A Methodology for Turbocharging Single Cylinder Four Stroke Internal Combustion Engines

by

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Abstract

This thesis presents a method for turbocharging single cylinder four stroke internal combustion engines, a model used to evaluate it, an experimental setup used to test it, and the findings of this experiment. A turbocharged engine has better fuel economy, cost efficiency, and power density than an equivalently sized, naturally aspirated engine. Most multi-cylinder diesel engines are turbocharged for this reason. However, due to the timing mismatch between the exhaust stroke, when the turbocharger is powered, and the intake stroke, when the engine intakes air, turbocharging is not used in commercial single cylinder engines. Single cylinder engines are ubiquitous in developing world off grid power applications such as tractors, generators, and water pumps due to their low cost. Turbocharging these engines could give users a lower cost and more fuel efficient engine. The proposed solution is to add an air capacitor, in the form of a large volume intake manifold, in between the turbocharger compressor and the engine intake to smooth out the flow.

Thesis Supervisor: Amos G. Winter, V.
Title: Assistant Professor

Acknowledgments

I would like to thank Professor Amos Winter V. for coming up with the method for turbocharging single cylinder engines discussed in this thesis, hiring me, funding me, and advising me. I also used his 2005 master's thesis as a guide for writing this thesis.

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Nomenclature

A = Cross sectional area of connecting tube

A_F = Cross sectional area of fin

C_p = specific heat of air

D = Diameter of connecting tube

D_C = Diameter of capacitor

F = Friction factor

h_c = Convective heat transfer coefficient for air inside capacitor

h_O = Convective heat transfer coefficient for air outside capacitor

K = Minor losses

k_a = Conductivity of air

k_{al} = Conductivity of aluminum

L = Length of connecting tube

L_C = Length of capacitor

m_c = Mass of gas inside the capacitor

$\dot{m}_c$ = Mass flow rate of gas into the capacitor

$\dot{m}_t$ = Mass flow rate of gas at turbocharger

m_f= fin constant 1

M_f= fin constant 2

n = number of fins

Nu = Nusselt number

P_C = Pressure inside the capacitor

P_e = Pressure in the engine at the end of the intake stroke

P_t = Pressure at the turbocharger

P_0 = Initial pressure inside the capacitor, same as ambient pressure

Pr = Prandtl Number

$\dot{P_C}$ = Rate of pressure change inside the capacitor

$\dot{Q_C}$ = Rate of heat transfer out of the capacitor

$\dot{Q_f}$ = Rate of heat transfer out of the fin

$\dot{Q_{tot}}$ = Total heat transfer out of the capacitor and fin

R = Specific gas constant

r = Capacitor pressure/turbo pressure

R_{in} = Thermal resistance inside the capacitor

R_{wall} = Thermal resistance of the wall of the capacitor

R_C = Thermal resistance of the wall of the capacitor minus fins

Re = Reynolds number

T_b = Fin base temperature

T_C = Temperature inside the capacitor

T_{drop} = Temperature drop due to heat exchange

T_t = Temperature at the turbocharger

T_0 = Ambient temperature

V_C = Volume of the capacitor

V_e = Volume of the engine

v_s = Velocity of air in connecting tube

v_c = Velocity of air in capacitor

μ = Viscosity of air

γ = Heat capacity ratio (1.4 for air)

ρ_C = Density of air inside the capacitor

ρ_t = Density of air at the turbocharger

ρ_0 = Atmospheric density of air

Chapter 1

Introduction

This thesis presents a method of turbocharging single cylinder four stroke engines, originally proposed by Professor Amos G. Winter, V. The proposed method is to add a buffer in the form of a large volume intake manifold, which we call an air capacitor, in order to store compressed air between the intake strokes (Fig. 1-1) [30]. This method for turbocharging single cylinder engines was first presented at the 2014 IDETC conference [2]. The goal of this research was to validate our methodology for turbocharging single cylinder, four stroke, internal combustion engines. This paper will review the theory, explain the series of computational models used, describe an experiment designed to validate the method, and the initial experimental results.

A turbocharger is a device that is fitted to an engine to improve performance. It pressurizes the intake stream of an engine causing it to combust more fuel and produce more power than an equivalently sized naturally aspirated engine. It consists of a turbine and a compressor connected by a shaft. The turbine is powered by the high temperature and pressure exhaust gas leaving the engine during the exhaust stroke [29, 14]. The compressor, which is powered by the turbine, compresses the air going into the intake of the engine.

There are three main advantages of a turbocharged engine over a naturally aspirated engine.

1. A turbocharged engine costs less than a naturally aspirated engine with the

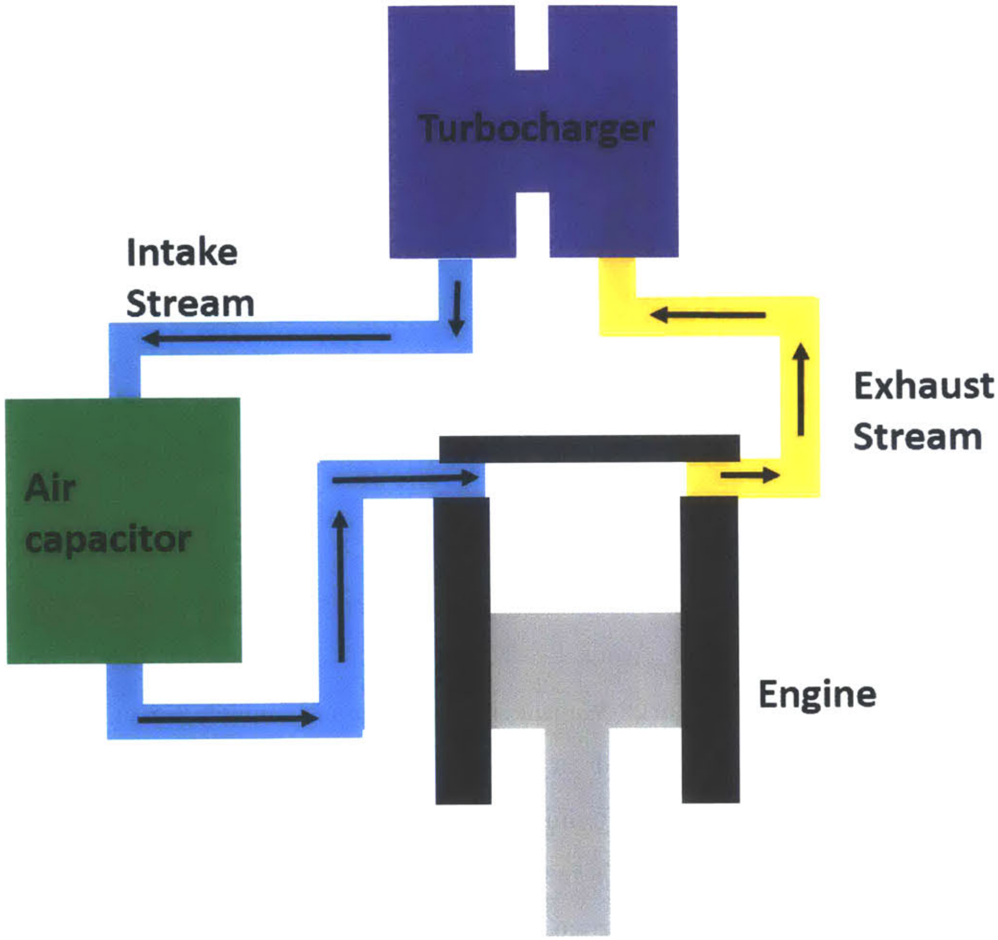

Figure 1-1: Block diagram of engine flow with air capacitor system

same power rating. According to our partner companies, adding a turbocharger is twenty percent of the cost of adding another cylinder. [1].

2. A turbocharged engine is more fuel efficient than a naturally aspirated engine with the same power rating. This is because turbocharged engines are smaller and, as a result, have fewer frictional losses[17]

3. A turbocharged engine has a higher power density compared to a naturally aspirated engine [3].

Most modern multi-cylinder diesel engines are turbocharged due to the multiple benefits of turbocharging listed above. A multi-cylinder engine can be designed in such a way that when one cylinder is going through the exhaust stroke, powering the turbocharger, another cylinder is intaking [29, 14]. In a single cylinder engine there is a phase difference between the intake and exhaust stroke. This means when the engine is exhausting, which is when the turbocharger is powered, the engine is not intaking and the air has nowhere to go. Due to the varying nature of single cylinder exhaust pulses and the phase mismatch commercial single cylinder engines are not currently turbocharged despite the numerous advantages of turbocharging.

1.1 Motivation

The main application being targeted for our technology is low-cost agricultural equipment. Turbocharged single cylinder engines could replace naturally aspirated engines in water pumps, generators, and tractors, making more power available for farmers at lower costs. This technology could open up a new market of small scale farmers who, in the past, could not afford mechanization. Research in the Indian market has shown a direct correlation between food yield and a number of factors including rain fall, fertilizer available, available farm power, and median income [22]. This study showed that farm yield is most closely correlated to available farm power (Fig. 1-2). In addition to low cost agriculture, this technology could be used in single cylinder applications in wealthy countries, such as small UAVs, motorcycles, and household generators.

Farming in India is extremely inefficient [22]. Fifty-five percent of the population in India is engaged in agriculture [22]. The percent of labor involved in agriculture is made more alarming when one considers that agriculture accounts for only fifteen percent of India's GDP [11]. The gap between work force and farm GDP is resulting in farm workers migrating to urban centers for higher paying jobs [11], leading to an increase in farm labor cost. The goal of my project is to spur farmers to adopt mechanization by making it cheaper resulting in, improved economic efficiency of small scale farmers.

India is lagging behind the developed world in farm mechanization [6, 11] which

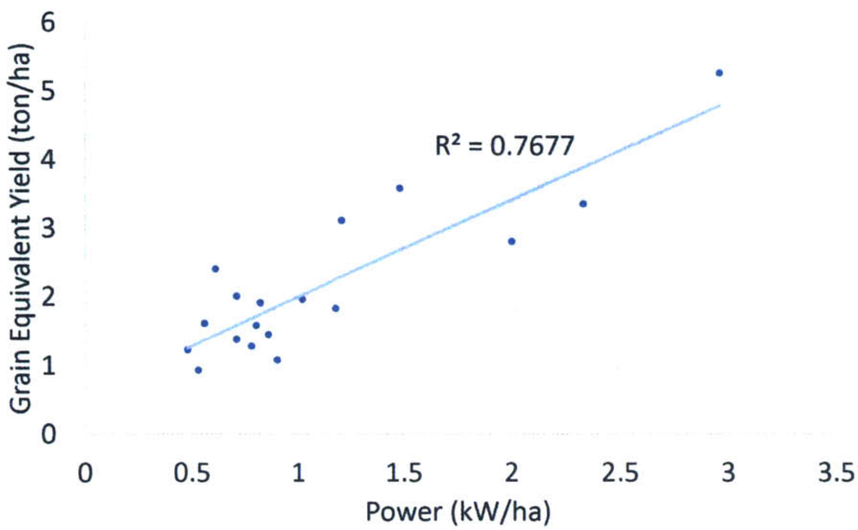

Figure 1-2: Correlation between farm power and yield

results in a significant percent of the population engaged in agriculture. Figure 1-3 shows the percent of the population engaged in agriculture and the level of farm mechanization in different regions of the globe, demonstrating obvious correlation between the two [6].

In addition to being correlated to farm yield by province (Fig. 1-2) [22] mechanization is also correlated to farm yield over time [19]. Figure 1-4 shows how farm mechanization, crop yield, and population change over time [19], showing the relationship between crop yield increases and mechanization. Mechanization is made more urgent since crop yield has to increase by a third, over the next couple of decades, in order to feed the growing population of India [11].

Figure 1-5 shows the evolution of farm power from different sources over the last forty years [19]. In the last seven years tractors and diesel engines went from approximately twenty five percent of available farm power to forty two percent [19]. While diesel engines have penetrated the low end market at approximately 1.2 million units a year [6], sub 15hp tractors have failed to gain traction accounting for less than one percent of tractor sales in India [19]. This means there is a large untapped market for small tractors and engines to power them.

Multiple studies have been commissioned by the Indian government to generate

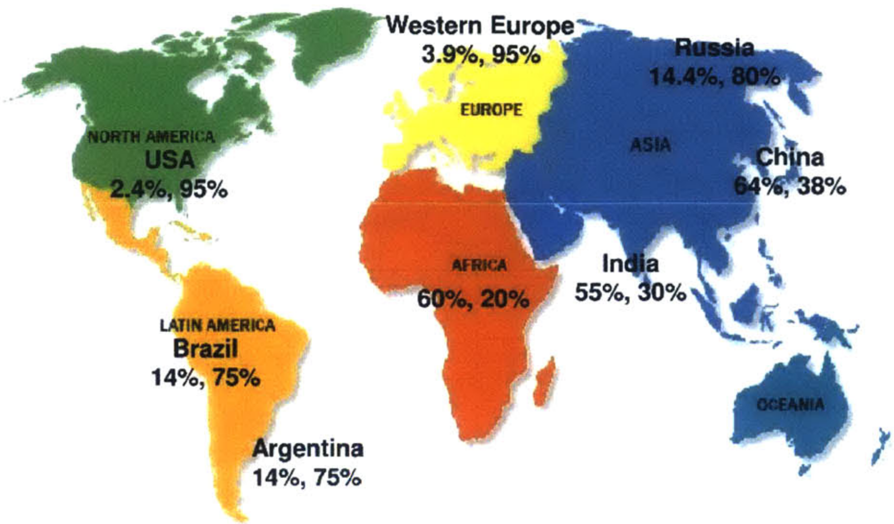

Figure 1-3: Map showing percent of population engaged in agriculture and percent of agriculture that is mechanized [6]

ideas on how to address agricultural issues in India. Two of the key studies were done by Bhopal Indian Counsel of Agricultural Research and by the Indian Department of Agriculture [22, 19]. It was found that while there was large growth in farm power in the 1980's and 1990's, over the last decade growth in available farm power has slowed to a little over two percent per year [22]. The other key finding was that farm mechanization could save seeds by fifteen to twenty percent, save fertilizer by fifteen to twenty percent, increase crop intensity by five to twenty percent, save time by twenty to thirty percent, reduce manual labor by twenty to thirty percent, and increase farm productivity by ten to fifteen percent [22]. These studies also showed that small farmers have difficulty affording tractors and getting access to lines of credit to buy farm equipment [22, 19]. This is key because while tractors have significantly higher capital cost over bullocks (40,000Rps for a pair of bullocks Vs. 250,000 Rps for a small tractor) [25] they have significantly lower running costs. It costs about 12,000 Rps a month to maintain a pair of bullocks while it only costs about 7,000 Rps a month to buy fuel for a tractor and maintain a tractor [25]. This means there is a large untapped market for small tractors and engines to power them.

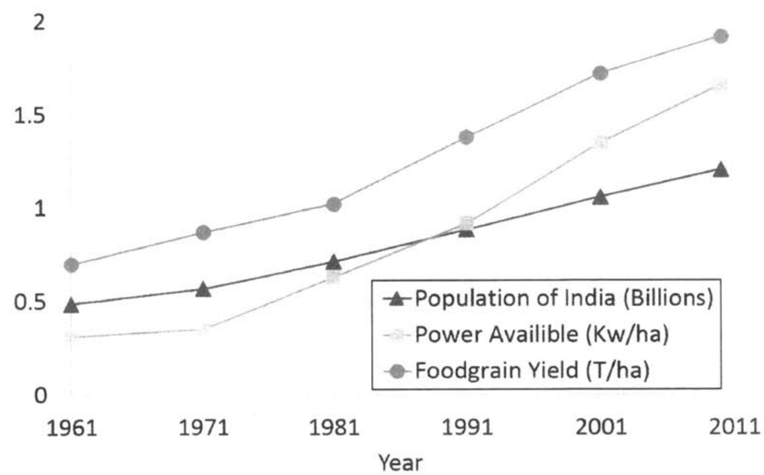

Figure 1-4: Plot showing population, farm power, and food yield over time

1.2 Research Purpose, Scope, and Summary

1.2.1 Research Questions

The key questions that this thesis addresses are:

1. Will adding a large volume intake manifold result in a working turbocharged single cylinder four stroke engine?

2. Can a simple model be created to approximate the power gain from turbocharging a single cylinder four stroke engine?

3. Can this model predict how air capacitor size affects performance?

4. How important is it to cool the intake air post turbocharging?

5. Is it possible to build a low cost dynamometer to test the feasibility of turbocharging a single cylinder four stroke engine using an air capacitor?

6. Can this dynamometer be used to test how capacitor size affects peak power, intake air density, and manifold pressures?

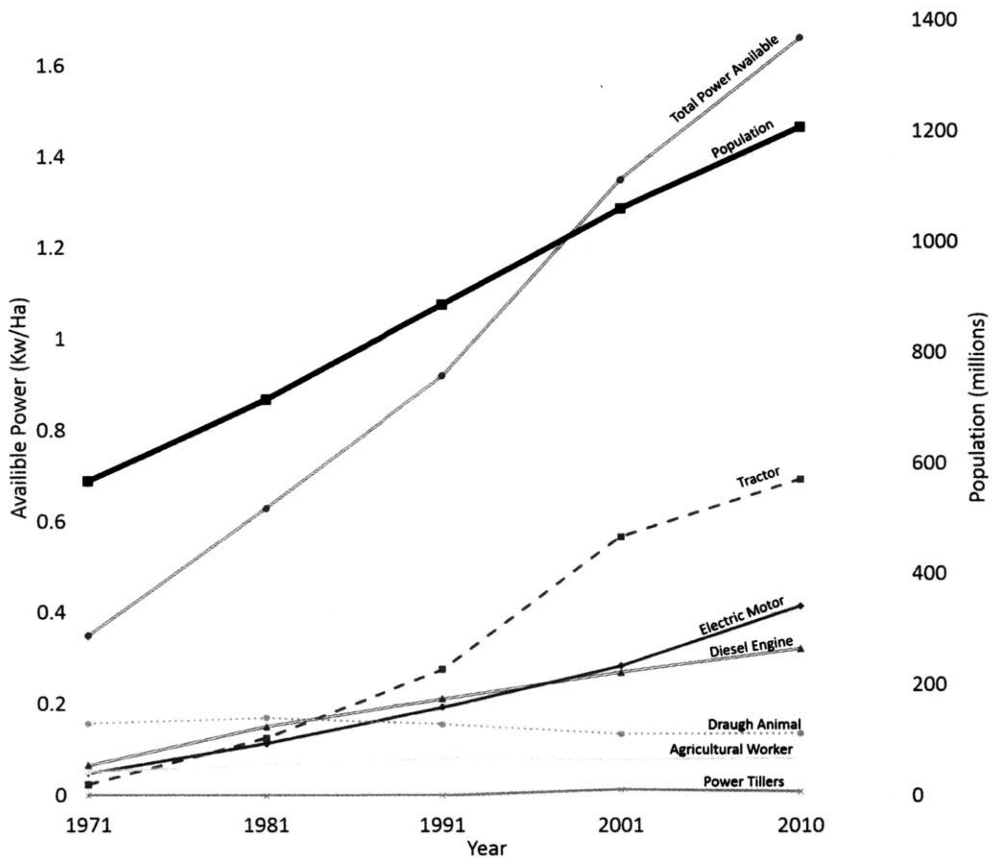

Figure 1-5: Plot of farm power from different sources in India over time compared to population [19]

1.2.2 Research Tasks

In order to answer the above questions the following research tasks were performed:

1. A series of models of increasing complexity were created to model the flow of air through the air capacitor in MATLAB.

2. A single cylinder four stroke engine that is typical of engines found in developing countries was obtained. This engine was already connected to a generator that was used for power dissipation.

3. The engine was modified with a custom designed exhaust manifold that accommodated a turbocharger and a custom designed intake manifold that allowed for different sized air capacitors to be easily swapped out.

4. The engine was fitted with sensors that measured the intake air pressure and temperature, pressure of air leaving the compressor, exhaust pressure, engine speed, engine power, and the fuel mass. The engine was also fitted with a power dissipater.

5. Constant speed engine tests were run. Data were collected for seven different capacitor sizes ranging from approximately two to seven times the engine volume. Naturally aspirated tests were also run to establish a baseline.

6. The results were then compiled and compared to the MATLAB models.

1.2.3 Scholarly Contribution of Research

The work presented in this thesis makes the following scholarly contributions:

1. A series of models describing flow through the intake manifold of a single cylinder turbocharged four stroke engine are presented. The assumptions made in this model are tested by comparing the model to data from an engine test bed.

2. A method for creating an accurate and low cost dynamometer to measure key engine running characteristics is presented.

3. Determine the viability of turbocharging a single cylinder four stroke engine using an air capacitor by modifying an existing engine.

4. The effect that manifold size has on the maximum power and manifold pressure of a single cylinder four stroke turbocharged engine is demonstrated through experimentation.

1.2.4 Summary of Results

In the first stage of this research, a theoretical model of how air flows through the intake manifold was developed. This model started with a simple constant pressure source fill model and was extended into a model that describes flow through the manifold. The model predicted that the turbo lag due to the capacitor is less than a second. Without heat exchange there is a density gain of approximately thirty seven percent, and with perfect heat exchange there is a density gain of approximately sixty percent. Note that density gain is proportional to power gain on the first order.

In the second stage of this research, an experimental set up was created around a single cylinder four stroke diesel engine. A typical developing world engine was chosen and was equipped with a turbocharger. A series of sensors were attached to the engine to measure pressure, temperature, and power output. The engine was run naturally aspirated and turbocharged with seven different sized intake manifolds. Using this experimental setup tests were conducted that showed that a turbocharger could be fitted to a single cylinder engine using an air capacitor to substantially increase intake air density and the peak power of the engine. It was found that peak power output increases with manifold size and that peak power increased by as much as twenty nine percent with the largest sized manifold (approximately seven times engine volume).

This is a significant increase in power and it proves that this method for turbocharging single cylinder engines works. However, it is less than the model predicted. This is probably due to back pressure effects that were not accounted for and the fact that the engine was not designed for positive pressure in its internal intake manifold which resulted in air leaking out. The other key takeaway from the experiment was that the temperature increase due to compression had a significant effect on performance, indicating that the system would benefit greatly from intercooling.

1.3 Thesis Organization

The second chapter goes over the chosen method [30] for turbocharging a single cylinder four stroke engine, turbocharger background, and theory significant to the project. The third chapter describes the models used to characterize the flow of air through the intake manifold of a turbocharged single cylinder four stroke engine. The fourth chapter goes over how a single cylinder four stroke generator was modified to be turbocharged with an air capacitor, and how a simple low cost dynamometer was built around this engine. The fifth chapter presents the results of experiments run on the dynamometer, discusses their meaning, and compares the results to the MATLAB models. The sixth and final chapter summarizes the work done in this thesis, explains its impact, and presents a plan for future work on turbocharging single cylinder four stroke engines. Additionally, there are two appendices; the first one contains the MATLAB code for the models and the second one contains the raw data from the experiment.

Chapter 2

Background

The goal of this chapter is to give the reader of this thesis an overview of the relevant technology and key prior art in order for the reader to have a better understanding of the proposed system. This chapter goes over how the proposed method works, how a diesel engine operates, how a turbocharger works, the physics behind the advantages of turbocharging, and the advantages of single cylinder engines over multi cylinder engines. Keep in mind that the history and development of internal combustion engines spans over a hundred and fifty years, so a significant amount of prior art exists. As a result, this chapter just touches on the relevant technology. For more information on internal combustion engines John Haywood's book Internal Combustion Engine Fundamentals is recommended [13].

2.1 Diesel cycle

The diesel cycle was first conceived by Rudolf Diesel in 1892 [13]. It consists of four strokes (Fig. 2-1) [15]. First, an intake stroke where fresh air is pulled into the engine. Then a compression stroke where the intake air is compressed to the point where it is hot enough to cause diesel fuel to undergo combustion. Then, the power stroke where the diesel fuel is injected and combustion occurs forcing the piston to expand. Finally, the exhaust stroke happens where the combusted air is expelled from the engine [13].

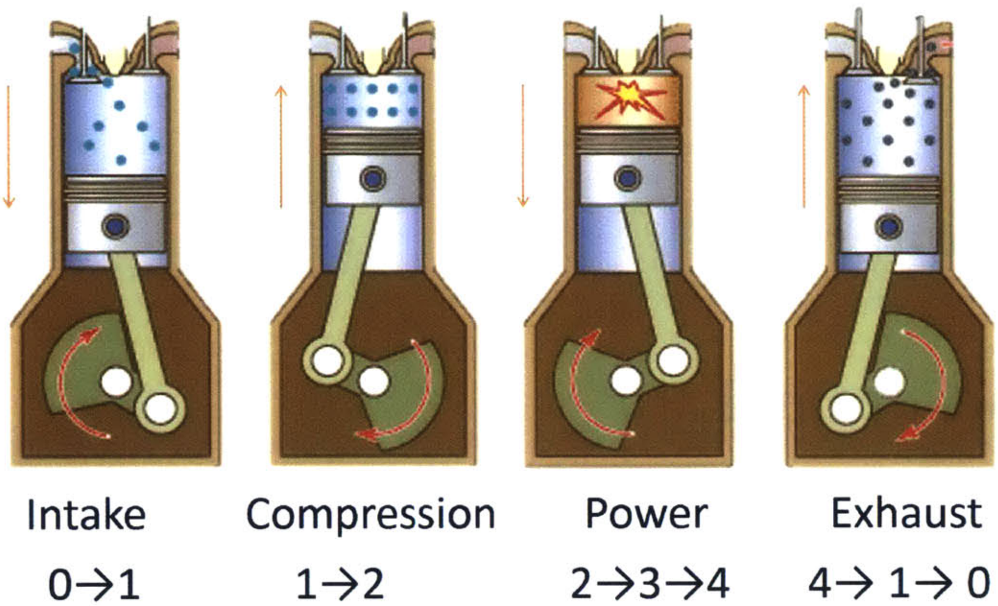

Figure 2-1: How the diesel engine cycle works from encyclopedia Britannica kids edition [15]

The ideal diesel cycle from a thermodynamic standpoint is shown in figure 2-2 [9]. The line from zero to one represents the intake stroke. The compression stroke is adiabatic and represented by the line that connects point one to two. The first part of the power stroke is isobaric expansion and represented by the line that connects point two to three. The second part of the power stroke is adiabatic expansion and represented by the line connecting point three to four. The expansion stroke consists of isochoric pressure drop as the valve opens which is shown by the line connecting point four to one and isobaric contraction.

The energy that is generated through this cycle is represented by the area encapsulated by the lines that connect points one through four. The loop between points one and zero is called the pumping loop. This is caused by the fact that the exhaust pressure is greater than the intake pressure. The area within this loop represents the energy that is lost due to moving the air into and out of the engine.

There were two main reasons that the diesel cycle was chosen to be the focus of this initial research over the Otto cycle (the cycle that gasoline engines use). The

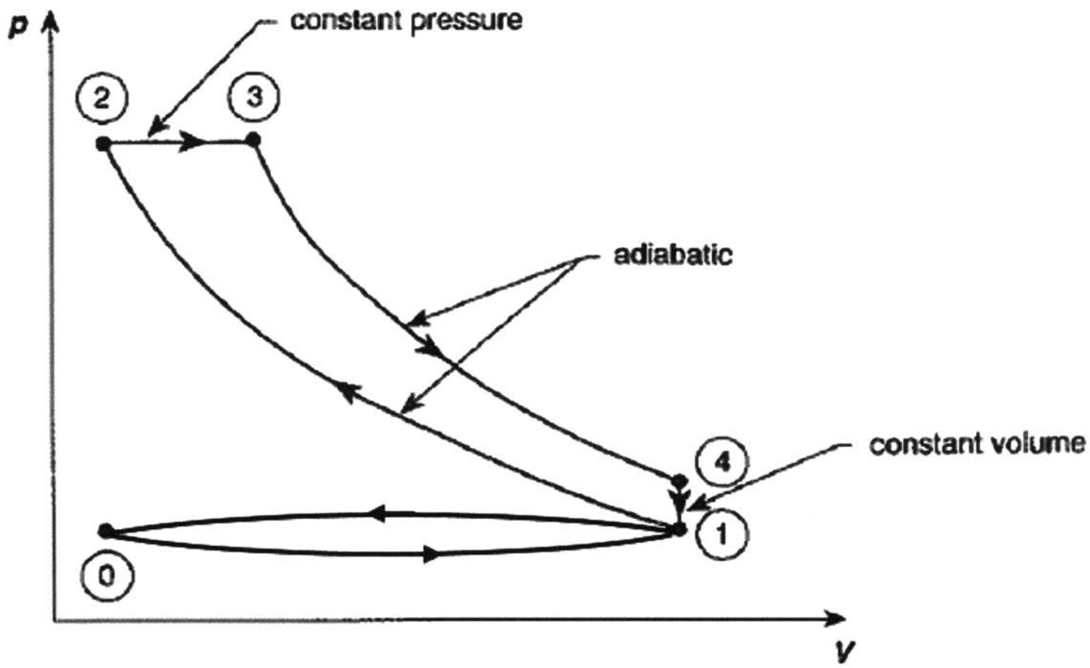

Figure 2-2: Figure showing the PV diagram for the diesel engine cycle from learneasy.info [9]

first is its prevalence in the Indian agricultural sector [6, 1]. The second is that, due to the nature of diesel combustion, the intake stream can be pressurized, increasing the effective compression ratio, without having to worry about knock [13].

One of the key things to look at when trying to optimize an engine is where the fuel energy goes. Only 34-38% of the energy released during combustion is converted into break power in a typical diesel engine [13]. The energy balance for a diesel engine is outlined in John Haywood's book as follows:

Break power: 34-38%

Exhaust losses: 22-35%

Cooling losses:16-35%

Incomplete combustion losses: 1-2%

Other thermal losses: 2-6%

2.2 Turbocharging

The amount of air that the engine can intake limits the amount of fuel that the engine can burn. This in turn limits to the amount of power an engine can produce. If an engine can intake more air it can burn more fuel and produce more power. Currently commercially available single cylinder engines are naturally aspirated. A naturally aspirated engine intakes air directly from the atmosphere. This limits the amount of air it can intake and thus its peak power.

A turbocharger compresses the intake air and thus increases its pressure and, as a result, the intake air density rises. This allows the engine to burn more fuel per cycle, which increases the power output of the engine. Figure 2-3 shows how a turbocharger works. A turbocharger consists of two parts, a turbine and a compressor. The turbine is connected to the exhaust stream of the engine. It uses the energy from the high pressure and temperature exhaust stream to power a compressor. The compressor pressurizes the atmospheric intake air, increasing its density. The engine intakes this higher density air and, as a result, can burn more fuel. Through this process, a turbocharged engine can produce more power than a naturally aspirated engine.

In order to optimize the system, it is important to understand how a turbocharger behaves under different engine operating conditions. At constant load, the cylinder pressure will vary with speed, but not significantly, and the cylinder temperature will increase with speed then decrease at a certain point [3]. At constant speed, both the cylinder pressure and temperature increase with load [3].

2.2.1 Fuel Economy Advantages

There are two places where having higher density intake air leads to fewer losses. First, turbocharging a smaller engine gives the same power output as a larger engine, allowing for fewer frictional losses. This is because a smaller engine has less frictional area between the piston and the cylinder. Note that in the engine energy balance outlined in section 2.1 frictional losses in the cylinder are counted as cooling losses. Second, due to larger mass flow rates of air, there is better heat transfer in the en-

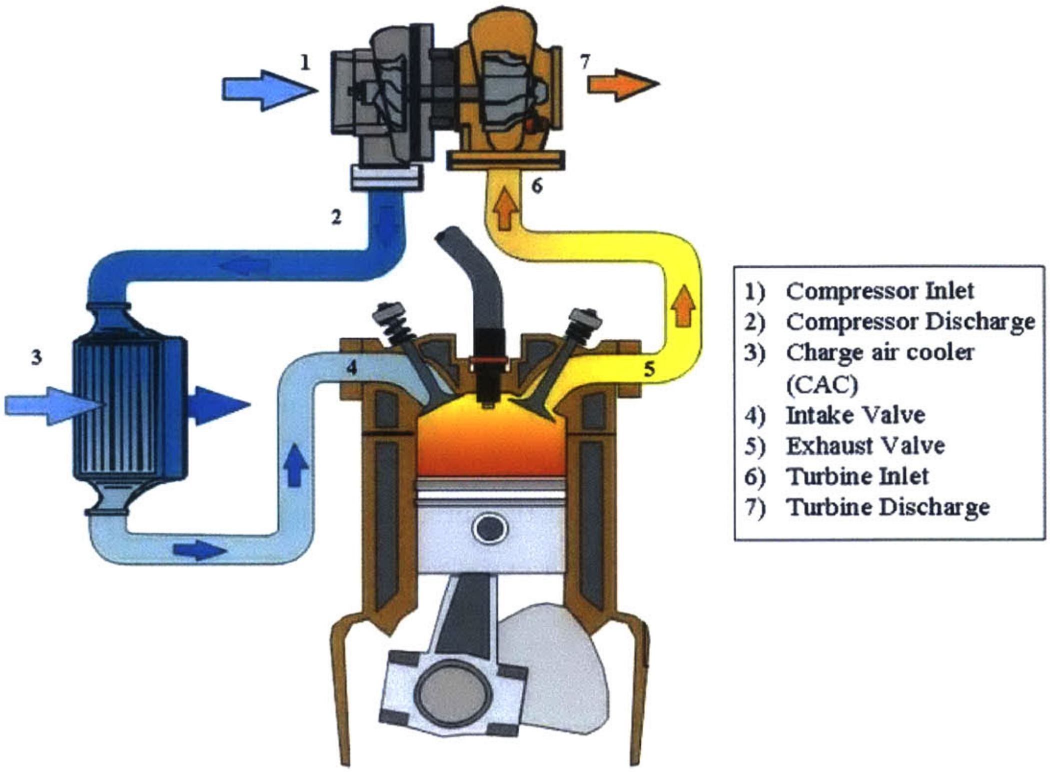

Figure 2-3: Figure showing how a turbocharger operates, figure from Turbo by Garrett [10]

gine, which reduces cooling losses. According to a textbook by Andrei Makartchouk, turbocharging a diesel engine increases the mechanical efficiency by somewhere between two and ten percent [17]. This translates into a reduction in mechanical losses ($\Delta\eta/(1-\eta)$) of between eight and fifty percent. Realistically, the true value will probably be in-between (somewhere around a five percent efficiency gain, and a twenty-five percent reduction in mechanical losses).

A study done on commercial vehicles found that turbocharging had a significant positive effect on fuel economy [8]. In this study several hundred turbocharged and naturally aspirated passenger cars were compared. It was found that turbocharging commercial vehicles resulted in a 8-10% boost in fuel economy and a 30% reduction in engine size [8]. Furthermore, the study concluded that turbocharging had the potential to provide up to an 18% fuel economy boost [8]. An additional study looked at how a single cylinder diesel engine performed under different intake pressures and temperatures. This study showed that fuel economy improved with higher intake density [27].

2.2.2 Cost Advantages

Turbocharging is also inexpensive relative to other methods of increasing power. Colleagues from OEM engine manufacturers have suggested that adding a turbocharger to an engine costs between 10-20% of what it would cost to add another cylinder [1]. The study of passenger cars also found that there was a $300 cost advantage to turbocharging, despite the added cost of the turbocharger [8]. This sets the goal of producing at least 20% more power with our system. If this is possible, the system becomes financially feasible.

2.2.3 Emissions Advantages

Turbocharging has also been shown to reduce emissions. The same study that looked at single cylinder engines under different intake pressures and temperatures showed that increasing the intake pressure also reduces nitrogen oxide emissions [27]. A study that looked at commercial vehicle emissions asserted that turbocharging can reduce carbon dioxide emissions in diesel engines by 30-50% [8].

2.2.4 Turbocharger Lag

The main drawback to turbocharging is turbo lag. This is the time lag that happens while the turbocharger is getting up to speed. With the proposed system, this lag would be increased due to the fact that the turbocharger has to both spool up to operating speed and pressurize the air capacitor to reach steady state operating condition. As a result, the main applications that are being targeted for this technology are steady state ones. These include water pumps, generators, and farm equipment.

2.2.5 Turbocharger Operation Under Pulsating Conditions

An important characteristic of turbochargers that will effect the performance of the proposed system is their behavior under pulsating conditions. This is because a turbocharger that is attached to a single cylinder engine will be powered by exhaust pulses that happen once per engine cycle. Studies suggest that the turbocharger acts like a pulsating pressure source [12]. Another experiment looked at pressure variation in the manifolds of a turbocharged engine as a function of crank angle [20]. In this experiment it was found that the pressures in the inlet and exhaust manifold fluctuate significantly during the engine cycle, which also implies that the turbocharger acts as a pulsating pressure source [20].

2.2.6 Turbocharging for the Indian Small Engine Market

A study was done to look at how turbocharging could effect the small diesel engine market in India. It specifically looked at turbocharging a two-cylinder, four-stroke diesel generator engine and compared it to a naturally aspirated three cylinder engine [23]. It did not consider a single cylinder engine due to the difficulty of turbocharging a single cylinder engine [23]. The goal of this study was to try to find a way to meet new emissions standards set by the Indian government for generator sets [23]. It found that the mechanical efficiency of the turbocharged engine was 1.5% larger (mechanical losses were reduced by about 10%). It also showed that there was a 24% power increase [23]. The study suggested that a greater power increase was possible but the peak power was limited by the fuel pump. The study also demonstrated a decrease in smoke, particulate, and carbon dioxide emissions [23]. However, no difference in nitrogen oxide emissions performance was observed. The turbocharged two cylinder engine also had the advantage of being 12% lighter than the three cylinder engine [23].

2.2.7 Single Cylinder Engine Turbocharging

The idea that turbocharging could improve single cylinder engines is not new. The same study that looked at turbocharging two cylinder engines for the Indian market recognized that turbocharging a single cylinder engine could have many advantages [23]. Turbocharged engines are traditionally multi-cylinder because they can be timed in such a way that when one cylinder is exhausting, and thus powering the turbo, another cylinder is in the intake process [29, 14]. In a single cylinder engine, the exhaust and intake strokes are out of phase, which makes turbocharging difficult because when the turbocharger is powered and tries to deliver air to the cylinder, the intake valve is closed.

Turbocharged single cylinder engines do exist. Hobbyists have posted examples of engines they modified to internet forums [26]. In these cases, they use a large inter-cooler with the turbocharger that probably creates an air capacitor effect. This effect has not been mentioned or characterized on any of the postings that were found. Other boost schemes have also been attempted. A group in India used a combined turbocharger/supercharger system to turbocharge a single cylinder engine [24]. While this eliminates the turbo lag problem and the pulsating flow issue, it is significantly more expensive than just using a turbocharger.

2.3 Overview of Method

The proposed method of turbocharging single cylinder engines is to add a volume to the intake manifold, referred to as an air capacitor, that acts as a buffer to store compressed air between intake strokes and smooth out the peaky nature of a turbocharger operating under pulsing inlet conditions (Fig. 1-1) [30].

The size and shape of the air capacitor are critical to its performance. A volume that is too large will cause excess turbo lag due to the pressurization time of the capacitor. Whereas a volume that is too small will cause a large pressure drop in the intake manifold pressure during the intake stroke, negating the benefits of the turbocharger. The shape of the intake manifold is important to minimize pipe losses and resonances in the system [31]. In this work the theoretical feasibility of the air capacitor is analyzed, focusing on fill time, optimal volume, density gain that can be achieved by the system, and thermal effects due to adiabatic compression of the intake air. The theory will then be tested experimentally.

2.4 Summary

This chapter gives a brief overview of diesel engines, turbocharging, and the proposed method of turbocharging single-cylinder, four-stroke engines. The diesel cycle and its characteristics are discussed in the first section. The second section presents an overview of how turbocharging works. It then discusses turbocharging effects on fuel economy, cost, emissions and engine lag time, and turbocharger behavior under pulsating inlet conditions. The second section concludes by reviewing turbocharging for the Indian small engines market and the issues involved in turbocharging a single cylinder engine. Finally, the third section presents the proposed method for turbocharging a single cylinder engine.

Chapter 3

Modeling and Analysis

This section goes over a series of models that were created to analyze flow through the air capacitor. The goal of these models was to predict the feasibility of using an air capacitor to turbocharge a single-cylinder, four-stroke engine. There were three factors that could effect feasibility: capacitor size, capacitor charge time, and density gain. The turbocharger lag time is the amount of time it takes for the system to reach steady state pressure. If the lag time is too large, then the system will not work in cases where quick power response is needed. The system needs to provide a large enough power gain to justify its increased cost. If the capacitor is too large it will be costly to manufacture and will not be feasible to use in a large number of devices.

The models built on each other in order to create an increasingly accurate picture of how the air capacitor/turbocharger system behaves. As the models got more complex, assumptions had to be made. Instead of trying to exactly predict how the engine will behave, the models looked at two opposite and extreme assumptions. The engine/turbocharger/capacitor system should operate within the boundaries of the operating conditions specified by the assumptions. If the two extremes meet the previously stated goals, then the engine/turbocharger/capacitor system is feasible.

3.1 Pressure Drop Calculation

The capacitor has to be small enough to minimize turbo lag, but large enough not to experience a significant pressure drop during the intake stroke. Our target is to design a capacitor that experiences no more than a twenty percent drop in pressure, compared to its peak pressure, during the intake stroke when the engine is running at steady state. Before investing any resources into this project, the simplest possible model was created. The air is considered to expand isotropically from the capacitor into the cylinder. The capacitor is assumed to start at an elevated pressure and fixed volume. Then during the intake stroke the volume increases to the sum of the volume of the capacitor and the engine. The necessary capacitor volume as a function of engine capacity to maintain at least 80% of the initial pressure throughout the intake stroke is calculated.

Equation 3.1 describes isotropic expansion of gas, where the product of pressure and volume raised to the heat capacity ratio is constant [16]. The left side of the equation represents the pressure and volume of the capacitor before the intake stroke. The right side of Eqn. 3.1 represents the state at the end of the intake stroke, where the capacitor pressure is eighty percent of the turbocharger pressure and the volume is equal to the combined volume of the capacitor and the engine. Equations 3.2-3.6 solve equation 3.1 step-by-step in order to find the capacitor volume as a function of engine volume. This analysis shows that the capacitor volume necessary to maintain the desired pressure throughout the intake stroke should be 5.8x the engine volume. This means that for a 0.4 liter engine, the capacitor volume would be approximately 2.3 liters, which is a reasonable size in comparison to other engine components, such as the fuel tank or muffler. Figure 3-1 shows how pressure drop in the capacitor relates to relative capacitor size; this plot is described by equations 3.7 and 3.8.

$$P_C V_C^{\gamma} = (0.8P_C)(V_C + V_e)^{\gamma} \tag{3.1}$$

$$V_C^{1.4} = 0.8(V_C + V_e)^{1.4} \tag{3.2}$$

$$\frac{V_C}{V_C + V_e} = .853 \tag{3.3}$$

$$1.172V_C = V_C + V_e \tag{3.4}$$

$$.172V_C = V_e \tag{3.5}$$

$$V_C = 5.8V_e \tag{3.6}$$

$$r = CapacitorPressure/TurboPressure \tag{3.7}$$

$$r = \frac{V_C}{V_C + V_e}^{1.4} \tag{3.8}$$

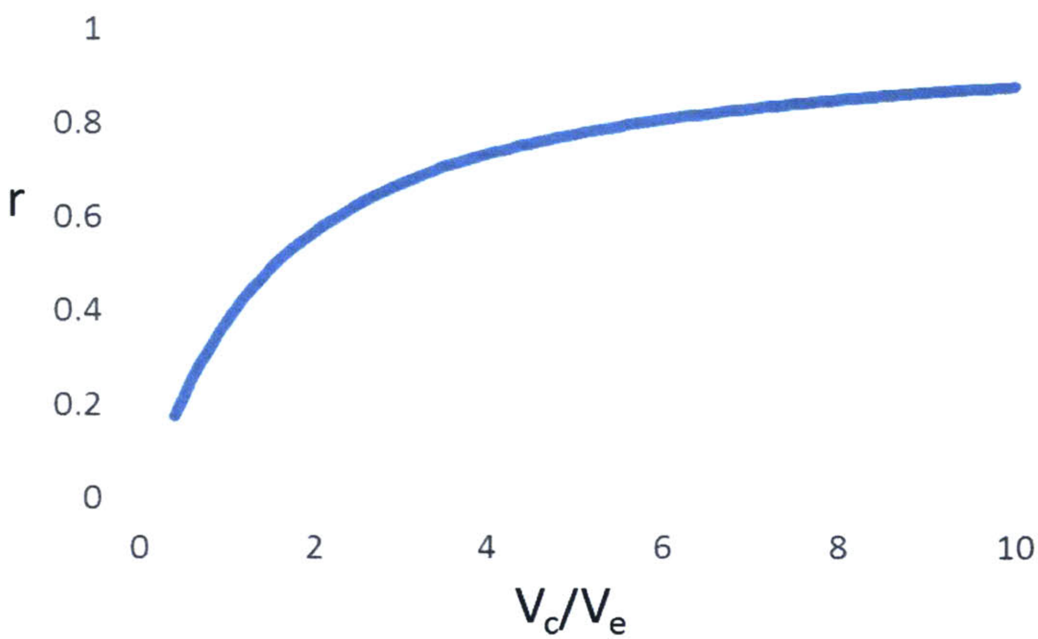

Figure 3-1: Plot showing pressure drop during the intake stroke as a function of non dimensional capacitor size

3.2 Fill Model

The time required to pressurize the capacitor from ambient to operating pressure is an important factor for its performance in transient applications. If the air capacitor takes too long to reach operating pressure, it will negatively contribute to turbo lag. A basic model of the turbocharger, intake manifold, and air capacitor is shown in

Fig. 3-2. The model treats the turbocharger as a constant pressure source that fills the capacitor through a connection which has viscous losses.

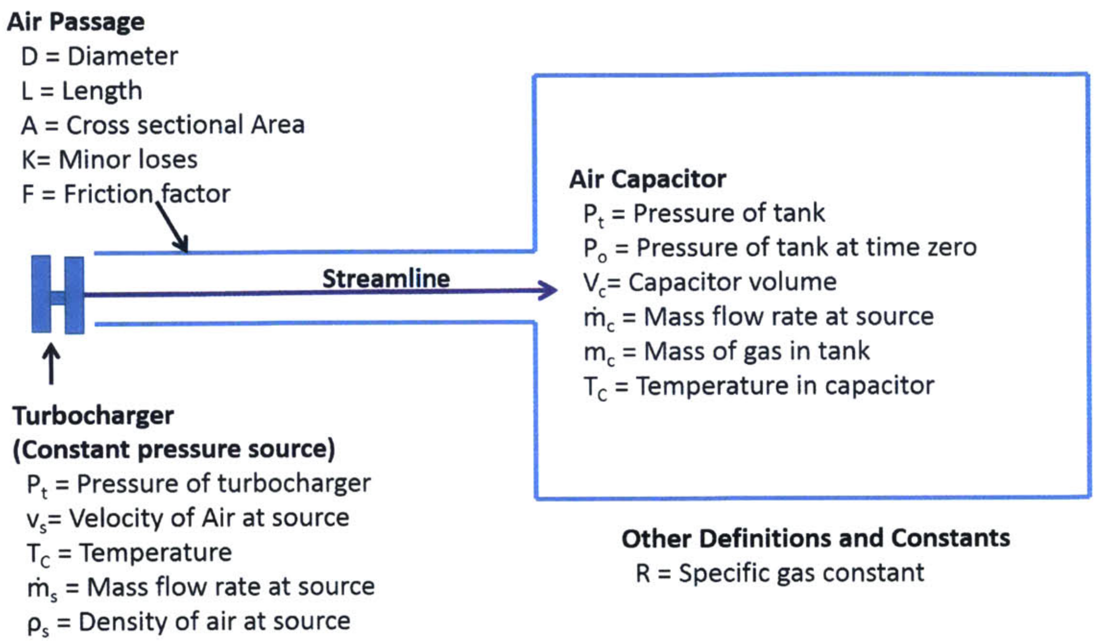

Figure 3-2: Diagram of constant pressure source system

Using conservation of mass, the ideal gas law, and the Bernoulli equation along the streamline that runs from the pressure source to the center of the air capacitor where the velocity of air is zero (Fig. 3-2), we can derive a nonlinear first order partial differential equation that describes how the pressure in the capacitor changes over time, under a constant turbocharger pressure. In the following derivation, equation 3.9-3.20, the flow is assumed to be steady and incompressible [29, 20, 16].

$$\frac{P_t - P_C}{\rho_t} = \frac{v_s^2}{2} + \frac{FL}{D}\frac{v_s^2}{2} + k\frac{v_s^2}{2} \tag{3.9}$$

$$\dot{m}_s = \rho_t v_s A \tag{3.10}$$

$$\dot{m}_C = \frac{\dot{P}_C V_C}{RT_C} \tag{3.11}$$

$$\dot{m}_C = \dot{m}_S \tag{3.12}$$

$$\rho_t v_s A = \frac{\dot{P}_C V_C}{RT_C} \tag{3.13}$$

$$v_s = \frac{\dot{P}_C V_C}{\rho_t A R T_C} \tag{3.14}$$

$$\frac{P_t - P_C}{\rho_t} = (\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}) v_s^2 \tag{3.15}$$

$$\frac{P_t - P_C}{\rho_t} = (\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2})(\frac{\dot{P}_C^2 V_C^2}{\rho_t^2 A^2 R^2 T_C^2}) \tag{3.16}$$

$$P_t - P_C = (\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2})(\frac{\dot{P}_C^2 V_C^2}{\rho_t A^2 R^2 T_C^2}) \tag{3.17}$$

$$C = \frac{\rho_t A^2 R^2 T_C^2}{(\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}) V_C^2} \tag{3.18}$$

$$P_t - P_C = \frac{1}{C} \dot{P}_C^2 \tag{3.19}$$

$$\dot{P}_C^2 + C P_C - C P_t = 0 \tag{3.20}$$

Equation 3.9 is the modified Bernoulli equation along the streamline shown in Fig. 3-2, which accounts for pipe losses [16]. Equation 3.10 defines mass flow from the pressure source as a function of density, velocity, and the cross sectional area of the connecting tube [16]. Equation 3.11 is the mass flow into the air capacitor as defined by the ideal gas law [16]. Equation 3.12 shows that due to mass conservation, the mass flow rate out of the constant pressure source will be equal to the mass flow rate into the air capacitor [16]. Equation 3.13 substitutes equations 3.10 and 3.11 into equation 3.12. Equation 3.14 rearranges equation 3.13 in order to isolate the velocity term. Equation 3.15 rearranges equation 3.9. Equation 3.16 combines equation 3.14 and 3.15 to eliminate the velocity term. Equation 3.17 simplifies equation 3.16. In equation 3.18 we define a variable C that includes the friction factor and velocity, which vary with the pressure differential. However, the variations in these factors are small enough that we can treat C as a constant for small changes in pressure. Since the equation is solved by breaking up the problem into small time steps this should not result in large error. In equation 3.19 we substitute the constant C into equation 3.17. Finally in equation 3.20 we rearrange Eqn. 3.19 to form the nonlinear first order differential equation that describes the pressure inside the capacitor as a function of the turbo pressure.

Using MATLAB and equation 3.20, we numerically solved for the time required to fill the air capacitor to 80% of the turbocharger pressure for a range of diameters and resistances for the tube between the turbocharger and air capacitor. The results of this analysis are given in Fig. 3-3, where the turbo pressure is two atmospheres, the initial capacitor pressure is one atmosphere, and the capacitor has a volume of four liters. The horizontal axis of Fig. 3-3 shows the sum of the dimensionless resistances: the major losses, the minor losses, and the velocity losses. The fill time is very sensitive to tube diameter; for diameters greater than two centimeters, fill time is less than one second. This analysis indicates that the turbo lag due to flow losses is minimal.

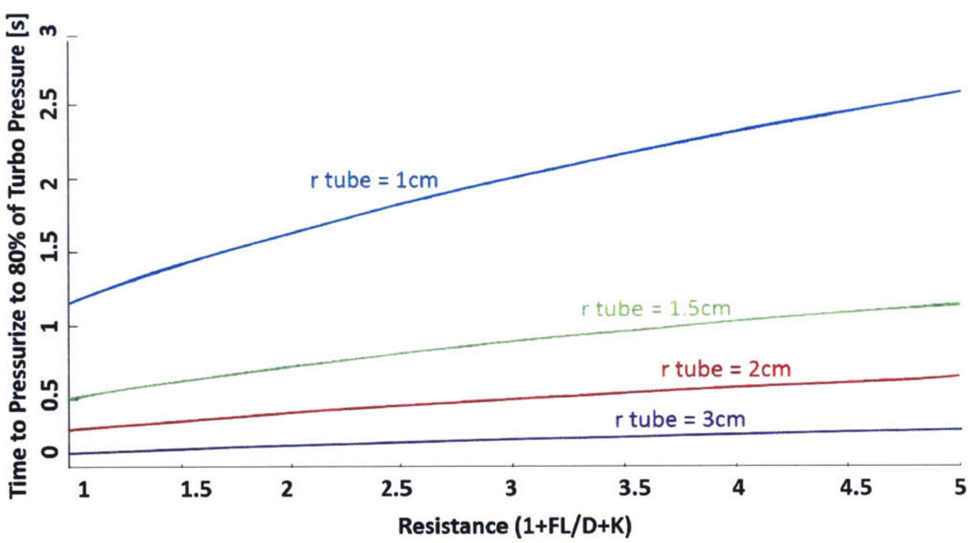

Figure 3-3: Fill time as a function of the resistance and tube radius

3.3 Infinite and Zero Inertia Model

The model presented in this section accounts for the air leaving the capacitor and entering the engine. It is built on the constant pressure source model described in the previous section by using a form of equation 3.20 to characterize flow going from the air capacitor to the engine in addition to the flow from the turbocharger to the capacitor. The air capacitor is treated as a varying pressure source. The engine is treated as a varying volume of air whose volume is based on the phase angle of the crankshaft and is opened and closed with the intake valve (Fig. 3-4).

We consider two ways of modeling the turbocharger. The first is to treat it as a constant pressure source, as presented in the previous section. We refer to this case as the infinite inertia model, since if the rotor had infinite inertia, the turbocharger would never slow down and never vary its pressure. The second is to treat the turbocharger as an intermittent pressure source that only acts like a pressure source when the engine is powering it during the exhaust stroke. We refer to it as the zero inertia model, since with zero inertia the turbocharger would instantly spool up and slow down with the exhaust stroke. In reality the turbine will act somewhere in between these two models. However, literature predicts that it will probably behave closer to the zero inertia mode [12].

The analysis presented in this section uses the following parameters: two-liter capacitor, 13 psi turbo pressure, 0.418 L engine, 3000 RPM engine speed, and compression ratio of 18:1. These parameters were chosen based on the small diesel engine that we selected to use for our experiment. The aim of this analysis is to determine the operating pressure and the time required for the capacitor to reach steady state. The model was built in MATLAB; the code is in Appendix A.

Figure 3-5 demonstrates the transient response of the air capacitor using the infinite inertia turbocharger model. The capacitor reaches steady state in approximately 0.15 seconds with a pressure nearly equal to the turbocharger's pressure. In addition, figure 3-5 shows the response of the system during one engine cycle (note that the engine pressure is only plotted during the intake stroke). During the intake stroke the pressure in the capacitor drops until it becomes equal to the engine pressure combined with the pressure drop due to flow losses. Then pressure in the engine and the capacitor decrease together for the remainder of the intake stroke. During the compression, power, and exhaust strokes, the pressure in the capacitor increases rapidly. A limitation of this model is that it does not capture the spool-up time required for the turbocharger to reach its operating angular velocity.

Figure 3-6 shows the transient pressure response for the air capacitor using the zero

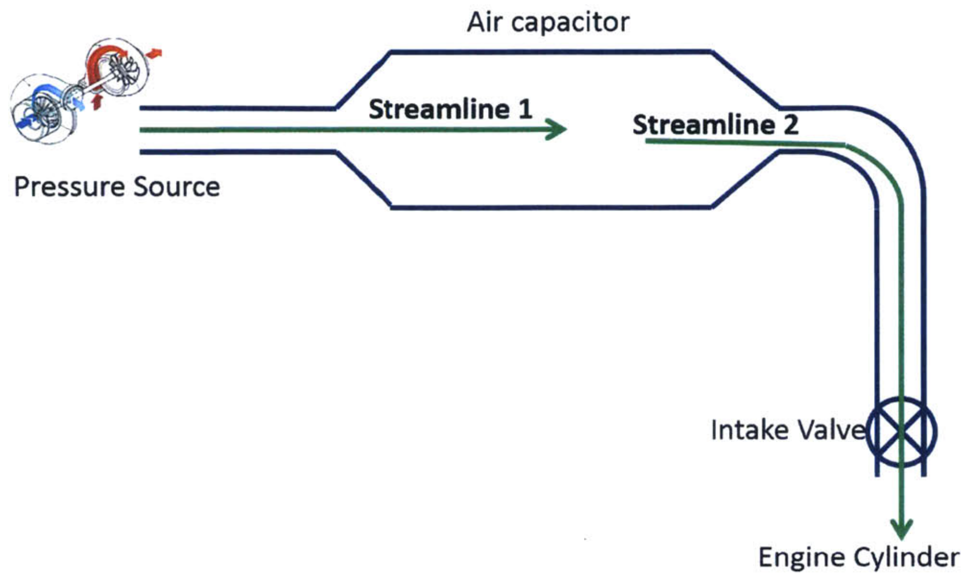

Figure 3-4: Diagram of the full intake model

inertia turbocharger model (note that the engine pressure is only plotted during the intake stroke). The capacitor reaches steady state in approximately half a second and provides an intake cylinder pressure of about fifty percent over atmospheric pressure. In addition Figure 3-6 shows the pressure response in the engine and capacitor over a single engine cycle. During the intake stroke, the pressure in the capacitor drops until it becomes equal to the engine pressure, and then drops further as the piston moves downward and the cylinder fills with air. The capacitor pressure remains constant during the compression and power stroke, and then increases rapidly during the exhaust stroke.

3.4 Temperature Effects

This section describes how heat transfer out of the air capacitor can change its performance. As the pressure inside a fixed volume increases, the density will stay constant and the temperature will increase. In the air capacitor, isotropic compression by the turbocharger results in a pressure gain that is proportionally larger than the density gain in the intake air. This effect of isotropic compression on the temperature of an

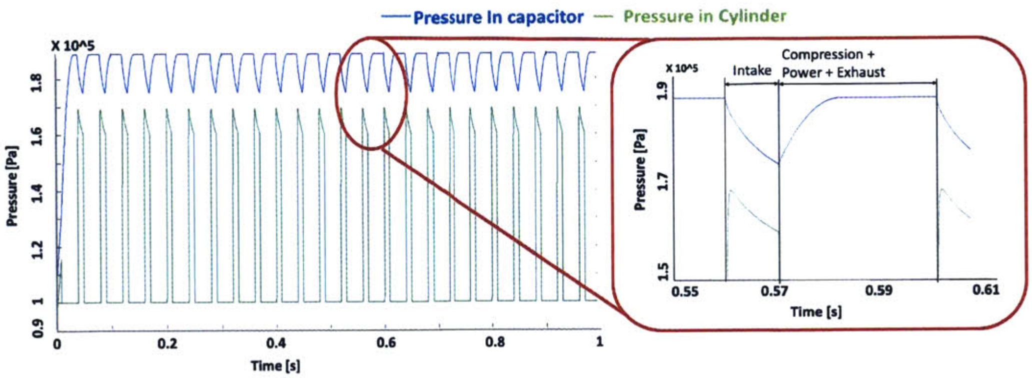

Figure 3-5: Engine and capacitor pressure profile using the infinite inertia turbocharger model; single engine cycle for the infinite inertia model shown on right. Note that engine pressure is only shown for the intake stroke.

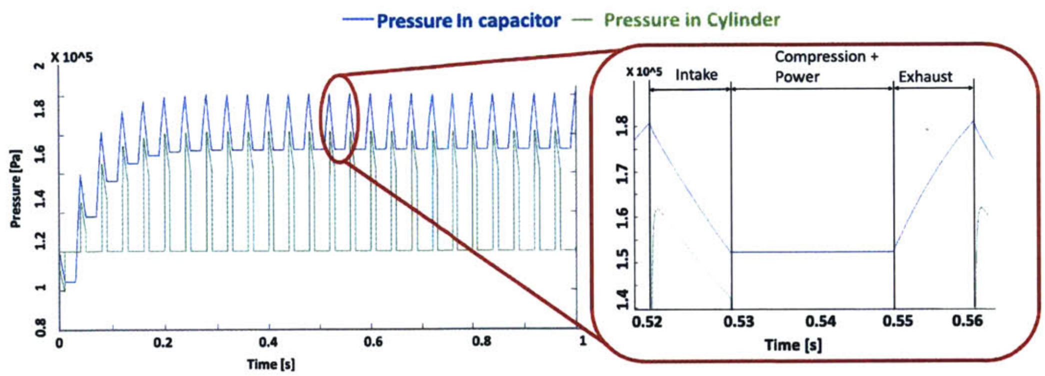

Figure 3-6: Engine and capacitor pressure profile using the zero inertia turbocharger model; single engine cycle for the zero inertia model shown on right. Note that engine pressure is only shown for the intake stroke.

ideal gas is described in a form of the ideal gas law (Eq. 3.21) [16]. How temperature affects density is defined in Eq.3.22 [16]. Equations 3.21 & 3.22 can be combined to get Eq. 3.23, which represents density changes as a function of pressure change in isotropic conditions. If the air is cooled to atmospheric temperature, the density gain would match the pressure gain. The amount of fuel that can be burned in the engine, and thus the power that can be produced, is proportional to the density of the intake air.

$$T_C = T_0(\frac{P_C}{P_0})^{\frac{\gamma-1}{\gamma}} \tag{3.21}$$

$$\rho_C = \frac{P_C}{RT_C} \tag{3.22}$$

$$\frac{\rho_C}{\rho_0} = (\frac{P_C}{P_0})^{\frac{1}{\gamma}} \tag{3.23}$$

The zero inertia MATLAB code is extended in order to create a model of density gain in the air capacitor with and without cooling (App. A.2). The zero inertia model was chosen over the infinite inertia model because it is the more conservative case and literature suggests that this is closer to the way the turbocharger behaves [3]. Section 3.3 operating parameters were used.

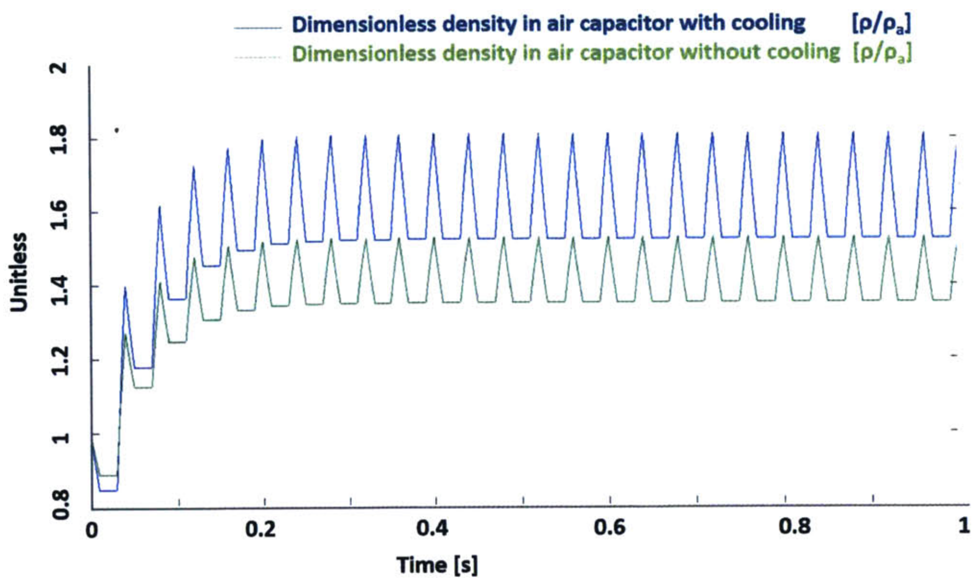

Figure 3-7: Plot showing density decrease due to thermal effects of isotropic compression

The results of this model are shown in Figure 3-7. The graph describes how density in the air capacitor varies under the zero inertia turbocharger model for two cases: assuming no heat transfer and assuming perfect heat transfer. In the no heat transfer case, the intake air density would be approximately 35% higher than atmospheric. To a first order approximation, this increase in density would correspond to a 35% increase in engine power, compared to natural aspiration, as the amount of fuel that can be burned is proportional to the amount of air in the cylinder. Adding an intercooler to the intake system, which would cool the air from the turbocharger, would further densify the the intake air and increase engine power. Perfect cooling of the air to ambient temperature would, to a first order approximation, increase engine power by approximately 65%, compared to natural aspiration. In reality, the system should operate somewhere between these two fringe cases. This proves that the system is feasible from a cost perspective since adding a turbocharger is approximately 20% the cost of adding a second cylinder (doubling the power) [1]. This means that even in the worst case heat transfer scenario, turbocharging is half the cost of adding a second cylinder per unit power gained.

3.5 Fin Cooling Model

Cooling the intake air can provide up to a 30% power gain, therefor methods to transfer heat out of the capacitor were investigated. Additionally, because system cost is one of the most important parameters and the amount of air flow is small compared to larger turbocharged engines, we ruled out traditional intercoolers (cross flow heat exchangers designed to work with larger turbochargers). For the same reasons we ruled out closed loop liquid cooled systems. This left us with looking at using a simple fin system to convect away some of the excess heat. This section describes how a fin system was modeled in MATLAB (model code is in App. A.3).

The MATLAB model that was created starts by defining the geometry of the capacitor, the volume of air flowing through the capacitor, the properties of the air leaving the capacitor, the properties of the atmospheric air, and the geometry of the fins. The model assumes a cylindrical capacitor with equally spaced fins. The Reynolds Number is calculated to determine if the flow through the capacitor is laminar or turbulent (Eqn. 3.24). If the flow is laminar (Re$<$2300), equation 3.25 is used to calculate the Nusselt number for the flow through the capacitor [16]. If the flow is turbulent (Re$>$2300), equations 3.26 & 3.27 are used to calculate the friction factor and Nusselt number respectively for the flow through the capacitor [16]. Using the Nusselt, the number convective heat transfer coefficient for air inside capacitor was calculated (equation 3.28) [16]. Using a similar method, the coefficient for air outside capacitor was calculated (equation 3.29 & 3.30) [16]. Using these coefficients the thermal resistance of the system (equation 3.31) and the heat transfer out of the capacitor wall can be calculated (equation 3.32).

At this point the code calculated the heat transfer of a series of adiabatically tipped fins of different lengths using equations 3.33 - 3.36 [16]. Finally, the total heat transfer rate was calculated (3.37) and, from it, the temperature drop of the air flowing through the capacitor can be calculated (3.38).

$$Re = \frac{\rho * v_C * D_C}{\mu} \tag{3.24}$$

$$Nu = 3.66 + \frac{0.065 * (D_C/L_C) * Re * Pr}{1 + .04 * ((D_C/L_C) * Re * Pr)^{\frac{2}{3}}} \tag{3.25}$$

$$f = (0.79 * log(Re) - 1.64)^{-2} \tag{3.26}$$

$$Nu = \frac{(f/8) * (Re - 1000) * Pr}{1 + 12.7 * (f/8)^{.5} * (Pr^{2/3} - 1)} \tag{3.27}$$

$$h_C = \frac{Nu * k_a}{D_C} \tag{3.28}$$

$$Nu = 0.664 * Re^{1/2} * Pr^{1/3} \tag{3.29}$$

$$h_O = \frac{Nu * k_a}{L_C} \tag{3.30}$$

$$R_C = \frac{1}{h_C * 3.14 * D_C * L_C} + \frac{T_C}{k_{al} * 3.14 * D_C * L_C} + \frac{1}{h_0 * 3.14 * D_C * L_C} \tag{3.31}$$

$$\dot{Q_C} = \frac{T_C - T_0}{R_C} \tag{3.32}$$

$$m_f = \sqrt{h_O * P_0/(k_{al} * A_f)} \tag{3.33}$$

$$M_f = \sqrt{h_O * P_0 * k_{al} * A_f} \tag{3.34}$$

$$T_b = \frac{n * M_f * tanh(m_f * L_C) * (R_{in} + R_{wall}) * T_0 + T_C}{1 + n * M_f * tanh(m_f * L_C) * (R_{in} + R_{wall})} \tag{3.35}$$

$$\dot{Q_f} = n * (T_b - T_0) * M_f * tanh(m_f * L_C) \tag{3.36}$$

$$\dot{Q_{tot}} = \dot{Q_f} + \dot{Q_C} \tag{3.37}$$

$$T_{drop} = \frac{\dot{Q}_{tot}}{v_c * \rho_0 * C_p} \quad (3.38)$$

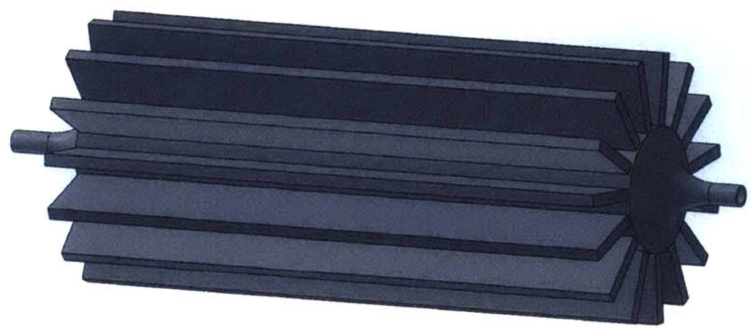

Figure 3-8: Example of a capacitor that could be modeled with the method presented in this section

The results for this model were plotted for a 2 liter aluminum capacitor that was half a meter long, in a 6m/s wind, had 6.7 L/s of air flowing through it, and was covered in fins. Figure 3-8 is a solid model of the type of capacitor that MATLAB is modeling. The results of the MATLAB model are plotted in Figure 3-9. It becomes apparent that there is some benefit to having a fin and, while the performance of the fin increases with fin length, there is a diminishing marginal return once the fin reaches 3-4 cm. However, the temperature drop due to adding fins is only about 10-12 degrees Celsius. This corresponds to 7-8% increase in density according to the ideal gas law which would mean a 7-8% increase in engine power to a first order approximation. While this might seem like a significant increase in power, it requires an aluminum capacitor with fins which is significantly more expensive than a cylindrical steel capacitor. In addition, the finned capacitor is significantly larger than the plain capacitor making it impractical for a lot of applications. As a result, fins were not used on the initial prototype.

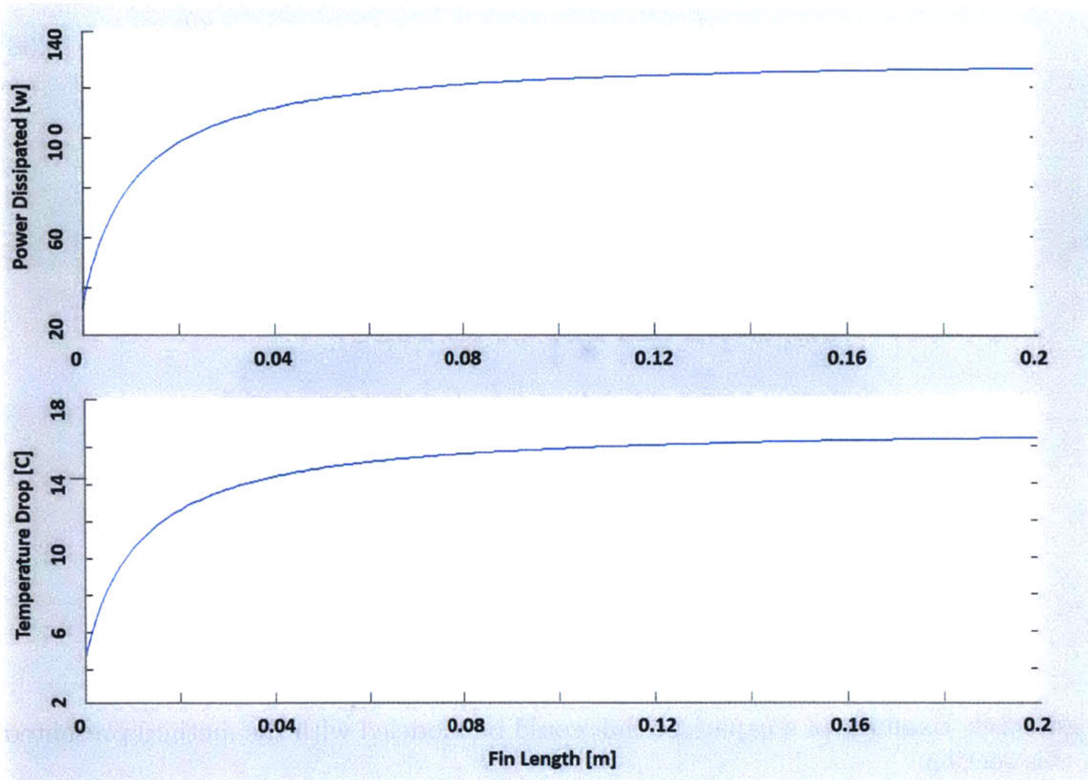

Figure 3-9: Plot showing heat transfer out of air capacitor and temperature drop as a function of fin length

3.6 Summary

In this chapter we consider turbocharging a four-stroke single cylinder internal combustion engine by adding an air capacitor to the intake manifold. The analysis presented addresses four critical design factors: capacitor size, capacitor charge time, pressure variation, and density gain of the air fed into the engine. Our results show that the optimal air capacitor volume is between four and five times the engine capacity, the charge time is less than two seconds, and the density gain of intake air that could be realized is at least 40% greater than ambient without cooling, and up to 70% with ideal cooling.

Chapter 4

Experiment Overview

The next step in this research was to test the theoretical model through experimentation. It was determined that it made more sense to design and build an experiment than to attempt to further develop the computational model. This is due to the fact that to develop further the model, either advanced computational methods would be needed to model combustion and exhaust or assumptions would have to be made that would compromise the accuracy of the results. A simple experiment was designed and built by modifying a commercially available four-stroke, single-cylinder diesel generator. This section goes over the goals and the design of the experiment.

4.1 Experiment Goals & Scope

The goal of this experiment was to demonstrate that a single cylinder engine could be turbocharged and to test how the engine performs with different size air capacitors. The scope of the experiment was kept intentionally narrow in order to minimize cost and time invested. The experiment was planned such that it would only take a year to build, run, and analyze the results. Specifically, this experiment was built to show that the air capacitor system works. If successful, a more complex experiment will be built in order to optimize the system.

The key goals of this experiment were to investigate:

1. The viability of increasing an engine's peak power by using an air capacitor

2. The relationship between peak power output and capacitor size
3. The effect of turbocharger and capacitor size on intake and exhaust manifold pressures
4. The system's effect on intake temperatures in order to determine if an inter-cooling scheme is necessary in the future

In order to keep the experiment setup simple, eight key variables were chosen: air capacitor size, intake air density, engine speed, intake pressure, exhaust pressure, compressor pressure, intake temperature, and power output. The two variables that the operator could control were capacitor size and power output. The other six variables were measured outputs of the experiment. From these metrics viability of the air capacitor could be verified. Other variables which would be interesting to study but costly and not relevant to the testing of the turbocharging scheme are emissions quality, capacitor geometry, and fuel efficiency. These variables will be measured in a more complex future experiment.

4.2 Experiment Design

The experiment was designed to be as simple as possible. The result was a basic dynamometer that cost less than five thousand dollars (Fig. 4-1) and was able to give us the exact data that were required. Since a properly ventilated engine test room was not available, the experiment needed to be mobile in order to be moved outside for operation due to noxious exhaust fumes and brought into the laboratory for storage so that the sensors would not get damaged by the environment. As a result, the entire system was built on a custom cart.

To keep the system as simple as possible, a four stroke single cylinder diesel generator was obtained. This was done so that there was no need to fit a power dissipation unit to the engine. An electrical load can be applied to the generator to apply a load to the engine. The load was applied through a series of space heaters. The exhaust and intake manifold were removed, and a custom system that contained a

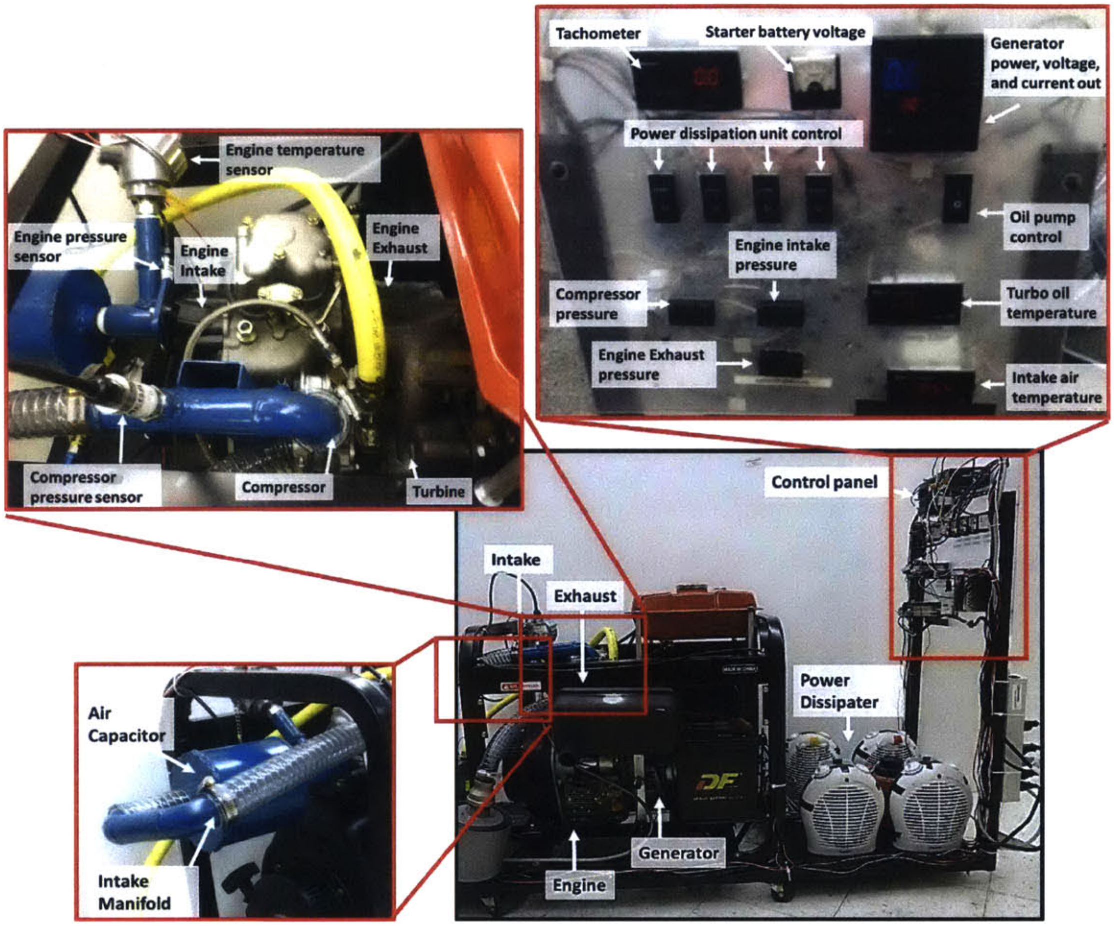

Figure 4-1: Diagram of the dynamometer used

turbocharger was fitted. The new manifold system was designed to allow the operator to quickly switch between different capacitor sizes. Sensors were also fitted to the system to allow the operator to view and log key engine operating variables.

4.2.1 Engine Choice

The diesel engine selected for the dynamometer was a Koop model KD186FA based on the following criteria; a large volume (0.418 L), an 18:1 compression ratio [4, 5], it is a four stroke single cylinder diesel engine, it is similar to engines used in the developing world (low cost, made in China), and it was available from Home Depot already integrated into a generator (Fig. 4-2). The engine is rated for 10 horsepower and the generator unit is rated for a peak power output of 6500 watts [7]. This is approximately the size of engines that our partner companies wanted to turbocharge for use on tractors. Once all the modifications were made, the naturally aspirated peak power was remeasured. The system was designed to run continuously at 3700 RPM to produce a 60 hz AC power source making it easy to find an appropriate model of space heater to act as an electrical load [7].

Figure 4-2: The generator that was used before any modifications were made (picture from Home Depot website [7])

4.2.2 Turbocharger Choice

The Garrett GT0632SZ turbocharger was chosen to be fitted to this engine. It is one of the smallest available turbochargers, designed for engines between 0.1 and 0.5 liters, has a peak turbo pressure of 13 psi [10], and a version of this turbocharger is already used by Tata Motors [28]. Tata Motors, an Indian vehicle manufacturer, uses this engine on some of their two cylinder products including the Tata Super Ace, a 40 hp, 2 cylinder, 0.8 l light truck retailing for approximately 380,000 Rps ($6,000) [28]. This turbocharger's current use on a two cylinder engine indicates that it should be able to operate under pulsating conditions. A separate oil system was fitted to the turbocharger in order to lubricate the bearings. The turbocharger map (Fig. 4-3) shows that for the operating case (13 psi inlet pressure, engine intake density of about 1.7 kg/m^2, and a 0.42 liter engine with a speed of 3700 rpm which results in a corrected air flow of about 2.9 lbs/min) the turbocharger's compressor is about 64% efficient and operates at 230000 RPM.

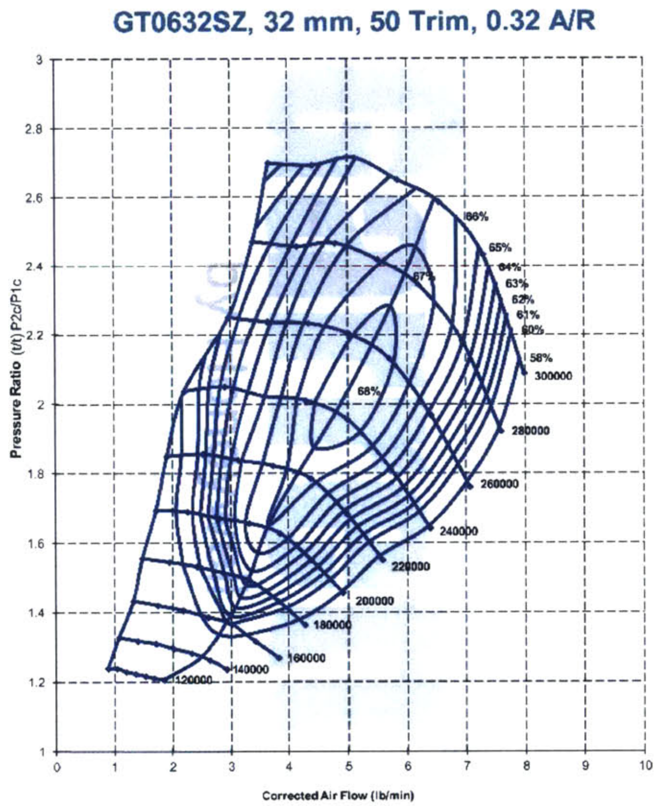

Figure 4-3: Turbocharger map of the Garrett GT0632SZ (map provided by Garrett [10])

4.2.3 Manifold & Capacitor Design

In order to fit a turbocharger to this engine, the intake and exhaust system were stripped. Then a custom exhaust manifold was designed to couple the turbocharger to the engine. It was designed to be as small as possible in order to prevent the air from cooling between the engine and the turbocharger. The manifold was cut out of steel plates with a water jet then welded together. An adapter was built to couple the turbocharger outlet to the muffler (Fig. 4-4). This adapter had a tube welded to it with a pressure sensor fitted to its end that enable the measurement of exhaust pressure without burning out the sensor.

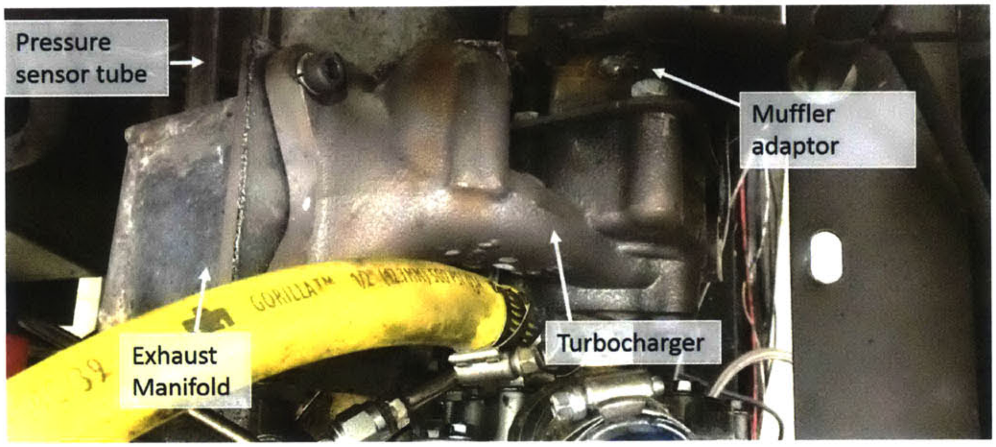

Figure 4-4: Custom exhaust system that was fitted to the engine

An air filter was chosen and connected to the turbocharger's compressor intake using 1.25" ID rigid tubing. The engine intake was modified with a custom made adapter that converted the stock engine intake fitting into a female one-inch NPT pipe fitting. The adapter was also fitted with a temperature sensor and a pressure sensor, placed as close to the engine as possible in order to obtain intake air density (Fig. 4-6).

Another pressure sensor was installed close to the turbocharger's compressor outlet. The NPT fitting allowed for different sized air capacitors to be attached to the engine (Fig. 4-5). The capacitors were made by welding together water jetted steel end plates to steel tubing. NPT pipe nipples were then welded into the middle of the end plates. The capacitors were screwed into the engine intake and then connected to the compressor with rigid tubing and hose clamps. This created a system where capacitors could be easily manufactured (about 2 hrs per capacitor) and could be swapped out in about 5 minutes. Six capacitors were made with sizes ranging from 0.4 litters to 2 litters (total intake manifold volume of 1.3 -2.9 litters). The system was also tested without a capacitor (total intake manifold volume of 0.9 litters). The result of these modifications was an engine with a new intake and exhaust system that included a turbocharger (Fig. 4-7).

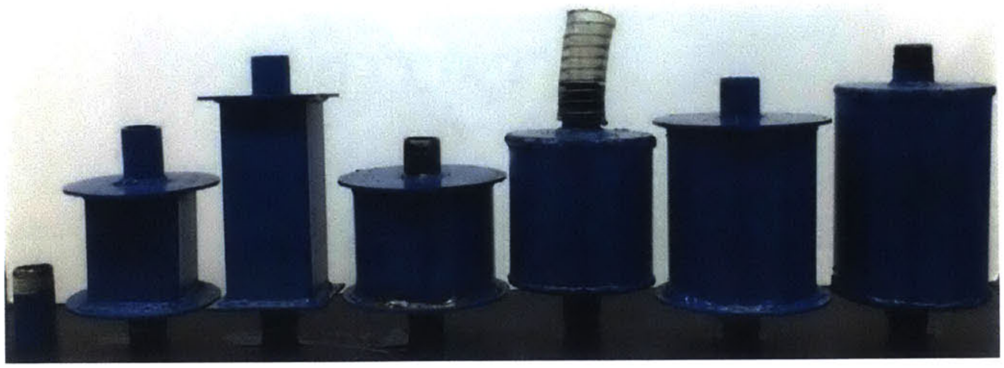

Figure 4-5: Air capacitors built for this setup that allowed for different intake manifold volumes (0.9 L, 1.3L, 1.6 L, 2L, 2.3L, 2.6 L, 2.9 L total manifold volume)

4.2.4 Power Dissipation

Space heaters were attached to the generator to provide a controlled load on the engine. Six space heaters were used and each could deliver a load of up to 1500 W. Three of the space heaters were wired to the generator through switches that the operator could control and two were plugged directly into the generator. Each of these heaters was set to draw 1500 watts. The final space heater was wired to the generator through a variable AC power transformer that the operator could control, allowing it to generate a load between zero and 1500 watts. As a result, the power dissipation system could provide a load of anywhere between 0 and greater than 9000 watts on the engine (while the space heaters were rated for 1500 watts they actually dissipated between 1500-1600 watts). The internal circuit breakers on the space heaters were removed to prevent them from tripping under the extreme conditions created by this experiment.

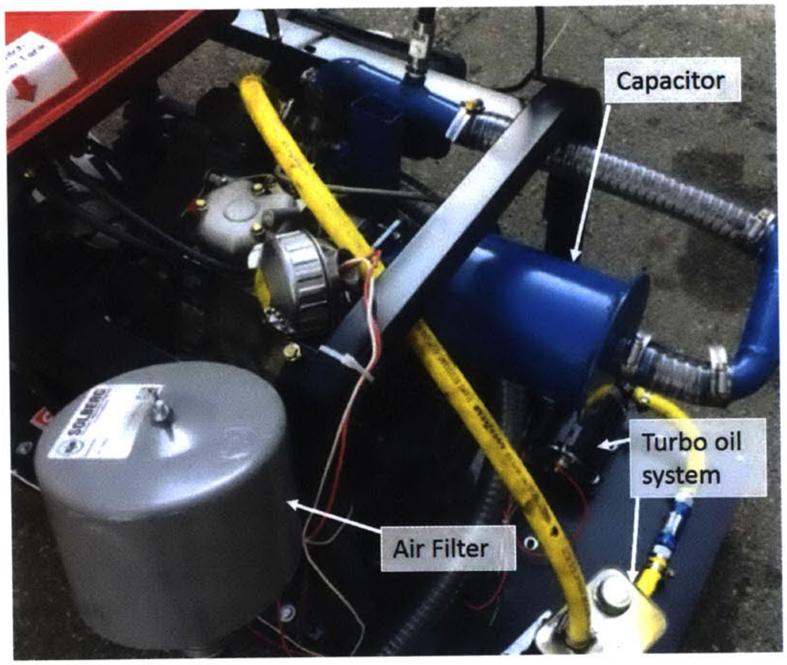

Figure 4-6: Intake manifold with a medium sized capacitor, 2.6 L total manifold volume

4.2.5 Sensor Choices

Pressure Sensors

Multiple pressure sensors were needed to monitor compressor pressure, intake pressure, the exhaust pressure. All three sensors had similar requirements; they had to be accurate, temperature resistant, and operate between 0 and 20 psi gauge. Omega PX309-030GV sensors were selected. These sensors were chosen because they are accurate to within 0.25%, can withstand temperatures of up to 85 degrees Celsius, have a range of 0-30 psi gauge, and can easily interface with a digital read out [21].

Temperature Sensors

To calculate air density, the intake temperature is monitored. An air tight 4-20 mA RTD probe from McMaster was chosen. The probe can withstand temperatures of up to 400 degrees Fahrenheit and can easily interface with a digital readout [18]. To construct an airtight seal, the probe was installed using an NPT fitting. The

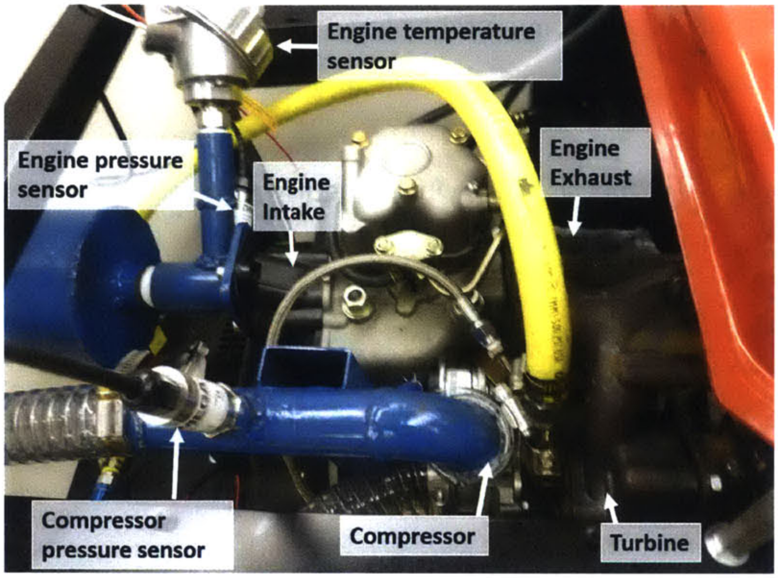

Figure 4-7: New manifold system designed around the turbocharger

turbo oil pressure was monitored to prevent damaging the turbo. A similar RTD probe, without the air tight housing, was chosen to measure the turbocharger oil temperature.

Tachometer

A tachometer was installed to measure engine speed. An optical tachometer was used because there was no location where a contact tachometer could be fitted. An optical tachometer that came with a digital readout was purchased from McMaster. A strip of aluminum foil was epoxied to the coupling that connected the diesel engine to the electric generator for the tachometer to read. The tachometer reading indicated that the engine was always operating between 3650-750 RPM. However, the vibrations from the engine resulted in occasional errors where it was unable to read the engine speed. These were addressed by adjusting the location of the optical sensor.

Power Sensors

The load on the engine was measured by a power-meter that displayed the total power, voltage, and current coming out of the generator. The meter worked by detecting the generator's output voltage from a probe placed on one of the space heater's electrical cords. The probe was able to get the generator's output current from an induction current sensor. One of the power wires from each space heater was run through this loop. This provided a robust and accurate method to measure the power output of the generator.

4.2.6 Control Panel Design

The whole dynamometer system is monitored and controlled through a central control panel (Fig. 4-8)), which had ease of use as the primary design requirement. The panel was laser cut out of a sheet of acrylic with a hole for each of the sensor read outs and switches. It allows the operator to easily vary the load on the engine, view the engine operating condition, and log the data coming from the experiment.

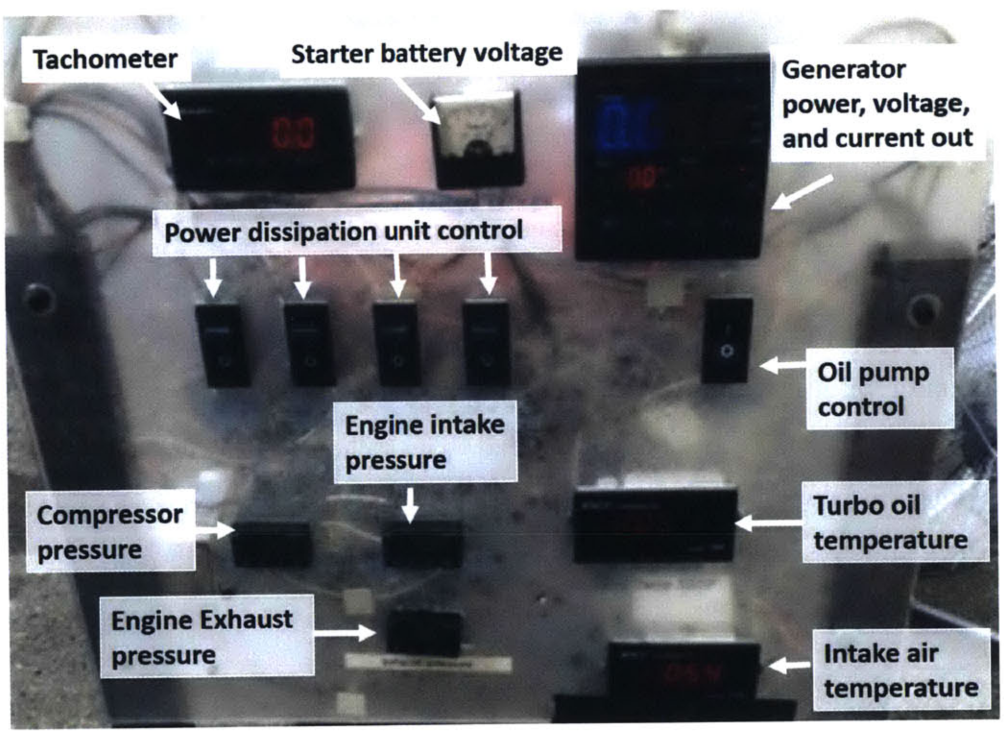

Figure 4-8: Control and monotoring panel for the dynamomiter

4.3 Running the Experiment

The experiment was run outside in order to avoid releasing noxious gas indoors. For each capacitor size, two experiments were run. The cold peak power test measured the peak power of the engine after cold start. The manifold characterization test measured the hot peak power by slowly increasing the load on the engine and logging the engine characteristics until manifold pressures started to vary significantly, at which point the load was increased until the engine stalled. In addition to this, the turbocharger system was removed and a naturally aspirated test was run to establish a baseline. The raw data from these tests are in appendix B.

4.3.1 Test procedure for the cold peak power test

1. Log the outside temperature and humidity
2. Make sure the engine is cold (it can take up to half an hour for it to cool off)
3. Turn on the turbocharger oil pump and wait until the oil flows smoothly through the system (approximately a minute)
4. Turn on the engine with no load on it and wait until it reaches steady state operation (less than a minute)
5. Gradually increase load by turning on space heaters and adjusting the variable AC controller
6. Log the generator stall load

4.3.2 Test Procedure for the Engine Characterization Test

1. Log the outside temperature and humidity
2. Make sure the engine is cold (it can take up to half an hour for it to cool off)
3. Turn on the turbocharger oil pump and wait until the oil flows smoothly through the system (approximately a minute)

4. Turn on the engine with no load on it and wait until it reaches steady state operation (less than a minute)

5. Log the load on the generator, intake temperature, exhaust pressure, intake pressure, and compressor pressure

6. Increase the load by between 700-800 W and wait until the engine reaches steady state operation (less than a minute)

7. Log the load on the generator, intake temperature, exhaust pressure, intake pressure, and compressor pressure

8. Repeat steps five and six until the manifold pressures vary by more than one PSI

9. Increase the load on the engine until it stalls

10. Log the engine stall power

4.4 Summary

In this section, the design of an experiment to test the air capacitor method for turbocharging a single cylinder four stroke was described. The experiment was designed to be low cost and built around a commercially available diesel generator. From this experimental set up it was possible to obtain data on peak power, manifold pressures, intake air density, and engine speed. It took approximately a year to build and run the experiment.

Chapter 5

Results and Analysis

This section goes over the results of the experiments described in Chapter 4. The experiments yielded data that showed how capacitor size affected peak power, intake pressure, compressor pressure, exhaust pressure differential, and intake air density. The conclusion is that intake manifold size has a significant effect on engine performance and that it is feasible to turbocharge a single-cylinder, four-stroke engine with an air capacitor. Additionally, the results were compared to the temperature adjusted zero inertia model. The raw data from the experiments are contained in Appendix B.

5.1 Peak Power Results

The key factor that determines the economic feasibility of the system is the peak power gains achieved by turbocharging. Cold and hot peak power were measured for seven different intake manifold sizes and the naturally aspirated case. The results are plotted in Figure 5-1. Peak power increases of up to twenty-nine percent were observed for the cold case and up to eighteen percent for the hot case.

The system had the same trend that the zero inertia temperature adjusted model predicted. As capacitor size increased, peak power increased. There was also a diminishing marginal return for increasing peak power past a certain point. However, the peak power was less than the model predicted. This could be due to a combination of

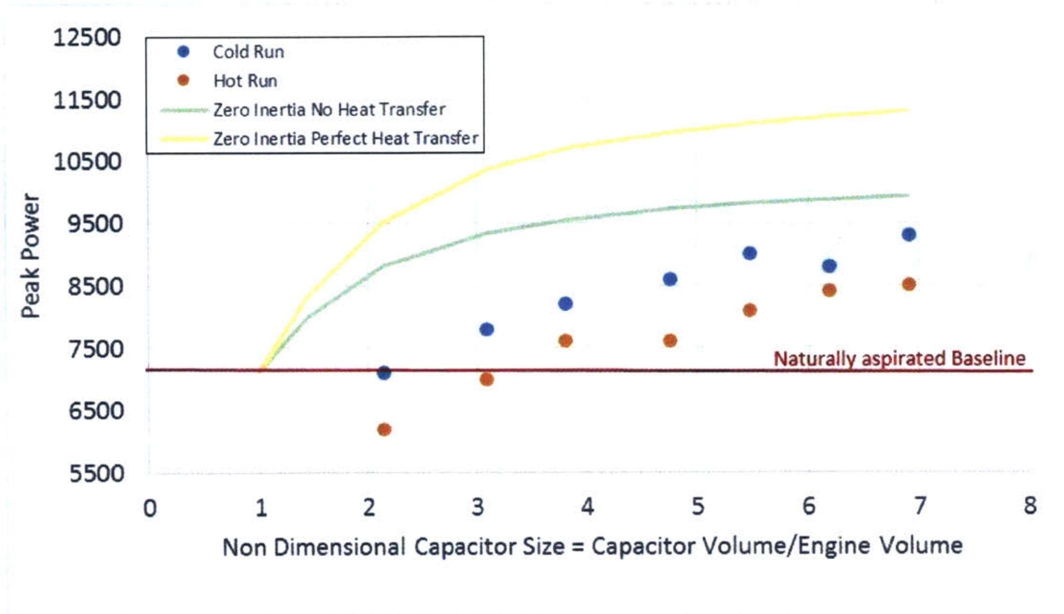

Figure 5-1: Plot of measured peak power values compared to temperature adjusted zero inertia model

factors including: ambient conditions, the engine intake leakage, exhaust gas drifting back into the engine, and elevated exhaust pressure. Adjusted peak power increases of up to thirty three percent were observed for the cold case and up to twenty three percent for the hot case. The ambient density effect is discussed in this section. The rest of the issues are examined in the results and discussion section (Section 5.4).

Since tests were run outdoors on different days, the ambient density of the air during the tests was not constant and could vary by as much as five percent. It is possible to adjust for ambient density to the first order using Eq. 5.1. The results, after adjusting for ambient temperatures, are shown in Figure 5-2. The adjusted results are closer to, but still noticeably less than the predicted power gain.

$$PeakPowerAdjusted = PeakPower\frac{AmbientDensityPeakPowerTest}{AmbientDensityNaturallyAspiratedTest} \quad (5.1)$$

Additionally, it was also observed that for the naturally aspirated case the cold and hot peak powers were the same, while in the turbocharged case the cold peak

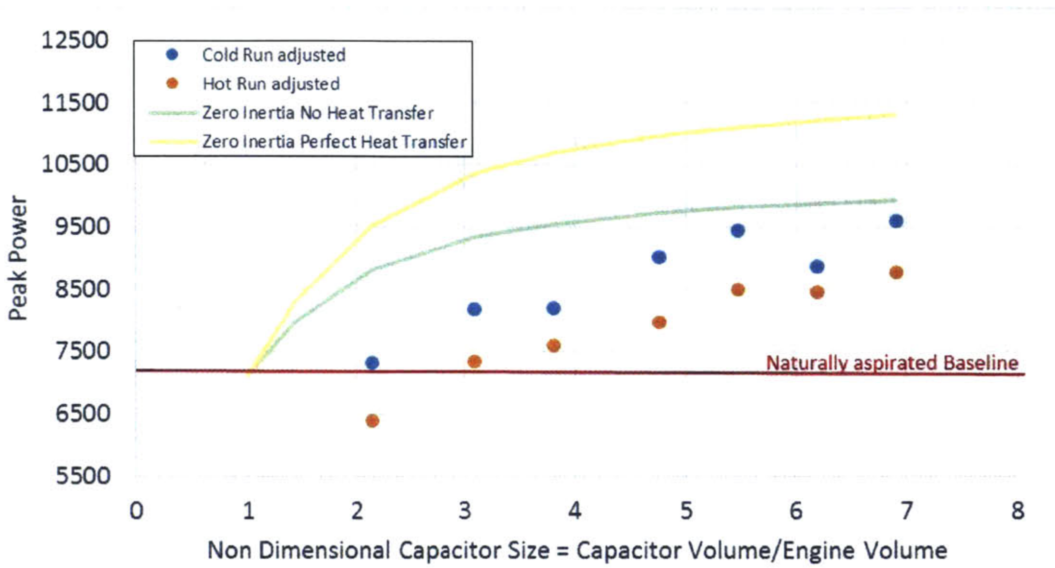

Figure 5-2: Plot of adjusted peak power values compared to temperature adjusted zero inertia model

power could be over ten percent greater than the hot peak power. The difference is probably due to the turbocharger getting hot and heating up the intake air as a result. This further shows the importance of cooling.

5.2 Density Gain Results

Density gain is important because it indicates how well the turbocharger is working and should be the same as peak power gain on the first order. It is calculated from the intake temperature and pressure using the ideal gas law (Eqn. 5.2) and is the percent increase of the intake density over the ambient air density (Eqn. 5.3). The experiment was able to measure intake air density gain for all capacitors up to about seventy percent of peak power at which point the pressure in the capacitor fluctuated too significantly to get accurate readings. Figure 5-3 shows the maximum measured density gain for different sized capacitors. The number next to each point is the load on the generator at which the maximum density gain measurement was taken.

$$IntakeDensity = \frac{IntakePressure}{IntakeTemperatureXSpecificGasConstant} \quad (5.2)$$

$$PercentDensityGain = (\frac{IntakeDensity}{AmbiantAirDensity} - 1)X100 \quad (5.3)$$

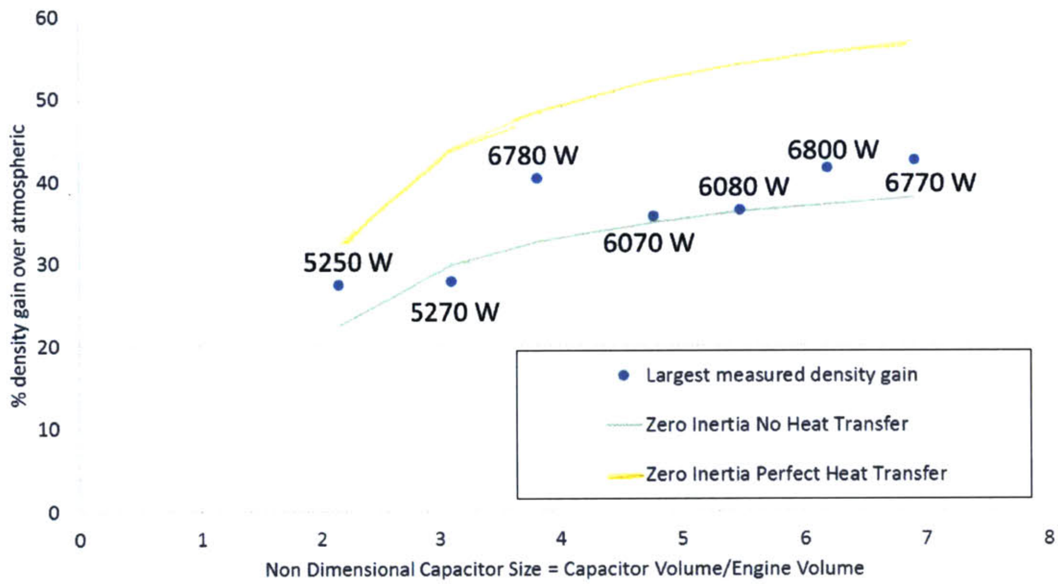

Figure 5-3: Plot of measured maximum density gain compared to temperature adjusted zero inertia model. The number next to each point represents the power at which peak density gain was measured which is less then the peak power output.

Note that because the measurement was taken at less than peak power, the measured peak density is probably less than the actual peak density, which should be the density at peak power. This also means that the points on the plot were taken at different power levels and, therefore, do not represent a trend of how density varies with capacitor sizes. These trends are shown in Figures 5-4 and Figure 5-5.

Figure 5-4 shows how the intake density gain is related to the load on the generator. This relationship is close to linear; as the load on the generator increases the intake density also increases. The naturally aspirated density gain is negative due to flow losses over the air filter and temperature gains due to the engine. At the maximum measured point, the turbocharged engine's density gain is about forty-five percent greater than the naturally aspirated engine's density gain. However, the adjusted peak power of the engine, as shown in figure 5-2, is about twenty nine percent. This suggests that density and power gain are not related as closely as predicted. This could be due to exhaust pressure effects, exhaust gas getting blown back into the engine, and/or the engine's internal intake manifold leaking. These potential sources of error are discussed later in this chapter.

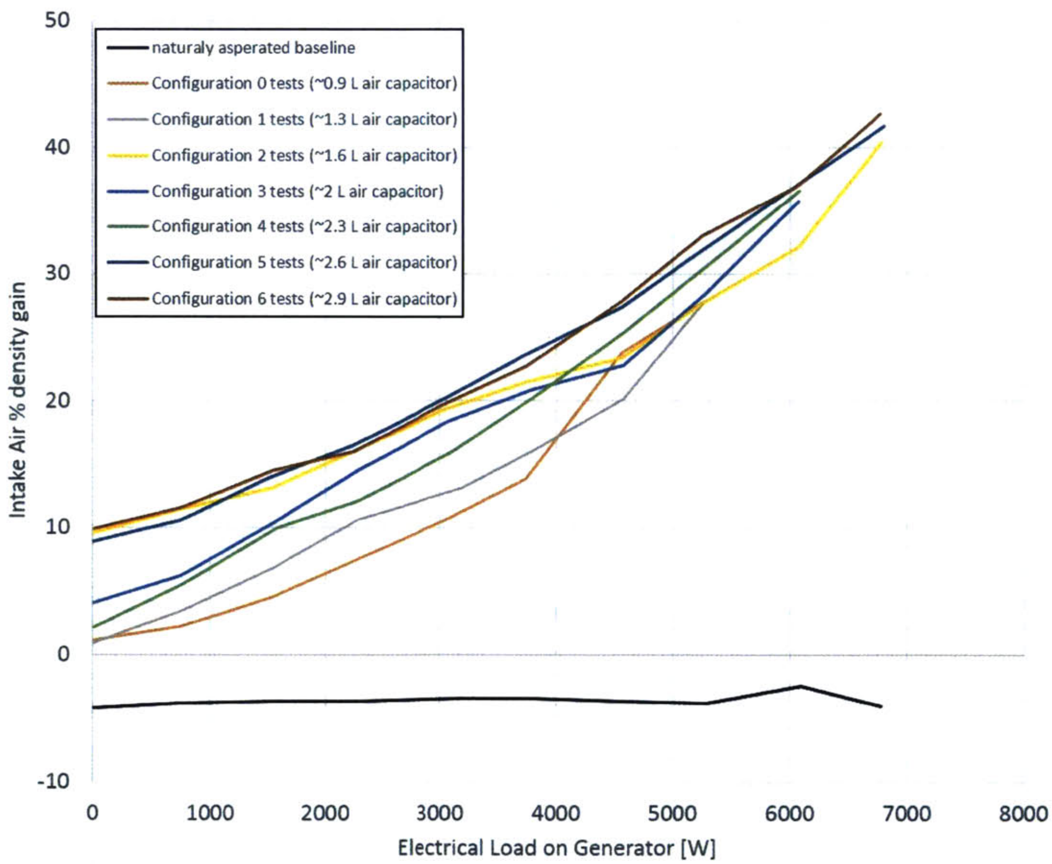

Figure 5-4: Plot of percent density gain of the intake air plotted as a function of power dissipated.

In order to show the relationship between capacitor size and density gain more clearly, the same data are plotted with capacitor size as the x axis. Figure 5-5 shows the same data as figure 5-4 but with lines of constant engine load. From this plot one can see a clear positive correlation between capacitor size and density gain. However, significant variation in the low power measurements makes it difficult to draw any meaningful conclusions.

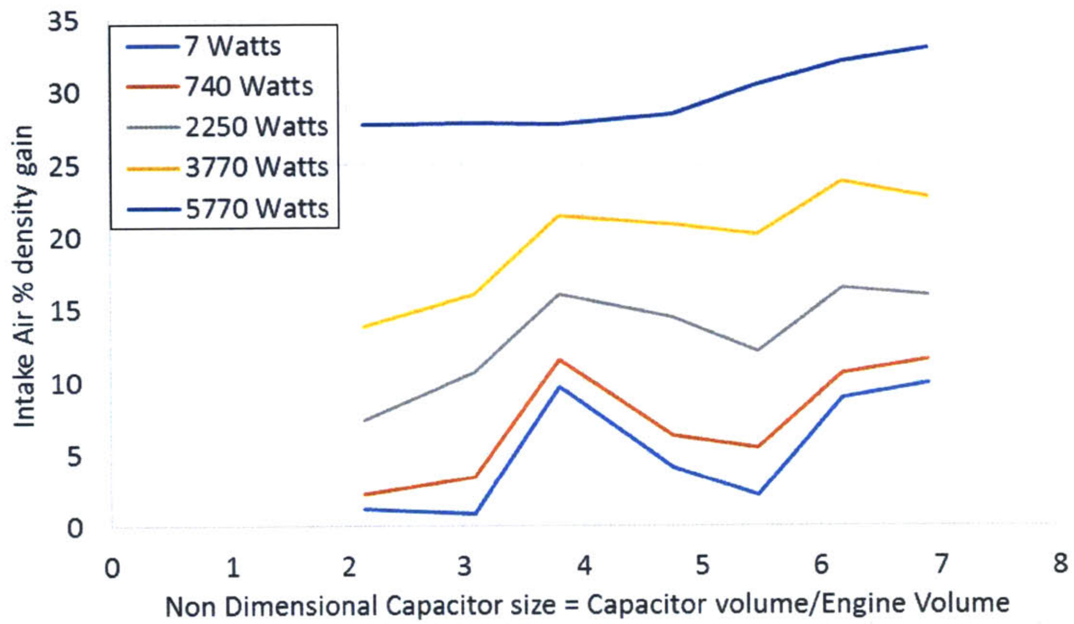

Figure 5-5: Plot of percent density gain of the intake air plotted as a function of capacitor size

5.3 Manifold Pressure Results

This section focuses on the pressures in the engine manifolds. The data were obtained from three pressure sensors: after the capacitor at the engine's intake, right after the compressor, and in the exhaust manifold. These pressures indicate how the turbocharger is working, how much of an effect back pressure has on the system, and how much pressure the compressor is providing.

Figure 5-6 shows the gauge pressure in the capacitor as a function of load on the generator. This view of the data is useful for understanding how pressure varies with load and comparing the turbocharged data with the naturally aspirated baseline. It is clear that as load on the generator increases so does the intake pressure. Note that the naturally aspirated intake pressure is negative due to flow losses across the air filter. The peak measured pressure is about 9.6 psi, which is 74% of the turbochargers waste-gate pressure (13 psi). This is less than the actual peak pressure in the cylinder due to the fact that the pressure was only measured with generator loads of about 6800 W while the turbocharged engine was able to produce outputs of up to 9300 W.

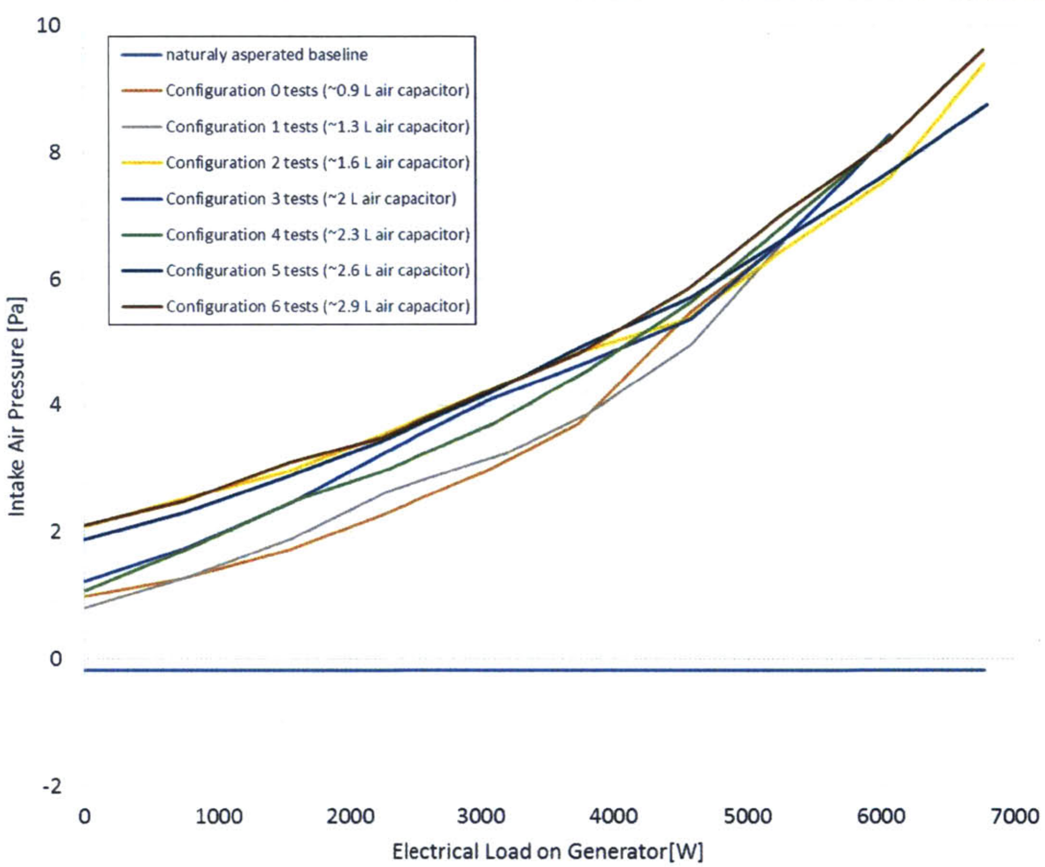

Figure 5-6: Plot of intake pressure as a function of power dissipated

Figure 5-7 shows the same data as figure 5-6, but with lines of constant engine load (+/- 2%) and the x axis as capacitor size. This view is useful for seeing how intake pressure varies with capacitor size. There seems to be a clear positive correlation between capacitor size and intake pressure. However, significant variation in the low power measurements makes it difficult to draw any meaningful conclusions as to the magnitude of this correlation.

Figure 5-8 shows how compressor pressure varies with capacitor size. It is clear that there is a positive correlation between capacitor size and compressor pressure. However, significant variation in the low power measurements makes it difficult to

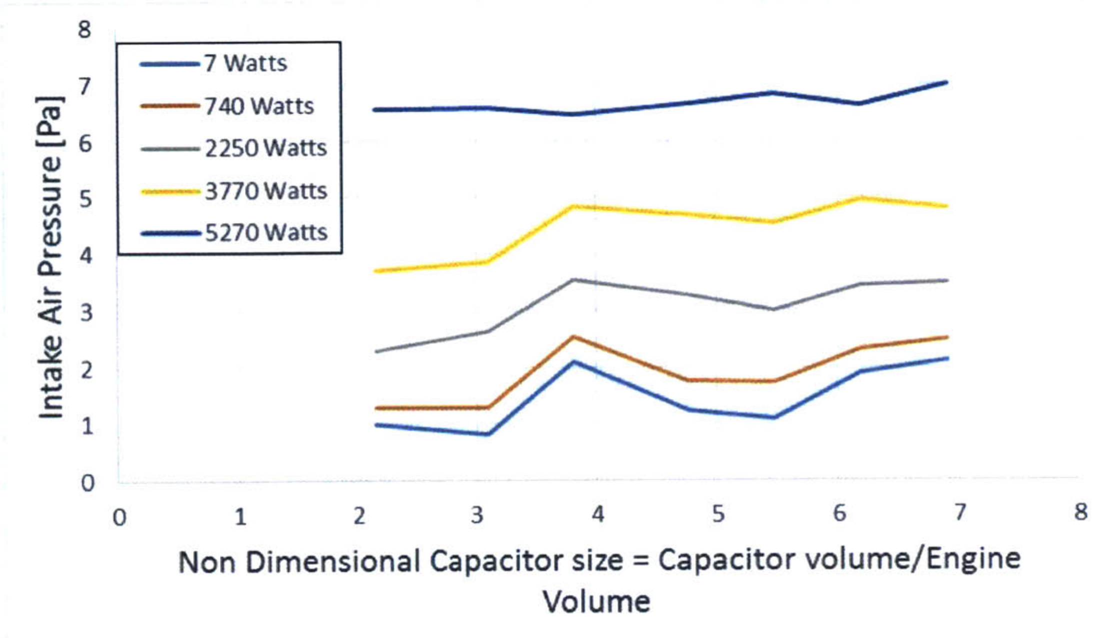

Figure 5-7: Plot of intake pressure as a function of capacitor size

draw any meaningful conclusions as to the magnitude of this correlation. These data are not plotted as a function of engine load because there is no naturally aspirated baseline to compare these data to; therefore, using power as the x axis provides little added value.

The final experimental result that were examined is how the exhaust pressure differential correlates with generator load and capacitor size. The exhaust pressure differential is caused by flow losses in the exhaust, making the exhaust pressure larger than the intake pressure. This is relevant because increased exhaust pressure differential will have two negative effects. First, it will increase pumping losses, taking energy out of the system and reducing the overall peak power. Second, because the exhaust gas is at a higher pressure, more exhaust gas will be left in the cylinder (due to incomplete scavenging) after the exhaust stroke, which will also reduce peak power.

Figure 5-9 shows how the exhaust pressure differential changes with load on the generator and the naturally aspirated exhaust pressure differential. It is clear from the data that, for the turbocharged case, as load on the generator increases the exhaust pressure differential decreases. It is the opposite case for the naturally aspirated

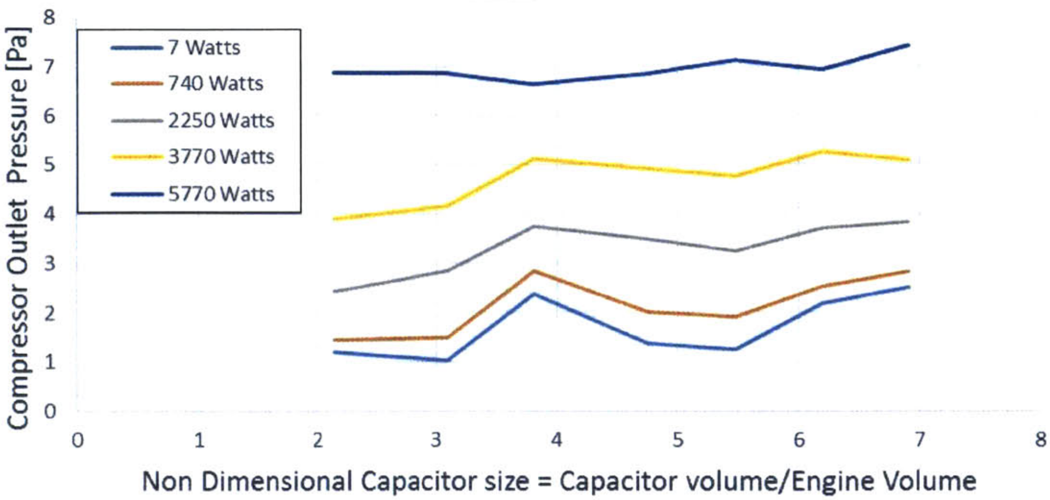

Figure 5-8: Plot of compressor pressure as a function of capacitor size

baseline, where the exhaust pressure differential increases as the load on the generator increases. For the turbocharged case the exhaust pressure differential most likely decreases with increased generator load due to higher enthalpy exhaust gas increasing the intake pressure. For the naturally aspirated case, the exhaust pressure differential increases with generator load probably because the increase in exhaust temperature that happens at higher engine loads increases the exhaust pressure.

However, for all cases the exhaust pressure differential is noticeably larger in the turbocharged case than it is in the naturally aspirated case. The difference ranges from approximately 3 Psi, when there is no load on the generator, to about 1.5 Psi at 6.8 KW of load on the generator. This increase is probably due to the added flow resistance of the turbine, the new exhaust manifold, and the muffler connector. This partially could account for the mismatch between the peak power and density gains.

In order to see how exhaust pressure varies with capacitor size, the same data are reploted with capacitor size as the x axis. Figure 5-10 shows the same data as 5-9, but with lines of constant engine load (+/- 2%). There seems to be no clear correlation between capacitor size and exhaust pressure differential. This means that the exhaust pressure ratio decreases consistently with the generator load independent

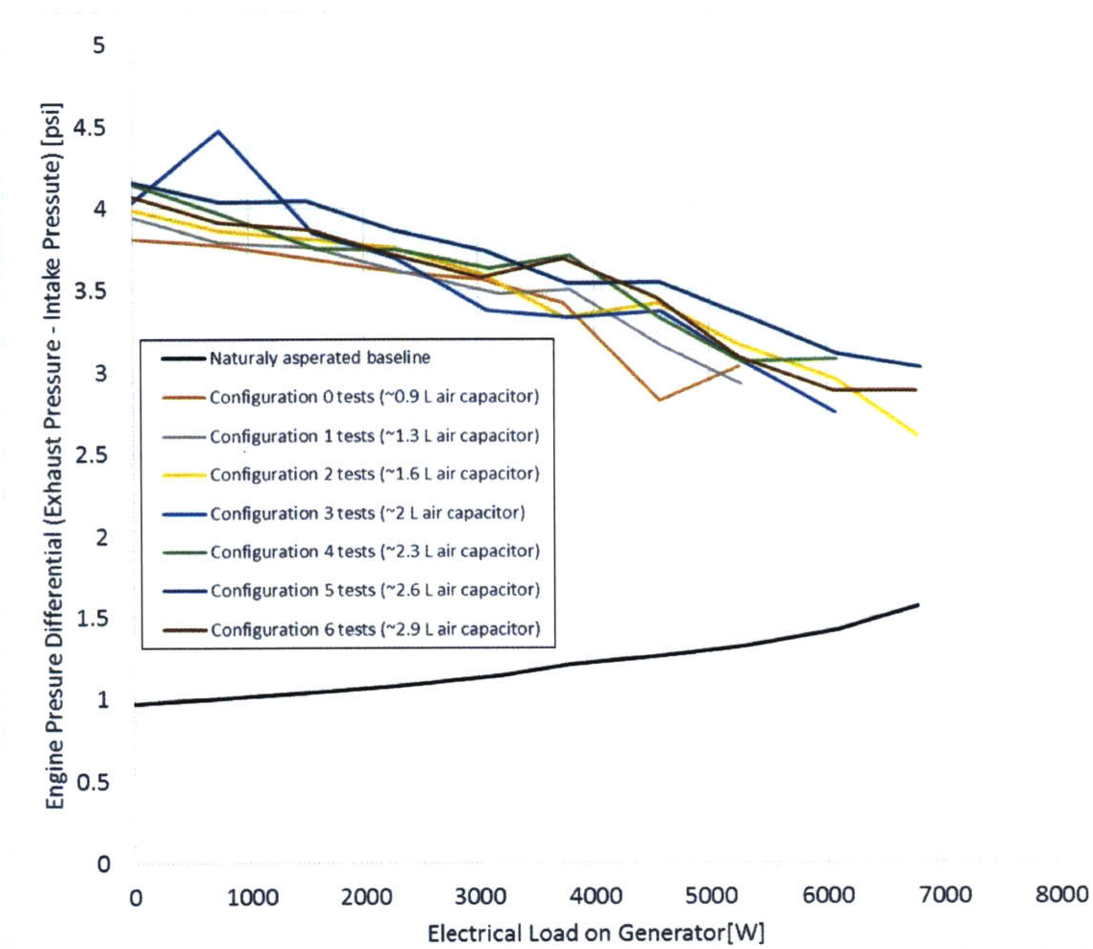

Figure 5-9: Plot of exhaust pressure differential (difference in pressure between the exhaust and intake manifolds) as a function of power dissipated

of capacitor size.

5.4 Error and Discussion

The results of this experiment confirm our two main hypotheses. First, it is possible to turbocharge a four stroke single cylinder engine using a large volume intake manifold. Second, the capacitor size has a key effect on the performance of the engine. It was also found that the model did a reasonable job of predicting peak density gain. However, the peak power gain did not match our model and did not correspond to the density gain to the first order. There were a few possible sources of error that

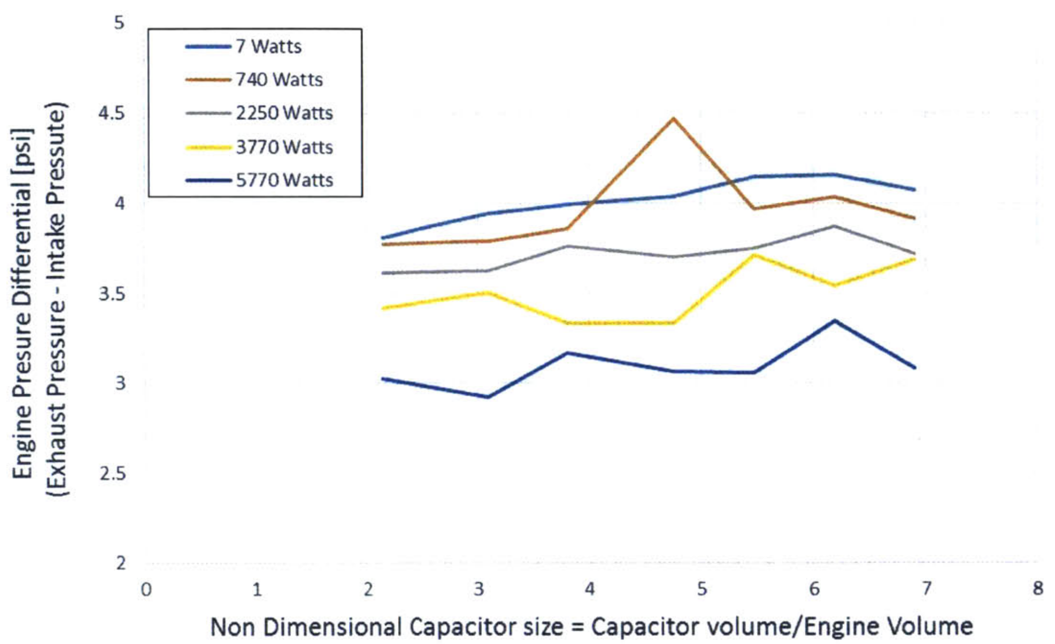

Figure 5-10: Plot of exhaust pressure differential (difference in pressure between the exhaust and intake manifolds) as a function of capacitor size.

could have contributed to the experimental results not matching the theory exactly.

The first possible source of error was that the theoretical model did not take into account exhaust pressure differential. An elevated exhaust pressure differential would cause an increase in pumping losses and, as a result, a decrease in peak power. In addition, it would lead to more exhaust mass left in the cylinder after the exhaust stroke, which would also decrease power.

A second possible source of error was air leakage. The passage that led from the custom built intake manifold to the intake valves was the same one that the engine came with and was not modified in any way. Since this engine was designed to be naturally aspirated, it was not designed to be under significant positive pressure. This led to leakage in this passage that was significant enough that the operator was able to observe it. This leakage would result in a density reduction of the air going into the engine cylinder, which would cause a reduction in peak power.

The third potential source of error was exhaust gas getting pulled into the intake.

While the tests were conducted outside, the area they were conducted in had poor air circulation. Therefor, exhaust gas could have gotten pulled into the intake stream. This would cause less oxygen to enter the engine, resulting in a decrease in peak power. The operator noticed this effect at higher power levels when dark exhaust gas could be seen near the intake.

Overall, the experiment is considered a success, and these errors will be addressed in future iterations of the experiment. The experiment showed that the proposed turbocharging system is feasible and that intake manifold properties significantly effect the engine operating characteristics.

5.5 Summary of Results

In this section the results of an experiment to test the feasibility of turbocharging a single-cylinder, four-stroke engine were presented. It was ascertained that peak power increases with capacitor size, but not exactly as the model predicted. Potential errors that could have caused this discrepancy were discussed. It was also established that peak intake density gain increase with capacitor size, as the model predicted. In addition to this it was observed that turbocharging increases the exhaust pressure differential, and as the load on the engine increases, the pressure differential decreases. As the theory predicts, intake pressure and capacitor pressure were found to increase as load on the engine increases and as capacitor size increases. Overall, the experiment could be deemed a success and a future study on this project is recommended.

Chapter 6

Summary

This thesis presents a novel method for turbocharging four-stroke, single-cylinder internal combustion engines. First, the motivation for turbocharging a four stroke single cylinder engine is covered. Second, the new method, using a large volume intake manifold to store the air between strokes, is reviewed. Third, computational models of the system are described. Fourth, the design and construction of an experiment to test the turbocharging method and the accuracy of the computational models is written about. Finally, the results of the experiment are presented and analyzed.

6.1 Scholarly Contributions

This thesis makes the following scholarly contributions:

1. Computational models are defined that predict how a single cylinder four stroke engine with different sized air capacitors behaves. The models showed that turbocharging a single cylinder four stroke engine is feasible and that air capacitor size effects key operating characteristics.

2. A low cost and accurate dynamometer that can test the turbocharging method was designed and built. This made it possible to compare how the engine behaved turbocharged versus naturally aspirated. The dynamometer also allowed

for measurements of key operating characteristics without having to invest the significant time and cost that a large scale dynamometer would require.

3. Utilizing the dynamometer, it was shown that it is possible to significantly increase the peak power of a single cylinder four stroke engine by using and air capacitor/turbocharger system.

4. The experimental results were compared to the model. It was found that while the experiment behaved with the same general trends as the model predicted there were multiple sources of error that were identified.

5. The effect that capacitor size has on peak power output, manifold pressures, and density gain was demonstrated through experimentation. It was shown that as air capacitor size increases so dose peak power, intake air density gain, and intake pressure.

6.2 Engineering and Social Impact

Turbocharged engines have significant advantages over naturally aspirated engines. They are lighter, more fuel efficient, and less expensive. However, due to the timing mismatch between the intake and exhaust strokes commercial single cylinder engines are not turbocharged. The method presented is a simple, low-cost way to turbocharge single-cylinder, four-stroke engines. This thesis presents data that show that this method is feasible and could have a broad impact.

Single cylinder engines are used in many applications in both the developing and developed world. The technology presented in this thesis could create a broad impact on equipment such as tractors, light vehicles, generators, and water pumps, increasing their specific power per unit cost and thus making them economical for new users. Improvements to this technology could significantly enhance quality of life for small engine users.

6.3 Future Work

The goal of this thesis is to create a technology that can become commercially viable in multiple sectors. Subsequent work will focus on creating a general method for optimizing a single cylinder turbocharged engine around three key factors: peak power, fuel economy, and emissions.

To achieve this, the next steps in this research are laid out as following. First, the current computational model needs to be refined in order to more accurately predict how turbocharging affects peak power output. Second, new models will be created in order to predict emissions impact and fuel economy. Based on these models a more complex experiment will be devised that will allow for measurement of fuel economy and emissions impact. From this point on, the experiment and the model will undergo iterations until it is possible to predict engine behavior and the system is optimized. Finally, with the help of corporate partners, the system will be implemented in commercial engines.

Appendix A

Model Code

This appendix contains select MATLAB code described in section three. Note that some models is not included because the code used in some of the simpler models are just simpler forms of the code used in the more complex model.

A.1 Basic Fill Model

This is the code used to create the model described in section 3.2. It is a function that outputs the time required to fill the capacitor. This function was run in a simple loop to create the plot shown in Fig. 3-3.

```
function [t] = basicFillT(n, rpm, Dt, Lt)
%basic flow model for the turbo charger air capasitor system, it asumes
%that no flow is leaving the capacitor and calcultes time to reach full
%charge

%n is the ratio of Capacitor volume to engin volume
%rpm is the engin speed
%Lt is the leangth of conecting tube in meters
%Dt is the diameter of conecting tube in meters
```

```
Ta=30+273; %ambiant temperature in C
Rair= 287; %specific gass constent for dry air J/KgK
At= ((Dt^2)/4) *pi ; %cross sectional area of the tube in m^2
Pa= 100000; %atmospheric pressure in pascal
we = rpm/60; %engin rps
Ve=.625; %engin capacity in liters
Vcap= Ve * n; %capacitor volume in liters
Pc(1)=Pa; %pressure inside the capactitor,
            %sets it to atmospheric pressure at time = 0
patm=Pa/(Rair*Ta); %density of air

e= 0.00001; %roughness of the pipe
u=2*10^-5; %viscosity of ait in kg/ms

Qe=we/2*Ve/1000; %volume flow rate out of the engine in m^3/s
Me=Pa*Qe/(Rair*Ta); %mass flow rate of air out of the engine kg/s
n=.95; %turbo charger efficiency
r=4; %compression ration
r=r/2; %adjusted compression ratio to acount for tubing and other volume
        %between exaust and turbo
PeDrop=r*Pa;
We=Me*PeDrop/patm; %work created in the turbo charger as a
                    %result of exaust air flow in watts

Pt=200000; %Turbo presure in pascals this will depend on amount of volume between
%the cylender and the turbo and the compression ratio, for now it will be
% estamated to be two times the ambiant pressure
pturbo= Pt/(Rair*Ta); %air densitr in the turbo in kg/m^3
Vc=((Pt-Pa)/Pa)*Vcap/1000; %volume that needs to be added to the capacitor in m^3
                            %to fill it to Pt

Vin= We/(Pt-Pa); %volume flow rate out of the turbo in m^3/s


Vair= Vin/(At); %velocity of air through the tube m/s
```

```
Re=(pturbo*Vair*Dt)/u;
f=(1.8 * log10((6.9/Re)+((e/Dt)/3.7)^1.11))^-2; %friction factor
PdropF = f*(Lt/Dt)*(pturbo*Vair^2)/2;
Vin1= We/(PdropF+Pt-Pa+(pturbo*Vair^2)/2); %mass flow rate out of the turbo in kg/s

while(abs(Vin1-Vin)>.0001)
    Vin=Vin1;
    Vair= Vin/(At); %velocity of air through the tube m/s
    Re=(pturbo*Vair*Dt)/u;
    f=(1.8 * log10((6.9/Re)+((e/Dt)/3.7)^1.11))^-2; %friction factor
    PdropF = f*(Lt/Dt)*(pturbo*Vair^2)/2;
    Vin1= We/(PdropF+Pt-Pa+(pturbo*Vair^2)/2);
                        %volume flow rate out of the turbo in m^3/s
end
Vin=Vin1;

t=Vc/Vin;


end
```

A.2 Zero Inertia Model With temperature Effects

This is the code used to create the model described in section 3.4 it outputs the plot shown in Fig. 3-7.

```
ps = 1.2; %density of air in atmmosphere [kg/m^]
V= 0.002; %volume of tank in [m^3]
Rc=18; %compression ration
r=.02; %radius of conecting tube [m]
D = 2*r; %diameter of conecting tube [m]
L = .75; %leangth of conecting tube [m]
A = 3.14*r^2; %cross sectional area of conecting tube [m^2]
R = 287; %specific gas constant [J/molK]
```

```
T = 300; %tempriture of gas in kelvin
F = 0.7; %darcy friction factor (an aproximation for now, I know its not constant)[1]
k = 3; %Minor losses(an aproximation for now, caused by piping conections and valves)[1

Pa = 100000;
Pt = 100000; %initial presure in tank (atmosphiric pressure) [pa]
Ps = 189000; %presure of constant presure source [pa]

C=(ps*A^2*R^2*T^2)/(V^2*.5*(1+F*(L/D)+k));

Vc= .000418; %Volume of cylender in m^3
RPM = 1500; %engine speed

tstep = .000005; %Time step [s]
trun = 1; %Time of run [s]
taray = [0:tstep:trun];

i=1;
PtArray(1)= Pt;
PcArray(1)= Pa;
PowerArray(1)= 0;
mc(i)=(Pa*(Vc/Rc))/(R*T);
Vcc=Vc/Rc;

while(i<length(taray))
    i=i+1
    Phi = rem(taray(i)*RPM*2*pi/60, 2*pi);
    if(Phi<(.5*pi))
        Pt=PtArray(i-1);
        Vcc=(Vc/Rc)+((Vc-(Vc/Rc))/(pi/2))*Phi;
        Pc=mc(i-1)*R*T/Vcc;
        Roh=mc(i-1)/Vcc;
        Cc=(Roh*A^2*R^2*T^2)/(Vcc^2*.5*(1+F*(L/D)+k));
        PcDt=0;
        if(Pt>Pc)
            PcDt = sqrt(Cc*(Pt-Pc));
```

```
        end
        dm=PcDt*Vcc/(R*T);
        mc(i)=mc(i-1)+dm*tstep;
        Pc=Pc+PcDt*tstep;
        PtArray(i)= PtArray(i-1)-(dm*tstep)*R*T/V;
        PcArray(i)= mc(i)*R*T/Vcc;
    elseif(Phi>(1.5*pi))
        PtDt = sqrt(C*(Ps-Pt));
        Pt=Pt+PtDt*tstep;
        PtArray(i)= Pt;
        PcArray(i)= Pa;
        mc(i)=(Pa*(Vc/Rc))/(R*T);
    else
        PtArray(i)= PtArray(i-1);
        PcArray(i)= Pa;
        mc(i)=(Pa*(Vc/Rc))/(R*T);
    end
end

densitycool=PtArray/Pa;
densityhot= (PtArray/Pa).^(0.714);
plot(taray, densitycool,taray, densityhot);
title('Temperature Effect on Intermittent Pressure Model');
xlabel('time[s]');
ylabel('Dimensionless density [P/Pa]')
```

A.3 Fin Cooling Model

This is the code used to create the model described in section 3.5 it outputs the plot shown in Fig. 3-9.

```
%thermo code should output one results, heat transfer as a function of fin
%leangth starting at zero for no heat fins
```

```
L = .5; %Leangth of capacitor in [m]
V = .002; %volume of capacitor in [m^3]
A = V/L; %cross sectional area of capacitor
r = sqrt(A/3.14); %radius of capacitor [m]
D = 2*r; %diameter of capacitor [m]
Vdot = .0067; %volume flow rate of air [m^3/s]
Vin = Vdot/A; %velocity of air [m/s]
R = 287; %gass costant for ait
Kair = 0.027; %conductivity of air in w/mk
Kal = 237; %conductivity of aluminium in w/mk
uair = 2*10^(-5); %viscosity of air
Cpair  = 1000; %specific heat of air
Tin = 380; %tempriture of the air coming out of the compressor [k]
Tatm = 300; %atmospheric temp [k]
Pr= Cpair * uair/Kair; %prandlt number of air
Vmove = 3; %velocity of tractor/wind speed in [m/s]
tf=.01; %Thikness of fins in [m]
tc=.02; %thikness of capacitor [m]
g=tf; %gap between fins [m]
l=[0:.001:.2]; %leangth of fins in [m]
SAtot= 3.14*D*L; %total outer surface area of capacitor [m^2]
n= floor(3.14*D/(tf+g)); %number of fins
SAcap=SAtot-(n*L*tf); %Surface area not finned capacitor [m^2]

%calculating h of air flowing through the capacitor
Pin = 1.8*10^5; %pressure of air inside cap
pin = Pin/(R*Tin); %density of air inside [kg/m^3]
Re = pin*Vin*D/uair; %reynolds number of air flowing through the pipe

if(Re<=2300)
    Nu=3.66+(0.065*(D/L)*Re*Pr)/(1+.04*((D/L)*Re*Pr)^(2/3));
end
if(Re>2300)
    f=(0.79*log(Re)-1.64)^(-2);
    Nu=((f/8)*(Re-1000)*Pr)/(1+12.7*((f/8)^(.5))*((Pr^(2/3))-1));
```

```
end

hin= Nu*Kair/D; %convective heat transfer coefficient for air inside capacitor

%calculating h of air flowing around the capacitor, treat it as laminar
%flow over a plate
Pout = 1*10^5; %pressure of air outside
pout = Pout/(R*Tatm); %density of air outside[kg/m^3]
Re = pout*Vmove*L/uair; %reynolds number of air flowing outside the pipe

Nu=0.664*(Re^(1/2))*(Pr^(1/3));
hout= Nu*Kair/L; %convective heat transfer coefficient for air outside capacitor

%thermal resistance calcularion for no fins
Rair = 1/(hin*SAtot); %resistance of air inside [k/w]
Rwall= tc/(Kal*SAtot); %resistance of wall [k/w]
Rtube=1/(hout*SAtot); %resistance of air outside [k/w]

Rtot=Rair+Rwall+Rtube;
Qbase=(Tin-Tatm)/Rtot;

%thermal resistance with fins
%assume dt between capacitor outside and
Ac=L*tf;
m=sqrt(hout*pout/(Kal*Ac));
M=sqrt(hout*pout*Kal*Ac);

Rin=Rair+Rwall;

i=1;
while(i<(length(l)+1))
    Rtube=1/(hout*SAcap);
    Rtot=Rair+Rwall+Rtube;
    Qminnor=(Tin-Tatm)/Rtot;

    Lfin=l(i);
```

```
    Tb=((n*M*tanh(m*Lfin)*Rin*Tatm+Tin)/(1+n*M*tanh(m*Lfin)*Rin))
    Qfin(i)=n*(Tb-Tatm)*M*tanh(m*Lfin);
    Qtot(i)=Qminnor+Qfin(i);
    Tdrop(i)= Qtot(i)/(Vdot*pout*Cpair);
    i=i+1;
end

a1 = subplot(2,1,1);
plot(l, Qtot);
ylabel('Power Dicipated [W]');
title('Heat Transfir in 0.5m long 2L Capacitor');
xlabel('Fin Leangth [m]');
a2 = subplot(2,1,2);
plot(l, Tdrop);
ylabel('Tempriture Drop [C]');
xlabel('Fin Leangth [m]');
```

Appendix B

Experiment Data

This appendix contains all the raw data from the experiment

Naturally aspirated Baseline Data

Max power cold: 7200W
Max power hot: 7200 W
Ambient air temperature: 41 F / 5 C
Ambient humidity: 61%
Ambient Density: 1.27 kg/m³
Speed Range: 3650-3750 RPM

Load (w)	Engine intake pressure (psi)	Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	-0.17	0.80	50	10.00	1.22	-4.19
740	-0.17	0.83	48	8.89	1.22	-3.81
1560	-0.17	0.87	47	8.33	1.22	-3.62
2270	-0.17	0.91	47	8.33	1.22	-3.62
3180	-0.17	0.97	46	7.78	1.23	-3.43
3800	-0.17	1.04	46	7.78	1.23	-3.43
4590	-0.17	1.09	47	8.33	1.22	-3.62
5290	-0.17	1.15	48	8.89	1.22	-3.81
6090	-0.17	1.25	41	5.00	1.24	-2.47
6780	-0.17	1.39	49	9.44	1.22	-4.00

Turbocharged Test 0.9L capacitor

Max power cold: 6200 W
Max power hot: 7100 W
Ambient air temperature: 52 F / 11 C
Ambient humidity: 94%
Ambient Density: 1.23 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	1.16	1.24	0.97	1.01	1.2	0.99	4.77	4.83	4.8	79	26.11	1.24	1.17
740	1.41	1.48	1.26	1.28	1.445	1.27	5.03	5.06	5.045	83	28.33	1.26	2.24
1540	1.84	1.94	1.68	1.76	1.89	1.72	5.38	5.44	5.41	86	30.00	1.29	4.58
2250	2.38	2.48	2.25	2.31	2.43	2.28	5.84	5.95	5.895	90	32.22	1.32	7.40
3040	3.11	3.19	2.92	3.01	3.15	2.965	6.49	6.57	6.53	96	35.56	1.36	10.58
3740	3.84	3.94	3.63	3.79	3.89	3.71	7.04	7.22	7.13	103	39.44	1.40	13.86
4570	5.33	5.68	5.25	5.71	5.505	5.48	8.2	8.4	8.3	108	42.22	1.52	23.82
5250	6.79	6.94	6.5	6.6	6.865	6.55	9.76	9.4	9.58	120	48.89	1.57	27.75

Turbocharged Test 1.3L capacitor

Max power cold: 7800 W
Max power hot: 7000 W
Ambient air temperature: 63 F / 17 C
Ambient humidity: 20%
Ambient Density: 1.21 kg/m^3
Speed Range: 3650-3750 RPM

Load (W)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	0.99	1.06	0.79	0.83	1.025	0.81	4.73	4.78	4.755	83	28.33	1.22	0.90
740	1.45	1.56	1.25	1.31	1.505	1.28	5.01	5.13	5.07	86	30.00	1.25	3.42
1560	2.03	2.12	1.86	1.94	2.075	1.9	5.61	5.71	5.66	89	31.67	1.29	6.90
2280	2.8	2.89	2.59	2.67	2.845	2.63	6.19	6.32	6.255	94	34.44	1.34	10.65
3190	3.38	3.58	3.18	3.3	3.48	3.24	6.62	6.81	6.715	101	38.33	1.37	13.15
3800	4.07	4.25	3.72	4	4.16	3.86	7.3	7.43	7.365	106	41.11	1.40	16.07
4580	5.1	5.39	4.9	5.04	5.245	4.97	8.06	8.2	8.13	120	48.89	1.45	20.11
5270	6.8	6.93	6.51	6.65	6.865	6.58	9.43	9.58	9.505	130	54.44	1.55	27.84

Turbocharged Test 1.6L capacitor

Max power cold: 8200 W
Max power hot: 7600 W
Ambient air temperature: 39 F / 4 C
Ambient humidity: 65%
Ambient Density: 1.27 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	2.36	2.42	2.05	2.13	2.39	2.09	6.04	6.12	6.08	56	13.33	1.39	9.62
740	2.82	2.87	2.51	2.56	2.845	2.535	6.35	6.44	6.395	61	16.11	1.42	11.48
1540	3.22	3.25	2.95	2.96	3.235	2.955	6.72	6.81	6.765	66	18.89	1.44	13.14
2260	3.72	3.78	3.49	3.6	3.75	3.545	7.26	7.35	7.305	70	21.11	1.47	16.08
3040	4.4	4.5	4.14	4.3	4.45	4.22	7.75	7.89	7.82	75	23.89	1.52	19.30
3750	5.05	5.22	4.79	4.9	5.135	4.845	8.13	8.23	8.18	83	28.33	1.54	21.46
4560	5.54	5.72	5.31	5.42	5.63	5.365	8.69	8.88	8.785	89	31.67	1.57	23.36
5260	6.58	6.71	6.34	6.55	6.645	6.445	9.54	9.68	9.61	99	37.22	1.62	27.74
6070	7.75	8.06	7.56	7.62	7.905	7.59	10.44	10.65	10.545	110	43.33	1.68	32.12
6776	9.42	9.61	9.22	9.56	9.515	9.39	11.8	12.2	12	120	48.89	1.78	40.41

Turbocharged Test 2L capacitor

Max power cold: 8600W
Max power hot: 7600 W
Ambient air temperature: 60 F / 16 C
Ambient humidity: 30%
Ambient Density: 1.21 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	1.34	1.41	1.15	1.32	1.375	1.235	5.24	5.3	5.27	81	27.22	1.26	4.08
740	1.95	2.05	1.67	1.81	2	1.74	6.68	5.74	6.21	87	30.56	1.29	6.24
1560	2.65	2.73	2.43	2.52	2.69	2.475	6.28	6.37	6.325	90	32.22	1.34	10.44
2270	3.4	3.56	3.2	3.31	3.48	3.255	6.91	7	6.955	95	35.00	1.39	14.48
3080	4.25	4.4	4.05	4.15	4.325	4.1	7.38	7.57	7.475	102	38.89	1.43	18.43
3800	4.83	4.99	4.6	4.77	4.91	4.685	7.98	8.06	8.02	108	42.22	1.46	20.86
4580	5.58	5.77	5.34	5.4	5.675	5.37	8.66	8.82	8.74	119	48.33	1.49	22.79
5280	6.78	6.9	6.61	6.68	6.84	6.645	9.68	9.74	9.71	129	53.89	1.55	28.45
6070	8.42	8.78	8.2	8.33	8.6	8.265	10.91	11.11	11.01	140	60.00	1.64	35.75

Turbocharged Test 2.3L capacitor

Max power cold: 9000W
Max power hot: 8100 W
Ambient air temperature: 63 F / 17 C
Ambient humidity: 20%
Ambient Density: 1.21 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	1.2	1.32	1.04	1.12	1.26	1.08	5.18	5.27	5.225	86	30.00	1.24	2.11
740	1.8	2.03	1.66	1.76	1.915	1.71	5.62	5.74	5.68	90	32.22	1.28	5.47
1580	2.68	2.8	2.45	2.54	2.74	2.495	6.36	6.14	6.25	93	33.89	1.33	9.97
2290	3.2	3.29	2.96	3.01	3.245	2.985	6.68	6.79	6.735	98	36.67	1.36	12.13
3100	3.84	4.05	3.67	3.75	3.945	3.71	7.29	7.4	7.345	102	38.89	1.40	15.94
3800	4.64	4.9	4.52	4.57	4.77	4.545	7.9	8.61	8.255	107	41.67	1.45	20.19
4580	5.75	5.96	5.57	5.7	5.855	5.635	8.87	9.05	8.96	115	46.11	1.52	25.30
5270	6.98	7.23	6.78	6.88	7.105	6.83	9.81	9.97	9.89	125	51.67	1.58	30.46
6080	8.66	8.86	8.18	8.32	8.76	8.25	11.25	11.4	11.325	136	57.78	1.65	36.57

Turbocharged Test 2.6L capacitor

Max power cold: 8800W
Max power hot: 8400 W
Ambient air temperature: 41 F / 5 C
Ambient humidity: 61%
Ambient Density: 1.27 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	2.15	2.21	1.86	1.91	2.18	1.885	6.01	6.07	6.04	57	13.89	1.37	8.92
740	2.53	2.54	2.26	2.34	2.535	2.3	6.27	6.4	6.335	62	16.67	1.39	10.61
1500	3.07	3.11	2.84	2.88	3.09	2.86	6.89	6.92	6.905	64	17.78	1.43	13.85
2250	3.69	3.7	3.4	3.47	3.695	3.435	7.21	7.4	7.305	69	20.56	1.47	16.51
3060	4.39	4.46	4.15	4.22	4.425	4.185	7.8	8.05	7.925	74	23.33	1.52	20.24
3780	5.15	5.36	4.92	4.99	5.255	4.955	8.49	8.5	8.495	80	26.67	1.56	23.81
4570	5.99	6.08	5.67	5.73	6.035	5.7	9.22	9.27	9.245	85	29.44	1.60	27.37
5280	6.87	6.95	6.57	6.7	6.91	6.635	9.93	10.03	9.98	90	32.22	1.66	32.05
6090	8.08	8.13	7.7	7.72	8.105	7.71	10.75	10.88	10.815	96	35.56	1.73	37.26
6800	9.08	9.21	8.7	8.8	9.145	8.75	11.84	11.71	11.775	104	40.00	1.78	41.65

Turbocharged Test 2.9L capacitor

Max power cold: 9300W
Max power hot: 8500 W
Ambient air temperature: 52 F / 11 C
Ambient humidity: 94%
Ambient Density: 1.26 kg/m^3
Speed Range: 3650-3750 RPM

Load (w)	Compressor outlet pressure low range (psi)	Compressor outlet pressure high range(psi)	Engine intake pressure low range (psi)	Engine intake pressure high range(psi)	Average Compressor outlet pressure (psi)	Average Engine intake pressure (psi)	Engine exhaust pressure low range (psi)	Engine exhaust pressure high range (psi)	Average Engine exhaust pressure (psi)	intake temp(f)	intake temp(C)	Density (kg/m^3)	density gain (%)
7	2.48	2.51	2.07	2.14	2.495	2.105	6.11	6.24	6.175	72	22.22	1.35	9.88
740	2.78	2.86	2.44	2.52	2.82	2.48	6.29	6.5	6.395	76	24.44	1.37	11.52
1540	3.35	3.45	3.06	3.12	3.4	3.09	6.88	7.03	6.955	81	27.22	1.41	14.46
2250	3.76	3.88	3.43	3.55	3.82	3.49	7.17	7.25	7.21	86	30.00	1.43	15.99
3030	4.5	4.56	4.16	4.24	4.53	4.2	7.71	7.85	7.78	90	32.22	1.47	19.69
3740	5.03	5.14	4.77	4.85	5.085	4.81	8.45	8.55	8.5	94	34.44	1.51	22.70
4550	6.1	6.16	5.77	5.88	6.13	5.825	9.24	9.32	9.28	100	37.78	1.57	27.76
5250	7.36	7.44	6.97	7.04	7.4	7.005	10.04	10.14	10.09	109	42.78	1.64	33.04
6060	8.43	8.66	8.09	8.28	8.545	8.185	10.98	11.15	11.065	123	50.56	1.68	36.96
6770	10.05	10.38	9.5	9.73	10.215	9.615	12.33	12.66	12.495	135	57.22	1.75	42.65

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