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Automotive Application of Turbo-Expansion

A Thesis Presented

by

Jean Christian Brutus

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The Graduate School

in Partial Fulfillment of the

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Abstract of the Thesis

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This research project focuses on a charge-air cooling concept, Turbo-Expansion, which offers promise to bring the application of Variable Geometry Turbine (VGT) technology toward its original aim of improving gasoline internal combustion (IC) engine performance and efficiency.

In a conventional turbocharger, as the pressure increases, the exhaust gas bypasses the turbine through a wastegate. This represents a loss of energy since the exhaust gas goes directly to the ambient surroundings without performing any useful work within the system. The Turbo-Expansion concept can eliminate the need for the exhaust bypassing system by using a VGT as the exhaust turbine. It can also eliminate the turbo lag that exists in regular turbochargers by functioning as a "small turbo" at low engine speed and a "big turbo" at high engine speed.

In addition, due to reduced operating temperatures the application of the Turbo-Expansion cooling system to the gasoline engine will not lead to knocking, an abnormal and engine-damaging combustion phenomenon that plagues naturally aspirated and conventionally turbocharged engines. Normal operation of gasoline engines produces higher exhaust gas compared to Diesel engines where turbocharger application is common. These higher temperatures lead to turbocharger component failures, which can be eliminated by the application of Turbo-Expansion technology.

Engine performance modeling software, Gamma Technologies' GT-Suite, was previously used to prove the theoretical benefits of the new concept. The results of the investigation show that Turbo-Expansion will increase the power output of current gasoline engines. It will also save fuel costs and lower emissions by lowering intake air-fuel mixture temperature below ambient. These changes will not require the use of advanced combustion technology, high-grade fuel, or advanced materials. Fuel economy will result from engine downsizing and higher compression ratios. Reliability and durability will increase as well, due to lower engine speeds.

Using expansion-cooling to attain these benefits has been questioned since it

opposes the main idea of turbocharging, which is based on the compression of inlet air. In the present effort, an analysis of the components involved is presented along with theoretical calculations, which prove that the concept presents a certain potential. Suggestions are made on critical requirements concerning the effectiveness of the intercooler and the control system needed to realize that potential.

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Preface

Born and raised in Haiti, I moved to Stony Brook University after graduating from High School in Port-au-Prince. My extra curricular activities (go-kart / motorcycle design and fabrication) both in Haiti and at Stony Brook led to my interest in pursuing an advanced degree conducting research on an automotive related project.

This study, begun in August 2006 and completed in May 2010, was carried out at Stony Brook University and neighboring facilities. This thesis by no means symbolizes the end of this research project but rather, a detailed summary of all the effort devoted to this particular subject to date and the beginning of a one-year reevaluation period.

I would first like to thank my parents who have given me the opportunity to travel to the United States to further my education and go beyond many limitations I would have faced in Haiti. Their support and that of my entire family has given me the confidence I needed to succeed.

I also wish to express my sincere appreciation to my advisor, Dr. Lin-Shu Wang, for intrusting me with decisions related to this project and for his valuable support in the course of countless informal discussions. He even provided space in his garage at home for storage and testing purposes before the Mechanical Engineering Department and the College of Engineering and Applied Sciences got together to create some space on campus for the laboratory. Professor Wang's former students, Shiyou Yang, Antonio Cacavale and Travis Wentz have brought important contributions to the realization of this work and are worth mentioning here as well.

It is very important to mention that my educational path as been shaped by the influence of different groups and programs at Stony Brook University. Such programs include but are not limited to the Collegiate Science and Technology Entry Program (CSTEP), the Minority Access to Research Careers (MARC) Fellowship, the Alliance for Graduate Education and the Professoriate (AGEP) and the W. Burghardt Turner Fellowship. In CSTEP, I found myself surrounded by people who made me feel like I was home. The moral and financial support received during the early stages of my undergraduate career was essential to my introduction to research possibilities after graduation. The MARC program sponsored my junior and senior years of undergraduate education and also funded this research project. AGEP guided me through the graduate experience and supported me when I needed it the most. Through their Summer Research Institute (SRI), I had the pleasure to meet Maleshia Jones from the University of Maryland Baltimore County, who under my guidance, performed intercooler testing at our laboratory. Finally, the Turner Fellowship program funded two years of my graduate research and education. My activities within Stony Brook Motorsports also contributed to a major part of my formation as an engineer and I am grateful for the support received from the team.

During the one-year reevaluation period mentioned above, I will assess the possibility of pursuing an advanced degree in Automotive Engineering while working on

U.S. army vehicle related projects at the U.S. Army Evaluation Center (AEC) located at Aberdeen Proving Ground (APG) in Maryland. APG is one of the largest and oldest United States Army research/testing facilities.

I am thankful for being able to complete my master's degree at Stony Brook University with the support of the MARC and Turner Fellowship programs, but I believe it may be in my best interest to investigate automotive programs at other institutions since the equipment and facilities required to complete my research project on time are not available at Stony Brook University. All engine tests had thus far been performed off campus at the Marcovicci and Wenz Racing facility where critical data was collected. None of this would have been possible without the use of their equipment at no charge.

I will always continue to assume my responsibilities as a leader, educator and researcher here in the United State as per the long-range goal of the Turner and MARC programs. My education does not stop here.

Introduction

Turbocharging is very common in the automobile industry. However, its application in spark-ignition (SI) engines is limited. Limiting factors include fuel consumption and emissions. A more important factor is the knocking phenomenon, which is the pre-detonation of the air fuel mixture, which damages engine components. High temperature due to the compression of air through the compressor is at the origin of all these problems. In order to decrease charged air temperature, experiments such has using water vapor or ethanol in the combustion chamber have been performed [1]. Although these methods have somewhat been successful, eliminating the need of adding different products to the combustion chamber seems more convenient. A more permanent design change, which would facilitate the decrease in charge air temperature, is the addition of an expansion device between the intercooler and the intake manifold. In the work presented in this thesis, a readily available turbine is used as the expander.

Expansion-cooling was introduced in the mid 1900’s. From then until today, expansion cooling for turbocharged internal combustion engines is a concept that has been approached by multiple research teams around the world, but no marketable prototype has been presented for the automotive industry. The major expected benefits are fuel economy, performance enhancement and emission reduction. Although the application of the concept is not seen in the automobile industry today, improvement in electronics, control systems and intercooling technologies since the mid 1900’s may help make this idea a reality to help today’s problems.

Two of today’s main crises having a worldwide impact are the constant concern of running out of oil, and pollution control. The concern of running out of oil or affecting Mother Nature never seems to improve. One might then wonder if it is time to talk about turbo systems when most people see such systems as ways to get more power while burning more fuel and emitting more toxic gases. The work presented in this thesis addresses the performance of internal combustion engines when subjected to high boost pressures and attempts to clarify any misconception. The expansion concept is explored in detail and its limitations are presented based on various experiments and numerical computations.

1. Theory

1.1 Turbocharging

1.1.1 Basics

In a naturally aspired internal combustion engine, an air filter is connected to the inlet of the intake manifold. The filtered air is mixed with fuel at a predetermined air fuel ratio to be able to combust in the cylinder and provide the power in the form of a rotating shaft. The products of the combustion, exhaust gases, escape from the cylinder through the exhaust system and straight to the atmosphere (cf. Fig. 1-1). Energy from the hot gases that could be used as a means to produce work is wasted. The key characteristic of a turbocharger is the use of this energy.

The idea behind turbocharging is the increase of inlet air density in internal combustion engines. If the mass of air is increased in the combustion chamber, more fuel can be added to maintain the same air fuel ratio, thus providing more power. This is the most basic advantage of turbocharging. Before proceeding with the diver's benefits of turbocharging an engine, it is necessary to understand how turbochargers work.

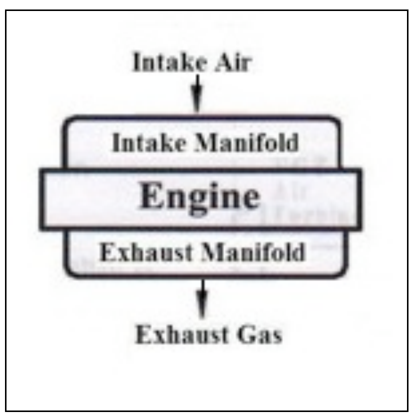

Fig. 1-1. Naturally Aspired (NA) engine setup

The basic turbocharging system is shown in Fig. 1-2. A turbocharger can be divided in three main sections: turbine, bearing block and compressor. The exhaust gases energy is recovered by expanding through the turbine. In other words, the exhaust gases from the engine power the turbine. A shaft supported through the bearing block is connected to the turbine blades. In general, the shaft and the turbine blades are fabricated in one piece. The bearings, seals and compressor blades are mounted on the shaft to form the rotating assembly. The rotational energy gained from the expansion is then recovered at the compressor, which pulls air from ambient surroundings and forces it into the intake manifold at a higher pressure than atmospheric. An air filter connected to the compressor inlet keeps particles from entering the intake system and damaging the components downstream. The air filter is exposed to ambient conditions. The air is brought to a higher pressure after going through the compressor but also, a higher temperature. To lower the air temperature before it reaches the intake manifold, an intercooler is used to remove some of the heat added at the compression stage.

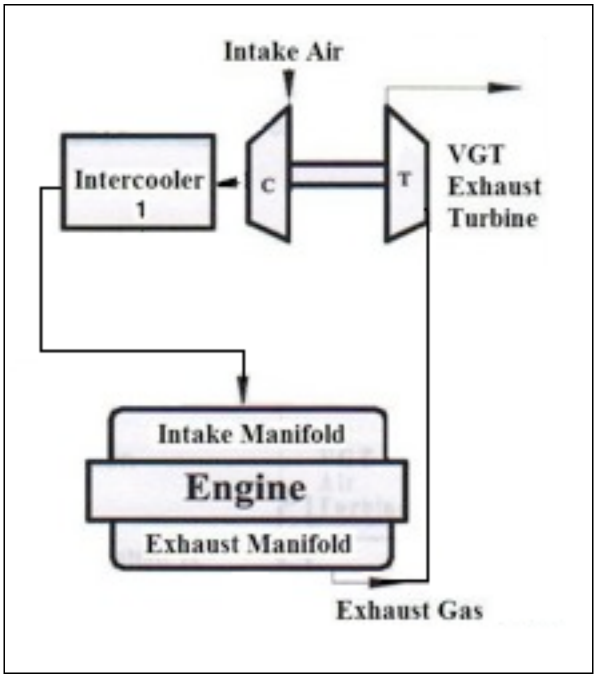

Fig. 1-2. Basic turbocharged engine setup

Notice that the temperature after the intercooler depends on the temperature of the cooling medium (i.e. air, water...). Using this technique, if the cooling agent is at ambient temperature or higher, the charge air cannot be cooled to temperatures below ambient unless cooler mediums are used. Different types of intercoolers are used and will be discussed in the intercooler section.

1.1.2 Benefits

There are different ways of looking at the benefits of turbocharging. Two benefits were already mentioned in previous sections. First, the exhaust energy is used to do useful work and increase air density through compression which in turn increases power. A simple way of summarizing these benefits is to look at the power equation (cf. Eq. 1-1) [2].

$$P = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i} (F/A)}{2} \text{ [hp]} \tag{1-1}$$

The parameters affecting the power generated are the fuel conversion efficiency η_f, the volumetric efficiency η_v, the speed N [rpm], the engine displacement V_d [ft^3], the fuel heating value Q_{HV} [hp/lb], the intake manifold air density $\rho_{a,i}$ [lbm/ft^3], and the fuel air ratio F/A.

As can be seen from Eq. 1-1, the power is increased proportionally with air density. The power generated by an engine can be increased proportionally with any of the parameters in the numerator. However, there are limitations.

Improving power is great but what is the impact of a turbocharger on the torque vs. rpm curve. In Heywood [2], it is experimentally proven that a flatter torque curve is obtained from a turbocharged system compared to the same naturally aspired system. Although the theoretical reasons are not derived, this could be understood by the improvement in mass flow of air, which accompanies the high pressure supplied by the turbocharger. Heywood does derive the torque equation (cf. Eq. 1-2).

$$T = \frac{\eta_f \eta_v V_d Q_{HV} \rho_{a,i} (F/A)}{4\pi} \text{ [lbf.ft]} \tag{1-2}$$

As described previously, η_v is the volumetric efficiency, which is, as in Eq. 1-3, the volume flow rate of the system divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N}. \tag{1-3}$$

In general, air intake systems are optimized for either low or high speeds unless a variable intake design is implemented. If designed for low speeds, the restriction in the flow decreases the torque at high speed as volumetric efficiency decreases with the mass flow of air. If designed for high speed, poor mass flow will be obtained at low speed, having the inverse impact. Having the mass flow controlled by the amount of boost flattens the curve. Engine manufacturers aim for a flat torque curve. Many engine enthusiasts wonder why this is very important for engine or vehicle performance and how it is obtainable. While it is impossible to generate a perfectly flat torque curve from 0 RPM to red line, designing an engine that has a fairly flat torque curve for a given broad range of engine speed is favorable for various reasons.

Eq. 1-4 shows the dependency of Horse Power on torque [2],

$$P = \frac{N \times T}{5250} \text{ [hp]}. \tag{1.4}$$

Therefore, power is directly proportional to torque. With a flat torque curve, and increasing engine speed, the power will increase as well. The broader the range, the faster a vehicle can accelerate because maximum torque is available at any speed within the range. The gear ratios can also be increased to provide more ground speed at high RPM within that range. If the maximum torque value is known, the power delivery would be predictable for a broad RPM range and would also result in more peak power.

When torque curves have a peak, this limits the engine applications or requires a large number of gear ratios to keep the engine performance at it's maximum. The problem with torque curves having a sharp peak is that torque will only be available at a specific RPM point. Low RPM for day-to-day drivers and short distance racers, high RPM for long distance racers. A sharp peak curve is produced vs. a flat curve because of design limitations such as intake system efficiency being optimal at either high or low charged air velocity through the ports, timing, carburetion, and exhaust limitations. Modification or on demand variations of these components can help generate flatter curves. Flat curves are essential for hauling and towing applications since performance can be appreciated at a variety of engine speeds. Boosting a system flattens the curve as explained above and will permit to open the throttle at any given RPM within the range without having to down shift. This advantage would be helpful when cornering where more torque is usually required.

1.1.3 Limitations

Fuel consumption efficiency can obviously be improved by increasing the compression ratio, r_c, of the engine since the second term of Eq. 1.5 will decrease [2].

$$\eta_{f,i} = 1 - \frac{1}{r_c^{\gamma-1}}. \tag{1-5}$$

$\Upsilon = c_p/c_v$ is the ratio of specific heat at constant pressure over that at constant volume. This requires piston or cylinder head modifications but is still a possibility. However, temperature also increases with higher compression ratios leading to knocking.

Volumetric efficiency, η_v, is a representation of how well the engine breathes. It is highly dependant on the manifold and port design. This efficiency is closely related to engine RPM, *N*. From experiments, a volumetric efficiency increase from lower RPM to peak torque RPM can be observed while a decrease from peak torque RPM forward is evident [2]. Internal design modifications are required and engine manufacturers try to optimize the volumetric efficiency in the detail design stages of their design process.

The fuel heating value, Q_{HV}, varies based on the type of fuel used. If no additional ingredients are to be added or fuel economy is a concern, no changes are possible in this area. This would be the same constraint seen when looking at the F/A ratio.

The intake manifold air density, $\rho_{a,i}$, will increase when air is compressed before entering the combustion chamber. However, density is not only dependent on pressure but also temperature [4]. The turbocharger compressor increases pressure therefore increases density. The higher temperature that results from the compression however decreases density as shown in Eq. 1-6.

$$\rho_{a,i} = 2.7 \frac{P_{a,i}}{T_{a,i}} \quad [\text{lb/ft}^3] \tag{1-6}$$

$P_{a,i}$ [psia] and $T_{a,i}$ [R] are the intake manifold pressure and temperature respectively.

1.2 Turbo-Expansion idea

A proposed solution to these limitations is the turbo-cool concept. Using expansion-cooling right before the intake manifold, the temperature of the charged air mixture can theoretically be brought to temperatures lower than ambient. As mentioned in the introduction, readily available turbines from turbo machinery manufacturers can be used as expanders. In a similar manner as described in the turbocharger basic section, an expander will extract work from the charged air by expanding it. This is not to be confused with the expansion of the exhaust gases described in the turbocharger section. As air expands, its temperature decreases. The energy extracted from the charged air at this point can be used to produce useful work. The main difference between the exhaust turbine and the expander (air turbine) is their purpose. The exhaust turbine purpose is to extract useful energy from the hot gases and as a consequence, the exhaust gas temperature decreases. The air turbine purpose is to decrease the charged air temperature and as a consequence, useful work is extracted.

For simplicity, a turbocharger can be installed in such a way that its turbine, used as an expander, will transfer the energy to the compressor attached to it as an assembly. Fig. 1-3 and 1-4 show two arrangements that emanate from this idea. In Fig. 1-3, the compressor attached to the expander is part of the intake system and will use the expansion energy to further compress the charged air. In Fig. 1-4, the compressor can be equipped with a valve that would regulate the flow of air from ambient and back to ambient.

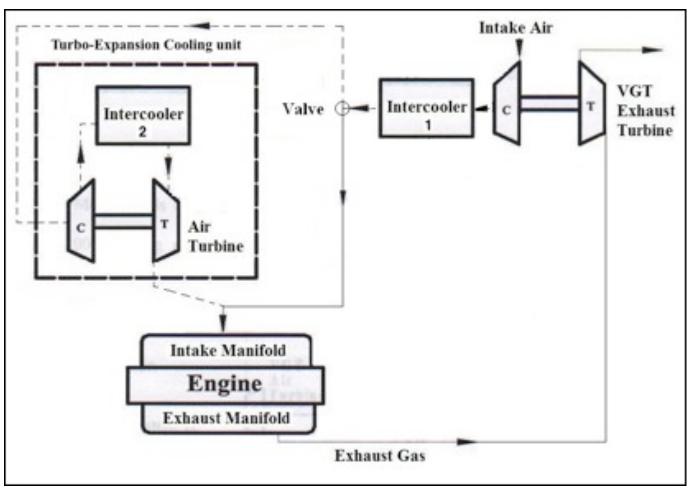

Fig. 1-3. Turbo-Expansion concept with double compression

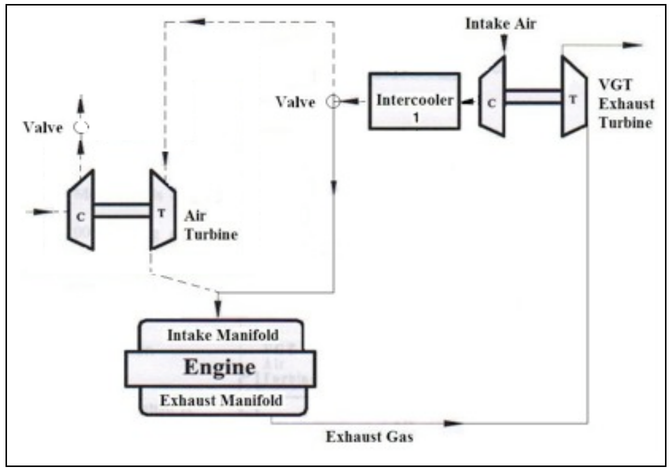

Fig. 1-4. Turbo-Expansion concept with single compression

If the second configuration is adopted, variation in the flow valve can regulate the expansion of the air by allowing more or less work to be extracted from it. Different types of turbines can also be used to regulate the expansion.

There are three common types of turbines, variable geometry, wastegated and free-floating turbines. Variable geometry turbines are also called VGT. They are equipped with a mechanism that adjusts the angle of the turbine vanes so that the torque transmitted to the turbo shaft is greater at low engine RPM. This mechanism can be controlled pneumatically or/and electronically. The wastegated turbine has an internal flap, a valve that opens when there is a need to bypass the turbine blades. This mechanism can also be controlled pneumatically or/and electronically. A free-floating turbine is not outfitted with any adjusting device and requires an external valve and additional piping.

For the expansion-cooling process, if a VGT is used as an expander, the vanes of the turbine will be adjusted to vary the expansion ratio. If a wastegated turbine is used, the wastegate opening will be adjusted to regulate how much of the charged air is expanded through the turbine. If a free-floating turbine is used, controlling the external valve coupled to the path that bypasses the expander blades will lead to the same end result. Theoretically, the charged air temperature can be reduced to temperatures below ambient using either of these setups. For an engine designer, this means that the compression ratio can be raised to higher values. For an engine already in use, this means that the boost pressure can be increased.

Expected outcome

a) Fuel efficiency

Increasing the fuel conversion efficiency can increase the power of a given engine. By the same token, less fuel could be burnt for the same work produced. This can be accomplished by increasing the compression ratio. The effect of the compression ratio on the fuel efficiency is clear when the later is expressed in terms of the compression ratio and the ratio of constant volume and constant pressure specific heat [2] as introduced in the turbocharging limitation section

If the engine is more efficient, it is evident that for the same amount of fuel burnt, the work or power produced will be greater. Unfortunately, although an increase in air density or compression ratio promises a certain increase in power, the charge air mixture is subjected to a rise in temperature as well. This was also introduced as a limiting factor for a turbocharging system. These parameters can be tuned to the limits of the knocking phenomenon. At certain high temperatures, gasoline will auto ignite possibly damaging the engine.

Most people believe that once a turbocharger is installed on a vehicle, the vehicle in question will burn more fuel. Here is an interesting thought: Given a certain engine overall efficiency, a precise amount of energy is needed for a task to be completed in a specific time. It should not matter if the engine is turbocharged or not. The energy needed will be provided by the same amount of fuel. More fuel is burnt when the additional power delivered by the turbocharger is used to accomplish the task faster. Without considering knock, less fuel will be used if the compression ratio is higher in a different engine since the fuel will be converted more efficiently [2].

The application of higher compression ratios results in engine downsizing (decrease in weight and specific volume), which results in vehicle downsizing. This is what makes the case for turbo application today. The lighter the vehicle, the less mass needs to be moved and the less fuel is burnt. If a small turbocharged SI engine can meet the power requirements of a bigger engine, fuel economy at part load will be better. At part load, the mechanical efficiency of a small engine is greater than that of a bigger engine [2].

Lastly, the racing enthusiast, or the drivers who regardless of the consequences still want to run their engines at high boost, simply add more fuel. It is understood that by adding more fuel, some of the heat is used to raise the temperature of the liquid fuel, therefore decreasing the overall temperature of the combustion chamber. However, for a regular driver, the additional fuel would only mean more money to be spent on a regular basis.

b) Emissions

In addition to reducing fuel costs, the Turbo-Expansion concept will help lessen pollutant formation. This will be possible for two reasons. Primarily, the objective of the concept is to reduce temperature of the charged air. As it will be shown in the subsequent paragraphs, pollutant production decreases with decreasing temperature. Secondly, adding more fuel to absorb some of the heat in the combustion chamber will no longer be necessary. A controlled fuel/air equivalence ratio results in reduced emissions since more pollutant is formed with a richer air fuel mixture.

When talking about emissions, the concerned products are carbon monoxide (CO), hydrocarbons (HC) nitric oxide (NO) and nitrogen dioxide (NO_2). NO and NO_2 are generally combined and called NO_x. According to John B. Heywood [2], these products are formed at high temperatures during the different phases of the engine cycle. For a four-stroke engine, the four phases would be compression, combustion, expansion and exhaust. Without elaborating on the details, one particular phase, the combustion phase, is of interest to this work. At this point of the engine's cycle, the in cylinder temperature is the highest.

During combustion, NO is formed around the spark plug, behind the flame propagating through the chamber, where the high-temperature burnt gas is present. The higher the temperature, the more NO is formed. Within that same region, CO is also formed when there is more fuel than oxygen as compared to the amount expected from a balanced chemical equation. Therefore, the richer the mixture, the more CO is produced has a result of the incomplete reaction which should have resulted in the production of CO_2 instead. Finally, HC production is observed at different phases but the main source is the extra fuel in the combustion chamber. As opposed to NO and CO, HC is formed away from the higher temperature region. This means around the periphery of the chamber if the spark plug is in the center. HC is seen on the cylinder walls and small volumes where the flame does not reach or is extinguished.

Ultimately, the expansion of the charged air before it reaches the combustion chamber will be the source of an overall drop in temperature across the engine cycle, which will result in emission control and render the necessity of adding fuel to the mixture obsolete. CO and NO production will drop with the temperature. A leaner fuel air mixture will control the formation of CO and HC.

c) Performance

The way turbochargers affect power generation was introduced in the turbocharging benefit section. It was shown that engine power was proportional to different factors including fuel conversion efficiency, η_f, inlet air density, $\rho_{a,i}$, and fuel air ratio, F/A.

The Turbo-Expansion concept will permit the increase of the compression ratio, r_c , which in turn will increase the fuel conversion efficiency as explained in the fuel efficiency section above. The concept will also allow higher boost pressures. As air is compressed, its density increases. The denser the inlet air, the more powerful the engine.

Also explained in the fuel efficiency section is the reason why an engine is some times ran with air fuel mixture richer than stoichiometric. Leaning the mixture will decrease the fuel air ratio, which in turn increases power. At the end, even the racing enthusiasts will be able to reduce their fuel consumption.

2. Turbo-expansion history

Now that the reader has a thorough understanding of the limitations of current turbocharging technology and the benefits of the Turbo-Expansion concept, it is important to emphasize that Turbo-Expansion is not a concept that was invented or discovered in recent years.

The first published text that introduces the idea of expanding internal combustion charged air is from 1959 [4]. In that publication, William R. Crooks evaluated various systems of combustion air conditioning, with a special highlight on expansion cooling of air by making use of the surplus of energy from the turbo-supercharging system. The supercharging system is similar to a turbocharging system except that the energy used to compress the charged air is taken from the engine shaft or external device and not an exhaust turbine. Supercharging will be addressed in later sections.

Since the early 50's, it was common knowledge that cooler air would increase maximum output of IC engines. The successful results presented in 1959 where related to the application of Turbo-Expansion to large, heavy duty, industrial engines using natural gas, natural gas and diesel oil or just diesel oil. The result evaluations where judged based on exhaust smoke and temperature, fuel consumption and knock. It is somewhat surprising that the concerns we have today were the same 50 years ago. However, Crooks seemed to believe that although air conditioning was not new technology in the late 50's, it had become a necessity based on changes in economic conditions, fuel prices and engine development.

The common methods of cooling combustion air back then were evaporative cooling, aftercooling and refrigeration cooling. Evaporative cooling involved spraying water mist in the air inlet but proved to be ineffective then. Aftercooling is similar to what we now call intercooling system. Further discussion on intercooler efficiency will be the main focus of later sections. Turbo-Expansion was then known as refrigeration cooling and the expected outcomes were higher specific loading and lower specific fuel consumption.

In 1961, Crooks filed a patent [5] with the US patent office. He then claimed to be the original inventor of a refrigerating concept using expansion-cooling. In his invention, the power generated by the expansion turbine was used to power a centrifugal Freon compressor. Freon was the refrigerating fluid used in the system. This was closely related to his work published in 1959 and addressed the issues of large, heavy duty, industrial engines using natural gas, natural gas and diesel oil or just diesel oil. His patent was issued in 1964.

In 1965, M. J. Helmich, from the same corporation as Crooks, published a paper covering the successful development and application of the design covered in Crooks' patent [6]. Basic laboratory development and results obtained from a field installation were presented. The results were collected over a period of ten years. It was then concluded that the study had proven the feasibility of Crooks' concept of Freon refrigeration using expansion cooling. The benefits of increase in power and decrease in fuel consumption were also proven. It is important to realize that the main objective of the work so far was not to reduce emissions but to increase power and lower fuel consumption. Additionally, the refrigeration system was so big that it had to be installed outside the building containing the engine. Therefore, an automotive application of this concept was not possible. A simplified concept using the idea of expansion resurfaced ten years later.

In 1975, Charles E. McInerney filed a patent with the US patent office [7]. In McInerney invention, there was no refrigeration cycle and it was the first attempt at a smaller setup that could eventually be applied to automotive. The power recovered from the expansion was used to power a fan blowing on the intercooler to maximize the heat rejection at that stage. McInerney claimed that recent exhaust pollution control devices were designed to reduce the production of carbon monoxide and hydrocarbons by increasing the operating temperature of the engine. This resulted in a more complete combustion of the fuel but produced more oxides of nitrogen. This agrees with the description given in the section exploring the benefits of Turbo-Expansion. The patent was issued and assigned in 1977 to the Garrett Corporation, which is the same company who manufactures the components used for the expansion-cooling research at the Stony Brook University laboratory. After contacting Garrett, now own by Honeywell, it was concluded that the concept was never pursued because the corporation had different priorities at that point. It was advised that the lack of electronics then, might have turned the corporation away from the idea. With the advances in electronics and controls today, this is less of an obstacle.

In 1991, Roy C. Meyer and S. M. Shahed presented a turbocooling system in which two-stage compression and intercooling is used [8]. Through the use of the Joule-Thompson effect, compressed and cooled charged air is expanded before reaching the intake manifold. The positive work extracted from the expansion is used in the first stage compression process. It was shown that the concept would be feasible using a basic thermodynamic model. More specifically, it was revealed that a minimum intercooler effectiveness of 55 % was required.

In 1992, Harry A. Cikanek and Vemulapalli D. N. Rao filed a patent with the US patent office [9]. Their invention is very similar to that of Meyer & Shahed. The one difference is that Cikanek & Rao made use of a refrigeration cycle. The compressor for the refrigerating fluid is driven by one of the turbines already in use by the charged air system. The refrigerating system's evaporating coils are place in close proximity or inside the intake manifold to further reduce the intake temperature. The goal of this invention was to reduce emission in diesel powered vehicles. The patent was issued in 1993 and assigned to the Ford Motor Corporation, which is the well-known American automotive company.

In 2003, Lotus engineers in England showed their interest in the concept. Lotus engineers, J. W. G. Turner, R. J. Pearson and M. D. Bassett partnered with J. Oscarsson from Opcon Autorotor AB in Sweden to perform an initial study on Turbo-Expansion [10]. Reducing inlet air temperature in order to increase compression ratio and go beyond the limits set by engine knock was the objective of these European engineers. On a bigger picture, fuel economy was their goal. They also focused on engine downsizing. The reduction in vehicle size and mass would also help achieve their goals. Running the downsized engines at higher brake mean effective pressure (BMEP) would also contribute since brake specific fuel consumption (BSFC) is improved due to reduction of throttling loss. The authors pointed out that the basic philosophy of Turbo-Expansion was similar to an aircraft air conditioning system. In addition to simulations, they collected data based on a test rig equipped with a positive displacement screw expander. It was concluded that Turbo-Expansion could improve power and fuel economy.

In 2004, the Lotus group partnered with the Wolfson School of Mechanical and Manufacturing Engineering, University of Loughbotough in the UK to develop a first law of thermodynamic model used to characterize the performance of the Turbo-Expansion concept [11]. The model was verified against experimental testing data collected by

Turner et al. [10] and the effects of the compressor, expander and intercooler effectiveness variation were quantified. It was shown that the intercooler effectiveness was the most critical parameter. The authors' concluding remarks noted that the constant mass flow rate thermodynamic analysis of the system was valuable as a test for the viability of the concept. The next step would be the engine simulation at constant BMEP. The same year, the Lotus group published a review of various technologies for fuel economy improvement in gasoline engines and discussed how they interacted [12]. They then introduced the NOMAD development engine that was being built to investigate the concept and other technologies.

In 2005, the Lotus group presented their disappointing dynamometer testing results of the NOMAD engine [13]. Honeywell Turbo Technologies had also provided assistance in the realization of the NOMAD charging system. It was concluded that the efficiency of the components used on the NOMAD engine were not able to deliver significant temperature reduction. The reductions were so minimal that even convection heat transfer from the engine compartment could offset any benefit. It was proposed that research concentrating on the improvement of expanders and intercoolers could help make the Turbo-Expansion a reality. Notice that the water-cooled intercooler used was set to maximum performance, which corresponds to an effectiveness of 96%.

Although the results were unsatisfactory, the model developed led to some interest in pursuing a single-cylinder study [14]. The capability of Turbo-Expansion to extend the knock limit was proven based on experimental data. It is important to understand that although successful results were obtained with the single cylinder engine, externally conditioned air was used with the plenum charge air temperature as the control variable.

Still in 2005, another group in the UK presented successful modeling results [15]. Increase power levels of a downsized engine using Turbo-Expansion were presented as results of a case study. At the time of publication, a rig-based experimental program had begun. Completion of the rig, testing and data analysis would be the basis of future publications. Chris Whelan, the main author, claims that in the mid-1980's, he carried out experimental work as part of a Formula 1 racing engine program. Rig test were performed to establish feasibility and engine designs prepared. However, the system was never vehicle tested and the work was never published. Although apparently successful, even today, he prefers to keep the Formula 1 information confidential. He plans however to publish the results of his most recent activities this year (8^{th} IMechE Turbocharger Conference in London, May 2010). Some of these activities include the successful design, fabrication and test of an expander.

In 2006, Stony Brook University brings some advancement to expansion-cooling with a first publication by Dr. Lin-Shu Wang and Shiyou Yang [16]. This work focused on the simulation of an air conditioning system for a turbocharged engine, using a variable geometry turbine as the exhaust turbine. Part of the compressed air is expanded and used as a heat exchanger cooling medium in a second intercooler before being compressed again to use the expansion energy and finally be rejected to the atmosphere. The other part of the charged air exchanges heat with ambient at a primary intercooler and with the expanded air at the secondary intercooler before reaching the intake manifold. The expander, which is a turbocharger turbine along with its compressor and the secondary intercooler, form the turbo-cool unit. The results of the simulation showed that the VGT turbocharger and turbo-cool are indispensable to each other and that the resulting temperatures could be below ambient. Improved break mean effective pressure and lower break specific fuel consumption were also reported.

The group Lotus and the University of Loughborough also published in 2006, the work performed as a continuation of their publication in 2004 [11, 17]. The results of their improved thermodynamic model, new engine cycle simulations, combustion and knock modeling showed an improvement in engine break specific fuel consumption based on Turbo-Expansion. Notice that even if a constant value of 0.8 was used for the effectiveness or efficiency of all components, the benefits were greatly reduced due to heat transfer between the expander and the manifold. A second explanation was based on the amount of residual gases left in the cylinder after the exhaust stroke. It was proposed that the use of a mechanically driven compressor, a supercharger, could help reduce the backpressure, therefore helping with residual gas reduction.

Following their successful simulation results, Dr. Wang et al. filed a patent with the US patent office in 2006 [18]. The concept presented in the patent is a variation of the concept presented in their publication in the sense that the expanded air would also be used in the combustion. This patent created a basis for all the work presented in this thesis, starting with the proof of concept.

It is important to mention that different names have been used for the concept since 1950: turbocooling 1959 and 1991, turboexpansion 2003 turbo-cooling 2005 and turbo-cool 2006. Some of the “inventions” or concepts are physically different or have different primary goals but they all make use of an expander to ultimately reduce the intake temperature. As presented in the benefit section, at the end, lower emissions, lower fuel consumption and increase power are the common expected outcome. “Turbo-Expansion” is used in this text.

3. Laboratory Experiments and Studies

As mentioned in the history section above, Dr. Lin-Shu Wang and Shiyou Yang suggested in their 2006 publication, that the VGT turbocharger and turbo-cool are indispensable to each other and that the resulting temperatures could be below ambient. Also, improved brake mean effective pressure and lower brake specific fuel consumption were reported. The theoretical simulation was the first step to Turbo-Expansion research at Stony Brook University. The next step was to physically verify the simulation results.

3.1 Engine test

A Nissan Silvia engine was acquired for the turbo-cool proof of concept since it matched the specifications used in the case study. The engine specifications are listed in Table 3-1 below.

Table 3-1. Parameters of the baseline turbocharged SR20DET DOHC 16V engine

Parameter	Unit	Data
Intake valve seat diameter	In	1.34–1.35
Exhaust valve seat diameter	In	1.18–1.19
Exhaust valve opening	Crank angle degree (BBDC)	53
Exhaust valve closing	Crank angle degree (ATDC)	7
Intake valve opening	Crank angle degree (BTDC)	−6
Intake valve closing	Crank angle degree (ATDC)	66
Throttle body bore	In	2.36
Bore	In	3.39
Stroke	In	3.39
Connecting rod length	In	5.37
Compression ratio		8.5
Rated power	Hp	205
Rated speed	Rpm	6000
Firing order	1–3–4–2	
Compressor type	T-25, 60 trim, 56 mm BCI-1	
Turbine type	T-25, 62 trim, 53.8 mm 0.64 A/R turbine housing	
Ignition timing - basic (static)	15 ±2° CA BTDC at 800 r/min	

The technicians at Marcovicci & Wenz Racing, where all engine tests would be performed, recommended the use of an aftermarket Engine Control Unit (ECU). After evaluation, a E11-2 Haltech ECU was chosen. The reasons for this choice was the very well known reputation of the brand name, and also, the online and free availability of the software and manuals needed. Two of professor Wang's students, Travis Wentz and Antonio Cacavale, were in charge of the upgrade. The author became involved with the project at the last stages of the upgrade. The upgraded engine was brought to Marcovicci & Wenz Racing for dynamometer testing (cf. Fig. 3-1).

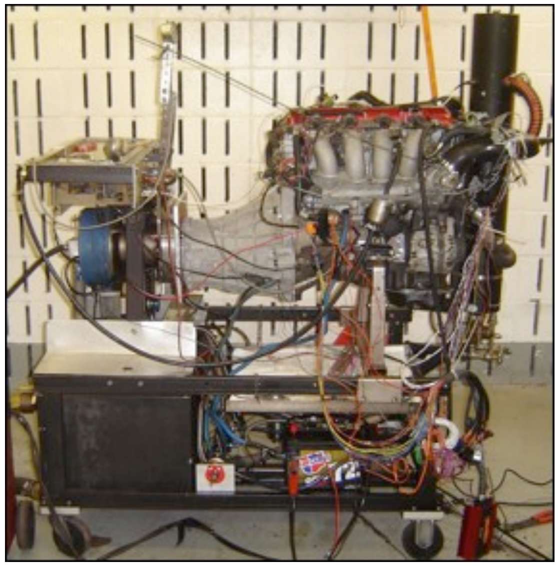

Fig. 3-1. Dynamometer testing of SR20 DET DOHC 16V engine with new ECU

The purpose of the first test was to record the performance of the chosen engine (cf. Fig. 3-2) and establish a point of reference. The engine produced 215 horsepower. The torque curve started at 190 lb-ft and power at 130 hp at low rpm. Maximum torque was reached at around 4750 rpm and maximum power around 5750 rpm. The power band of this particular engine is defined between those two peaks. Since the engine is rated at 205 hp, these results show that the engine was in good condition before performing further upgrades.

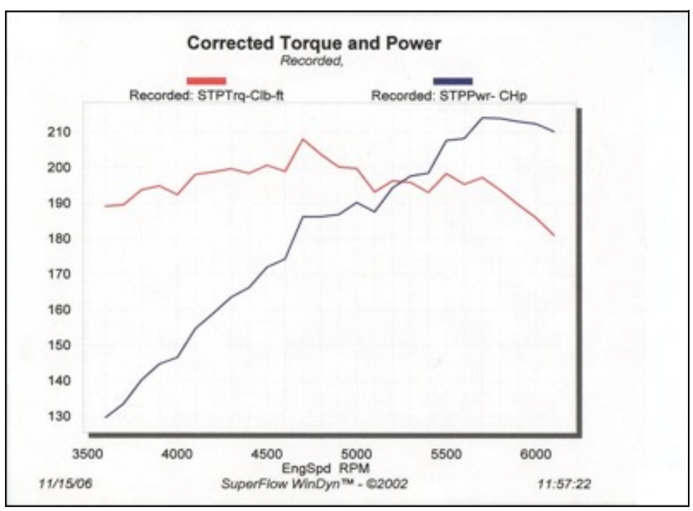

Fig. 3-2. Power vs. torque diagram

During testing, instead of using a stoichiometric air-fuel mixture for complete oxidation, a rich mixture was used. As covered in the earlier fuel economy section, using a rich fuel mixture means adding more fuel than needed. The extra fuel cools the mixture down and prevents knocking. This also increases power, but lowers fuel economy and increases emission. This explains the 10 hp over the rated value. The fuel map developed was also saved for comparison (cf. Fig. 3-3).

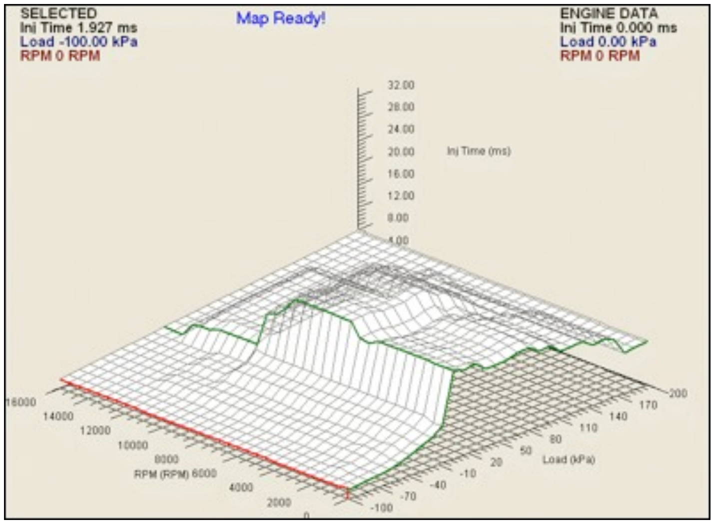

Fig. 3-3. Fuel map from first dynamometer test

The engine was then brought back to the lab for modifications. First, the stock Garrett turbocharger (TB2541) with wastegate was replaced by a VGT Garrett turbocharger (TD2503). Then, the setup shown in Fig. 1-3 was implemented utilizing the TB2541 within the turbo-cool unit. The completed setup with the pressure gauges and temperature sensors is shown in Fig. 3-4. The two electric valves used to regulate the airflow can be seen on the left side of the engine setup on the dynamometer stand (cf. Fig. 3-5).

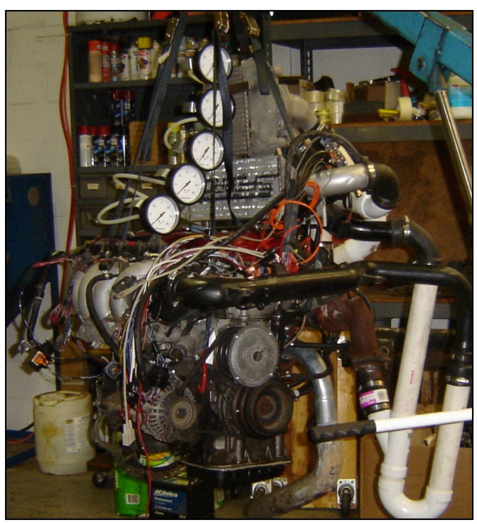

Fig. 3-4. SR20 engine with turbo-cooling unit installed

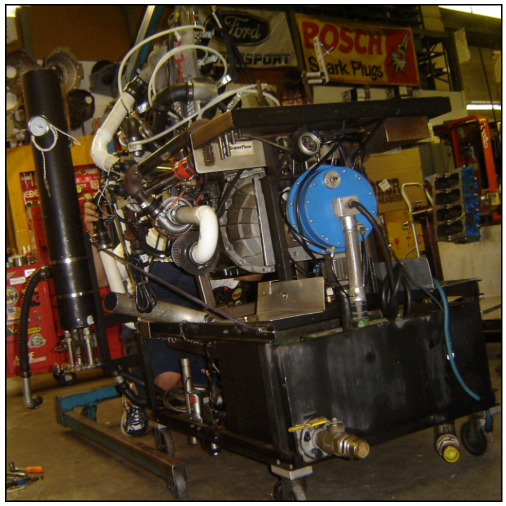

Fig. 3-5. Engine installed on dynamometer stand

At the second visit to Marcovicci & Wenz Racing, a new testing plan was designed. First, the valves would be switched so that the charged air would bypass the turbo-cooling unit. New data would then be collected and compared to the first results. This would point out any difference caused by the VGT installed after the original point of reference test. Then, the cooling unit circuit would be opened for collection of the final set of data. The data collected would then be compared to the two previous sets to evaluate the benefits of the turbo-cooling system. The objective was to use a stoichiometric mixture to increase fuel economy while decreasing emission.

Unfortunately, within the first half hour of testing, the dynamometer room was filled with smoke and testing had to be aborted. The engine and the VGT were taken apart for evaluation. It was concluded that a coked-up bearing in the VGT caused the smoke. Noticeable wear was observed on the turbo shaft (cf. Fig. 3-6) and both the compressor and turbine wheels. The bearing failure had led to the infiltration of oil in the exhaust turbine. The burnt oil produced the smoke at the exhaust pipe.

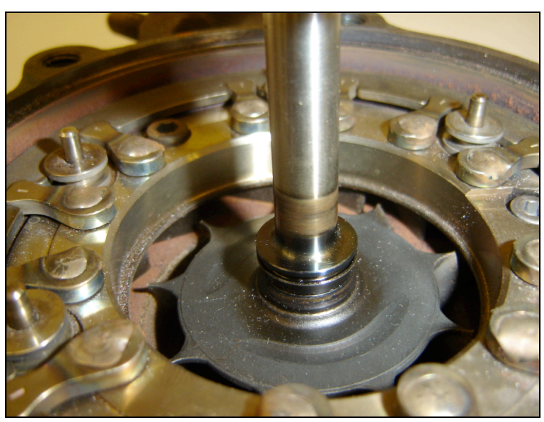

Fig. 3-6. Wear showing on coked-up VGT

After evaluation, two reasons that may have caused the failure were identified. First, although it was verified before testing that it was ok to split the oil supply line from the engine in order to cool and lubricate both turbochargers, insufficient lubrication might have caused the damage. It was decided to install a separate lubrication unit for future testing. The second reason was a high backpressure. During the test, high variations in pressure between 10 and 30 psig were observed at the exhaust turbine. High pressure could also cause bearing failure due to high loads and oil leakage. High pressure can result from a turbine witch is too small for a specific application. Because of the failure, it was decided to take a step back, and conduct a feasibility study on the components at hand. This would also push back any future test and allow time to learn more about the limits of each component.

3.2 System feasibility

The feasibility study had two different aspects. Firstly, as if the prototype phase was starting from scratch, with the performance of the engine during the first successful dynamometer test in mind, a turbocharger component selection procedure was executed. Each selection was compared to the components used in the failed test to validate their use in future test. Also, the specifications of the components already purchased or readily available on the market were used to reevaluate the turbo-cool concept in its entirety.

3.2.1 Components involved

a) Compressor

When choosing a turbocharger for an application, some basic goals are to be set and several estimations are necessary. The compressor is the first component to be analyzed. Each knowledgeable expert uses his/her own method to determine if a compressor is appropriate for a specific application [19, 20, 21]. Although some methods appear to be completely different, they all have the same goal. Some are faster than others based on the amount of assumptions. The method chosen for this project made use of the knowledge acquired from each method to reduce as many assumptions as possible.

First, a horsepower target is expected at a certain RPM. The mass flow of air $\dot{m}_a$, needed to attain this target is given in Eq. 3-1.

$$\dot{m}_a = HP \times A/F \times BSFC/60 \quad \text{[lb/min]} \tag{3-1}$$

Where:
HP = Horsepower Target (flywheel)
A/F = Air/Fuel Ratio
$BSFC$ = Brake Specific Fuel Consumption [lb/(Hp * hr)] ÷ 60 (to convert from hours to minutes)

The Air/Fuel Ratio and the brake specific fuel consumption are to be assumed based on realistic values obtained in practice. A stoichiometric mixture for gasoline is 14.7. This means that the right amount of air is mixed with the fuel to result in complete combustion. A higher A/F ratio will result in a lean mixture. An excess of air will be in the combustion chamber. The consequences of a lean mixture include an increase in temperatures. A lower A/F number, a rich mixture, describes an excess of fuel. The excess of fuel is accompanied by a decrease in fuel economy and an increase in emission. In practice, gasoline engines tend to run a little rich for best performance. Garrett/Honeywell's engineers recommend using values around 0.5 and 0.6 for BSFC based on experiments. A well-tuned engine will have a BSFC in the lower range.

After the mass flow needed is calculated using Eq. 3-1 with the assumption as suggested above, the intake manifold pressure is computed using Eq. 3-2.

$$P_{a.i} = \frac{\dot{m}_a \times R \times (460 + T_{a.i})}{\eta_v \times N/2 \times V_d} \text{ [psia]} \tag{3.2}$$

Although the optimal turbocharged setup would not experience any pressure losses, it is unavoidable to have a pressure drop through the intercooler and pipes between the compressor and the intake manifold. These losses in an intercooler may vary between 0.5 an 2 psi. An additional 0.5 psi can be assumed for the piping. The pressure at the outlet of the compressor, P_{2C}, can be calculated by adding the mentioned pressure losses, $\Delta P_{intercooler+piping}$ in Eq. 3-3.

$$P_{2C} = P_{a.i} + \Delta P_{intercooler+piping} \text{ [psia]} \tag{3.3}$$

Many car enthusiasts do not include a pressure loss through the intake air filter to simplify calculations. The pressure at the inlet of the compressor can be calculated by subtracting the pressure loss through the air filter and pipe connections (cf. Eq. 3-4). These losses can usually be assumed to be around 1 psi. Finally, the required compressor pressure ratio can be calculated as the ratio of the pressure at the outlet of the compressor over that of the inlet (cf. Eq. 3-5).

$$P_{1C} = P_{Ambient} - \Delta P_{filter+piping} \text{ [psia]} \tag{3-4}$$

$$\Pi_{compressor} = P_{2C} / P_{1C} \tag{3-5}$$

With the knowledge of the pressure ratio and the mass flow required for maximum-targeted horsepower, reference to a compressor map can be made to verify that the operating point at maximum power will fall within an acceptable efficiency zone. Unfortunately but also unavoidable, the air temperature increases as it is compressed. The efficiency of the compressor is at the source of this effect. Assuming that there is no heat transfer through the compressor walls, the compression is adiabatic [22]. Also, assuming that at these low temperatures and pressure, the ideal gas law still applies, the temperature increase through the compressor can be calculated (cf. Eq. 3-6).

$$\Delta T_c = \frac{T_{1c} \times Y_c}{\eta_c} \tag{3-6}$$

η_c is the isentropic compressor efficiency. A value of 70% is usually assumed for simplified numerical computations. Y_c is the air calculation constant for the compressor and is calculated using the pressure ratio as follows (cf. Eq. 3-7).

$$Y_c = \left(\frac{P_{2c}}{P_{1c}}\right)^{.283} - 1 \tag{3-7}$$

b) Intercooler

The idea behind using an intercooler is to lower the temperature as much as possible before reaching the intake manifold. The effectiveness of an intercooler is a measure of how much heat is extracted from the charged air as compared to how much is added by the compressor [19]. One can compute the intercooler effectiveness as shown in Eq. 3-8.

$$E_{IC} = \frac{T_{co} - T_{io}}{T_{co} - T_a} \tag{3-8}$$

Where:
T_{co}=Compressor outlet temperature
T_{io}=Intercooler inlet temperature
T_a=Ambient air temperature

Although most intercooler manufacturers and resellers do not explain how the effectiveness is derived, the intercooler is no different than a heat exchanger. The intercooler effectiveness can therefore be derived from the heat exchanger's effectiveness calculation. The effectiveness is then understood as the actual heat transfer over the maximum heat transfer possible with the temperature difference available. The actual heat transfer Q can be measured in two different ways (cf. Eq. 3-9) depending on the parameters available. Q_c represents the heat transferred to the coldest medium and Q_h represent the heat transferred from the hottest medium.

$$\begin{aligned} Q_h &= \dot{m}_h c_h (T_{h,in} - T_{h,out}) \\ Q_c &= \dot{m}_c c_c (T_{c,out} - T_{c,in}) \end{aligned} \tag{3.9}$$

In Eq. 3-9, $\dot{m}$ represents mass flow through the heat exchanger, c represents specific heat, and T temperature. The subscripts h and c stand for hot and cold. These represent the parameters of the mediums entering and exiting the heat exchanger. The maximum heat exchange possible is calculated in a similar way by multiplying the minimum product between $\dot{m}_c c_c$ and $\dot{m}_h c_h$ by the maximum temperature difference possible between all four available temperatures $T_{h,in}$, $T_{h,out}$, $T_{c,out}$ and $T_{c,in}$ (cf. Eq. 3-10).

$$Q_{\max} = (\dot{m}c)_{\min}(T_{hottest} - T_{coldest}) \tag{3-10}$$

In an intercooler application, it is understood that Q_h is the heat transferred from the charged air to the ambient surroundings. Assuming that the intercooler is infinitely long, $(\dot{m}c)_{\min}$ would be that of the charged air. The hottest temperature is the temperature out of the compressor and the coldest is the ambient temperature. The effectiveness is therefore simplified to Eq. 3-8 for an intercooler.

c) Turbine

Turbines are more forgiving than compressors and therefore do not require as much attention as the compressor. On the same token, turbines are more complicated than compressors. No simple equations are capable of determining the right turbine for an application given that too many unknowns are present. Exhaust gas exact temperatures and pressures are two examples. Turbocharger designers are well aware of these complications and spend a lot of time to come up with final designs. They also supply some guidelines to help determine if a turbine is right for an application.

Two main turbine parameters are used to help determine the turbine characteristics. First, the turbine exducer bore diameter can be measured. The exducer is the area where the exhaust gas exits the turbine. Experiments have shown [19] that the compressor mass flow capacity is highly dependent on such parameter. These experimental graphs [19] can be used as an approximation but do not guarantee performance.

The second aspect is the A/R ratio. This ratio is defined as the cross section area A of a turbine flow path over its distance from the center (i.e. its radius, R). This ratio is kept constant and affects the speed at which exhaust gas impact the turbine wheel. This results in more or less torque being applied to the compressor. Again, from experiments [19], is has been proven that a small A/R ratio provides more low speed response and a larger one provides more top end power. Although the compressor housing is designed in a similar way, the compressor A/R ratio does not have much effect on turbocharger response.

Overall, these two parameters are just used as approximations when switching from one turbine housing to the other for tuning purposes. When purchasing a turbocharger, manufacturers already match the compressor to appropriate turbines. It is recommended to size the compressor and choose a turbocharger assembly, which has a few turbine housings available for tuning. Adjusting a wastegate or variable geometry turbine actuator can further help tune the system.

Although not useful when selecting a turbocharger, with few assumptions, similarly to the compressor, specific equations can be used to determine the temperature drop through a turbine [22]. Assuming that there is no heat transfer through the turbine walls, the expansion is adiabatic. Also, assuming that temperature and pressure is low enough that the ideal gas law still applies, the temperature decrease through the turbine can be calculated (cf. Eq. 3-11).

$$\Delta T_t = \frac{T_{1t} \times \eta_t \times Y_t}{1 + Y_t} \tag{3-11}$$

η_t is the isentropic turbine efficiency. A value of 70% is usually assumed for simplified numerical computations. Y_t is the air calculation constant for the turbine and is calculated using the pressure ratio as follows (cf. Eq. 3-12).

$$Y_t = \left(\frac{P_{1t}}{P_{2t}} \right)^{.283} - 1 \tag{3-12}$$

The pressure ratio is obtained based on the simplified turbine maps provided by the manufacturer. Once the mass flow is approximated, a pressure ratio can also be approximated.

3.2.2 Feasibility study

The components described above are the only components used in the expansion-cooling concept intake system. The equations derived can therefore be used to compute the pressure and temperature changes through the system given a specific mass flow for a target horsepower value.

From the first engine test, an output of 215 horsepower was recorded at 5700 rpm. This was the target horsepower. Following the procedure as explained in the compressor section and using Eq. 3-1, the necessary mass flow of 18.9 lb/min was calculated as shown in Table A-1 in the appendix. Notice that the intake manifold of 7 psig was the factory setting of the stock turbocharged engine. Numbers in bold were known values, numbers with * were assumed base on values obtained in practice [2] and proposed by Garrett/Honeywell's engineers. The compressor ratio was also calculated as a reference for comparison purposes.

The following 7 steps were taken to determine if the same performance would be achievable using an expansion turbine to lower charged air temperature:

1- Find a turbine allowing for 19 lb/min or more
2- Find the expansion ratio of the selected turbines at 19 lb/min
3- Specify desired temperature at the intake manifold
4- Obtained temperature before turbine going backwards from desired manifold temperature
5- Obtained required compressor ratios going backwards based on desired manifold pressure

6- Compute temperature after intercooler going forward based on ambient temperature and the selected compressor from step 5
7- Adjust intercooler effectiveness until temperatures at steps 4 and 6 are equal.

Step 1.

Since the mass flow required for the target performance is known, an expansion turbine allowing for that mass is chosen based on the turbine map. Since Garrett/Honeywell turbochargers are being used, only turbines from that brand were considered for selection.

Step 2.

Five turbines were selected for computation purposes. At the specified mass flow rate, available turbine pressure ratios were found to be 1.26, 1.36, 1.44, 1.68 and 1.79.

Step 3.

Because regular intercoolers may be able to bring the manifold temperature close to ambient but never below, five temperature points at and below ambient were considered. Assuming a 70 °F ambient temperature, results were obtained for target temperatures from 60 to 70 °F at 2.5 °F intervals.

Step 4.

Using Eq. 3-11 and 3-12, the temperature drop through the turbine was calculated. Given a targeted manifold temperature, the higher temperature at the expansion turbine inlet was obtained.

Step 5.

Given a target pressure at the intake manifold (i.e. 7 psig) and an expansion ratio, the compressor ratio is obtained based on ambient pressure using Eq. 3-3 to 3-5.

Step 6.

Using Eq. 3-6 through 3-8, the temperature at the outlet of the intercooler was calculated.

Step 7.

For the targeted performance to be attained and the simulation to make sense, the temperature at one common section must be the same no matter what the starting point is. In other words, temperature at step 4 is obtained from the target temperature at the intake manifold and temperature at step 6 is obtained from the ambient temperature and the calculated compressor ratio, must be equal since they describe the same location. The only parameter that can be adjusted is the intercooler efficiency.

3.2.3 Observations from the feasibility study

The results of the feasibility study are reported in the appendix (cf. Table A-1 through A-5). Some observations from Tables A-2, A-3 and A-4 require some attention. The first observation was that the higher the expansion turbine ratio, the lower the intercooler effectiveness required. This is a good observation as intercoolers are limited to a certain range based on the space available. This space is usually limited in automobiles.

Second observation, the compressor ratio required to maintain the desired manifold pressure of 7 psig also increases with the turbine expansion ratio. As mentioned in the compressor section, compressors are limited by their performance described by the compressor map. For the given horsepower range, the compression ratio cannot exceed a certain limit because it will not be readily available on the market.

Third observation is that the lower the target temperature, the higher the intercooler effectiveness required. For the same reason mentioned as a positive aspect in the first observation, targeting too low of a temperature would therefore make it impossible to find an intercooler on the market.

3.3 Control system design

Given the complexity and the evident dependency of all parameters involved, it was obvious that a controller would be necessary in order to take full advantage of the concept. In this section, an analysis of the components involved from a control system standpoint is presented. The application of a H_∞ controller is also introduced.

The expansion-cooling concept is based on the idea of expanding part or all the charged air upstream from the intake manifold [23]. Readers familiar with Exhaust Gas Recirculation (EGR) [24] in diesel engines may relate the flow separator valve and expansion turbine added to a regular turbocharged engine to a recirculation valve and an exhaust gas heat exchanger. For this same reason, many ideas were sourced from EGR publications [25]. Additionally, both concepts essentially achieve the same goals of increasing fuel economy, and reducing emission).

Today, EGR is seen in multiple applications because controllers have been developed to maximize engine efficiency and minimize emissions. In EGR development, the goal is to mix enough exhaust gas to the intake manifold such that the combustion temperature is lowered as much as possible while not drastically reducing the amount of air induced, therefore affecting the power output. Similarly, in turbo-cooling, the goal is to expand as much air as possible while maintaining a certain pressure at the intake manifold. This would maximize cooling therefore also maximizing engine power output.

Considering a fuel injection Spark Ignition (SI) engine, with mechanically actuated throttle, the Engine Control Unit (ECU) is programmed to inject the right amount of fuel based on the amount of air induced. The spark timing is also programmed in the ECU. Adding the expansion-cooling system to a vehicle requires the ECU to also control the position of the wastegate or Variable Geometry Turbine (VGT) actuator along with that of the flow separator valve. Similar control algorithms are applied to EGR control and the hardware should be available for application.

The following subsections cover some of the modeling ideas developed and applied by different authors. Assumptions are discussed in terms of model representation and uncertainties.

3.3.1 Plant modeling

In order to implement a control system to a plant, the physical system must be modeled in such a way that the events affecting such system can be simulated. There are many ways to model a system for simulation purposes; however, since one main objective of this section is to apply the knowledge acquired from a state-space analysis methods and multi-variable controller design course, a state space model will be sought for in order to design a H_2 or H_∞ controller as presented in [26].

a) Modeling of Dynamics

Certain considerations are needed before introducing the components' model. Two main aspects can be considered in an internal combustion engine [27]:

-Combustion process
-Thermodynamic boundary conditions

This system, as stated earlier, controls the thermodynamic boundary conditions that govern the combustion process (i.e. intake pressure and temperature). The main assumption that will considerably reduce simulation time is that combustion will not vary if the boundary conditions are fixed. Consequently, a Mean Value Model (MVM) will be used versus a Discrete Event Model (DEM). In the so-called MVM, time *t* is the independent variable where as for the DEM, the crank angle is. Clearly, for a project such as the current one, it is more effective to use the MVM. Since most MVM are lumped parameter models, ordinary differentials are expected [27]. This gives a good basis for the state space application since the state space matrices are based on differential equations. Other ways to implement a model will not be discussed here but interested readers are advised to consult references [28,29,30,31,32].

After determining that a mean value would be used, two different but closely related methods were utilized to obtain the final differential equations namely the Cause & Effect Diagrams [27] and the Control Volume & Restrictions [33].

b) Cause and Effect Diagrams

The cause and effect diagram method begins with an abstract mean-value structure of the system. It is a basic figure with some representation of the real components. Then, two main classes of objects are taken into account:

-<u>Reservoirs</u>, thermal or kinetic energy of mass or information containing variables defining the content
-<u>Flows</u>, energy or mass flowing in between the reservoirs

A diagram representing all reservoirs and flows for a system is called the system's cause and effect diagram. The current project description started as a cause and effect diagram but a much simpler way is to use the following approach [33].

c) Control Volume & Restriction Concept

This method is not too different than the one previously presented. The only benefit is that it is more component driven. The method would also start with the abstract mean value structure.

Restrictions are defined as any component, which defines flow disturbances. Such components may be increasing or reducing flow like the compressor, the turbine, air filter and valves or any restrictions in the system. Mass flow and temperature are the transported properties through the restrictions.

In between the restriction components, control volumes are defined. Pipes and manifolds are representatives of such volumes. It is also made very clear in [33] that pressure and temperature from the control volumes should be taken as state variables since they can easily be measured for tuning purposes.

From this point on, it is a matter of gathering all the equations used for a specific system. The equations presented below are well known and proven equations based on physics and thermodynamics.

3.3.2 Components

a) Control Volumes

Control volumes are also known as restriction receivers as they receive the transported properties from the restrictions. To derive the differential equations for the pressure and temperature in the control volumes, some assumptions are made:

-no heat or mass transfer through the walls

-no substantial changes in potential or kinetic energy

-fluids can be modeled as ideal gases

-Temperature T in the control volume is equal to the temperature downstream of the control volume based on the lumped parameter approach (i.e. $T_{ds}(t) = T(t)$)

The following well-known equations [34] are used to derive the needed differential equations:

$$\frac{d}{dt}m(t) = \dot{m}_{us}(t) - \dot{m}_{ds}(t) \tag{3-13}$$

$$\frac{d}{dt}U(t) = \dot{H}_{us}(t) - \dot{H}_{ds}(t) \tag{3-14}$$

$$p(t)V = m(t) \cdot T(t) \cdot R \tag{3-15}$$

$$U(t) = c_v \cdot T(t) \cdot m(t) \tag{3-16}$$

$$\dot{H}_{us}(t) = c_p \cdot T_{us}(t) \cdot \dot{m}_{us}(t) \tag{3-17}$$

$$\dot{H}_{ds}(t) = c_p \cdot T(t) \cdot \dot{m}_{ds}(t) \tag{3-18}$$

Substituting Eq. 3-15 through 3-18 into Eq. 3-13 and 3-14 results in the wanted differential equations for pressure and temperature through the control volume:

$$\frac{d}{dt}p(t) = \frac{k \cdot R}{V} \cdot \left[\dot{m}_{us}(t) \cdot T_{us}(t) - \dot{m}_{ds}(t) \cdot T_{ds}(t) \right] \tag{3-19}$$

k is the ratio of specific heat and R is the gas constant.

$$\frac{d}{dt}T(t) = \frac{T_{ds} \cdot R}{P \cdot V \cdot c_v} \cdot \left[\begin{array}{l} c_p \cdot \dot{m}_{us}(t) \cdot T_{us}(t) \\ - c_p \cdot \dot{m}_{ds}(t) \cdot T_{ds}(t) \\ - c_p \cdot \left(\dot{m}_{us}(t) - \dot{m}_{ds}(t) \right) \cdot T_{ds}(t) \end{array} \right] \tag{3-20}$$

b) Flow rate through restrictions with pressure losses

Restrictions where a pressure loss can be observed with no particular change in temperature are the air filter, catalyst, exhaust system, and pipe bends. Also a pressure drop is seen across the throttle valve, recirculation valves, wastegate and compressor bypass. These two sets can be divided in the valve types that can be represented as incompressible and turbulent flow and those that cannot because they have higher-pressure losses and flow velocities. The later type requires a compressible flow through nozzle model [33].

i) Mass flow rate through air filter, catalyst, exhaust system and pipe bends

As stated above, the air filter, catalyst, exhaust system and pipe bends flow can be well described by the incompressible and turbulent flow equation (cf. Eq. 3-21).

$$\dot{m} = \begin{cases} \sqrt{\dfrac{p_{us}(p_{us} - p_{ds})}{C \cdot T_{us}}} & p_{us} - p_{ds} \geq p_{lin} \\ \sqrt{\dfrac{p_{us}}{C \cdot T_{us}}} \cdot \dfrac{p_{us} - p_{ds}}{\sqrt{p_{lin}}} & \text{Otherwise} \end{cases} \tag{3-21}$$

The linear term is included in the second definition to account for the fact that at low flow, laminar properties are observed. As the flow increases, the transition is made to the first definition for the turbulent range.

ii) Mass flow rate through throttle valve, recirculation valves, wastegate and compressor bypass

The valves in this section are subject to high-pressure losses and flow velocities and require the use of compressible flow through a nozzle:

$$\dot{m} = \frac{p_{us}}{\sqrt{R \cdot T_{us}}} A_e(u)\Psi(\Pi_{th}) . \tag{3-22}$$

$A_e(u)$ is the effective area based on input signal u and $\Psi(\Pi_{th})$ is defined as:

$$\Psi(\Pi_{th}^*) = \sqrt{\frac{2k}{k-1}\left(\Pi_{th}^{*\frac{2}{k}} - \Pi_{th}^{*\frac{k+1}{k}}\right)} \tag{3-23}$$

$$\Pi_{th}^* = \max\left(\frac{p_{ds}}{p_{us}}, \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}\right) \tag{3-24}$$

c) Intercooler

i) Intercooler mass flow

Although the intercooler is often modeled with Eq. 3-21, the fact that the air passes through very thin fins adds some discrepancy when compared to measured data [35]. To solve this problem, a linear term is added to the pressure drop equation and a better fit is obtained:

$$\Delta p_{ic} = C_{ic,1}\frac{T_c \dot{m}_{air}}{p_c} + C_{ic,2}\frac{T_c \dot{m}_{air}^2}{p_c} \tag{3-25}$$

ii) Intercooler temperature drop

The reader can reference any heat transfer book for the derivation of a heat exchanger's effectiveness, which is what the intercooler is. Basically, the heat transfer from the charged air is divided by the maximum heat transfer allowable based on the maximum temperature different between the charged air and the cooling medium. This equation was derived in the intercooler subsection of the system feasibility section and is listed here for the reader's convenience (cf. Eq. 3-26):

$$\varepsilon = \frac{T_c - T_{ic}}{T_c - T_{cool}} \tag{3-26}$$

The temperature downstream of the intercooler can then be calculated in terms of effectiveness (cf. Eq. 3-27):

$$T_{ic} = T_c - \varepsilon(T_c - T_{cool}) \tag{3-27}$$

In the case of an air-to-air intercooler, $T_{cool} = T_a$. To simplify simulations, the effectiveness is sometimes assumed to be a constant between 65 and 75%. However, two models described in [35], the standard NTU-model and the regression model, show that the effectiveness is of course related to the mass flow and temperatures of the mediums in question.

J.P. Holman derived the NTU model represented by Eq. 3-28 through 3-31:

$$\varepsilon = 1 - e^{\frac{e^{-C \cdot N^{0.78}} - 1}{CN^{-0.22}}} \tag{3-28}$$

$$N = \frac{UA}{c_{p,air} \cdot \dot{m}_{air}} = \frac{K}{c_{p,air}} \dot{m}_{air}^{-0.2} \mu_i^{-0.5} \tag{3-29}$$

$$\mu_i = 2.3937 \times 10^{-7} \left(\frac{T_c - T_{cool}}{2} \right)^{0.7617} \tag{3-30}$$

$$C = \frac{\dot{m}_{air}}{\dot{m}_{cool}} \tag{3-31}$$

From the important parameters observed in the NTU model, the regression model is derived:

$$\varepsilon = a_0 + a_1 \left(\frac{T_c + T_{cool}}{2} \right) + a_2 \dot{m}_{air} + a_3 \frac{\dot{m}_{air}}{\dot{m}_{cool}} \tag{3-32}$$

Experimental testing [35] shows that the regression model is a better fit than the NTU model. It is very important to understand that when tuning, the difference between the temperature should be minimized and not the efficiency specifically.

d) Engine

In accordance with the mean value model, the engine reciprocating behavior is ignored and represented by a continuously working volumetric pump that produced exhaust gas and torque [27]. So the standard model is used based on volumetric efficiency:

$$\dot{m}_{eng} = \frac{\eta_{vol} \cdot V_d \cdot N \cdot p_{im}}{2R \cdot T_{im}} \tag{3-33}$$

For simulation purposes, it is necessary to have available lookup tables or a parameterized function of the volumetric efficiency to calculate the mass flow rate though the engine, therefore power. As explained in [27], an approximation for the volumetric efficiency shows a dependency on two factors, one is a function of manifold pressure and the other is a function of speed.

$$\eta_{vol} = \eta_{vol,p}(p_{im}) \cdot \eta_{vol,N}(\omega_{eng}) \tag{3-34}$$

Assuming perfect gases with constant k and isentropic process, the part dependent on pressure due to the gas trapped at top dead center (TDC) can be estimated as:

$$\eta_{vol,p}(p_{im}) = \frac{V_c + V_d}{V_d} - \left(\frac{p_{em}}{p_{im}}\right)^{1/k} \frac{V_c}{V_d} \tag{3-35}$$

Therefore, when collecting data to find a parameterization for the volumetric efficiency, only one term needs to be fitted. Experimental fitting will also be necessary. The temperature upstream form the intake manifold can be parameterized in function of mass airflow through the engine [36].

$$T_{em} = f(\dot{m}_{eng}) \tag{3-36}$$

e) Turbocharger

The turbocharger is one of the components that complicate the modeling of the system. Not only is there a necessity for either curve fitting or lookup tables, but also relations between the turbine and the compressor through the shaft need to be taken into account.

i) Compressor

The compressor manufacturers provide a map that describes the characteristics of the compressor as shown in Fig. 3-7.

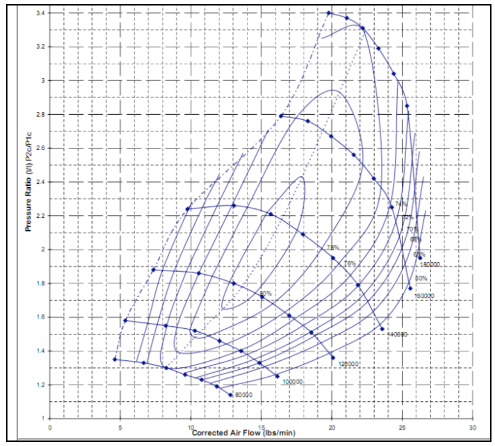

Fig. 3-7. Typical compressor map

The maps provided by the manufacturers are correct for a specific inlet pressure and temperature. This is done to make sure that future measurements are always calculated in reference to the settings that were used to obtain the maps.

If the pressure upstream and down stream of the compressor are known, the pressure ratio can be calculated (cf. Eq. 3-37). Measuring the actual compressor speed and calculating the corrected speed (cf. Eq. 3-38) will define the corrected mass flow corresponding to that specific operating point. The corrected mass flow can then be used to calculate the actual mass flow (cf. Eq. 3-39).

$$\Pi_c = \frac{p_{ds}}{p_{us}} \tag{3-37}$$

$$\tilde{\omega}_{tc,cort} = \sqrt{T_{us,ref} / T_{us,act}} \cdot \omega_{tc,act} \tag{3-38}$$

$$\dot{m}_c = (p_{us} / p_{ref}) \cdot \sqrt{T_{us,ref} / T_{us}} \cdot \dot{\mu}_c \tag{3-39}$$

Also available on the map is the compressor isentropic efficiency. It can be read at the same operating point. The efficiency is necessary to calculate the power consumption of the compressor:

$$P_c = \frac{P_{c,s}}{\eta_c} = \dot{m}_c \cdot c_p \cdot T_{us}\left[\Pi_c^{(k-1)/k} - 1\right]\cdot\frac{1}{\eta_c}. \tag{3-40}$$

$P_{c,s}$ is the isentropic power consumption of the compressor. Since the compressor speed is measured, the compressor torque can be calculated:

$$T_{qc} = \frac{P_c}{\omega_{tc}} \tag{3-41}$$

With the efficiency, and the upstream temperature known, the temperature downstream from the compressor can be calculated:

$$T_{ds} = T_{us} + \left[\Pi_c^{(k-1)/k} - 1\right]\cdot\frac{T_{us}}{\eta_c} \tag{3-42}$$

When running simulations, the computers need to be able to find the actual mass flow and efficiency at any particular operating point. For this purpose, two solutions have been considered:

-Lookup table (interpolation) [37]
-Curve fitting (parameterization) [33,35,38,39]

Using a lookup table is not more complicated than going through Eq. 3.40 and 3.41 and looking up the values in the table. For values that are not plotted, interpolation is used. It has been shown however that standard table interpolation routines are not continuously differentiable (leading to apparent discontinuities in the simulation), the extrapolation is unreliable and the table representation is not compact [39].

The unreliability comes from the fact that the range of data reported from a flow bench does not cover the entire engine operating range. As described in [39], two main problems are at the source of this range deficiency. At lower flows, the accuracy of the sensor readings is not appropriate. If the sensors are changed for low measurements, then they will not capture the higher flows. A step in the data, caused by such a change would not be acceptable either. Also, the heat transfer from the lubricating oil gets transferred to the compressor housing at low flows. This makes the compressor look worst and the turbine look better. This is again very inaccurate. Manufacturers just do not report lower ranges. Therefore, curves that will expand to the lower range are to be developed.

Curve fitting involves coming up with a basic model and tuning its parameters to get the best fit. The model is compared to both experimental values and manufacturer's map.

Different models have been developed and compared in the literature. In [39], four different models are described and compared:

-Jensen & Kristensen method
-Mueller method

-Zero Slope Line method (ZSLM)
-Neural networks

All methods output almost identical results. The Jensen & Kristensen method is the one mostly encountered in practice and therefore is described here.

For the Jensen & Kristensen method, four parameters are defined:

-Head parameter:

$$\Psi = \frac{c_p.T_a\left(\left(\frac{p_{out}}{p_{in}}\right)^{\frac{y-1}{y}} - 1\right)}{\frac{1}{2}U_c^2} \tag{3-43}$$

Here, y replaces k in terms of ratio of specific heat.

-The compressor blade tip speed:

$$U_c = \frac{\pi}{60} d_c N_{tc} \tag{3-44}$$

-The normalized compressor flow rate:

$$\Phi = \frac{W_c}{\rho_a \frac{\pi}{4} d_c^2 U_c} \tag{3-45}$$

-The inlet Mach number:

$$M = \frac{U_c}{\sqrt{\gamma R T_a}} \tag{3-46}$$

The head parameter and compressor efficiency are then expressed as function of the Mach number and the normalized compressor flow:

$$\Psi = \frac{k_1 + k_2\Phi}{k_3 - \Phi} \quad k_i = k_{i1} + k_{i2}M,\ i = 1,2,3 \tag{3-47}$$

$$\eta_c = a_1\Phi^2 + a_2\Phi + a_{3,}\ \ a_i = \frac{a_{i1} + a_{i2}M}{a_{i3} - M} \tag{3-48}$$

The parameters are determined through a least square fit on experimental data. Since Eq. 3-47 is inverted, we obtain:

$$\Phi = \frac{k_3\Psi - k_1}{k_2 + \Psi} \tag{3-49}$$

Therefore,

$$\dot{m}_c = \Phi \rho_a \frac{\pi}{4} d_c^2 U_c \tag{3-50}$$

ii) Turbine

Similarly to the compressor, turbine maps are provided and are based on corrected mass flow and expansion ratios. An example is shown in Fig. 3-8.

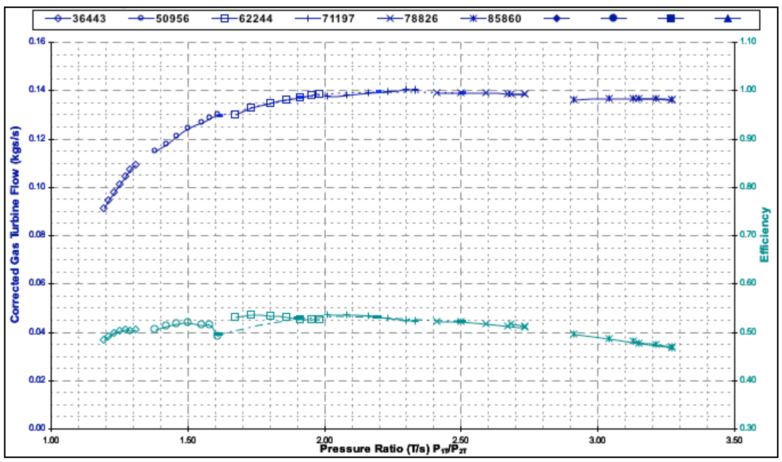

Fig. 3-8. Typical turbine map

Eq. 3-51 through 3-53 would be used to calculate the actual mass flow through the turbine.

$$\Pi_t = \frac{p_{us}}{p_{ds}} \tag{3-51}$$

$$\tilde{\omega}_{tc,cort} = \sqrt{T_{us,ref} \big/ T_{us,act}} \cdot \omega_{tc,act} \tag{3-52}$$

$$\dot{m}_c = (p_{us} \big/ p_{ref}) \cdot \sqrt{T_{us,ref} \big/ T_{us}} \cdot \dot{\mu}_t \tag{3-53}$$

Similarly to the compressor efficiency, the turbine efficiency can be found at the same operating point by reading is from the map. The turbine power can therefore be calculated.

$$P_t = \frac{P_{t,s}}{\eta_t} = \dot{m}_t \cdot c_p \cdot T_{us}\left[1-\left(\frac{p_{ds}}{p_{us}}\right)^{(k-1)/k}\right]\cdot \eta_t \tag{3-54}$$

Consequently, the turbine torque is derived,

$$T_{qt} = \frac{P_t}{\omega_{tc}} \tag{3-55}$$

The temperature downstream from the turbine can also be calculated knowing the efficiency, the upstream temperature and pressure on both sides.

$$T_{ds} = T_{us}\left(1-\eta_t\left(1-\left(\frac{p_{ds}}{p_{us}}\right)^{(k-1)/k}\right)\right) \tag{3-56}$$

In terms of look up tables, the problems encountered are the same as described in the compressor section. Therefore curve fitting is necessary. The turbine is not as complicated as the compressor therefore the curve fitting process is also simpler.

It was proven in [39] that the mass flow through the turbine could be modeled using Eq. 3-22 through 3-24 describing adiabatic flow through a nozzle.

Jensen & Kristensen proposed a simple fit for the effective area in the specified equations:

$$A_t = \frac{k_{t1}}{\left(\frac{p_{ds}}{p_{us}}\right)} + k_{t2} \tag{3-57}$$

Where the parameters k_{ti} are functions of the speed parameter:

$$k_{ti} = k_{1i}\tilde{N}_{tc} + k_{2i},\ \text{for } i=1,2 \tag{3-58}$$

ii) Turbocharger rotational dynamics

Using Newton's second law, we find the derivative of the rotational speed of the turbocharger [35]:

$$\frac{d}{dt}\omega_{tc} = \frac{T_{qt} - T_{qc}}{J_{tc}} \tag{3-59}$$

3.3.3 Controller design

a) Different Controllers

In the automotive world today, many control systems make use of a PID controller [40]. However, many researchers develop new control algorithms to see how they can optimize the performance of the PID. In reference [41], different controllers are used to control heavy-duty diesel engine turbocharged systems. These controllers include a PID and a fuzzy PID (FPID). Also, two model predictive control algorithms were tested: a generalized predictive controller (GPC) and a Dynamic matrix controller (DMC). Through different sets of tests, it was determined that the GPC performed better than all other controllers while the FPID was an improvement over the PID.

Other types of predictive [42] and adaptive [43] control algorithms are presented in automotive applications. One Ph.D thesis [38] presented a robust control approach to air path control of a turbocharged engine. Another Master's thesis used optimal control of a diesel engine with VGT and EGR [44]. A natural gas application used LGG/LQR controllers to optimize performance [45].

b) H_∞

From all the examples listed above, it was concluded that some controllers performed better than others but sometimes required a higher parameterization cost. Therefore, the choice of controller depended on the application, the expected performance, but also the background of the designer.

In the case of this project, since the H2 and Hinf controller were the two main controllers studied by the author, only these two were considered. Additionally, as it was explained in the modeling section, the Mean Value Model developed has multiple sources of uncertainties going from the curve fitting of the compressor and turbine to the volumetric efficiency of the engine and all the estimations made for the control volumes. For this particular reason, Hinf was chosen because of the robustness needed.

c) Design procedure

Several days were spent looking for published papers related to Turbo-Expansion. Since this is not a common topic, there was no success. Knowing that EGR is closely related in terms of components, several papers were examined in order to select similarities. Most papers found were considering the controls aspect but not the modeling, which is the first and most complicated part. Once the control volume and restriction method was found, it was easy to select the variables and set the differential equations for the components.

However, after all differential equations were collected, the nonlinearity of the system limited the ability to find the states space matrices by hand. The goal was then set to create a simulink model representing all the control volumes and flow restrictions. A similar idea is presented in [33].

The function *"linmod"* in Matlab can then be used to extract the state spaces matrices of the system. Then, the matrices can be implemented in a simplified simulink model, which would be used to design the controller using *"hinfsyn"* function in Matlab. Such simpler model is shown in Fig. 3-9.

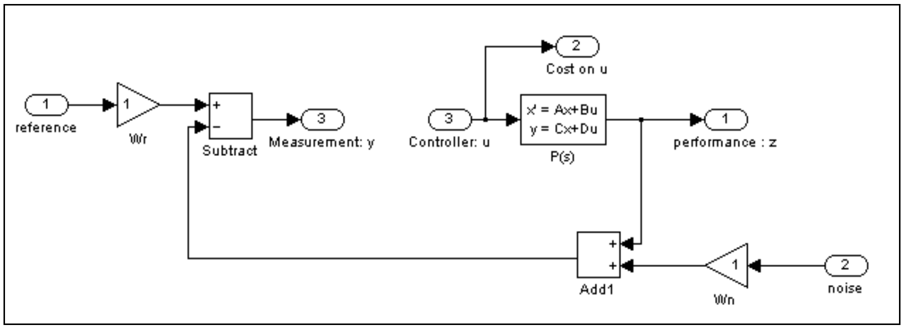

Fig. 3-9. Plant implemented in new simulink model with reference and noise added

Once the controller was obtained, it would be added to the simplified plant as shown in Fig. 3-10.

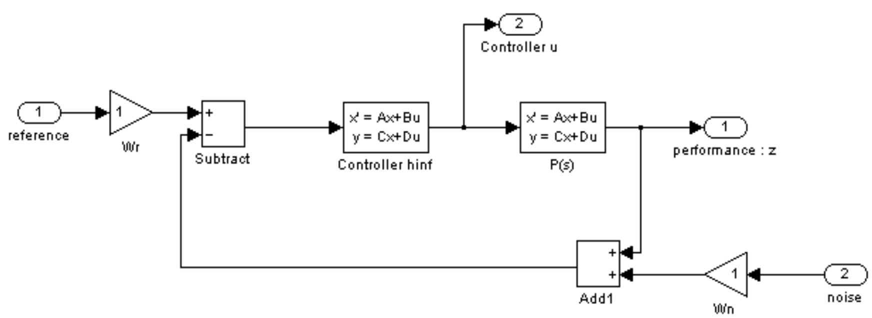

Fig. 3-10. Controller implemented

Performance could then be checked and modification would be made where necessary.

3.3.4 Problem setup

Before implementing the model in Fig. 3-9, the problem has to be defined in terms of controlled input to the plant, measured output variables, performances and also disturbances.

Based on the simulink model, 7 controlled volumes would be defined. The pressure and the temperature at these controlled volumes are taken as states for a total of 14 states. Additionally, the turbocharger speed is also taken as a state. In total, the model has 15 states.

Control inputs are the VGT and the flow separator valve. To simplify the model and not have to take the actuator dynamics into consideration, the effective area of the turbine and the valve are chosen as the controlled inputs.

Measured outputs are taken to be the intake manifold pressure, the air mass flow before the throttle and the engine speed. These are chosen as measurement outputs because they are readily available outputs in most engines today.

Performance outputs are chosen to be the difference between exhaust pressure and intake manifold pressure. This is to control pumping losses, which affect fuel consumption. An intake pressure map can be developed from engine testing and used as a reference based on engine rpm. The error between the actual intake pressure and the reference value is also chosen as a performance output. Similar error can be taken as performance output as well if maps are obtained for the mass airflow or power output. To minimize cost, the control inputs are also taken as performance outputs. This minimizes the effort required to achieve performance. Since the main objective of the expansion-cooling concept is to keep the temperature as low as possible, or around ambient temperature, one possible performance output could be the error between intake manifold temperature and ambient. The possibility of having the intake manifold temperature equal the ambient temperature at maximum performance needs to be experimentally tested before such a controller can be implemented.

Finally the throttle valve effective area is treated as a disturbance as it affects all the parameters and is activated by the driver.

Once all signals are designed in simulink, weighing factors are added to the input and output lines to guide the way the controller is designed. The weighing factors are tuned to attained performance requirements.

3.3.5 Observation from controller design

The objective of this project was to apply the tools learned in a controls course to a specific application that required the use of a controller. Expansion-cooling was chosen as the core of the project. Difficulties were found when trying to model the system. All differential equations pertaining to the realization of the state space model were obtained and presented in the modeling section. Drawbacks were the lack of experimental data, which were necessary to parameterize a lot of the components especially, the turbocharger. No assumptions were made because the equation contained too many variables and no simple assumption could have helped.

Many designers assume that the temperature is a slowly varying parameter. Therefore, to simplify the equations, its differential is set to 0. Since the objective was to control temperature, this was not an option. The intercooler effectiveness was shown to be a function of the mass flow rates and the temperatures of the hot charged air and cooler medium. A constant effectiveness would not be appropriate.

One aspect of the turbocharger was not even analyzed in the above model for simplicity, but surge protection is one main research goal for many control system designers.

3.4 Intercooler testing

From the feasibility study, it seemed as if it would be very difficult to prove that the turbo-cool concept could become a reality with today's technology unless a high performance intercooler was available. After various discussions with engineers from well-known intercooler manufacturers, such as Spearco and Bell intercoolers, it was clear that no detailed information about the effectiveness of any of their products would be made available to the public. It was judged necessary to run an experiment in the lab and compare the performance of two types of intercoolers, one of which would be selected for future test. Two types that are available for application in the automotive industry were chosen for the comparison, air-to-air versus liquid-to-air.

As their names imply, the cooling medium is air for an air-to-air intercooler type and liquid for the liquid-to-air type. Liquid-to-air intercoolers generally have greater thermal efficiency at low speeds, have a higher rated efficiency, and are more compact; however, they are usually more expensive than air-to-air intercoolers because additional components such as a liquid pump and a heat exchanger is required to cool the liquid. Air-to-air intercoolers are generally simpler to install since no additional components are needed. They also achieve greater thermal efficiency at high speeds, and cost less; however they may take up more space within the engine compartment due to the large surface area needed for effective heat convection to occur. In this section, a comparative test is presented where the effectiveness of both intercooler types are weighed against each other. Of course, the more effective one is, the most favorable it is for expansion cooling application.

This task was assigned to Maleshia Jones, a mechanical engineering student from the University of Maryland, Baltimore County. The experiment took place at Stony Brook University and was fully funded by the Alliance for Graduate Education and the Professoriate (AGEP) as part of their Summer Research Institute (SRI) program.

3.4.1 Component selection

The intercooler used for the preliminary engine tests was a Spearco air-to-air intercooler (cf. Fig. 3-11). The later was automatically selected as the air-to-air intercooler to be tested. The specifications on the Spearco intercooler are as listed in Table 3-2.

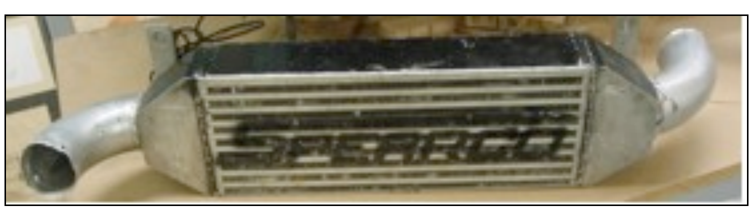

Fig. 3-11. Spearco intercooler

Table 3-2. Spearco intercooler specifications

Width	3″
Length	17″
Height	6″
Inlet & outlet pipe Diameter	2-1/2″

A PWR Performance Products liquid-to-air barrel intercooler was selected as the second intercooler. The decision was made based on the fact that the intercooler was available as a kit (cf. Fig. 3-12). The barrel kit included an electric water pump with 3/4" barbed fittings, a small radiator with fan and billet inline filler with cap. The specifications on the kit are listed in Table 3-3.

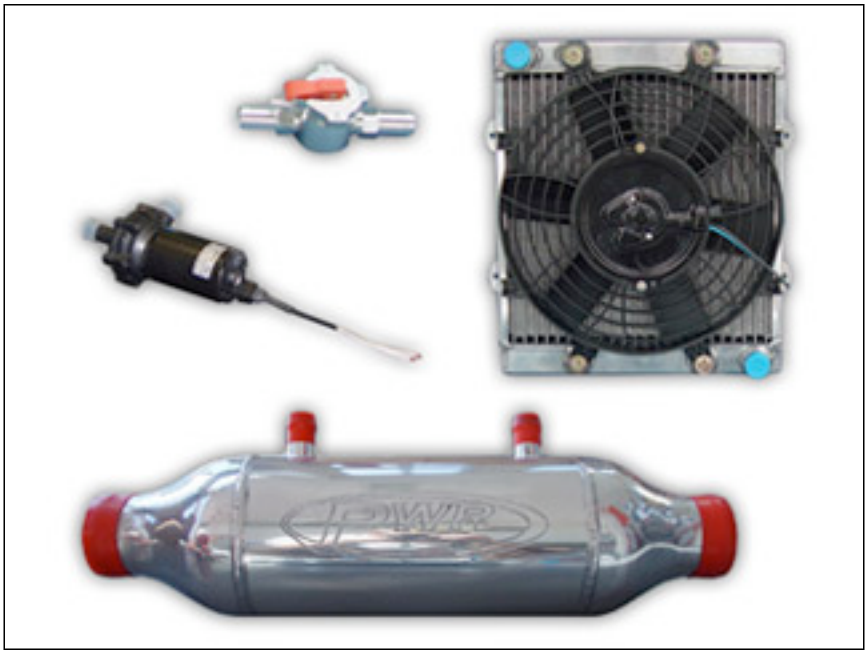

Fig. 3-12. PWR liquid-to-air intercooler kit

Table 3-3. PWR 2410041 liquid-to-air intercooler kit specifications

Length	15.75"
Inlet & outlet Hose Fittings	3/4 Push On
Inlet & outlet pipe Diameter	2.25"
Inlet & outlet Hose Fittings	3/4 Push On
Pressure Drop @ 7PSI	1.1 PSI
Radiator Dimensions	L=11.5" W=10" H=1.5
Dimensions	4" x 10"
Rated power	300 HP
Rated flow	390 CFM

3.4.2 Testing equipment

A test bench that would mimic the behavior of a real automotive turbocharging system was developed to gather data for intercooler effectiveness calculations. A Craftsman leaf blower replaced the turbocharger compressor, which usually provides the mass flow of air. The leaf blower however does not compress the air, therefore does not raise the pressure or the temperature. A space heater was used to solve the temperature problem. Although the pressure would not be high enough to replicate the automotive application, this was the only economical solution the lab could afford for the experiment and was judged acceptable for the comparative evaluation. The test bench was also outfitted with an Omega® 4-input thermometer plus data logger, a handheld digital manometer to measure gage pressure and a general purpose industrial air velocity/temperature transmitter/indicator to calculate mass flow rate. The specifications for the listed equipment are reported in Table A-6 through A-8 in the appendix.

3.4.3 Testing procedure

A large fan was set in front of the air-to-air intercooler to substitute for the flow of air that would be seen in a moving vehicle. Although no fan was necessary for the liquid-to-air intercooler itself, the small radiator that was part of the kit was equipped with its own. When starting the experiments, after powering the large fan for the air-to-air intercooler or the pump and the small fan for the liquid-to-air intercooler, the leaf blower and the space heater were powered. For each experiment, after approximately ten minutes, the intercooler inlet temperature would stabilize at around 125°F. The inlet temperature could be controlled by the amount of heat generated by the space heater. At maximum speed, the leaf blower could only generate 5 lb/min. This flow is considerably smaller than what could be expected in reality but was judged acceptable for comparative results.

Temperature measurements of the charge air were taken at the intercooler inlet and outlet, and at the water inlet and outlet (only for liquid-to-air intercooler). Pressure sensor and flow meter readings of the air were taken at the intercooler outlet. The process was repeated several times but only one measurement for each parameter was recorded each time.

From the temperature readings, the effectiveness was calculated. From the pressure and temperature readings, the air density was calculated in order to compute mass flow rate based on air velocity.

3.4.4 Observations from intercooler testing

The air-to-air intercooler effectiveness was found to vary between 60 to 70 percent depending on the speed selected on the large fan. This would confirm the comment made earlier about high thermal efficiency at high speed. The liquid-to-air intercooler efficiency was found to exceed that of the air-to-air intercooler by about 20 percent. However, these numbers were only recorded based on the liquid-to-air intercooler itself as an independent component. In reality, when the overall efficiency of the liquid-to-air system was considered, taking the radiator efficiency into consideration, the efficiency dropped to values similar to the air-to-air system.

4. Conclusion

From the theoretical study conducted at Stony Brook University and published in 2006, turbo-cool was presented as a successfully proven theoretical concept. This concept would utilize variable geometry turbine technology to permit higher compression ratios and boost pressures. There would be no need for higher octane or for higher fuel air ratios to lower the mixture temperature. In addition, due to the lower operating temperatures of the system, a reduction in emission could be expected. Products such as CO and NO are formed during the combustion process in the very high temperatures. Their production rate drops as the temperature drops. In addition, the richer the mixture, the more emissions are anticipated. Therefore, leaning the fuel air mixture can control the formation of CO and HC. Over all, the turbo-cool system would bring an improvement in fuel economy by permitting higher compression ratios in new engine designs and higher boost pressures with leaner fuel mixtures for engine currently in use.

The original aim of the phase following the first publication was to physically prove the simulation results. This was the author's main reason for getting involved in the project. For that purpose, a prototype was put together and tested in order to collect baseline engine performance data with an aftermarket ECU. The turbo-cool unit was then installed on the engine and a second dynamometer test was run. The prototype's configuration was modified from the original arrangement to the simpler patent representation. Unfortunately, before any data could be collected with the turbo-cool unit installed, the VGT was destroyed due to a lack of proper lubrication and possibly high backpressure. Because testing was set back and the event also correlated to the author becoming the main research project manager, it was decided to go back to the drawing board and verify if the components selected for the prototype would satisfy a first law of thermodynamics feasibility study. This idea was also pursued because high-pressure variations were observed at the exhaust. This could have occurred because the exhaust turbine was not sized properly and could have led to the damages as well.

As a result from the feasibility study, it was concluded that the intercooler was the one component that could limit the concept's potential. The required effectiveness for the intercooler was found to be around 85 to 90 percent. Since the advertised effectiveness range for regular air-to-air intercooler varies between 65 and 75 percent, an alternative would be to make use of the higher effectiveness of liquid-to-air intercoolers.

Assuming that the proper intercooler would be found, the best result was chosen to be the setup that required the lowest compression ratio to keep the air temperature at all point in the intake system as low as possible. Also, attaining ambient temperature at the manifold was set as a goal in order to lower the intercooler effectiveness required.

When the best expansion setup (i.e. 1.26 expansion ratio - 2.1 compression ratio) was compared to a regular system with the same intercooler effectiveness as shown in Table A-5, an increase in air fuel ratio was observed for the same power output. This proved that the application of the expansion concept could result in an increase in fuel economy.

These calculations were computed for one particular rpm. With the transient behavior of an engine, a very efficient control system would have to be adopted to control the turbocharger turbine actuator and the bypass valve for the expander. For this purpose, the system was modeled in MATLAB using simulink, a platform for multidomain simulation and model-based design of dynamic systems. Although the groundwork was laid out, any advancement into the simulation would require access to experimental results. Only then, would the controller be designed. No actuator dynamics were taken into consideration and all input were directly linked to the area variation of the valves or turbine. Publications on actuator and sensor noise and sensitivity are common and would definitely be useful in finalizing the controller design. One main parameter that was needed for the model was the exhaust manifold temperature, which would be measured and parameterized in function of the mass flow at different set points. The main simulink model was started but was not continued, as it would not help without the necessary parameters or would not have any use if the required intercooler was not found. The new objective was then set: Evaluate and compare two of the most commonly used intercoolers.

For the comparative experiment, a test bench was developed to gather temperature and pressure drops, as well as mass flow rates. It was found that under the conditions of 125°F inlet charge air temperature, and mass flow of 5 lb/min, the air-to-air intercooler effectiveness typically ranged between 60-70%, whereas the liquid-to-air intercooler effectiveness ranged between 87-90%. However, the entire liquid-to-air system effectiveness ranged between 65-67%. The test bench operating conditions were different from real operating conditions of an internal combustion system but was judge acceptable for a comparative experiment. More importantly, the liquid-to-air test bench effectiveness was found to depend on the effectiveness of the small radiator that cools down the liquid temperature. Further study of the liquid cooled system effectiveness at real conditions of higher temperatures and pressures *and* equipped with a high-performance radiator should be conducted. Chris Whelan from WDL ltd. in the UK claims that his company regularly produces total system effectiveness of 80 to 85 percent. Partnership with companies such as WDL and Lotus could definitely accelerate the turbo-cool proof of concept. These final results agree with those found by the Lotus engineers. Future research should involve verification of maximum achievable overall liquid-to-air effectiveness, and collection of engine test data to not only verify the feasibility of the concept but to also develop the necessary controller.

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Appendix

Table A-1. Calculating mass flow required for 215 hp at 5700 rpm

RPM	**5700**
Expected Horse power [HP]	**215**
Ambient Temperature [F]	**70**
Air fuel Ratio	10.6*
Brake Specific Fuel Consumption [lb/(hp*hr)]	0.5*
Volumetric Efficiency	0.9*
Mass Air Flow [lb/min]	18.9
Manifold Boost Pressure 7 [psig]	**7**
Manifold Temperature [F]	100.8
Engine Displacement [ci]	**121.9**
Pressure lost in Intercooler [psi]	0.5*
Pressure lost in Pipes	0.5*
Pressure Compressor Outlet [psia]	22.7
Pressure Loss from Air Filter [psi]	1
Pressure Compressor Intake [psia]	13.7
Compressor Pressure Ratio	1.7

Table A-2. Results based on 1.44 turbine expansion ratio

Expansion Turbine Pressure Ratio	1.44	1.44	1.44	1.44	1.44
Desired Pressure at the intake [psig]	7.0	7.0	7.0	7.0	7.0
Desired Temperature [F]	60.0	62.5	65.0	67.5	70.0
Ambient Temperature [F]	70.0	70.0	70.0	70.0	70.0
Ambient Pressure [psia]	14.7	14.7	14.7		14.7
Calculation constant for turbine	0.1	0.1	0.1	0.1	0.1
Turbine efficiency	0.7	0.7	0.7	0.7	0.7
Temperature Turbine Intake [F]	98.3	101.0	103.7	106.4	109.1
Pressure at the turbine intake [psig]	16.5	16.5	16.5	16.5	16.5
Pressure outlet compressor [psig]	17.5	17.5	17.5	17.5	17.5
Pressure Ratio (Compressor)	2.4	2.4	2.4	2.4	2.4
Calculation constant for compressor	0.3	0.3	0.3	0.3	0.3
Compressor Efficiency	0.7	0.7	0.7	0.7	0.7
Temperature outlet compressor [F]	277.6	277.6	277.6	277.6	277.6
Intercooler Effectiveness	0.87	0.85	0.84	0.83	0.81
Temperature outlet intercooler [F]	98.0	101.1	103.2	106.3	109.4

Table A-3. Results based on 1.36 turbine expansion ratio

Expansion Turbine Pressure Ratio	1.36	1.36	1.36	1.36	1.36
Desired Pressure at the intake [psig]	7.0	7.0	7.0	7.0	7.0
Desired Temperature [F]	60.0	62.5	65.0	67.5	70.0
Ambient Temperature [F]	70.0	70.0	70.0	70.0	70.0
Ambient Pressure [psia]	14.7	14.7	14.7		14.7
Calculation constant for turbine	0.1	0.1	0.1	0.1	0.1
Turbine efficiency	0.7	0.7	0.7	0.7	0.7
Temperature Turbine Intake [F]	92.2	94.9	97.5	100.2	102.8
Pressure at the turbine intake [psig]	14.8	14.8	14.8	14.8	14.8
Pressure outlet compressor [psig]	15.8	15.8	15.8	15.8	15.8
Pressure Ratio (Compressor)	2.2	2.2	2.2	2.2	2.2
Calculation constant for compressor	0.3	0.3	0.3	0.3	0.3
Compressor Efficiency	0.7	0.7	0.7	0.7	0.7
Temperature outlet compressor [F]	262.6	262.6	262.6	262.6	262.6
Intercooler Effectiveness	0.89	0.87	0.86	0.84	0.83
Temperature outlet intercooler [F]	91.2	95.0	97.0	100.8	102.7

Table A-4. Results based on 1.26 turbine expansion ratio

Expansion Turbine Pressure Ratio	1.26	1.26	1.26	1.26	1.26
Desired Pressure at the intake [psig]	7.0	7.0	7.0	7.0	7.0
Desired Temperature [F]	60.0	62.5	65.0	67.5	70.0
Ambient Temperature [F]	70.0	70.0	70.0	70.0	70.0
Ambient Pressure [psia]	14.7	14.7	14.7	14.7	14.7
Calculation constant for turbine	0.1	0.1	0.1	0.1	0.1
Turbine efficiency	0.7	0.7	0.7	0.7	0.7
Temperature Turbine Intake [F]	84.1	86.7	89.3	92.0	94.6
Pressure at the turbine intake [psig]	12.6	12.6	12.6	12.6	12.6
Pressure outlet compressor [psig]	13.6	13.6	13.6	13.6	13.6
Pressure Ratio (Compressor)	2.1	2.1	2.1	2.1	2.1
Calculation constant for compressor	0.2	0.2	0.2	0.2	0.2
Compressor Efficiency	0.7	0.7	0.7	0.7	0.7
Temperature outlet compressor [F]	242.9	242.9	242.9	242.9	242.9
Intercooler Effectiveness	0.92	0.90	0.89	0.87	0.86
Temperature outlet intercooler [F]	83.8	87.3	89.0	92.5	94.2

Table A-5. Decrease in air fuel ratio

Pressure Ratio (Compressor)	2.1	1.7
Calculation constant for compressor	0.2	0.2
Compressor Efficiency	0.7	0.7
Temperature outlet compressor [F]	242.9	186.3
Intercooler Effectiveness	0.86	0.86
Temperature Intake Manifold [F]	70	86.3
Air fuel Ratio (Garrett)	11.2	10.9
Brake Specific Fuel Consumption [lb/(hp*hr)]	0.5	0.5
Volumetric Efficiency	0.9	0.9
Mass Flow rate at 5700 RPM	20.0	19.4
Horse Power Generated at 5700 RPM	215	215

Table A-6. Omega HH309A, 4-input thermometer plus data logger

Range	-328 to 2498°F
Resolution	0.1° between -200 and 200°C/F, else 1°C/F
Operating Conditions	32 to 122°F, 0% to 80% RH (32 to 95°F), 0% to 60% RH (95 to 122°F)
Battery	9 V (included)
Weight	Approx 8.8 oz
Typical Accuracy	±0.2% rdg +1°C
Sample Rate	3 seconds
Storage Temperature	-4 to 140°F
Dimensions	7.25” H x 2.5” W x 1.2" D

Table A-7. Omega HHP241-015G, Handheld Digital Manometer for Gage Pressure

Accuracy	±0.25% FS
Storage Temperature	14 to 122°F
Operating Temperature	-40 to 140°F
Temperature Effects	No temperature effect over the entire operating range
Engineering Units	8 user selectable (InH_2O, mmH_2O, mmHg, InHg, kPa, mBar, bar and psi)
Proof Pressure	3x full scale or 14 bar (200 psi) whichever is less (high side only on differential units)
Pressure Port	1/8 FNPT
Media Compatibility	Clean, dry non-corrosive gases
Battery	3 "AA" batteries (included)
Battery Life	>300 hours continuous
Dimensions	6.0" H x 3.0" W x 1.0" D
Weight	12 oz

Table A-8. Omega FMA1003R-V1, General Purpose Industrial Air Velocity/Temperature Transmitter/Indicator

Air Velocity Range	0 to 10000 FPM
Air Temperature Range	-40 to 250°F
Air Velocity Accuracy	1.5% full scale
Air Temperature Accuracy	0.5% full scale
Air Velocity/Temperature Probe	Stainless steel, ¼OD x 12" L
Air Velocity/Temperature Sensor	Three RTDs, 100 and 1000 Ω
Display	Backlit LCD, 1.25 x 2")
Air Velocity Analog Output	0 to 5 Vdc
Air Temperature Analog Output	0 to 5 Vdc
Air Velocity/ Temperature Probe Pressure	150 psig maximum
Operating Relative Humidity	Less than 80% RH without condensation
Sensor Probe Operating Ambient Temperature	-40 to 250°F
Electronic Case Operating Ambient Temperature	32 to 122°F
Alarms	High and low alarm voltage outputs, corresponding to air velocity
Power	15 to 24 Vdc, 200 mA
Case Dimensions	4.5" H x 3.5" W x 1.5" D
Weight	0.5 lbs