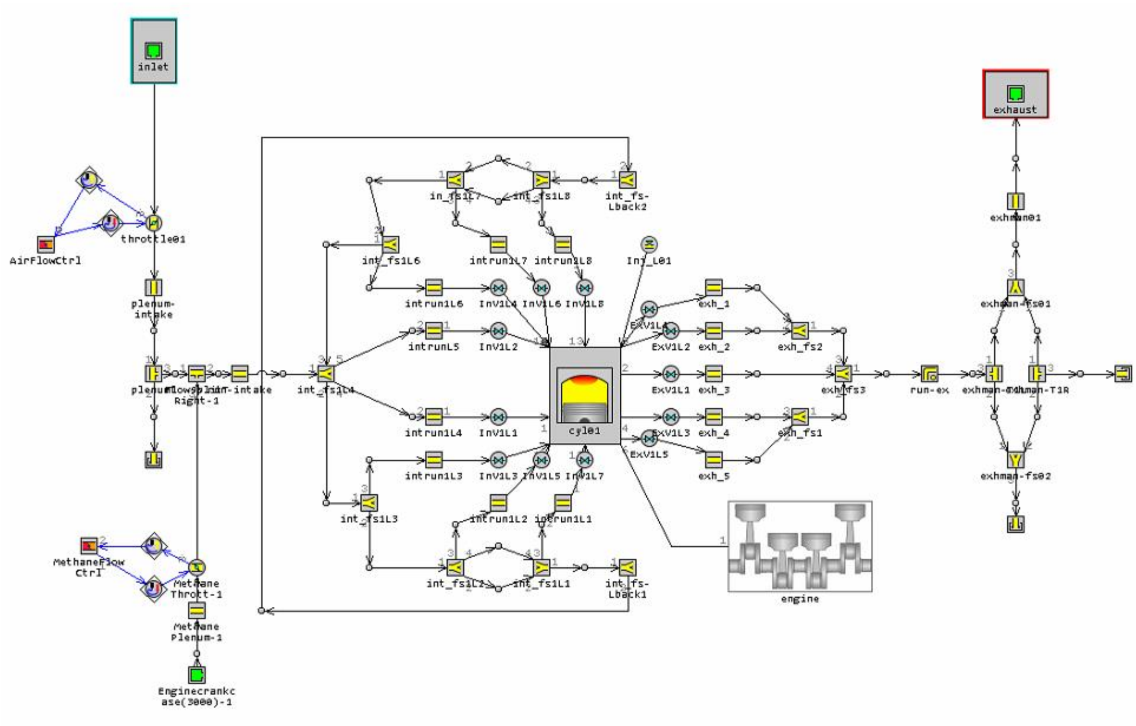

Figure 2-2: 1-Cylinder GT-Power model for rebreathed engine crankcase (3000 ppm) simulation

The GT-Power models for the rebreathed engine crankcase with 1500 ppm methane, and compressor vent gases are extremely similar to the model shown in Figure 2-2. The only difference physically between the two models is the box labeled "EngineCrankcase(3000)" changes for the different sources of fugitive methane emissions. These models can be seen in Appendix B.

2.4 Simulation Validation

The results from this study cannot be assumed accurate unless validated by experimental data from the Copper-Bessemer GMV-4 engine. CSU provided experimental data that was overlaid against the results of this study. This data can be found in more detail in Appendix A.

2.5 Methane Reduction Calculations

This study's main goal is to determine the reduction of fugitive methane emissions through combustion. There are several different flows of methane, so to help visualize these flows, a simple control volume schematic was created and can be seen below in Figure 2-3. On the left side of the cylinder, there is methane flow into the cylinder through the intake valves from the rebreathed gases. On top of the cylinder, methane is injected into the cylinder through the direct injection system. On the right side of the cylinder, two flows represent the methane that escapes through short circuiting, and unburned methane in the exhaust. On the bottom of the cylinder, the last flow represents the methane that escapes the cylinder through the blow-by gases. Since these gases are going to be collected and rebreathed into the engine air intake, this flow will be considered a part of the control volume.

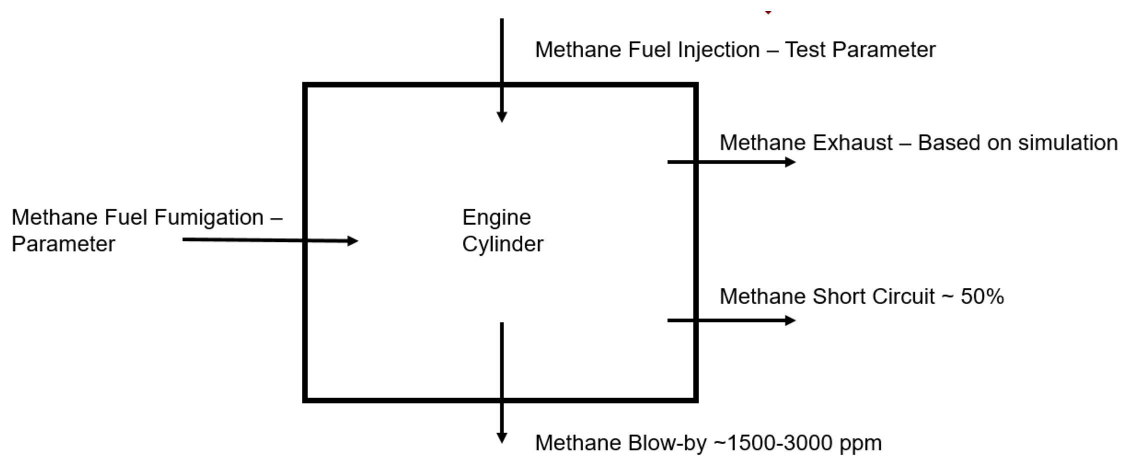

Figure 2-3: Schematic of control volume

From this control volume, Equation 2-5 below was used to calculate the percent methane reduction.

$$\text{Reduced Methane \%} = \frac{(\text{FM-SCM-EM})}{\text{FM}} * 100 \qquad (2\text{-}5)$$

This equation subtracts the methane from the fumigated source that escapes through cylinder by short circuit, and the methane from the fumigated source that is unburned in the exhaust, to determine how much methane, on a mass flow basis, was reduced through the combustion process. A few more steps need to be followed to determine the right value for the fumigated methane, methane in the short circuit, and methane in the exhaust which was originally from the fumigated methane source and not the fuel injected source.

Starting with the fumigated methane, if the compressor vents is the sources, then the composition of the gas will be methane. If the engine crankcase is the source, then the gas will be composed of methane, products of combustion for this simulation. It is assumed that methane is 1500-3000 ppm in blow-by and exhaust. Using the mass flow rate from the engine crankcase, the methane flow rate can be found using Equation 2-6 below.

$$m_{FM} = m_{crankcase} * X_{CH4,crankcase} \qquad (2\text{-}6)$$

It is assumed that the methane that short circuits the cylinder will be from the fumigated methane entirely. There is very little to no time for the methane injected into the cylinder to escape out the exhaust valve, so all the methane is the fumigated methane. The short circuit gases will also contain air, and other products of combustion, so the mass flow rate of methane in the short circuit gases needs to be determined.

Equation 2-7 below is used to calculate the mass flow rate of the short circuit gases. Equation 2-8 is the fraction of fumigated methane mass flow rate to the total mass flow rate. Assuming that the fumigated methane is evenly distributed throughout the cylinder and intake air, Equation 2-9 is used to determine the mass flow rate of fumigated methane that short circuited the cylinder.

$$m_{SC} = m_{total} * (1\text{-}TER) \tag{2-7}$$

$$\%_{CH4,\ intake} = \frac{m_{FM}}{m_{total}} \tag{2-8}$$

$$m_{SCM} = m_{SC} * \%_{CH4,intake} \tag{2-9}$$

The remaining flow that did not short circuit will be considered the exhaust flow. The test parameter for the particular case will determine what the methane concentration is in the exhaust. Similar to Equation 2-7, Equation 2-10 is used to determine the methane flow in the exhaust. Equation 2-11 is then used to find the ratio of fumigated methane trapped in the cylinder to the total amount of methane in the cylinder. The methane exhaust flow rate is multiplied by this ratio in Equation 2-12 to find the amount of fumigated methane that is in the exhaust.

$$m_{CH4,exhaust} = (m_{total}\text{-}m_{SC}) * X_{CH4} \tag{2-10}$$

$$\%_{FM,\ CH4} = \frac{m_{FM,trapped}}{m_{CH4,total}} \tag{2-11}$$

$$m_{EM} = m_{CH4,exhaust} * \%_{FM,CH4} \tag{2-12}$$

Now the values for all the terms for Equation 2-5 have been determined, and the percent methane reduction can be calculated.

3 RESULTS AND DISCUSSION

3.1 Engine Stability

Each simulation is run until the engine reaches steady state conditions. GT-Post was used to analyze and develop plots of the results. To observe how the sensitivity analysis would affect the combustion performance of the engine, the pressure curves were plotted against each other to compare the cases to themselves and the baseline case. Figures 3-1, 3-1, and 3-3 below plot the pressure curves for the engine crankcase gases with 1500 ppm methane cases.

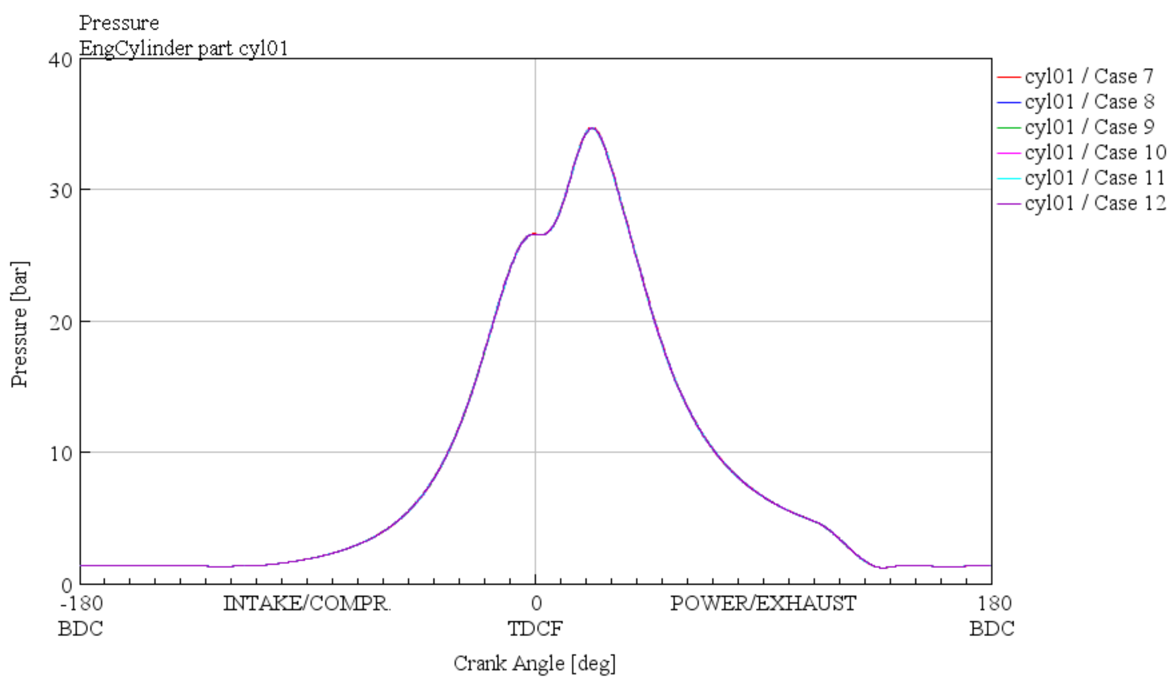

Figure 3-1: Pressure curve results for 1% engine crankcase gases with 1500 ppm methane

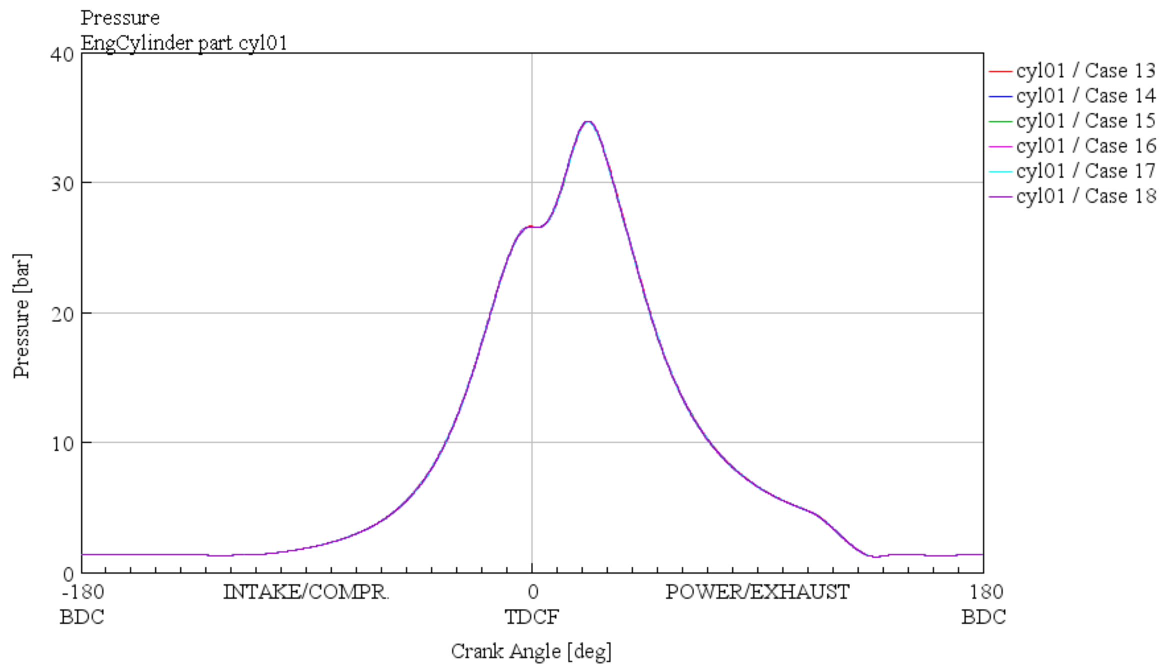

Figure 3-2: Pressure curve results for 3% engine crankcase gases with 1500 ppm methane

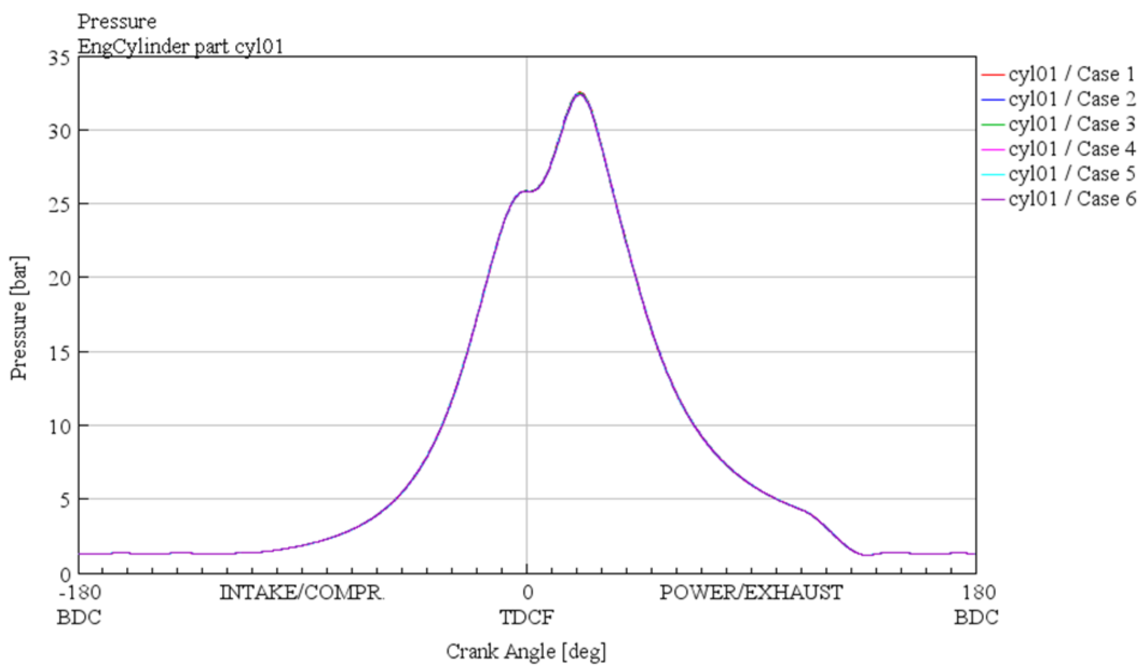

Figure 3-3: Pressure curve results for 5% engine crankcase gases with 1500 ppm methane

The results from the above plots are promising. The pressure curves show little to no variation between each case. It appears the curves line up on top of each other for all portions of the curve, compression stroke, exhaust stroke, and peak pressure. Looking at the three plots it also appears that there is no variation between the 1%, 3%, and 5% cases. These results will point to combustion not being affected with the addition of the engine crankcase gases with methane at 1500 ppm. If the pressure remains constant, then it can be assumed that the temperature and other engine performance measures such as IMEP are constant as well. A quick look into GT-Post proved this was true, and results for the IMEP for these cases can be found in Appendix C. There is not an increase in the probability of knock or pre-ignition in the cases above.

Similar results can be seen for the engine crank case gases with 3000 ppm. These pressure curves can be seen in Figures 3-4, 3-5, and 3-6 below. Once again, there is little to no variation between the pressure curves on each plot, and in between the three plots. In fact, the plots are almost, if not, identical to the pressure curves for the first three cases presented. It seems that the methane concentration in the exhaust and blow-by has little to no effect on the performance of the engine. Again, the probability of knock or pre-ignition does not increase with the addition of rebreathed gases from the engine crankcase. Extra tables regarding the IMEP values can be found in Appendix C.

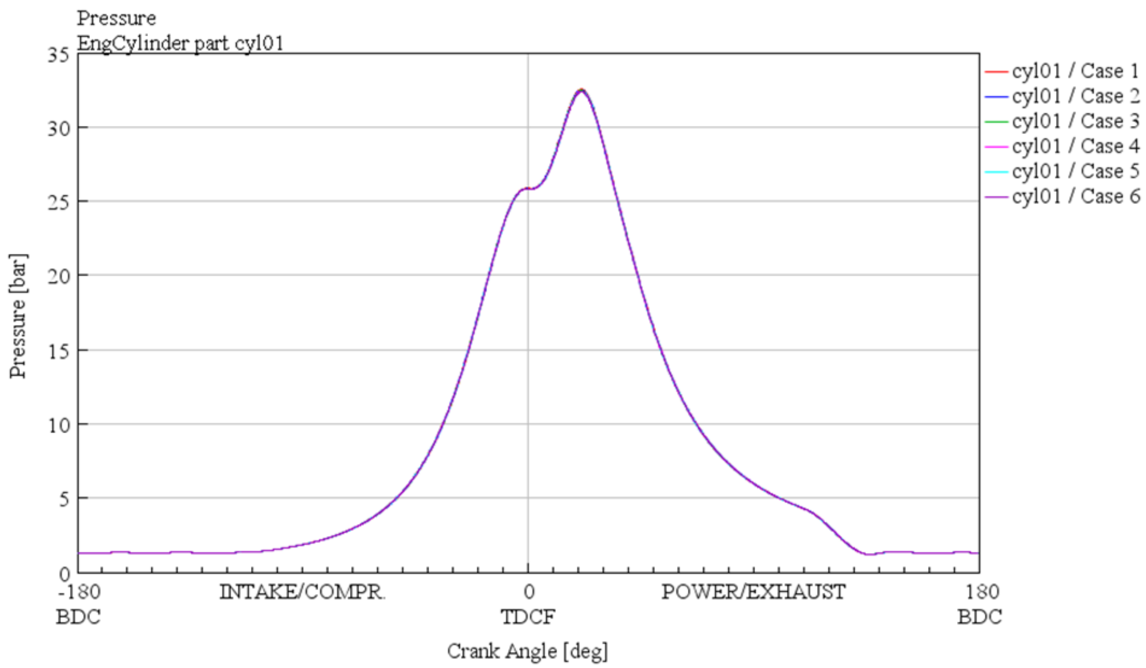

Figure 3-4: Pressure curve results for 1% engine crankcase gases with 3000 ppm methane

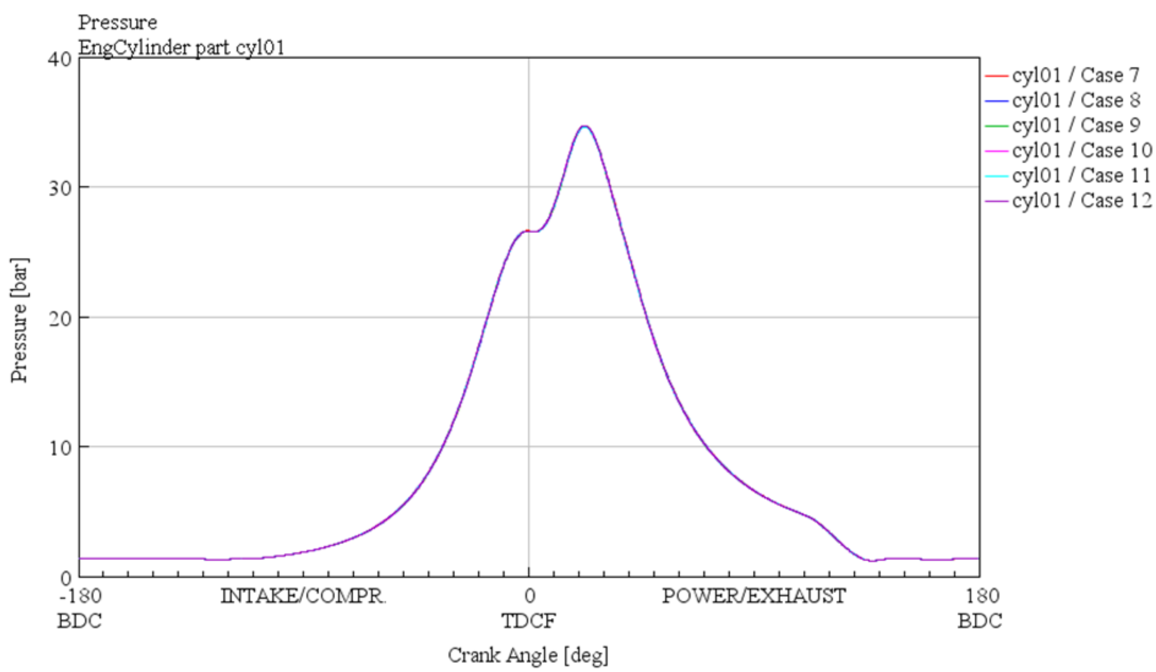

Figure 3-5: Pressure curve results for 3% engine crankcase gases with 3000 ppm methane

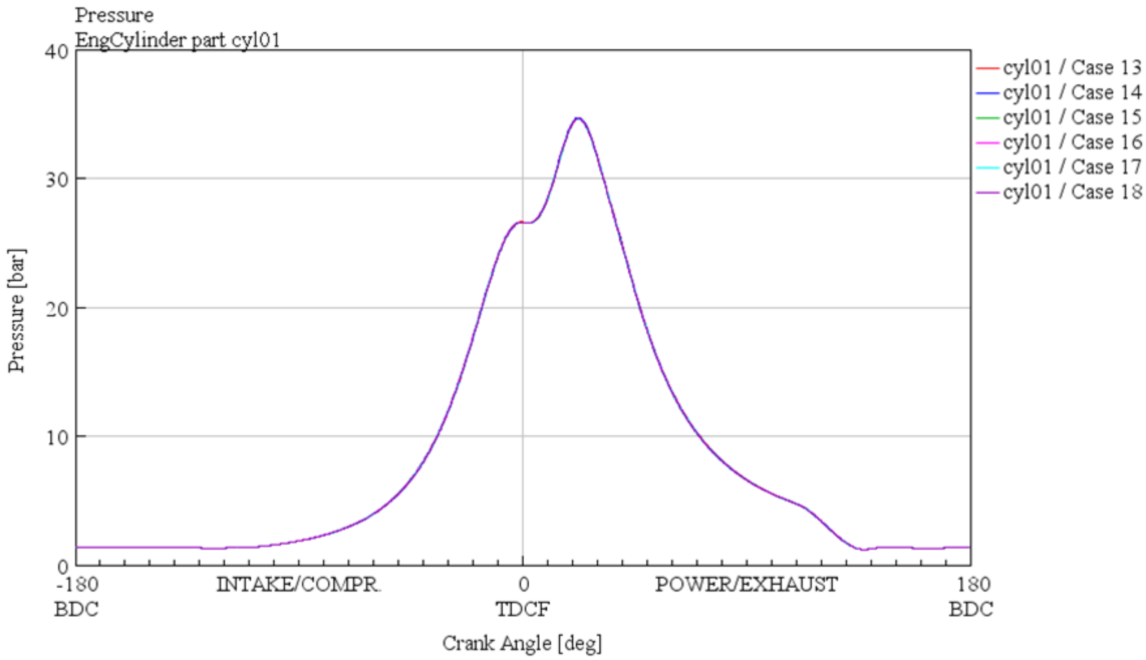

Figure 3-6: Pressure curve results for 5% engine crankcase gases with 3000 ppm methane

The pressure curve results for the rebreathed compressor vent cases can be seen below in Figures 3-7, 3-8, and 3-9. On all three plots, the compression and the exhaust strokes maintain the same pressure curves amongst all the 18 different cases. The only portion that changes is the peak pressures. The difference in peak pressures is relatively small reaching a maximum difference of 1.5% between the baseline and the largest fumigated mass flow rate case. Despite the differences in peak pressures, the engine performance remained unchanged. The IMEP for all 18 cases with compressor vent gas addition were constant. The increase in probability of pre-ignition and knock due to the higher temperatures is negligible since the change in temperature and engine performance is extremely small.

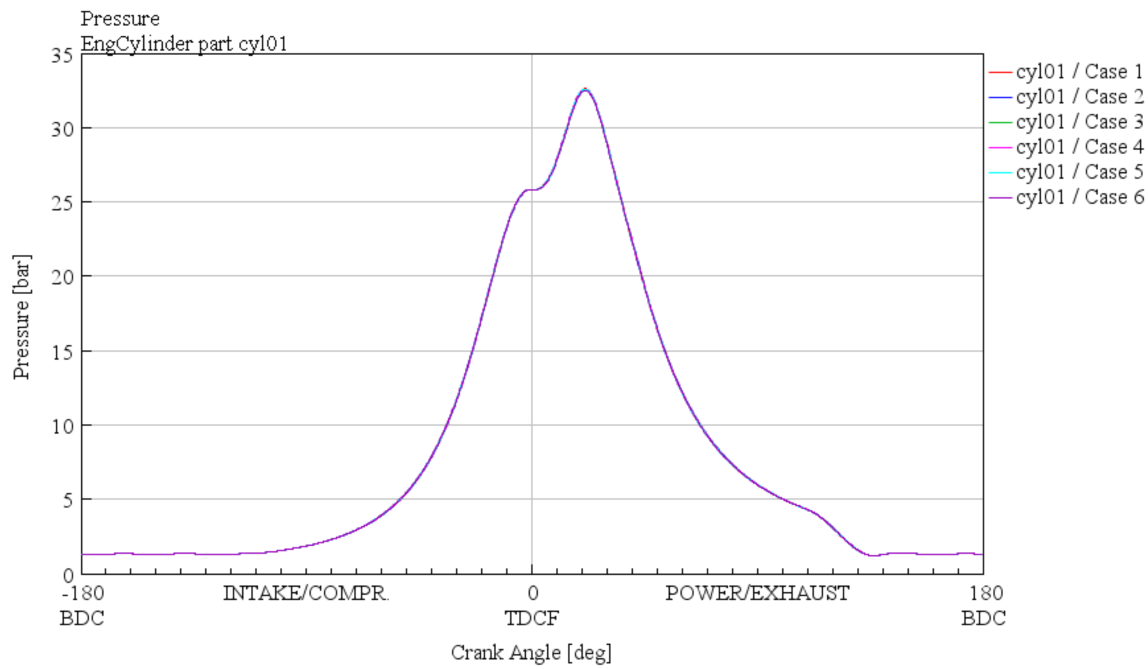

Figure 3-7: Pressure curve results for 1% compressor vent gases

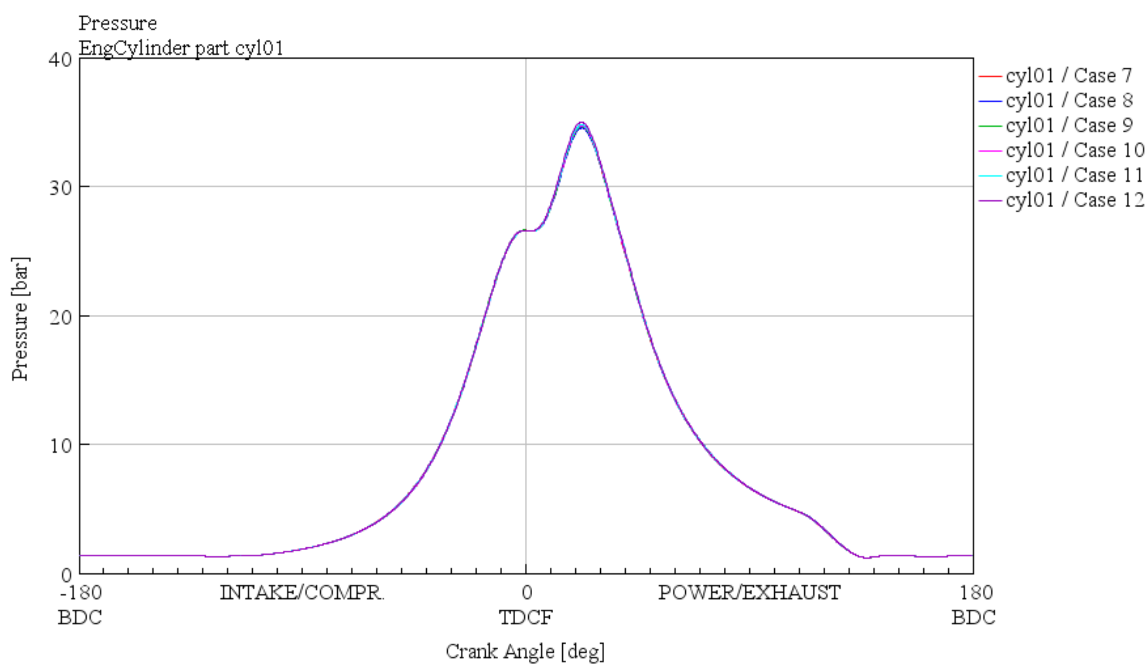

Figure 3-8: Pressure curve results for 3% compressor vent gases

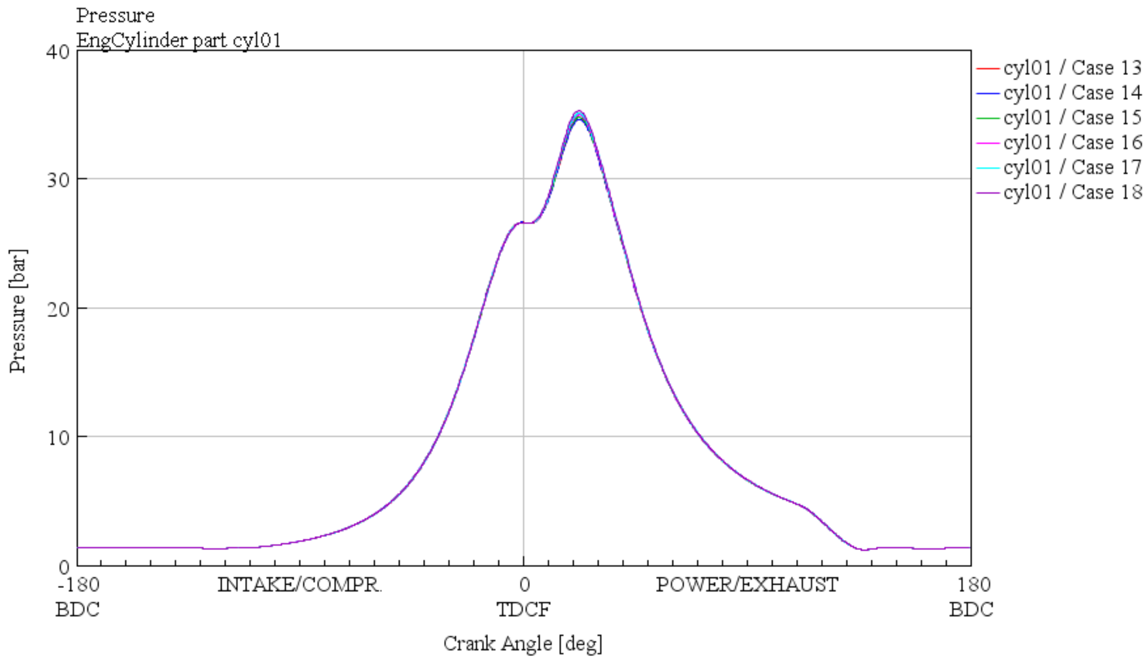

Figure 3-9: Pressure curve results for 5% compressor vent gases

The pressure results for all the test space conditions show that the addition of rebreathed gases, no matter the source, will have little to no effect on the engine operation and performance. It should be noted that a pre-ignition model was not used in the simulation as there was not access to a robust model, bust based on the results of the study, no increase in the probability of pre-ignition or auto-ignition was observed.

The above pressure traces are for when the source of rebreathed gases is restricted to one source. Realistically, these gases will be fumigated into the engine intake from both the engine crankcase and compressor vent simultaneously. Figure 3-10 below shows the pressure trace for 1%, 3%, and 5% rebreathed gas addition for compressor vent gases and engine crankcase gases with 3000 ppm methane. The pressure traces behave the same as the pressure curves when the source of gases is from one source. It can be said then that the results from the single rebreathed gas source cases will be representative of the simulation results for both sources fumigated into the engine air intake.

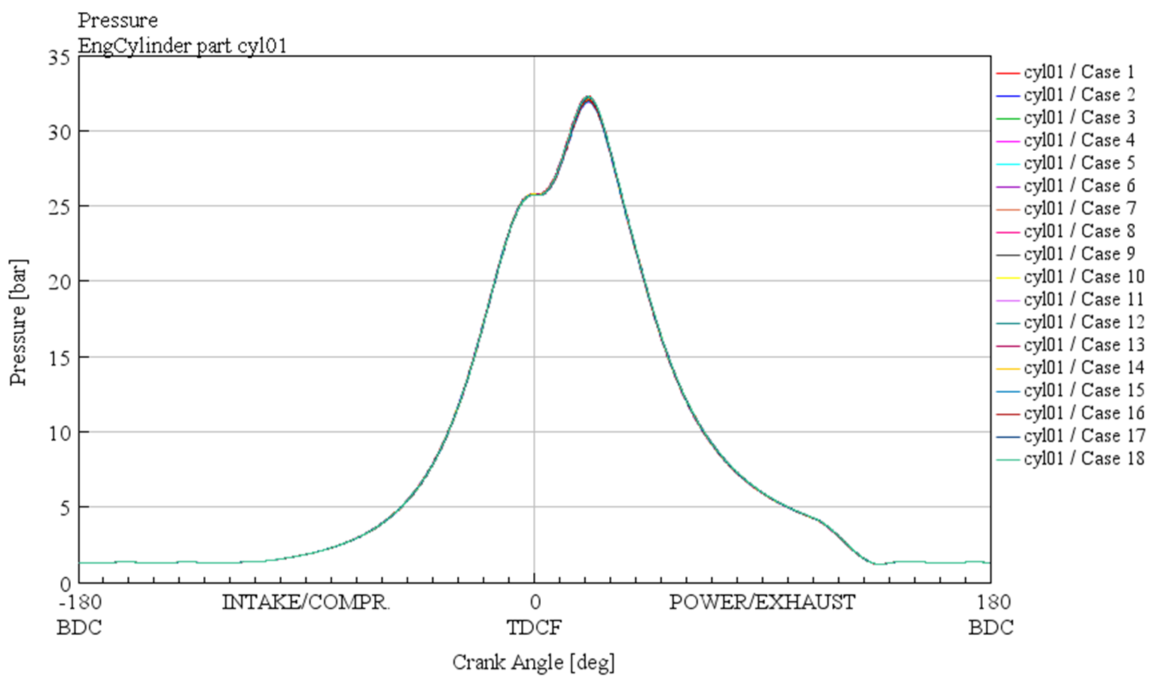

Figure 3-10: Pressure curve with rebreathed gases from engine and compressor simultaneously

These results cannot be proven accurate and reliable without verification data from the real engine. Figure 3-11 below overlays the experimental verification pressure curve for cylinder one with a randomly selected set of curves presented earlier. Overall, the verification curve is relatively close to the simulation data, and the simulation results are proven accurate and reliable. However, there are differences between the simulation and experimental pressure curves. The curves are identical in the compression stroke up to a few degrees before TDC. From the breakoff point, the experimental data increases with a smaller slope until TDC where the slope increases until peak pressure. The peak pressure is comparable in value, but is approximately 7 degrees earlier. The exhaust curves are identical in behavior until 60 to 70 degrees after TDC, but the experimental data is still offset from the offset in peak pressure. Again, the exhaust curve starts to decrease with a smaller rate of change until the two curves have the same value, and ultimately remain at similar values until compression starts again. These differences can most likely be attributed to the differences in geometry, and valve timings between the simulation model and the real engine. The real engine contains a pre-combustion chamber which the model used for this study does not. CSU is currently performing more detailed analyses of the engine to better define the GMV-4 geometry in their lab. The inlet and exhaust port timings were also not clearly known, and were estimated from drawings of the ports. The estimated area arrays for the ports can be found in Appendix A. These two engine parameters can have a significant difference between the simulation and experimental data, and once the geometry and port timings become better defined, the data for the simulation will become more accurate.

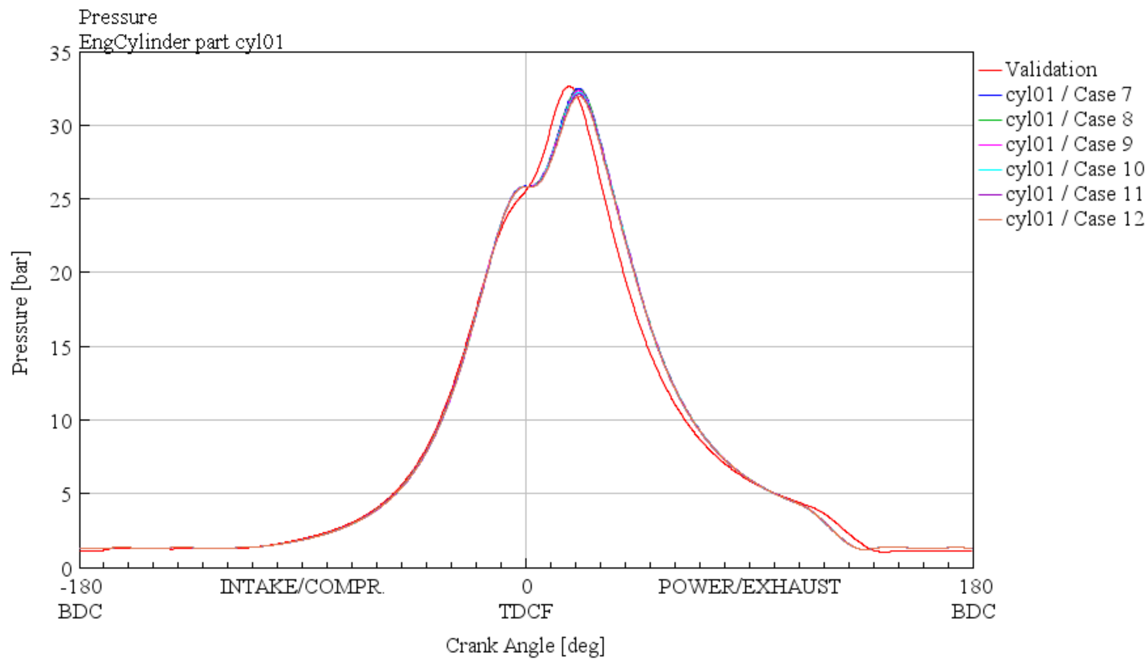

Figure 3-11: Validation curve overlaid on top of Figure 3-8

An uncertainty analysis was not performed, but an input parameter sensitivity analysis was conducted to see how the IMEP of the engine is affected by the varying parameter values. The three parameters in focus are the air intake, boost pressure, and fuel injection rate. The analysis increased and decreased the parameters by 10%, and the resulting IMEPs are compared to a baseline case with normal conditions. It is noted that there is zero rebreathed gas fumigation in all of the cases. Table 3-1 below summarizes the IMEP of the different cases run. The “Normal Conditions” case is the same as case 1 for the compressor vent rebreathed gas test space. The variation in the air intake mass flow rate has a negligible effect on the IMEP of the engine. The boost pressure, or AMP, has the largest effect on the IMEP, spanning from -0.74 to 7.26 bar. The increase in the fuel flow did not have an effect on the IMEP, but the decrease in the flow dropped the

IMEP by 0.08 bar. The engine performance is more sensitive to a change in the boost pressure compared to the other parameters varied.

Table 3-1: IMEP results of input parameter variation

Case	Normal Conditions	+ 10% Air Intake	- 10% Air Intake	+ 10% Boost Pressure	- 10% Boost Pressure	+ 10% Fuel Injection	- 10% Fuel Injection
IMEP (bar)	5.70	5.70	5.70	7.26	-0.74	5.70	5.62

3.2 Methane Reduction

Now that the engine performance is known to be safe and reliable with the addition of rebreathed gases from the engine crankcase and compressor vents, the next step is to determine how effective is the proposed solution. This was done following the methodology in the simulation set up section of the report. To determine the reduction value, the portions of the fugitive methane that escaped through short circuit or the exhaust were removed from the original value. Tables 3-2, 3-3, and 3-4 below show the total methane reduction as a flow rate, and as a percentage for the rebreathed engine crankcase gases. As expected, when the flow rate from the engine crankcase is increased, the total methane reduction is also increased. This behavior is seen for all cases. However, the percentage reduction remains almost constant at a value of 52%. Although this value is better considering the methane is vented directly into the atmosphere currently, this is a small reduction percentage. Ideally, the methane reduction rate should be close to the reduction rate of a flare (86.9%) or other highly efficient energy wasting device [12].

Table 3-2: Methane reduction for 1% engine crankcase gases with 1500 ppm methane

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Net Reduction (lb/hr)	0	0.005	0.011	0.016	0.021	0.027
Net Reduction (%)	0	52.36	52.34	52.32	52.30	52.28

Table 3-3: Methane reduction for 3% engine crankcase gases with 1500 ppm methane

	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
Net Reduction (lb/hr)	0.000	0.016	0.032	0.048	0.064	0.080
Net Reduction (%)	0.00	52.32	52.27	52.22	52.19	52.16

Table 3-4: Methane reduction for 5% engine crankcase gases with 1500 ppm methane

	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
Net Reduction (lb/hr)	0.000	0.027	0.053	0.080	0.106	0.133
Net Reduction (%)	0.00	52.28	52.21	52.16	52.16	52.16

The engine crankcase gases with 3000 ppm methane in the exhaust followed the same trends. Tables 3-5, 3-6, and 3-7 show the methane reduction for the engine gases with 3000 ppm. The total net reduction is almost exactly doubled the values for the engine crankcase gases with 1500 ppm methane which is what was expected. If the concentration is doubled, the net reduction should also roughly be doubled. However, the net methane reduction percentage remained fairly constant at the same value of 52%. For the engine crankcase, it does not matter what the methane concentration in the blow-by gases is, the net methane reduction rate will be around 52%.

Table 3-5: Methane reduction for 1% engine crankcase gases with 3000 ppm methane

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Net Reduction (lb/hr)	0.000	0.011	0.021	0.032	0.043	0.053
Net Reduction (%)	0.00	52.19	52.18	52.15	52.13	52.12

Table 3-6: Methane reduction for 3% engine crankcase gases with 3000 ppm methane

	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
Net Reduction (lb/hr)	0.000	0.032	0.064	0.096	0.127	0.159
Net Reduction (%)	0.00	52.15	52.10	52.06	52.02	52.00

Table 3-7: Methane reduction for 5% engine crankcase gases with 3000 ppm methane

	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
Net Reduction (lb/hr)	0.000	0.053	0.106	0.159	0.212	0.265
Net Reduction (%)	0.00	52.12	52.04	52.00	51.99	52.00

The total net methane reduction and percent reduction for rebreathed compressor vent gases are shown in Tables 3-8, 3-9, and 3-10 below. As expected, with an increase in compressor vent flow rates, which is pure methane for this study, the total net reduction was increased. Just as all the other simulations, the net methane reduction percentage was 52%.

Table 3-8: Methane reduction for 1% compressor vent gases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Net Reduction (lb/hr)	0.000	0.047	0.099	0.151	0.198	0.250
Net Reduction (%)	0.00	52.06	52.06	52.06	52.05	52.06

Table 3-9: Methane reduction for 3% compressor vent gases

	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12
Net Reduction (lb/hr)	0.000	0.150	0.300	0.448	0.599	0.750
Net Reduction (%)	0.00	52.06	52.06	52.06	52.06	52.05

Table 3-10: Methane reduction for 5% compressor vent gases

	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
Net Reduction (lb/hr)	0.000	0.250	0.500	0.750	0.999	1.249
Net Reduction (%)	0.00	52.06	52.06	52.06	52.05	52.05

No matter where the source of the rebreathed gases came from, nor the flow rate into the air intake, the methane reduction rate was always 52%. When looking into other parameters to see what could be influencing the net methane reduction, it was found that the trapping ratio controls the net methane reduction on a percentage basis. Table 3-11 below shows the trapping ratio for the 5% addition cases for the engine crankcase and compressor vent gases. The trapping ratio for all cases are around 0.525, or 52.5%. This trend is the same for the other cases not shown in this table. It appears that the methane reduction percentage depends solely on the trapping ratio.

Table 3-11: Trapping Ratio for 5% methane addition cases

		Case 13	Case 14	Case 15	Case 16	Case 17	Case 18
Trapping Ratio	Engine Crankcase (1500 ppm) Gases	0.525	0.525	0.524	0.523	0.523	0.523
	Engine Crankcase (3000 ppm) gases	0.525	0.525	0.524	0.523	0.523	0.523
	Compressor Vent Gases	0.525	0.525	0.525	0.525	0.525	0.525

As previously stated, 52% methane reduction is better than emitting the methane into the atmosphere, but these results are not significant enough to justify spending more time and resources into developing the proposed rebreathing system.

4 CONCLUSIONS

With increasing interest and efforts to reduce the human impact on the environment, the focus of this study was to determine the effectiveness of rebreathing fugitive methane emissions from engine crankcases and compressor vents into the air intake of a large bore, natural gas, 2-stroke engine. It was also desired to find the composition of the rebreathed gases from the engine crankcase and how the rebreathed gases will affect the combustion process and engine performance.

From the literature review, it is near impossible to estimate the composition of the engine crankcase gases without physically pulling of the gases and measuring it. The composition of the gases depends on a large variety of parameters that easily differ between each engine, and can change for the same engine over time. It is expected though, that the composition of the engine blow-by gases will consist of products of combustion, fuel, air, oil, and particulates. The composition of the compressor vent gases will be the same as the fluid (natural gas in this case) being compressed.

The effect of the rebreathed gases on the engine performance can be considered negligible. The pressure curves for the rebreathed engine crankcase gases remained unchanged through all 36 cases. The pressure curves for the compressor vent were also the same except for a small difference in the peak pressure. Fine tuning the controls to maintain the same AFR will solve this problem. The temperature curves will follow a similar behavior to the pressure curve. A pre-ignition model was not used in this study, but based on the results from the simulations, there was no observed increase in the probability of pre-ignition or auto-ignition.

The percentage methane reduction for every test case was around 52%. This reduction percentage is not desirable; the goal is to increase the reduction rate to 100%. The increased fumigated mass flow rates, and methane concentration had very insignificant effect on the methane reduction percentage. A 52% reduction rate would leave 48% of the methane emitting to the atmosphere through the engine exhaust, which is also unacceptable. A method to help further reduce the methane emissions is to run the engine exhaust through a waste heat recovery device that would need to be installed on site. The problem however, does not lie with the concept of rebreathing itself, but with the engine used to combust the fugitive methane emissions. It was discovered that the methane reduction greatly, if not entirely, depends on the trapping ratio of the engine; the trapping ratio of the GMV-4 is around 50% so the actual methane reduction will be small. At present, rebreathing the fugitive methane emissions into a 2-stroke engine is not feasible, it would be better to investigate other sources of combustion at a compressor station to route the rebreathed gases.

5 FUTURE WORK

The elimination of the proposed fugitive methane emissions rebreathing system coupled with a large bore, natural gas, 2-stroke engine, only means that more possibilities need to be explored to find the best method to effectively reduce fugitive methane emissions. There are two proposed solutions from conclusions of this study: rebreathe the fugitive emissions into a different device with a higher trapping ratio, and consider methods to reduce emissions from the compressor.

The problem with the 2-stroke engine is the low trapping ratio. Almost half of the air intake, which includes the fumigated emissions, short circuits the engine and escapes through the exhaust. Using the 2-stroke engine would have been convenient since it is already on site driving the compressors, but routing the emissions to a device with a higher trapping ratio should produce better results. Some compressor stations have a 4-stroke engine that runs a generator, or a boiler on site, and it would be worth the time and effort to perform this same study on the new equipment. If the results are desirable, the next test would be to run experiments on the 4-stroke engine and see if the experimental results match up with the simulations.

It was observed during the study that the majority of the fugitive methane emissions come from the compressor, which makes sense. The engine combustion starts with mainly air and a small amount of methane, so the blow-by gases will also consist of an extremely small concentration of methane. In the compressor however, methane is the only gas being compressed, so the fugitive missions will be entirely methane. Shifting the focus to piston rod packing designs can reduce the fugitive methane emissions at the source instead of looking at post emission reduction.

REFERENCES

1 Lashof, Daniel A., and Ahuja, Dilip R. “Relative contributions of greenhouse gas emissions to global warming”. *Nature*, vol. 344, no. 6266, 1990, pp. 529-531.

2 Raphael, John. "ALERT: Global Methane Emissions 110 Percent Higher Than Previously Thought." Nature World News. Nature, 06 Oct. 2016. Web. 21 Dec. 2016.

3 Hannon, Dan. “Compressor Cylinder Emissions Reduction Technology.” GMRC (2015): n. pag. Web. 5 Oct. 2016.

4 Johnson, Derek, and April Covington. "Potential Reduction of Fugitive Methane Emissions at Compressor Stations and Storage Facilities Powered by Natural Gas Engines." *Volume 6A: Energy* (2014): n. pag. Web. 5 Oct. 2016.

5 Candy, J., and L. Guerbe. "Blow-by Gases Coalescing Separation. Performances on Passenger Car Engines." SAE Technical Paper Series (2009): n. pag. SAE. Web.

6 Johnson, Derek, and April Covington. "Methane Leak and Loss Audits of Natural Gas Fueled Compressor." *Volume 1: Large Bore Engines; Fuels; Advanced Combustion; Emissions Control Systems* (2014): n. pag. Web. 5 Oct. 2016.

7 Páv, Karel. "Blowby Gas Composition in Si Engines." *Journal of Middle European Construction and Design of Cars* 13.3 (2015): 1-7. Web. 5 Oct. 2016.

8 Caterpillar. “Crankcase Ventilation.” Application and Installation Guide. n. pag. Web. 1 Aug. 2016

9 VanDerveer, R. and Hittler, D., "The American Motors Crankcase-to-Air-Cleaner Engine Ventilation System," SAE Technical Paper 630087, 1963, doi:10.4271/630087.

10 Krause, Wolfgang, Karl H. Spies, Laurence E. Bell, and Fritz Ebert. "Oil Separation in Crankcase Ventilation - New Concepts Through System Analysis and Measurements." SAE Technical Paper Series (1995): n. pag. SAE. Web. 21 Dec. 2016.

11 Koch, F., Haubner, F., and Orlowsky, K., "Lubrication and Ventilation System of Modern Engines - Measurements, Calculations and Analysis," SAE Technical Paper 2002-01-1315, 2002, doi:10.4271/2002-01-1315.

12 "Engine Oil Consumption." Engine Oil Consumption. AA1Car, n.d. Web. 21 Dec. 2016.

13 Xiao, F., A. Sohrabi, and G.a. Karim. "Reducing the Environmental Impact of Fugitive Gas Emissions through Combustion in Diesel Engines." SAE Technical Paper Series (2007): n. pag. SAE. Web. 5 Oct. 2016.

14 Parker, Todd. "Fugitive Methane Emission Reduction Using Gas Turbines." *Volume 3: Coal, Biomass and Alternative Fuels; Combustion and Fuels; Oil and Gas Applications; Cycle Innovations* (1998): n. pag. *ASME*. Web. 5 Oct. 2016.

15 Lentz, E., "Crankcase Ventilators which Eliminate Atmospheric Escape of Blowby," SAE Technical Paper 630086, 1963, doi:10.4271/630086.

16 Ephraim, M. and Koci, L. US Patent 3,754,538. August 28, 1973.

17 Tice, John K. Patent WO2011037755 A3. August 25, 2011.

18 Tracy, C. and Frank, W., "Fuels, Lubricants, and Positive Crankcase Ventilation Systems," SAE Technical Paper 640261, 1964, doi:10.4271/640261.

19 "SlipStream®." Spartan Controls. N.p., n.d. Web. 04 May 2017. <http://www.spartancontrols.com/rem-technology/rem-technology-products/slipstream/>.

20 Olsen, Daniel B., *ARTVOL2BCYL1&3* (1998) [Microsoft Excel spreadsheet]. Fort Collins, Colorado: CSU. [accessed 15 February 2017].

21 Olsen, Daniel B., Neuner, Benjamin and Ladd, John. “Variable Fuel Composition Air Fuel Ratio Control of Lean Burn Engines,” PRCI, 2016.

22 Olsen, Daniel B., *GMV Cyl Geometry* (2015) [Microsoft Powerpoint]. Fort Collins, Colorado: CSU. [accessed 15 February 2017].

23 Olsen, Daniel B., *Pressure Traces – 11inHg* (2000) [Microsoft Excel spreadsheet]. Fort Collins, Colorado: CSU. [accessed 15 February 2017].

APPENDIX

A – CSU Data

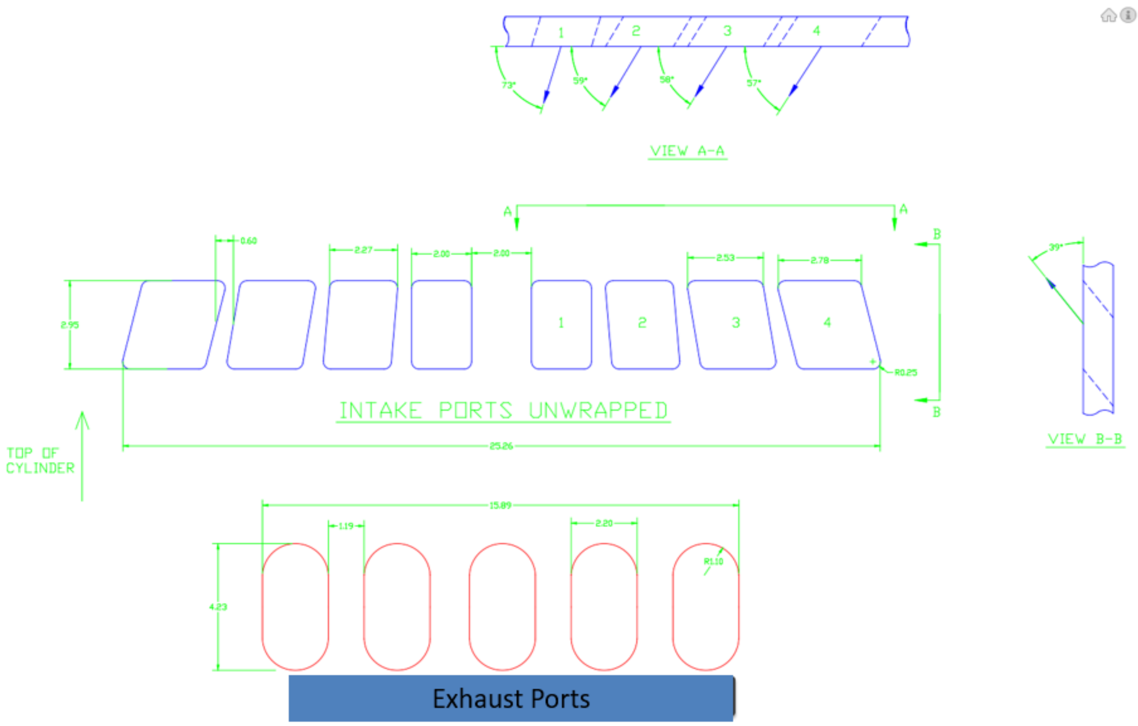

Figure A-1: Inlet and exhaust port geometry. Reprinted from [21]

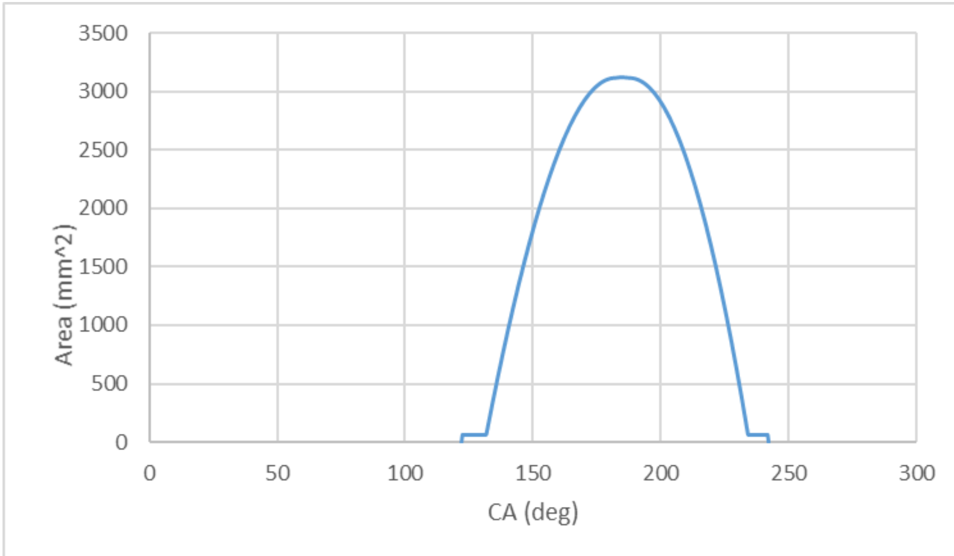

Figure A-2: Inlet port area array

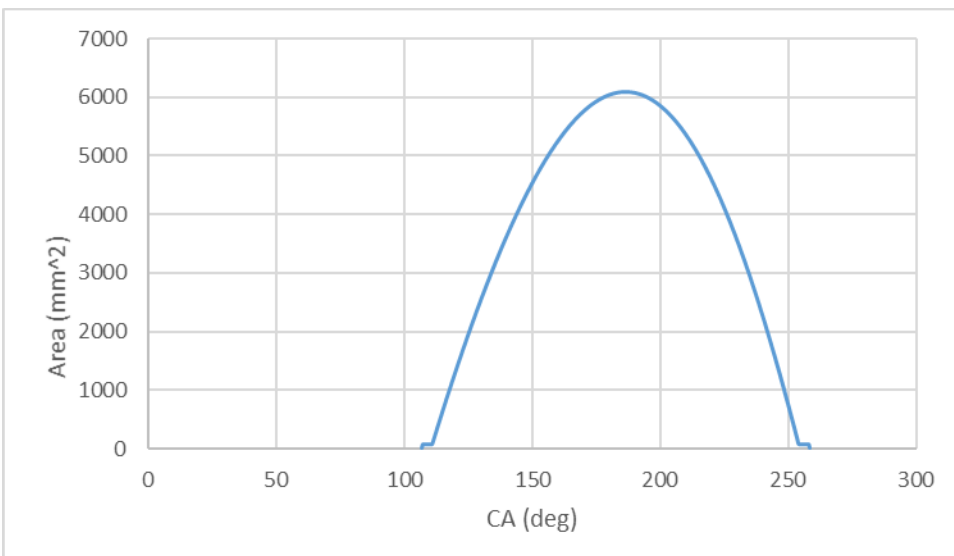

Figure A-3: Exhaust port area array

Table A-1: CSU Cooper-Bessemer GMV-4 Engine Parameters. Reprinted from [19]

AMP		22.5		ABS AMP		25.78585
EMP		18.75		ABS EMP		23.94401
BP		30		BP		14.73477
AMT		110		ABS AMT		569.67
Inlet Air Flow lb/hr		10846		Density Charge		0.113443
Fuel flow lb/hr		192		Ratio Scav.		1.312828
Exhaust flow lb/hr		11037		Eff. Scav.		0.730942
PCC Fuel Flow lb/hr		1.455		Theta		0.556769
Total Air/Fuel Ratio		56.7		A/F Trapped		31.5688
PCC Volume (cc)		48				
Cylinder Bore		14		Mass Trapped		0.08685
Cylinder Stroke		14		Mean Piston Speed		8400
PCC Throat Area		0.11045				

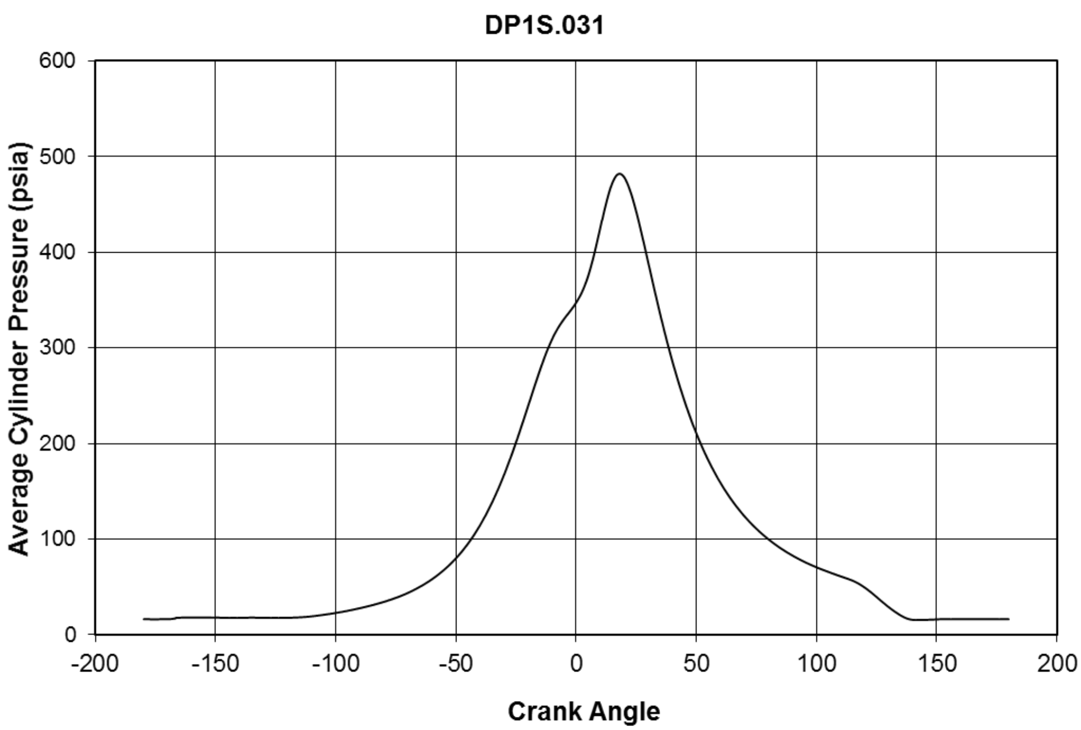

Figure A-4: Validation data. Reprinted from [22]

B – Simulation Models

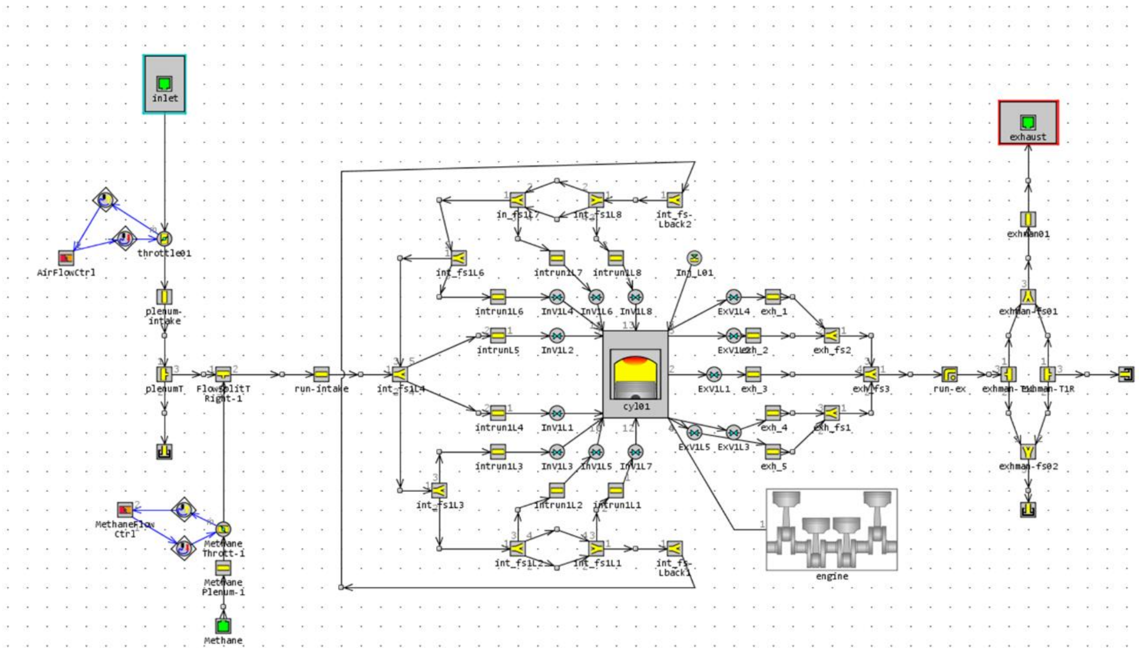

Figure B-1: Engine simulation model with compressor vent rebreathed gases

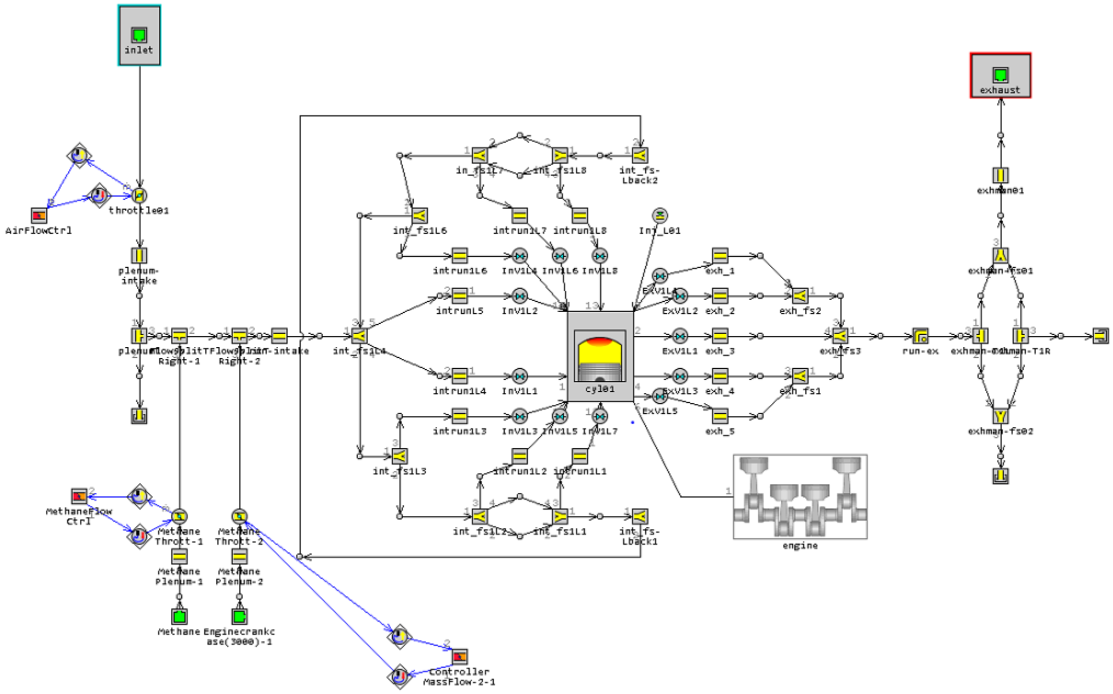

Figure B-2: Engine simulation model with both sources of rebreathed gases

C – Simulation Results

Table C-1: IMEP values for engine rebreathed gases with 1500 ppm methane

Case	1	2	3	4	5	6
IMEP (bar)	5.69	5.68	5.68	5.68	5.67	5.67
Case	7	8	9	10	11	12
IMEP (bar)	5.69	5.68	5.67	5.65	5.64	5.63
Case	13	14	15	16	17	18
IMEP (bar)	5.69	5.67	5.65	5.63	5.63	5.63

Table C-2: IMEP values for engine rebreathed gases with 3000 ppm methane

Case	1	2	3	4	5	6
IMEP (bar)	5.69	5.69	5.68	5.68	5.68	5.67
Case	7	8	9	10	11	12
IMEP (bar)	5.69	5.68	5.67	5.66	5.65	5.64
Case	13	14	15	16	17	18
IMEP (bar)	5.69	5.67	5.66	5.64	5.63	5.63

Table C-3: IMEP values for compressor rebreathed gases

Case	1	2	3	4	5	6
IMEP (bar)	5.70	5.70	5.70	5.70	5.70	5.69
Case	7	8	9	10	11	12
IMEP (bar)	5.70	5.70	5.69	5.69	5.71	5.72
Case	13	14	15	16	17	18
IMEP (bar)	5.70	5.69	5.71	5.73	5.77	5.79