

**BRAKE SPECIFIC FUEL CONSUMPTION AND POWER ADVANTAGES FOR A  
TURBOCHARGED TWO-STROKE DIRECT INJECTION ENGINE**

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**ABSTRACT**

The University of Idaho has been developing a clean two-stroke engine using gasoline direct injection for the Clean Snowmobile Challenge. The major benefit of gasoline direct injection is that fuel is introduced into the cylinder after transfer ports have closed. With traditional two-stroke engines, the time available for fuel to enter the cylinder is fixed based on the geometry of the transfer ports, resulting in a shorter fuel delivery window, allowing fuel to short-circuit unburned out the exhaust, and ultimately limiting the amount of fuel that can be trapped in the cylinder.

Another benefit of direct injection is the separation of the scavenging and fuel flows. This separation allows the use of turbocharging to efficiently increase the mass of air delivered to the engine, reducing the drawback of short-circuited fuel. The results show that turbocharging and direct injection can successfully increase specific power and lower specific fuel consumption across the operating range for a high performance two-stroke engine. Power was improved 40% at the peak engine speed and a BSFC gain of 20% was measured at part-load.

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**DEFINITION OF TERMS**

ABDC: After Bottom Dead Center

AFR: Air-Fuel Ratio

ATDC: After Top Dead Center

BBDC: Before Bottom Dead Center

BDC: Bottom Dead Center

BHP: Brake Horse Power

BRP: Bombardier Recreational Products

BSFC: Brake Specific Fuel Consumption

BTDC: Before Top Dead Center

CA: Crank Angle

CARB: California Air Resources Board

CDI: Capacitive Discharge Ignition

CFD: Computational Fluid Dynamics

CO: Carbon Monoxide

CSC: Clean Snowmobile Challenge

CVT: Continuously Variable Transmission

EFI: Electronic Fuel Injection

EGR: Exhaust Gas Residual

EGT: Exhaust Gas Temperature

EMM: Engine Management Module

GDI: Gasoline Direct Injection

IDI: Inductive Ignition

MAP: Manifold Absolute Pressure

NO<sub>x</sub>: Oxides of Nitrogen

OEM: Original Equipment Manufacturer

PM: Particulate Matter

PSI: Pounds per Square Inch

RPM: Engine Speed at the Output Shaft

SAE: Society of Automotive Engineers

SDI: Semi-Direct Injection

SOI: Start of Injection

SPR: Scavenging Pressure Ratio

TDC: Top Dead Center

UHC: Unburned Hydrocarbons

UI: University of Idaho

VNT: Variable Nozzle Turbine

WOT: Wide Open Throttle

$\lambda$ : Lambda – Actual AFR divided by the Stoichiometric AFR

$\phi$ : Equivalence Ratio - Stoichiometric AFR divided by the Actual AFR

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## **1.0 INTRODUCTION**

The focus of this work is to continue the development of the University of Idaho (UI) Clean Snowmobile Competition (CSC) direct injection two-stroke engine with research and testing focused on combustion stability, fuel delivery strategies, and the results of forced induction through the use of a turbocharger.

### **1.1 SNOWMOBILES AND THEIR USAGE**

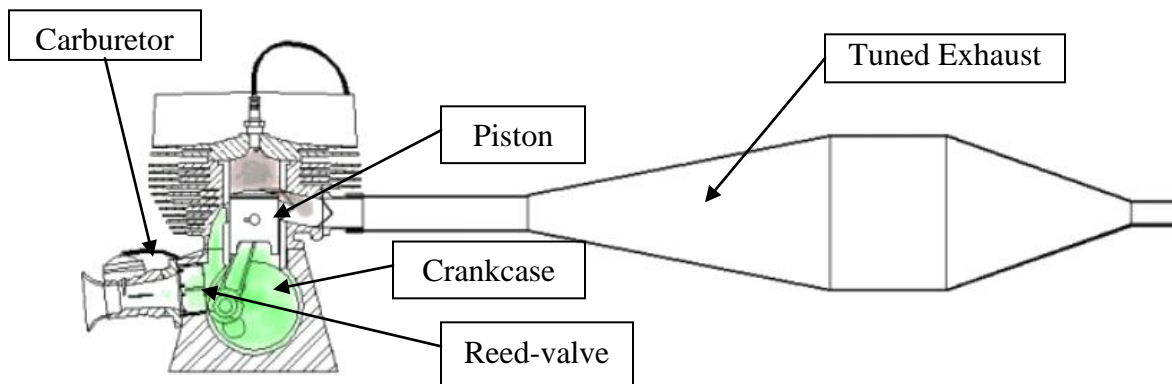
Snowmobiling is a popular method of winter transportation and recreation. Snowmobiles are used for several different purposes: utility, leisure, touring, and performance riding. Each of these types of riding can be conducted on groomed trails, deep snow, or mountainous terrain. The many possible combinations of snowmobile usage and riding conditions result in several different designs of snowmobile chassis and power plants. Current trail snowmobiles get low fuel economy, and have acceptable power output. Mountain snowmobiles also get poor fuel economy, and would benefit more than the trail snowmobiles from improved power. Both markets are in need of a reduction in emissions and noise to improve the image of snowmobiling as well as meet future regulations.

### **1.2 SNOWMOBILE ENGINES AND THEIR EMISSIONS**

Traditionally, snowmobiles have been powered by mid-sized (440 cc-700 cc) two or three cylinder two-stroke engines. Two-stroke snowmobile engines are low maintenance engines with unsurpassed power-to-weight ratios. Their low weight and high power make them well suited for over-the-snow travel. Furthermore, two-stroke engines have excellent cold start abilities down to temperatures as low as  $-40^{\circ}\text{C}$  [1]. Two-stroke engines also have a torque curve that is well suited for the belt-type continuously variable transmission (CVT) used in snowmobiles [1].

Typically two-stroke snowmobile engines are crankcase charged and loop-scavenged with exhaust-pipe tuning. The induction system is usually piston-port, rotary-valve, or reed-valve and the fuel metering is carburetion (fuel-air mixture into the intake), throttle body fuel injection (fuel injected after the throttle bodies), or semi-direct fuel injection (fuel injected

into the transfer ports). These engines also use a “total-loss” oil-injection system where the lubricating oil is either premixed with the fuel or injected into the air-fuel stream prior to entering the crankcase. The literature is full of alternative designs for charging, scavenging, fuel delivery, and ignition control, but none provide a lightweight and high-power combination like the simple two-stroke engine described in this paper [2, 3, 4]. Figure 1 is a schematic drawing of a carbureted, reed-valve, crankcase charged, loop-scavenged, and piston-controlled two-stroke engine with a tuned exhaust, typical of traditional snowmobiles.

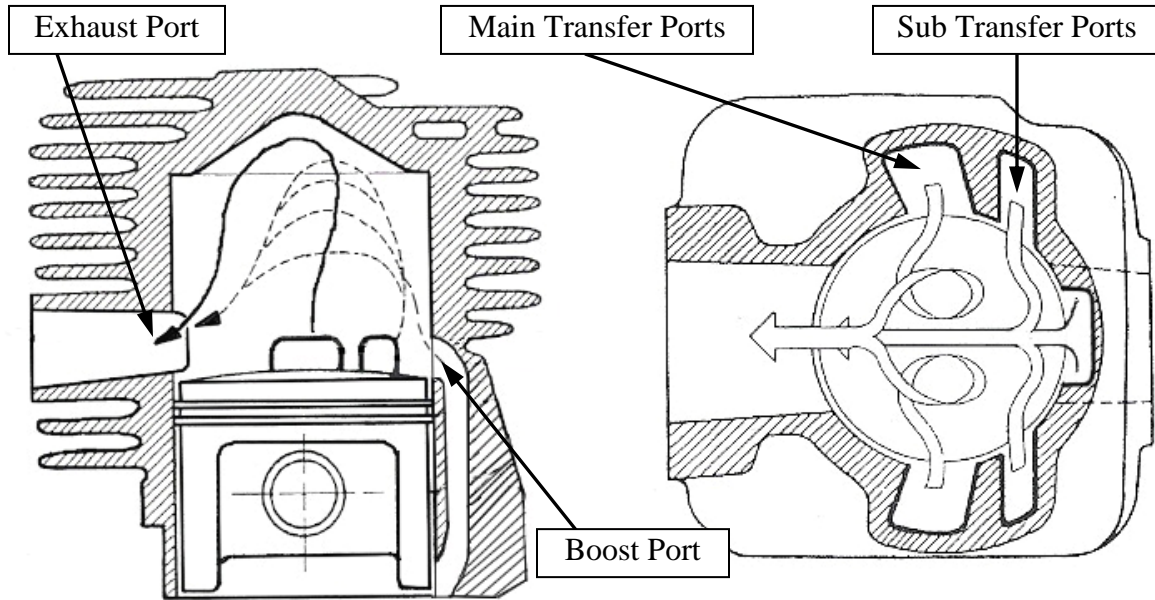


**Figure 1:** Traditional snowmobile two-stroke engine.

Since two-stroke engines do not have separate intake and exhaust strokes, a scavenging pump must be used to introduce the fresh charge into the cylinder to push the exhaust gases out during the scavenging process [2]. Two-stroke engines use one of the simplest forms of scavenging pumps referred to as crankcase scavenging. The bottom of the piston in conjunction with the crankcase is used for scavenging. The movement of the piston up and down produces the pumping action required to push the air-fuel mixture into the combustion chamber through the intake ports cut into the side of the cylinder. The geometry of the intake and exhaust ports determines the action of the scavenging flows used to displace the combustion products and provides a new intake charge.

The most widely used port layout is the Schnurle loop-scavenging system shown in figure 2 [3]. This type is used because the main scavenging ports are directed away from the exhaust ports and a piston with a flat top can be used [5]. The Schnurle system alleviates the

hot spots and allows for a more compact combustion chamber that leads to a more rapid and efficient combustion process [5].



**Figure 2:** Schematic of in-cylinder flows for a Schnurle loop-scavenged engine.

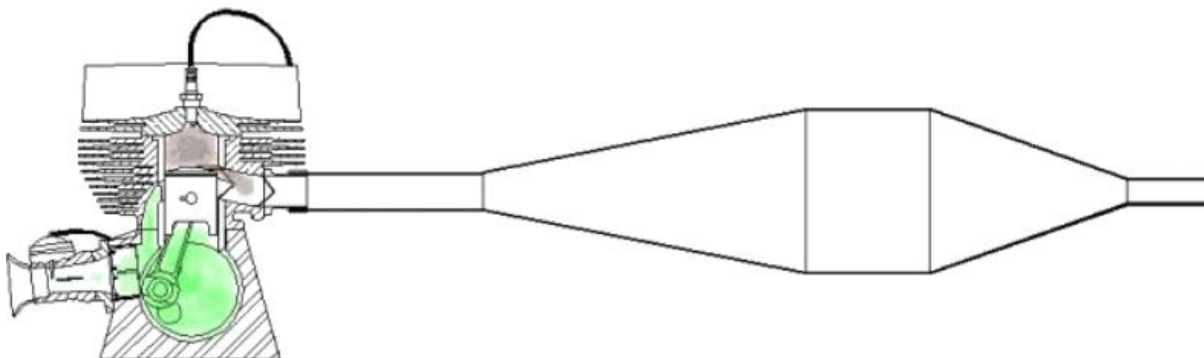
While the two-stroke engine is mechanically simple, the scavenging process is very complex. Because the intake and exhaust ports are open at the same time some unique problems arise related to intake and exhaust gas mixing. There has been extensive work aimed at optimizing the scavenging process through modeling and the use of computational fluid dynamics (CFD) [3, 5].

### 1.3 TWO-STROKE ENGINE CYCLE

The following description of the two-stroke engine cycle highlights some of the most important concepts of the gas-exchange process. The description is simplified and generalized [6]. Detailed descriptions can be found from Blair and Heywood [3, 5].

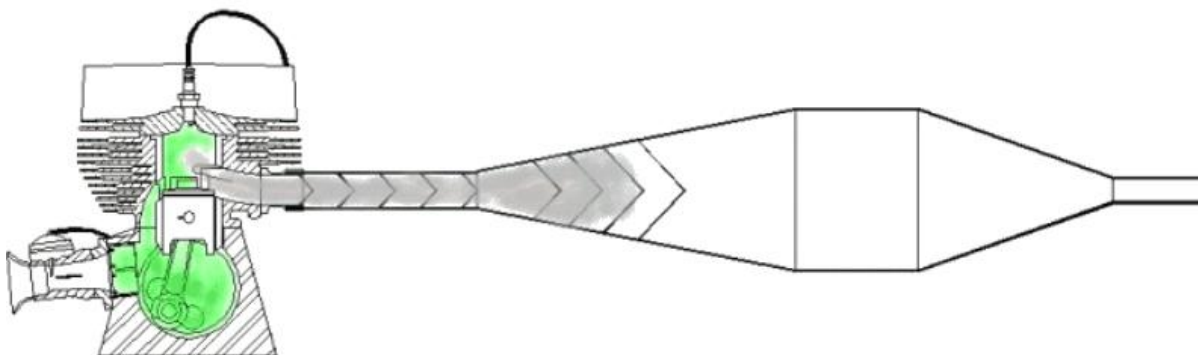
1. After ignition, there is about  $70^\circ$  crank angle (CA) of useful expansion and work. At about  $80^\circ$  before bottom dead center (BBDC), the piston uncovers the exhaust port and the combustion gases begin to evacuate the cylinder. The cylinder pressure drops

drastically and the negative pressure wave aids in pulling exhaust gases out. This is the end of the power stroke, figure 3.



**Figure 3:** The end of the power stroke and the beginning of blow-down.

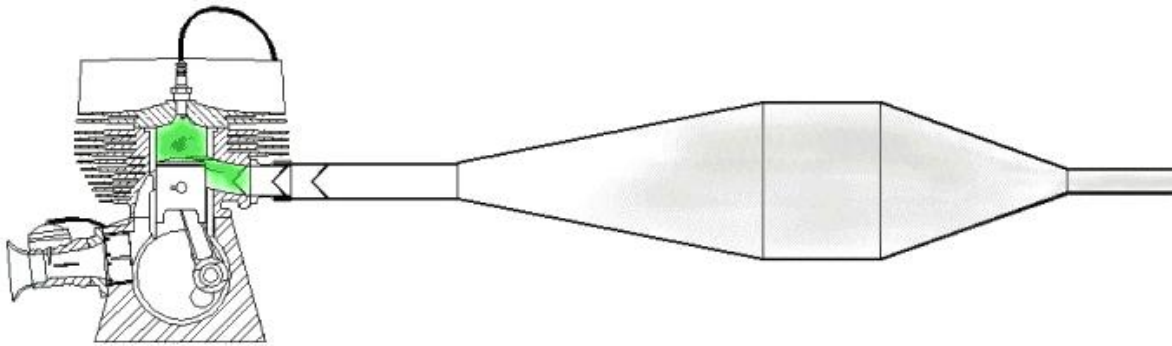
2. Some 10-20° later the intake ports open and a fresh air-fuel charge, compressed in the crankcase by the underside of the piston, begins to enter the combustion chamber. The incoming charge displaces and mixes with the exhaust-gas residuals. The tuned pipe continues to aid in pulling the cylinder contents out and some of the charge will be short-circuited figure 4.



**Figure 4:** Intake ports begin to open and scavenging begins while the tuned pipe continues to help evacuate combustion products.

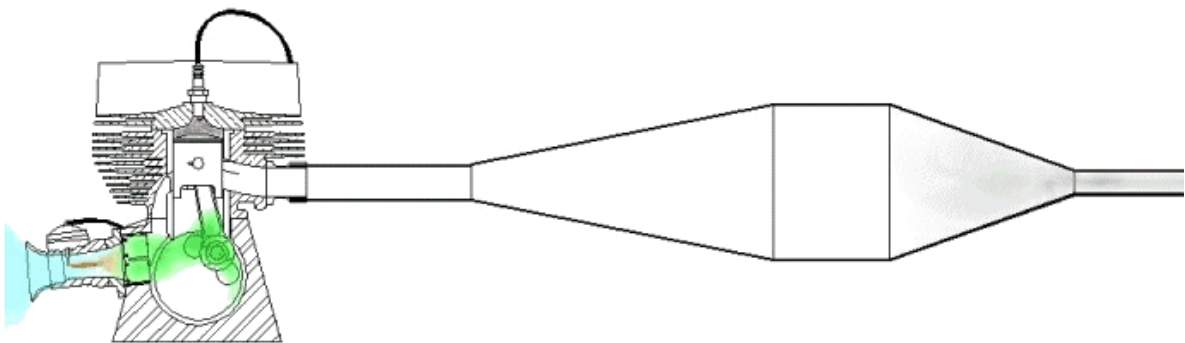
3. The scavenging process ends with both the crankcase and cylinder pressure close to the ambient pressure level once the inlet port is closed (about 60° after bottom dead center (ABDC)). Towards the end of the scavenging process, there can be a back flow of

charge and exhaust-gas residuals into the combustion chamber from the plugging pulse of the tuned pipe. The upward movement of the piston reduces the pressure in the crankcase and begins to open the reed valves, figure 5.



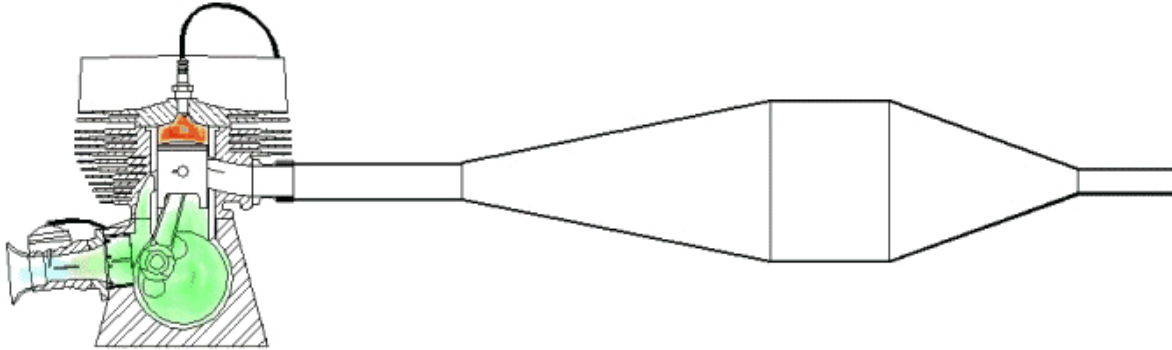
**Figure 5:** End of the scavenging process, pressure drops inside the crankcase, with the returning pressure wave in the exhaust providing the plugging pulse.

4. At about  $80^\circ$  ABDC, the exhaust port is closed, and the upward movement of the piston begins to compress the freshly scavenged charge. The pressure in the crankcase continues to drop until the pressure is low enough to draw the next cycle's fresh charge in through the reed valves, figure 6. A ram tuning effect similar to the one in the exhaust system can be designed into the intake system in order to trap more mass in the crankcase [3].



**Figure 6:** Exhaust ports close, compression begins, and a fresh intake charge begins to enter the crankcase.

5. Ignition typically occurs within the range of  $10\text{-}40^\circ$  before top dead center (BTDC). The burning fuel creates a pressure rise in the cylinder that is timed to peak around  $10^\circ$  after top dead center (ATDC) for maximum torque output. Work is done by the expanding gases on the piston, the power stroke, until the exhaust port opens and the cycle repeats, figure 7.



**Figure 7:** Ignition occurs and the power stroke begins.

It is evident that the two-stroke engine gas-exchange process is complex due to the in-cylinder flows as the fresh charge mixes and displaces the combustion products. During the scavenging process when the piston is near bottom dead center (BDC) the intake and exhaust ports are both open. This results in some fresh charge flowing directly into the exhaust system. The loss of the incoming fuel directly to the exhaust is called short-circuited fuel. The short-circuited fuel is the single largest contributor to poor fuel economy, excessive unburned hydrocarbon (UHC) emissions, and particulate matter (PM). Short-circuited fuel can account for a loss of as much as 50% of the supplied fuel, especially during off-design speeds and loads. However, the CVT used for snowmobiles keeps the engine operating conditions close to the designed engine speeds and loads, limiting the short-circuited fuel to around 10-30% [6].

Although the largest portion of UHCs is from short-circuited fuel, the combustion process also produces unburned-hydrocarbon emissions. Specifically, flame quenching, crevice-volume filling, and incomplete combustion or misfires are mechanisms for combustion related UHCs [4]. Flame quenching occurs when fuel adheres to cylinder walls



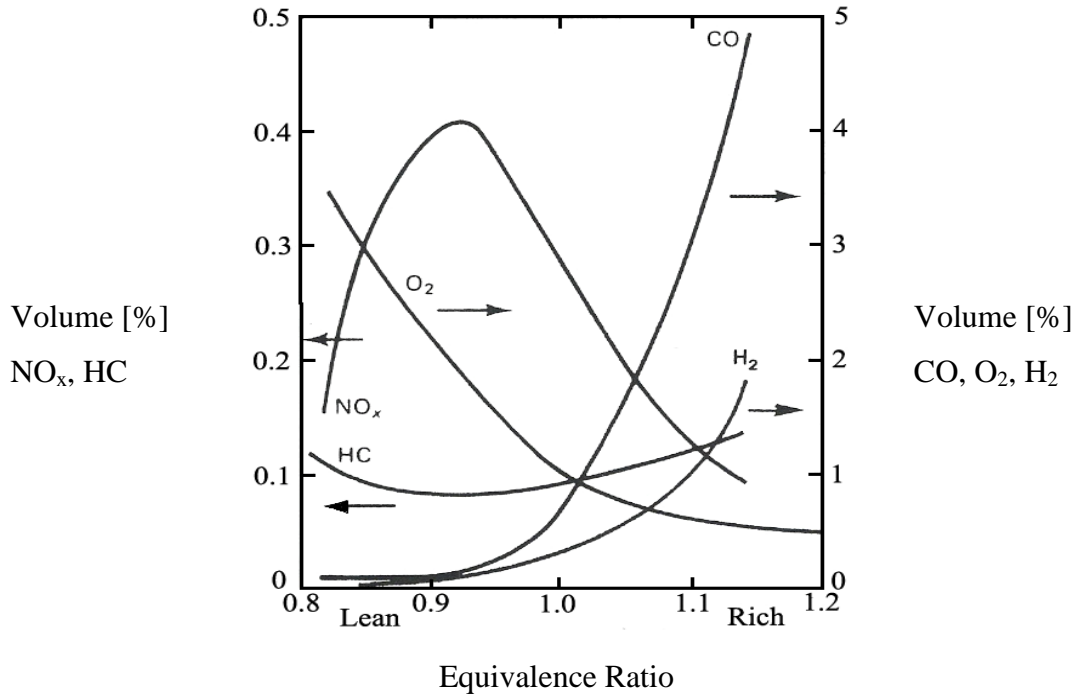
and combustion can only happen through surface evaporation and subsequent oxidation. Hydrocarbon emissions from flame quenching are many orders of magnitude less than short-circuited UHC [4]. Hydrocarbon emissions from crevice-volumes account for approximately 2.5 to 4% of total UHC emissions for a traditional two-stroke engine [4].

Incomplete combustion and/or misfire can be a major source of UHC emissions, especially during low load and engine speeds. Low delivery ratios, the amount of air/fuel mixture delivered from the crankcase through the transfer ports, associated with off-design loads and speed lead to large amounts of exhaust-gas residuals [4]. With the excessive exhaust-gas residuals (EGR), the combustion process becomes unstable to the point where several cycles can occur with partial or no heat release [5]. The partially combusted charge is then lost to the exhaust system.

Two-stroke engines are also known to have high carbon monoxide (CO) emissions. The formation process for carbon-monoxide in two-stroke engines is exactly the same as that of other engines. Extensive discussions on CO formation can be found from Blair and Stone [2, 5]. Simply stated, CO formation results from operating an engine fuel-rich. The lack of oxygen in the combustion chamber prevents the carbon from fully oxidizing to carbon dioxide and CO forms. Two-stroke engines are operated fuel-rich for three reasons: (1) excess fuel cools the piston crown and prevents engine seizure; (2) it increases power output due to a maximum heat release occurring just rich of stoichiometric air-fuel ratios (AFR), and (3) for rapid transient response [4]. The emission trend for spark ignited engines is shown in figure 8 [2]. The equivalence ratio ( $\phi$ ) is the stoichiometric air-fuel ratio divided by the actual air-fuel ratio. Later in the text this ratio is inverted and is referred to as lambda ( $\lambda$ ). Equation 1.3.1 shows the relationship of the equivalence ratio and lambda.

$$\phi = \frac{1}{\lambda} = \frac{AFR_{stoich}}{AFR_{actual}}$$

**Equation 1.3.1**



**Figure 8:** Emission trends for spark-ignition engines for different equivalence ratios.

Contrary to common belief, two-stroke engine lubricating oil is not a significant source of UHC [4, 5, 7]. Two-stroke engines do have excessive particulate matter (PM) emissions due to the total-loss lubrication system. UICSC testing has shown oil consumption with a direct injected engine can be reduced approximately 50% since the fuel does not dilute or remove the oil from the moving components. With improvements in oil refining, changes in the type of oils, along with recent technologies, manufacturers have been able to lower PM emissions to levels below the EPA 2004-2012 off-road Tier 3 regulations of 0.4g/kW-hr [7].

#### 1.4 TWO-STROKE ENGINE EXHAUST SYSTEMS

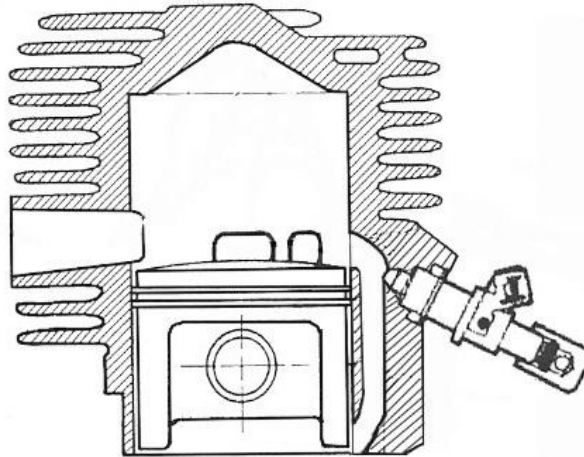
It was noted earlier that a tuned pipe is paramount to a high-performance two-stroke engine's performance. A well-tuned exhaust system aids in scavenging and reduces short-circuited fuel. Tuned exhaust systems are highly speed and load dependant, with designs ranging from one common pipe to individual pipes for each cylinder. Multiple pipes can increase power output, but they are costly, have a narrower power band, and are only found on high-performance engines. Tuned exhaust pipes consist of a header pipe that is connected

to the exhaust outlet of the cylinders by a branch-pipe or “y-pipe”. Following the header pipe is a diverging cone, straight section, and a converging cone. Just after the converging cone there is section of straight pipe called the stinger. In some designs the stinger will begin inside the converging cone for noise reduction.

The pipe works by reflecting pressure waves in the exhaust system [5]. When the exhaust port opens, a pressure wave created by combustion enters the divergent cone and a negative pressure wave is reflected back into the cylinder, which aids in removing exhaust gases from the cylinder and pulling the fresh air-fuel mixture in. When the original pressure wave reaches the converging cone at the end of the pipe, a positive pressure wave is reflected back towards the cylinder. The positive pressure wave, often referred to as the plugging-pulse, forces some of the exhaust contents just outside of the exhaust port back into the cylinder, along with some of the short-circuited fuel. Tuned pipes are designed to maximize charging efficiency and to lower brake-specific emissions and fuel-consumption at particular engine speeds and loads [5]. When the engine is operated at off-design engine speeds and loads the plugging-pulse is mistimed, and there is a significant loss in torque and an increase in emissions and fuel consumption.

## **1.5 FUEL DELIVERY STRATEGIES**

Recently, manufacturers have been developing new technologies for two-stroke engines to meet EPA regulations of 70g/kW-hr HC and 275 g/kW-hr CO [11]. The latest commercially available development for snowmobile two-stroke engines has been semi-direct injection (SDI), or boost-port injection. Bombardier introduced a SDI system, 2-TEC®, in 2004 and Polaris introduced their SDI system, Clean fire®, in 2006. These engines use low-pressure fuel-injectors, similar to those used in automobiles, to inject fuel into the boost-port. Figure 9 shows one possible location for an injector for an SDI application.



**Figure 9:** One possible arrangement for a SDI cylinder.

Semi-direct fuel injection can significantly reduce short-circuiting, decreasing the five-mode weighted UHC emissions by about 50%. The five-mode test represents the typical duty cycle for snowmobile engines while in operation, Table 1 [1]. SDI engines are still operated under slightly fuel-rich conditions and the reduction in CO is not as dramatic, with approximately 30% less CO production [6].

Mode Point	Speed [% of Rated]	Torque [% of Rated]	Weighting [%]
1	100	100	12
2	85	51	27
3	75	33	25
4	65	19	31
5	Idle	N/A	5

**Table 1:** Five-mode weighted testing points for snowmobile engines.

It is evident that SDI technology has reduced UHC and CO emissions enough to meet the 2012 EPA regulations [8]. At CSC 2006, the SDI engine saw a dramatic improvement in fuel economy by achieving 19 miles per gallon (mpg) compared to the snowmobile powered by four-stroke engines that achieved 15 mpg [8]. This is directly related to the reduction of short-circuited fuel. These fuel economy readings were performed at the same time under the same conditions for all snowmobiles in the competition.

## **2.0 GASOLINE DIRECT INJECTION**

Due to its simple construction, high power-to-weight ratio, and low cost the two-stroke engine remains the engine of choice for snowmobile riders, outfitters, and manufacturers. Gasoline direct injection (GDI) promises to improve this engine platform further.

### **2.1 STRATIFIED AND HOMOGENEOUS COMBUSTION**

As discussed earlier, the two main undesirable side effects of the two-stroke engine cycle are: (1) mixing of the fresh air-fuel charge with the exhaust-gas residuals, and (2) the short-circuiting of the fresh charge during the scavenging process. It is essential that these problems be overcome if the two-stroke engine is to meet the new recreational vehicle emissions standards without increasing engine complexity or weight.

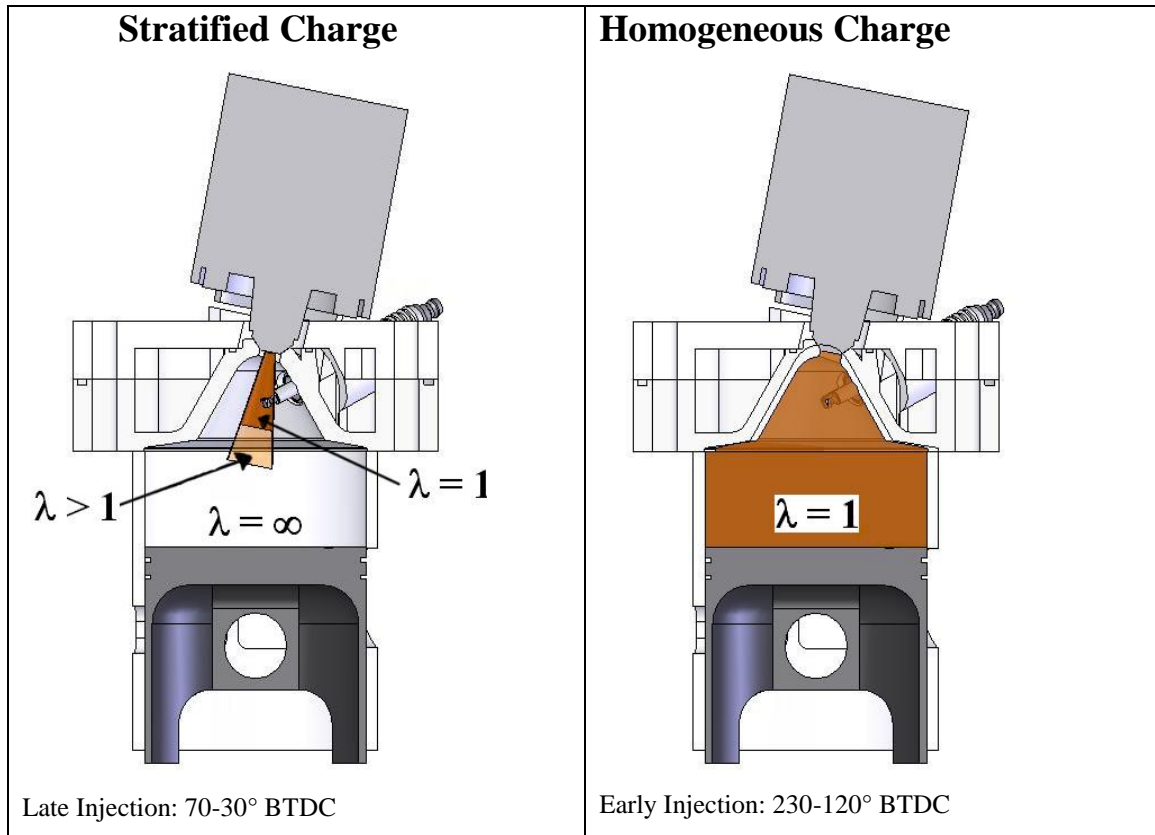
Over the past few decades a significant amount of work has been performed to overcome the two-stroke engine problems [3, 4, 5]. Of the concepts researched, GDI has the largest impact on two-stroke engine performance. GDI allows the fuel to be directly injected into the combustion chamber at an optimal time to promote efficient combustion. The obvious advantage of a GDI engine is that it delays the introduction of the fuel into the scavenging process until later in the cycle, thereby limiting the amount of fuel that can be short-circuited. GDI is also known to improve cold start reliability [2]. GDI provides a more ignitable mixture at the spark plug compared with a carbureted or SDI engine that requires the scavenging flows to deliver fuel to the spark plug area.

Another advantage is that the GDI system and combustion chamber can be designed to produce a non-uniform fuel distribution inside the combustion chamber, allowing the engine to operate in two different modes of combustion: stratified and homogeneous. The ability to use two different types of combustion makes GDI especially attractive to two-stroke engine designers. A stratified engine promises to significantly reduce fuel consumption, CO, and UHC emissions [2]. Stratified combustion is often referred to as a diffusion flame, which consists of a readily ignitable mixture near the spark plug with a weaker, often non-ignitable, mixture in the rest of the combustion chamber [2]. This is done

so that the power output of the engine can be controlled by varying the fuel supply rather than throttling the engine. The diesel cycle is the classic example of a stratified engine.

Stratified combustion in a two-stroke GDI engine occurs when fuel injection is timed late in the cycle and ignition is delayed until there is a fuel-rich mixture surrounding the spark plug. The rich condition occurring at the onset of combustion provides a reaction rate high enough to initiate combustion. The flame front, occurring at the interface between the fuel and oxidant, moves outward from the spark plug gap, burning the ever-leaner mixture until combustion can no longer be sustained. Stratified combustion can eliminate poor idle quality and reduce low load operation [2].

A GDI system can also create a homogeneously charged combustion chamber. For the GDI engine, homogeneous operation is accomplished when fuel is injected early in the cycle so there is time for the fuel to better atomize and mix with the fresh scavenging air. Homogeneous combustion is used for medium to high loads and is accomplished in two ways. The first is during medium loads, where the fuel is injected early, and an overall trapped lean air-fuel ratio with some EGR is desired to limit heat release [8]. The second is used during high loads, where the goal is to maximize air utilization and to operate the engine with a stoichiometric or slightly rich condition to maximize power [8]. The timing of the fuel injection, while much earlier than stratified injection, must be late enough to avoid any fuel from mixing with the scavenging flows that short-circuits the fuel [3]. Figure 10 shows the difference between in-cylinder lambda values for a stratified and homogeneously charged engine [6].



**Figure 10:** Schematic showing the difference between stratified and homogeneous lambda values and charge stratification.

The application of GDI to two-stroke engines has been successful in the outboard industry. Systems such as the Orbital air-assist system used by Mercury Marine and the E-TEC GDI system from BRP, have been shown to meet the California Air Resources Board's (CARB) "Ultra low" 3-Star emissions for outboard marine engines [10]. For this industry, the two-stroke GDI engines have been shown to have better fuel economy, less oil consumption, similar UHC+NO<sub>x</sub> (unburned hydrocarbons plus nitrogen oxides), and significantly less CO emissions than their four-stroke counterparts [7]. However, these GDI systems were developed specifically for outboard engines and their performance requirements, which differ significantly from the requirements and specifications of a snowmobile engine. The differences are the higher speeds, high specific power, and intense scavenging flows found in snowmobile engines.

## 2.2 SPECIAL CONSIDERATIONS FOR SNOWMOBILE GDI ENGINES

The main reason why development of a GDI system for a high-performance two-stroke engine is difficult is because these engines operate at significantly higher engine speeds with greater fuel demands. Snowmobile engines operate at speeds around 8000 rpm with specific power outputs of nearly 150 kW/liter, compared to outboard engines with rated speeds of 6000 rpm and specific power outputs of 70 kW/liter. This requires more fuel per injection event than the outboards. At peak loads, a very short period of time (1.2-3.5 ms) exists where a significant amount of fuel must be injected and fully atomized. The large peak load fuel requirements also pose a challenge for low load and idle fuel requirements. An injector nozzle designed to deliver high quantities of fuel quickly, typically has poor light load and idle fuel-spray qualities [9]. A two-stroke GDI engine at full power can use in excess of 40 kg/hr of fuel while at idle only need 0.7 kg/hr, leading to the difficult task of designing a precision nozzle capable of delivering high flow rates and precise fuel metering.

Beyond the performance requirements, another obstacle is the economic impact associated with producing a high-performance GDI two-stroke engine. Industries where two-stroke engines are attractive require low production costs. Even small increases in part costs or research and development costs may impact the overall production costs significantly, eliminating the low-cost production advantage the traditional two-stroke engine has over a four-stroke engine. Increases in cost passed on to the consumer will lead to higher performance expectations through increased fuel efficiency and/or power output.

With the goal of producing a clean, powerful, and fuel efficient two-stroke snowmobile in mind, the University of Idaho is exploring the feasibility of a single-fluid and spray-guided GDI turbocharged two-stroke engine for use in a snowmobile.



### 3.0 UNIVERSITY OF IDAHO GASOLINE DIRECT INJECTION SYSTEM

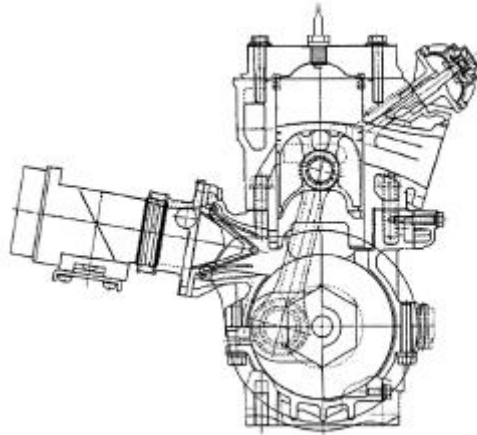
#### 3.1 BASELINE ENGINE

The baseline engine used for testing was a 2006 Rotax 593 cc two-stroke with variable exhaust valves and a tuned exhaust. This engine was chosen for two reasons: first, it represented a standard engine size for two-stroke snowmobile engines; and second, it was the maximum allowable two-stroke displacement for engines competing in the Society of Automotive Engineers (SAE) CSC competition [11]. Table 2 shows the specifications of the engine. [13].

Engine Type	Two-Stroke, Liquid Cooled
Number of Cylinders	2
Bore	72 [mm]
Stroke	73 [mm]
Displacement	593 [cc]
Compression Ratio	6.3 (Trapped)
Intake Type	Cylinder Reed Valve
Scavenging Type	Loop (Curtis-Type)
Rated Power	82 [kW]

**Table 2:** Rotax 593 cc Baseline Engine Specifications.

Figure 11 shows a cross section of an engine similar to the one used for research [12]. The cross-section also shows the exhaust valve above the exhaust port. The exhaust valve is a guillotine that changes the height of the exhaust port based on rpm, load and cylinder pressure. At lower engine speeds and loads, the valve is in the lower position to change the tuning of the pipe and improve part-load performance.



**Figure 11:** Cross section of typical two-stroke engine.

### 3.2 E-TEC DIRECT INJECTION SYSTEM

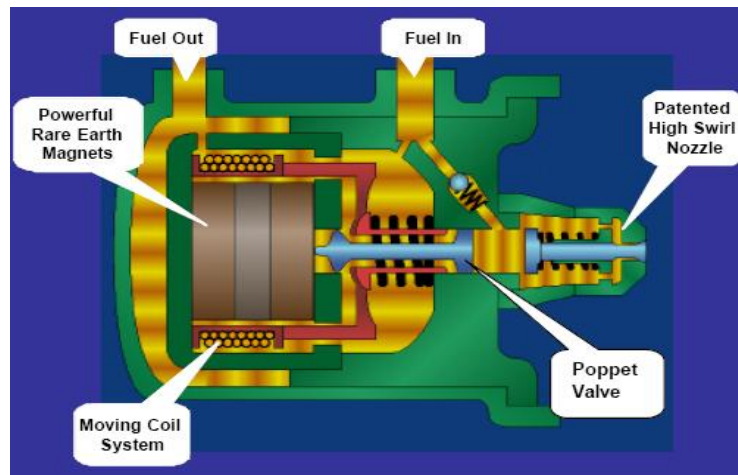
After researching many GDI systems the University of Idaho chose to develop a single-fluid and modulated GDI system. Modulated systems have the fewest components, good fuel atomization, low-pressure fuel supply, and can adapt to any number of cylinders or engine sizes [10]. The turbocharged research engine uses the E-TEC single-fluid and modulated GDI system.

The most appealing features of the E-TEC system were:

- The system was designed with two-stroke engines in mind
- It is easily adapted to any number of cylinders or displacements
- The injectors have high fuel flow and speed potential
- Very precise fuel metering is possible
- Few components are required to make the system work
- A significant amount of information about the system was available

The E-TEC injectors are second generation electromechanical injectors developed by BRP. These injectors are 25 percent lighter, and have 50 percent fewer moving parts than the previous FICHT injectors. The E-TEC injectors are vastly superior to the FICHT injectors

because they have faster operation, higher fuel flow rates, higher fuel pressures, and better fuel atomization [13]. Figure 12 shows a cutaway of the E-TEC injector [13].



**Figure 12:** Cutaway schematic of the E-TEC injector.

### 3.3 E-TEC SUPPORTING ARCHITECTURE

The adaptation of the E-TEC injectors requires a new electrical system. This includes the stator, flywheel, wiring harness, engine management module (EMM), all of which needed to be compatible with the 55 volts required to operate the injectors. The EMM must include an additional feature to drive and control the injectors. Electrical system descriptions can be found in the thesis by Nathan Bradbury [6].

### 3.4 COMBUSTION DISCUSSION

As described earlier, the short-circuiting of fuel and the mixing of the fresh air-fuel mixture with the exhaust-gas residuals have the largest negative impact on two-stroke engine fuel economy and emissions. While SDI systems have made significant improvements, a further reduction of emissions and an improvement in fuel economy can, and must be achieved if two-stroke engines are to have continued use in recreational vehicles. Gasoline direct injection (GDI) can greatly reduce short-circuiting as well as lessen the effects of charge/exhaust gas mixing.

Because most of the scavenging takes place when the piston is near BDC, an engine scavenged with only air would not lose any fuel during scavenging. Rather than rely on the transfer ports to bring the air-fuel mixture into the combustion chamber, the fuel is injected directly into the cylinder when the exhaust ports are nearly closed or completely covered, thereby eliminating the short-circuited fuel. Even though direct injection is considered the best technology available, as Blair describes, not many analytical or experimental studies have been carried out on gasoline direct-injected two-stroke engines.

It is evident from the few commercially available GDI two-stroke engines that it is difficult to develop. It is the desire to maintain mechanical simplicity, low cost of manufacture, high-power output, and low emissions that impedes the development of GDI two-stroke engines.

Two-stroke gasoline engines operate at much higher engine speeds than lower speed diesel engines that also use direct injection. This is another factor limiting the development of GDI in gasoline two-stroke engines. As engine speed increases, the amount of time available to inject the fuel into the combustion chamber decreases. Coupled with the fact that the compression part of the cycle is very short, designing injectors to meet the fuel requirements of high power output two-stroke engines has been difficult. With development of the E-TEC system by BRP, the opportunity for research of two-stroke direct injection engines became possible.

## **4.0 TURBOCHARGING A TWO-STROKE ENGINE**

This section describes turbochargers and how they are used to improve the performance of internal combustion engines. Specific design considerations that can significantly affect overall system operation for two-stroke engines are discussed. There is a description of how to select the proper turbocharger based on the desired performance and practical applications. Finally, there is a description of the turbocharger and system layout for the UI GDI engine. The design and operational goals for the GDI turbocharged engine were to increase the part-load operation efficiency, and use minimal boost pressure levels for reliability and durability while increasing the overall output and performance.

### **4.1 TURBOCHARGING INTRODUCTION**

The fundamental purpose of turbocharging is to increase available engine power while reducing specific fuel consumption. Power developed by an engine is determined by the amount of fuel that can be efficiently combusted in the cylinder. The amount of trapped air in the cylinders determines the amount of fuel that can be used. It follows that the power output of an engine can be increased by increasing the density of the delivered air (charge density). Turbocharging is one method often employed to achieve the desired increase in air density.

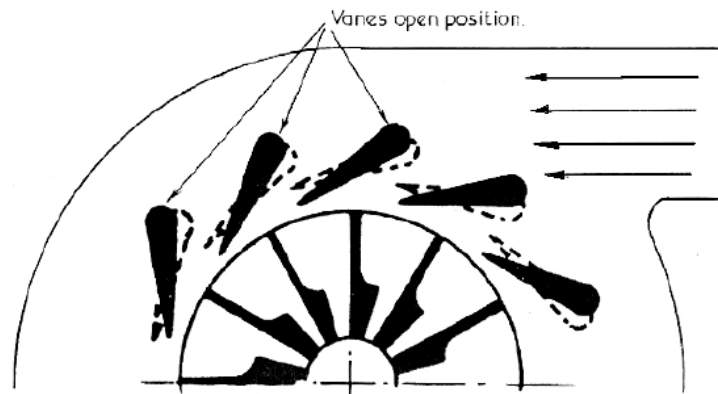
Turbochargers are most often used on four-stroke diesel engines in order to increase efficiency and fuel economy. They are also used in high performance applications with four-stroke gasoline engines. There are almost no applications of original equipment manufacturers (OEM) producing turbocharged two-stroke engines. Turbochargers utilize a turbine that uses the energy in the exhaust gas to drive a compressor, via a common shaft, to raise the inlet air pressure. The pressure rise of the intake air increases the density, and therefore the mass, of air delivered to the engine. Turbochargers usually feature a radial flow turbine connected to a centrifugal compressor. The turbine side consists of an inlet volute or scroll, nozzle vanes (may not be present), a turbine housing, and the turbine wheel. As exhaust gases enter the volute it accelerates radially inwards towards the turbine. The nozzle vanes, if present, further accelerate the flow. As the exhaust gases flow towards the turbine

they accumulate more kinetic energy. The high velocity and high temperature exhaust gases then enter the turbine wheel. The gases expand through the turbine wheel where energy is extracted and transferred to the turbocharger shaft. The gases exit the turbine housing axially, having been turned  $90^\circ$  as they traveled through the turbine housing [14].

Turbines are steady flow devices designed to operate at specific conditions. Turbines with fixed geometry cannot efficiently support an engine over a wide range of speeds [15]. Comparing turbines of different sizes highlights this problem. A large turbine will supply high pressures to the intake manifold during high speed operation, providing good fuel economy. However, at low engine speeds the mass flow through the engine will not provide sufficient energy to the turbine, resulting in low boost pressures and poor throttle response.

With a small turbine the inlet pressure can be boosted significantly providing good low-end torque and transient response. As can be expected, there is a drawback: at high engine speeds there will be too much energy in the exhaust, resulting in excessively high intake pressures or turbine over-speeding. To alleviate these problems a waste-gate can be used to bypass some of the exhaust gases past the turbine, or a blow-off valve can be used to release intake pressure if it is too high. Either method wastes useable energy, reducing fuel economy and thermal efficiency [15].

One proposed solution is to use a variable geometry turbocharger that allows the turbine nozzle area to be varied with engine speed and load. Variable nozzle turbines (VNT) offer many benefits, including improved fuel economy and throttle response [15]. The nozzle area of the turbine can be controlled by a single vane or by multiple vanes [16]. A schematic of a multiple vane VNT is shown in Figure 13 [16]. The vanes close when the exhaust flow is low in order to provide a small nozzle area. As exhaust gases speed up, the vanes progressively open to create a larger nozzle area. The movable vanes allow the turbocharger to provide the best characteristics of both small and large turbines: better throttle response and low-end performance coupled with high power output and improved fuel economy. A downfall of the multi-vane turbine is the added complexity that leads to higher manufacturing costs and an increased risk of component failure [16].



**Figure 13:** Schematic of a multiple vane VNT turbine.

The compressor side of a turbocharger consists of an inlet casing, compressor wheel, a diffuser with vanes (or vane-less), and a discharge volute or scroll. The shaft work created by the turbine is used to turn the compressor wheel at very high speeds, 120,000 rpm or greater. As the intake air enters the compressor housing axially through the inlet casing it is accelerated by centrifugal force [17]. The air then travels through the diffuser where it is slowed down and the kinetic energy of the air is converted to a static pressure rise. Finally, the compressed air flows through the volute to a pipe connected to inlet of the engine.

Unfortunately, the compression process causes an increased air temperature. The amount of temperature increase depends on the efficiency of the compressor at the operating conditions. An intercooler can be used to reduce the temperature of the inlet air.

Constant-pressure turbocharging addresses the issue of unsteady turbine performance. A large volume exhaust manifold is used to damp out the mass flow and pressure pulsations resulting from exhaust port opening. This essentially provides a steady flow at the turbine inlet. However, there is a loss of exhaust gas energy as the high-velocity gases exiting the cylinders mix with the low velocity gases in the exhaust manifold. Although this type of system operates more predictably, and more efficiently, the maximum amount of work available in the exhaust is not utilized [14].

## 4.2 TWO-STROKE ENGINE TURBOCHARGING

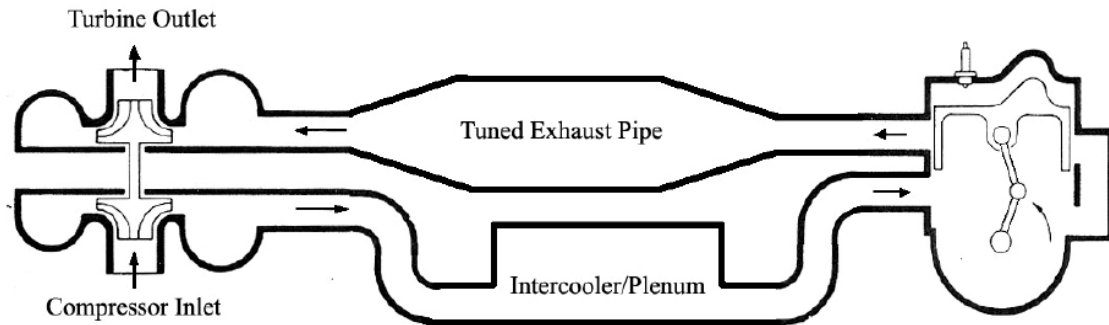
In two-stroke engines where the opening and closing of the ports are controlled by the piston, the exhaust ports close after the intake ports. Therefore, the trapped pressure is determined by the pressure of the exhaust system. Even if a large amount of air with a significant pressure rise is supplied to the engine, the trapped pressure will not increase without a proportional rise in backpressure. Fortunately, for simple two-stroke engines, the presence of a turbine in the exhaust system increases backpressure. However, a balance must be maintained. If the backpressure increases too much, scavenging efficiency and delivery ratio will be reduced, resulting in poor engine performance. If the backpressure is too low, high scavenging efficiency and delivery ratios may result, but the trapping efficiency will be reduced and performance will suffer. The pressure ratio of the exhaust backpressure to the intake pressure will be referred to as the scavenging pressure ratio (SPR) in this thesis.

A constant-pressure system was used because the engine is expected to operate with a broad engine speed range. Additionally, a constant-pressure system can retain the beneficial tuning characteristics of the exhaust pipe. When a two-stroke engine is turbocharged with a constant-pressure system, the turbine creates a larger backpressure in the exhaust manifold and an appropriately higher intake pressure is required [3]. During low loads the turbine may not have enough power to sufficiently increase the intake pressure, and an auxiliary compressor may be required. For the simple two-stroke engine, the crankcase pump already in series can be used to aid in scavenging. Additionally, Heywood points out that a constant-pressure system is preferred for simple two-stroke engines that use an under piston pump [18].

As with any type of scavenging pump, the unsteady flow associated with them may cause surge problems in the centrifugal compressor. To alleviate surge in the compressor, it is suggested that a reasonably large receiver be used between the compressor and the scavenging pump to maintain an adequate margin between the average mass flow rate and the surge limit [19]. The receiver is typically a plenum and/or an intercooler. Based on the available literature, a constant-pressure turbocharger system incorporating an under-piston



scavenging pump, intake plenum, and intercooler was used. Figure 14 shows the arrangement of the engine, pipe and turbocharger [17].



**Figure 14:** Constant-pressure and crankcase scavenged two-stroke engine.

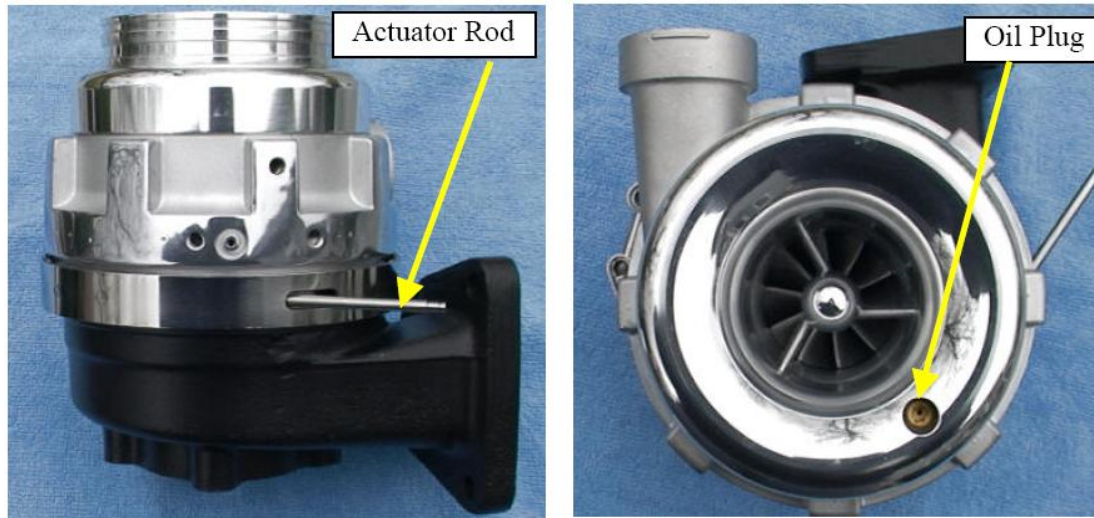
While OEMs have not turbocharged the simple two-stroke engine, it is often done by private parties using aftermarket parts. Typically, turbocharged snowmobiles are used for drag racing or deep powder mountain riding. Most often they are carbureted engines with displacements ranging from 600 cc to 1300 cc, producing between 180 and 350 horsepower. Some examples of turbocharged two-stroke snowmobiles are shown in Figure 15. These engines are designed to operate at high engine speeds with no concern for fuel economy or emissions. As a result, these engines have extremely high specific power, satisfactory run quality at off-design conditions, high fuel consumption, and poor emissions. The above-mentioned negative aspects, along with the increased cost and reduced durability, are why OEMs have not produced turbocharged two-stroke engines. With the advent of successful two-stroke GDI engine systems, turbochargers can now be utilized to produce clean and fuel-efficient two-stroke engines.



**Figure 15:** Aftermarket turbochargers installed on two-stroke snowmobile engines.

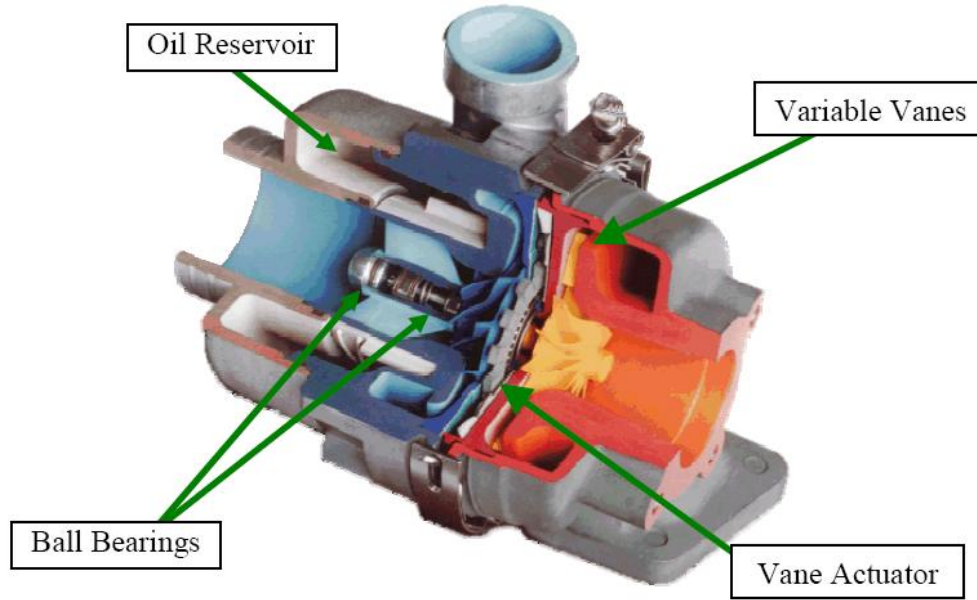
### 4.3 PRACTICAL SIZING

There are multiple options available to turbocharge a two-stroke snowmobile engine. The most common of these are the Garrett GTs and the Aerochargers. The Garrett GTs are vane-less turbochargers with internal waste-gates and require external cooling and oiling loops, typical of most turbochargers. The Aerocharger was chosen over the Garrett GT models for several reasons. First, the Aerocharger had a variable turbine housing, which improves turbocharger performance and reduces turbocharger-lag. Second, the Aerocharger did not require external cooling or oiling loops, which significantly reduces overall system complexity. Finally, the Aerocharger has worked well as an aftermarket add-on with many snowmobile engines. Aerocharger turbochargers were originally manufactured by Aerodyne of Dallas, Texas. At the time of this work, Aerodyne had sold the manufacturing rights to several smaller companies [20]. There are two turbocharger series designations, 66 and 53. The 66 series are larger than the 53 series and are often used on snowmobile engines with displacements greater than 900 cc. Each of the series has two turbine housing sizes; the 128 housing of the 53 series is smaller than the 158. The turbocharger used for the UI engine was the smallest and fastest responding Aerocharger available, 53 series with the 128 housing. Side and top views of an Aerocharger are shown in figure 16 and a cut-away view is provided in figure 17.



**Figure 16:** Two views of an Aerocharger turbocharger.

The turbine housing has multiple movable vanes similar to the ones described earlier in Figure 17, which are actuated by a steel rod that passes through the turbine housing, shown in Figure 16. The movement of the actuator rod is controlled by a pressure operated bellows with one side open to the atmosphere and the other side connected to the intake pressure at a convenient location. As boost pressure rises, the pressure differential across the bellows moves the actuator rod to open the vanes. The initial vane position is varied by adjusting a screw located on the vane-actuator housing, allowing precise control of the engine speed at which the turbocharger began to produce boost pressure. Aerocharger turbochargers are considered one of the most advanced and user-friendly turbochargers available [16]. Installation is very easy because there is no need to develop an oiling or cooling system. For quick spool-up time, they utilize high precision ball bearings that receive oil from a reservoir in the compressor housing that wicks to the bearings. The oil can be refilled by removing the brass plug located on the top of the compressor housing. However, without an external cooling or oiling system the turbocharger can be overheated. If it is allowed to get too hot, the oil will heat up and burn, causing the bearings to seize.



**Figure 17:** Cut-away view of an Aerocharger turbocharger.

Another feature is that the two housings halves of the turbocharger can be connected in eight different configurations, changing the relative position of the compressor outlet and turbine inlet. This offers a high degree of flexibility for turbocharger placement. However, the turbocharger must be mounted vertically to ensure proper oil distribution to the bearings.

As discussed earlier, the UI engine utilized a constant-pressure turbocharging system that retained the tuning characteristics of the exhaust pipe. Therefore, the turbocharger would need to be attached to the exhaust stream at the end of the tuned pipe. To connect the turbine inlet flange to the exhaust outlet, a socket flange for the turbocharger was cut from a stock silencer, allowing the use of springs to attach the turbocharger to the tuned pipe, similar to the stock system.

#### 4.4 COMBUSTION CHAMBER DESIGN

The combustion chamber volume for a turbocharged engine can be increased to reduce peak pressures and the onset of knock/detonation. This is even more important when using 87 octane fuel. Using 110 octane fuel the compression ratio can be kept closer to

stock, approximately 6.3:1. Brad Story is a respected turbocharged engine builder in the snowmobile industry and has over 20 year of turbocharging engines. His experience has shown that the engine can handle 4-6 psi of boost pressure without changing trapped compression ratio when running lower octane fuel [21]. The chamber used on the test engine had the spark plug on the intake side, and the injector angle was 11°, targeting the fuel towards the intake. The direct injection combustion chamber shape is taller than the carbureted versions to reduce fuel spray impingement on the piston and position the spark plug into the fuel spray. This shape and spark plug to fuel spray were shown in figure 10. The intake spark plug configuration does not allow later injection angles and limited the amount of boost pressure. The injectors cannot deliver the larger fuel quantity required by early injection angles. Testing has shown that with the spark plug location changed to the exhaust side of the chamber, the start of injection (SOI) can be delayed 30° before afterburn becomes excessive. More discussion of afterburn is given in section 5.1.

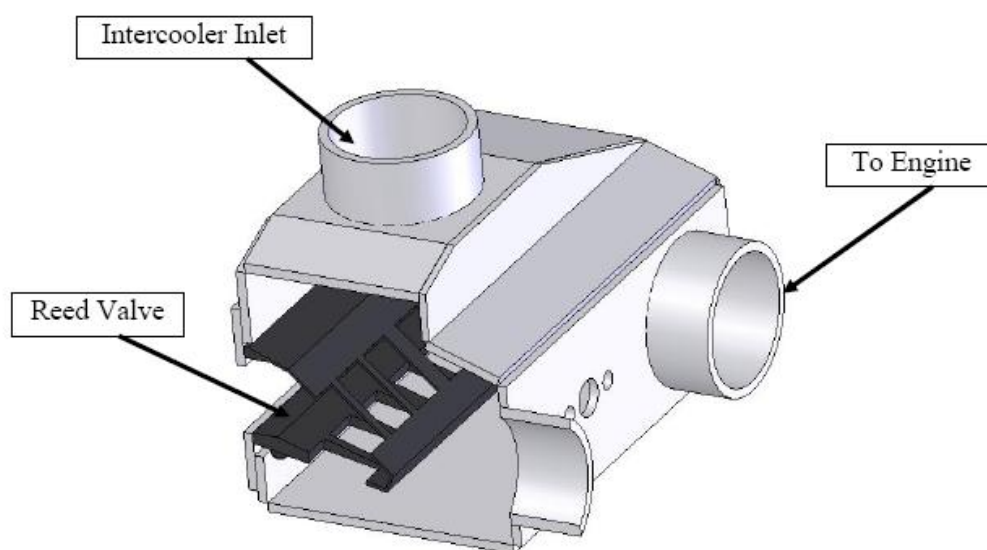
#### **4.5 INTERCOOLER AND PLENUM**

An intercooler was used to increase intake system volume and to remove heat from the compressed air, increasing the density of the delivered air. The intercooler is an air-to-air heat exchanger that relies on airflow through the fins to remove heat from the compressed air. Rather than designing and manufacturing an intercooler, one was purchased from a company that specializes in turbochargers. Bell Intercoolers built an intercooler for the engine using the predicted engine airflow and the dimensions of the space for the intercooler.

Brad Story's experience has shown larger intakes are better and an optimal size should be around 2.5 inches. It has also been shown to improve results using a wider intercooler compared to a longer one. Testing has shown that the first 4 inches does most of the cooling. Plenum sizes should range from 1.5 to 2 times the swept volume of the engine. The minimum size should be no less than the swept volume of the engine. The small plenums will have more throttle response but have the tradeoff of building more heat and higher intake temperatures [21].

In addition to the intercooler, an intake plenum was used to help damp out intake pulsations and provide a connection between the intercooler outlet and the engine intake.

The plenum's volume was 1.5 times the swept volume of the engine. In addition, a reed valve, similar to the engine intake reed valves, was installed in the plenum. The reed valve would ensure that the plenum never operated under vacuum. If the plenum were to experience a lower than atmospheric pressure, the reed valve would open and let air into the system. The added air would increase the mass flow through the engine, increasing the turbine spooling speed. The turbocharger would then provide more air to the engine. The plenum was connected to the same throttle bodies used by the Rotax SDI engine. A solid model showing the plenum design and the location of the reed valve is shown in Figure 18.



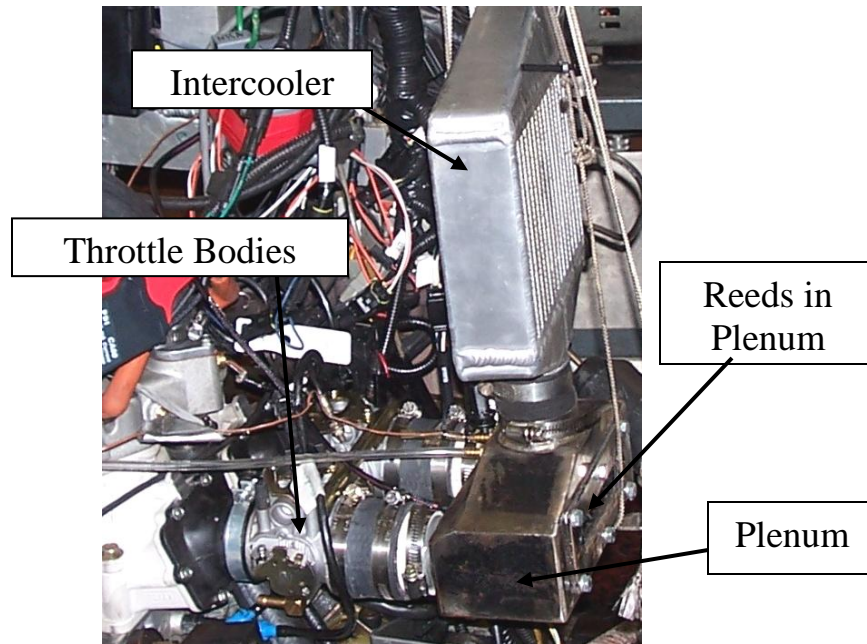
**Figure 18:** Solid model of the second plenum showing the reed valve.

Semi-direct injection and carbureted turbocharged engines with compressor inlet temperatures of 30° F ambient air have intake air temperatures ranging from 80-100°F at 8 psi of boost pressure to 140-150°F with 14 psi of boost pressure [21]. These temperatures are reached when the engine has been operating at steady-state for intervals of one minute. Most turbocharged applications are race type engines which do not require longer steady state conditions.

During testing, inlet temperatures were monitored in the plenum before the throttle bodies. High intake temperatures were experienced at extended steady state conditions lasting longer than two minutes, despite having a fan positioned directly behind the intercooler, therefore a spray bar system that misted water onto the intercooler was used to help reduce intake temperatures. Before the spray bar system, the temperatures were greater than 150°F, but with the spray bar system the temperatures were kept below 115°F. Table 3 compares the different intake temperatures. Figure 19 shows the intake installed on the engine.

Engine	Steady-State	Ambient Temperature	Boost	Inlet Temperature
DI w/out spray bar	> 2 min	30° F	5 psi	> 150° F
DI w/ spray bar	> 2 min	30° F	5 psi	<115° F
SDI/ Carb	< 1 min	30° F	8 psi	80-100° F
SDI/ Carb	< 1 min	30° F	14 psi	140-150° F

**Table 3:** Intake Temperature Comparison.



**Figure 19:** Intake and Intercooler.

## 4.6 EXHAUST SYSTEM

Most current turbocharged two-stroke engine exhaust systems retain the same geometry for the y-pipe and tuned pipe. Some are reinforced to help durability when running higher backpressures. The turbocharger is attached at the end of the expansion chamber, allowing the tuned pipe to still provide plugging pulses to the engine and damp out pulsation to the turbine inlet. After the turbocharger, some designs use a straight pipe, while others use a diverging/converging cone with an internal stinger. The internal stinger extends from the exit to the beginning of the converging cone. The diverging/converging cone design allows the swirling motion of the exhaust gases as they exit the turbine to expand and provide a more linear flow out of the exhaust. This helps in removing exhaust gases from the turbocharger allowing it to spool faster [20]. Figure 20 shows the turbocharger location and converging /diverging cone.



**Figure 20:** Turbocharger location.

Aftermarket SDI and carbureted turbocharger engines use a wideband air-fuel ratio/lambda sensor located after the turbocharger, 7-8 inches before the outlet, for lambda measurement [21]. Lambda is the actual air-fuel ratio divided by the stoichiometric air-fuel



ratio. This differs from the GDI setup where the lambda sensor is located at the beginning of the expansion chamber after the y-pipe. This location was chosen because of the changes in air-fuel ratio due to afterburn down the pipe, which was observed with the rapidly increasing exhaust gas temperatures (EGT) at the end of the pipe.

Typical EGTs in the manifold are in the 1200°F range and can reach 1600-1700°F at the turbine inlet [21]. The turbocharged GDI system became unstable when the turbine inlet temperatures exceeded 1300°F. Turbine inlet temperatures would rapidly increase to 1800°F+ and the engine would misfire. The temperature increase was from residual unburned fuel igniting in the exhaust pipe. Section 5.1 gives more discussion of afterburn.

#### **4.7 ENGINE MANAGEMENT MODIFICATIONS**

The engine management module (EMM) remained in stock configuration for the data presented in this thesis. Other UI designs have used a higher range (3 bar) pressure sensor for the manifold absolute pressure (MAP) input. This configuration allows the input of boost pressure correction values making it possible for the calibration to self-correct for injection quantity, injection angle, and ignition timing based on boost pressure. The data presented here were taken in a “manual mode,” which requires user input of injection quantity, injection angle and ignition timing values as the engine is running. Calibration values are changed based on boost pressure, torque, engine speed, lambda, and EGT, with changes to the calibration being fuel quantity, SOI, exhaust valve opening, and ignition timing. Having the self-correcting map is useful for obtaining data at all engine calibration points, but the time required to develop a full map takes longer. With limited testing time, the “manual mode” was used to get data at the load line points and full throttle. For testing in the snowmobile, the self-correcting map would have to be used.

#### **4.8 IGNITION SYSTEM**

An inductive ignition (IDI) replaced the previous capacitive discharge ignition (CDI) to increase combustion robustness. An inductive ignition discharges energy continuously into the fuel-air mixture as opposed to the multiple strike strategy of a capacitive discharge system. As the fuel-air mixture passes through the spark gap, the continual supply of ignition

energy aids in the initiation and development of a flame kernel into a fully developed flame capable of sustaining itself and growing. With the inductive ignition, the engine was able to run under a wider range of fuel quantities, allowing leaner calibrations, which further improved fuel economy and reduced emissions while enhancing run quality. The spark plugs used featured a smaller tip that held less heat and also reduced high speed misfire. The spark plugs were NGK model ZFR7F.

#### **4.9 INJECTOR CONSTRAINTS**

During testing, several limitations of the injectors were found. At mid-range engine speeds (5000-6500 rpm), the injectors can deliver an adequate amount of fuel. However, as engine speed increases to 8000 rpm and wide open throttle (WOT), the fueling limits of the injectors are limited to  $66\text{mm}^3$ . With this maximum fuel delivery, the boost pressure is limited to approximately 5 psi.

There are two reasons for the maximum fuel flow constraint from the injector. The first is the maximum forward pulse width of 3.5 to 4 ms. The stroke of the injector plunger limits this forward pulse width, as anything higher will bottom out the coil. At 8000 rpm one revolution takes 7.5 ms, thus requiring the fuel to be injected and have time to mix before the spark event at  $12^\circ$  BTDC. With an injection angle of  $245^\circ$  BTDC, and an ignition at  $12^\circ$  BTDC, there is 4.8 ms for fuel injection and mixing. With the injection event taking 3.6 ms, there is just over 1 ms to mix the fuel.

At 8000 rpm the cycles get close together and the injector does not have enough time to reset. An additional 3.5ms is required to reset the dynamic components inside the injector. Injectors not resetting causes instability, cyclic variation in fuel shot quantity, and ultimately power output variations and poor combustion/engine stability. The specific control of the injector is proprietary and is not discussed in this thesis. The injectors were supplied with fuel at 35 psi where it is increased to nearly 600 psi [13]. The injectors had  $115\ \mu\text{m}$  stroke nozzles. The other features of the injector remained stock, and were characterized on an individual basis to ensure proper fuel delivery. Table 4 summarizes the injector properties.

Maximum Fuel Flow at 8000 rpm (mm <sup>3</sup> , kg/hr)	Minimum Fuel Flow at 1000 rpm (mm <sup>3</sup> , kg/hr)	Line pressure (psi)	Injected pressure (psi)	Duration at max delivery (ms)
66, 40	4, 0.7	35	600	3.5-4

**Table 4:** Injector Properties

The fuel was chilled to prevent knock with stock compression and maintain injector stability at high speed. Testing has shown when the injectors are run at high speed, the heat created in the coils effects the injected fuel quantity. To combat this, the fuel was run through copper coils submerged in a bath of RV antifreeze inside a five cubic foot freezer/cooler. The temperature of the fuel was maintained at 7° Celsius. Temperature conditioning was used for both the 87 octane and the 110 octane fuel.

## 5.0 ENGINE TESTING

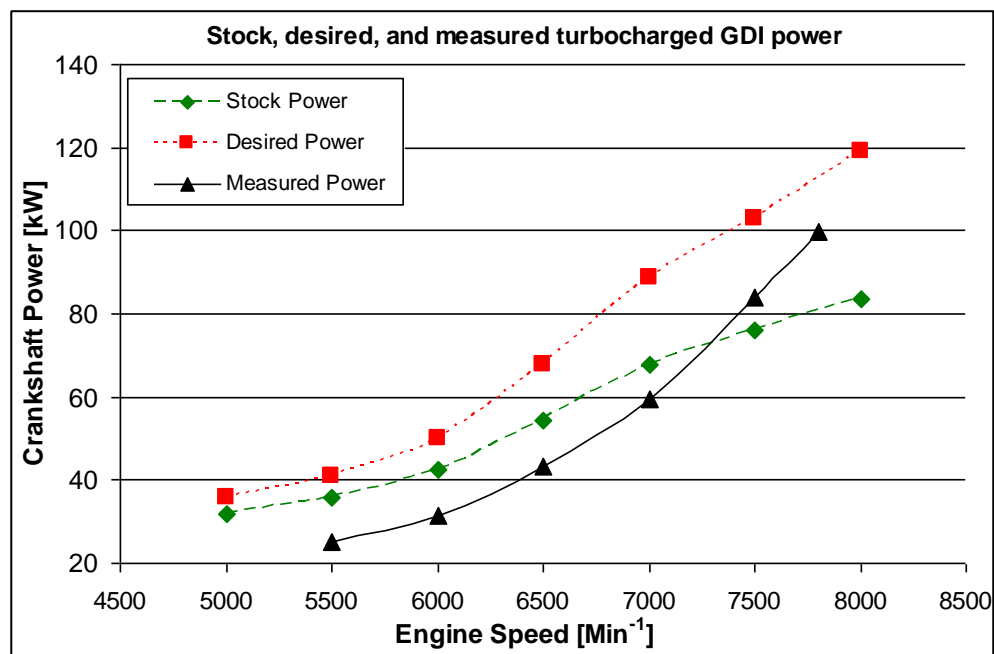
Previous turbocharging work was done with a Polaris 600 cc engine similar to the Rotax engine. Engine platforms were changed due to sponsorship and an offer to test our turbocharged engine from BRP.

### 5.1 PREVIOUS TURBOCHARGING ENGINE RESULTS

The turbocharged Polaris engine had mixed results since the engine was never fully mapped or able to reach full engine speed [6]. The low backpressure issues with the scavenging pressure ratio were thought to be from the turbocharger itself, and attempts were made to correct it with exhaust flow restriction and changing the boost pressure levels. One major reason for poor run quality with the turbocharged Polaris engine was the uncalibrated areas of the map. If the engine were run at any of these points, the engine would misfire, causing a drastic transient change in torque output and engine speed, and the dynamometer could not control it. This same scenario was experienced with the naturally aspirated engine during the early tuning stages before lambda measurement. Since implementing lambda measurement, engine tuning time has been drastically reduced and the engine runs much more smoothly throughout the map. Lambda measurement allows us to know the exact condition of rich/lean, giving a better idea of how to adjust fuel quantity. Previous testing was only based on EGTs and audible run quality, causing a slow and unsuccessful calibration process. Due to the success of the turbocharged Rotax engine, the Polaris engine was retested with lambda measurement, and ran much better throughout most of the operating range. The Polaris engine had higher fuel flow requirements and was unable to reach peak engine speed because the injectors were unable to reset while trying to deliver the large required fuel quantities.

The other major contributor to the success of the current engine was the understanding of the afterburn effects. Afterburn is when unburnt fuel in the exhaust system reaches a high enough temperature to ignite, causing combustion in the exhaust system, and creating more energy to spin the turbocharger. The higher exhaust temperatures change the tuning characteristics of the pipe. The excess energy causes the turbine to speed erratically,

building more boost pressure, changing the scavenging pressure ratio from the intake to the exhaust. More boost pressure, without the proportional increase in backpressure and injected fuel quantity, along with a “shorter” tuned pipe caused misfire and run instability. Previously, on the Polaris engine, a restriction was added before the turbocharger, which helped, but did not solve the problem. The restriction caused higher back pressure throughout the operating range and reduced part-load performance. Figure 21 shows the measured power values and inability to reach full engine speed with the Polaris turbocharged engine [6].

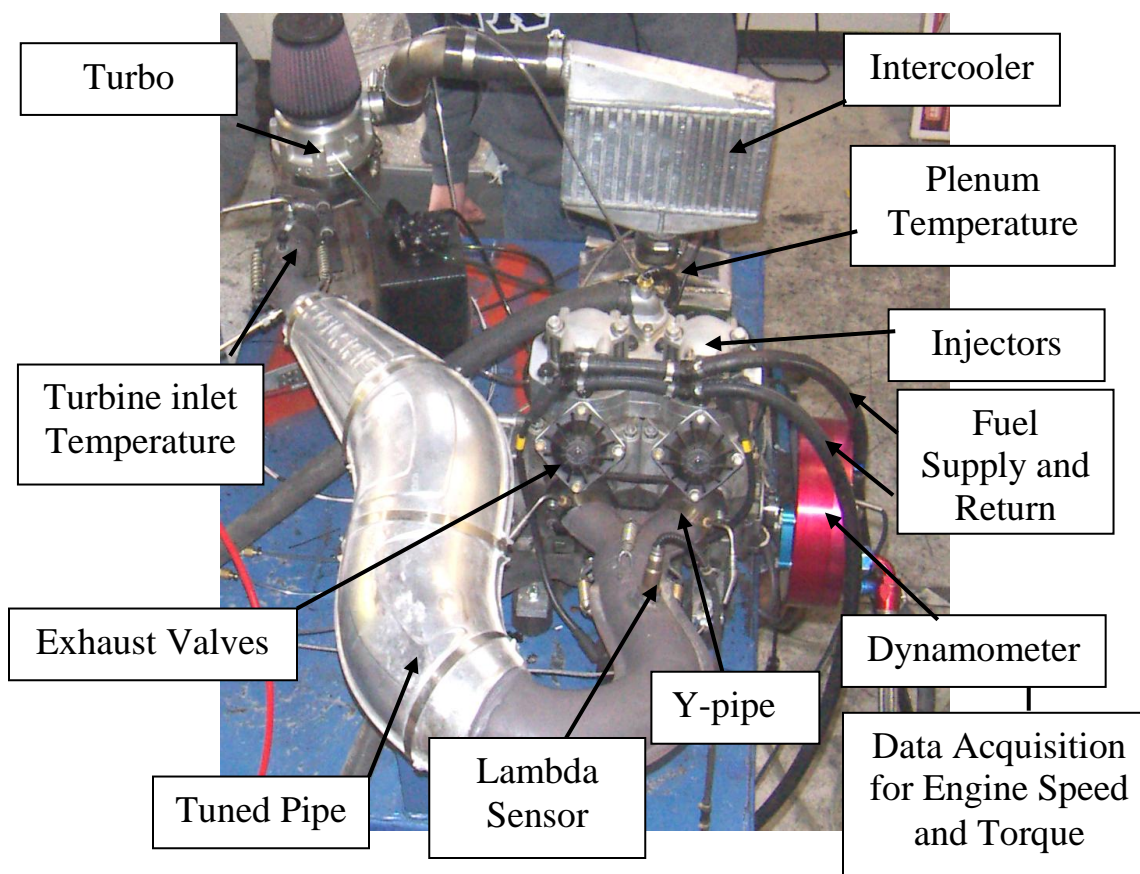


**Figure 21:** Polaris Turbocharger Testing Results.

Now that the presence and effect of afterburn are better understood, the engine is more stable. Afterburn can be controlled by reducing the amount of energy in the exhaust with earlier injection angles and larger fuel quantities. These changes lower combustion temperatures, and allow more fuel to short-circuit, effectively lowering EGTs. Afterburn is very sensitive to changes in the SOI, injected fuel quantity, and ignition timing. Hardware changes such as the location of the spark plug (exhaust or intake side) will also help the engine run with less afterburn. Testing has shown the exhaust side spark plug location allows reduced SOI angles up to 30° further than an intake side spark plug location.

## 5.2 DESCRIPTION OF TESTING EQUIPMENT

Engine testing was done at the Evinrude Product Development Center in Waukegan, Illinois. The dynamometer used was an eddy current SAJ Froude AG 150 with a range of 150 kW @ 2800 – 8000 rpm and 500nm @ 1800-2800 rpm. Typical error for the dynamometer is  $\pm 2\%$ . For fuel measurement, an AVL fuel balance was used with an error of  $\pm 2\%$ . The model number was 7130-06 with a 7030 balance control and fuel calculator. The fuel measurement gives values of kg/hr and was used to calculate brake-specific fuel consumption (BSFC) in g/kW-hr. A similar setup of the testing equipment is given in figure 22. This has the same turbocharger and intake setup on a Polaris engine with a Land and Sea dynamometer. The dynamometers attach at the same position on the output shaft of the engine. Not shown are the AVL fuel balance, and engine management module.

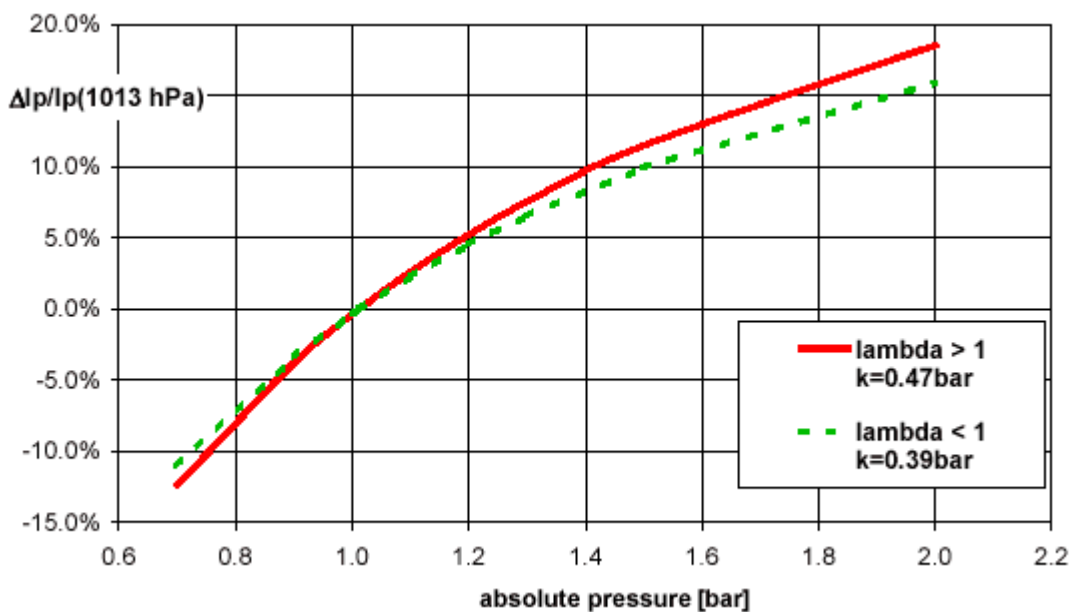


**Figure 22:** Testing Equipment Schematic

EGTs were measured in each branch of the y-pipe and at the turbine inlet. Intake

temperatures were monitored in the plenum. Both temperature devices were type K thermocouples. EGTs were used to monitor the amount of afterburn occurring and intake temperature was used to monitor intercooler effectiveness. Excessive intake temperatures will result in poor engine run quality because charge density will be reduced.

Lambda was monitored in the tuned pipe after the y-pipe, a high pressure portion of the exhaust system. The increase of pressure in the exhaust pipe will cause the lambda sensor to read 10% richer than actual [22]. Even though the actual value was incorrect, this provided a valuable tuning aid by knowing how fuel quantity adjustments affected the measured lambda value, aiming for lambda values around 1.0 at part load and 0.9 at WOT. Figure 23 shows the pressure effects on lambda values [22]. The x-axis shows pressure in bar (5 psi boost pressure is about 1.35 absolute bar) and the y-axis is % error of the actual reading versus the correct value. The k factor depends on the rich or lean conditions,  $I_p$  is the output signal at standard pressure and  $\Delta I_p$  is the change in output signal for a given pressure difference.



**Figure 23:** Pressure effects on lambda values.

Lambda measurement helped the tuning process by knowing the lambda values required by naturally aspirated engine, and calibrating the turbocharged engine's fuel quantity so the lambda value was the same or slightly richer than the naturally aspirated lambda values. Carbureted and SDI turbocharged engines typically install the lambda sensor after the turbocharger. This location was chosen to reduce the pressure effects since the exhaust products are more uniform after mixing rather than right out of the cylinder where there is the possibility of large gradients in the exhaust products.

Knock was monitored with individual Rotax sensors on each cylinder. The knock sensors could be adjusted for sensitivity. The knock sensors output an audible noise through headsets. Good run quality will be a consistent buzz with knock as a loud ping. Pressure transducers, installed in the side of the cylinder head, were also utilized to monitor knock and runability. No data were collected with the pressure transducers. Good runability was seen with multiple pressure traces that were similar and knock was seen as a large spike in the pressure trace.

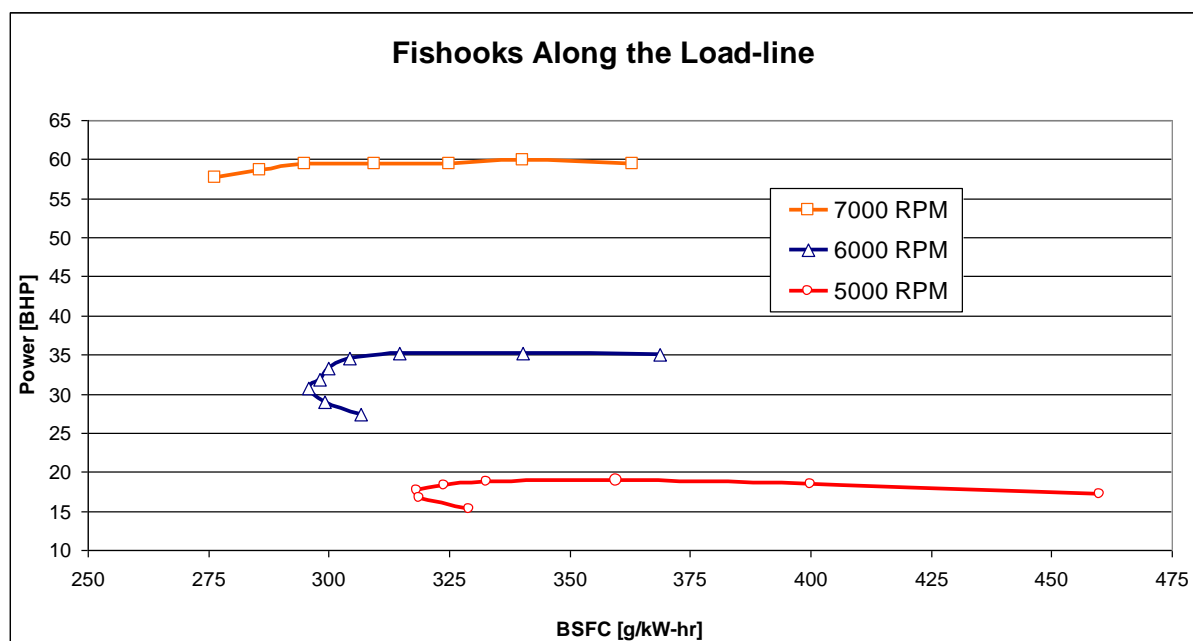
### **5.3 TUNING STRATEGY**

The goal was to optimize injection angle, fuel quantity, and ignition timing at the part-load points in order to minimize the BSFC, and then run power sweeps at WOT to observe how much power could be generated before reaching the limits of the injector. The E-TEC injectors have never been used on an engine requiring this large fuel quantity at such a high engine speed, as they were designed for outboard engines with lower engine speeds, specific power, and injected fuel quantities.

At part-load during testing, the best BSFC points were found by varying injection quantity, while maintaining a constant injection angle, engine speed, and throttle position. This technique is referred to as a fishhook and is shown in Figure 24. The injection angle was chosen by maintaining a constant throttle position, engine speed, lambda value of 1.0, and picking the injection angle with the lowest BSFC. Once the injection angle was chosen the fishhook was performed. During testing, it was concluded that the best BSFC occurs at the same point where the pipe begins to afterburn. A possible reason for this is the combination of higher EGTs and a slower burn rate due to lean air-fuel ratios at part-load. If



the lean burn is slow enough, the blow-down may include unburnt fuel and a flame front that ignites the excess fuel in the exhaust system. Lean burn conditions will produce y-pipe temperatures of 1300°F. If the afterburn is excessive, the turbocharger inlet temperatures rise upwards of 1800-2000°F (980-1100°C). Engine run quality will diminish from the afterburn and the turbine will be damaged from the high inlet temperatures. This explains why the fishhook at 7000 rpm does not hook around. When the fishhook turns after the minimum point of BSFC and power decreases, the lean limit of combustion is reached, the run quality of the engine diminishes, and afterburn goes away. The part-load (load-line) engine speeds and horsepower values are representative of actual riding conditions, and they correspond closely with the five-mode emission test for power and engine speed requirements for snowmobiles [1]. Figure 24 shows the BSFC trends at the various engine speeds. The x-axis shows BSFC in g/kW-hr and the y-axis is power in brake horsepower (BHP).



**Figure 24:** BSFC Fishhooks.

The WOT points began at 5000 rpm and were swept to 8000 rpm. These points were not optimized for BSFC. Since the primary goal was to make the maximum power possible without engine damage, fuel quantity and injection angle were determined as a function of lambda measured after the y-pipe and the turbine inlet temperature. If the turbocharger inlet

temperature was excessive, the start of injection (SOI) was increased (earlier injection angle) along with the injected fuel quantity, which gave more time for the fuel to mix with the incoming air, reducing the onset of afterburn. Since the turbocharger is delivering more air, the injected fuel quantity is also increased to maintain a lambda value of approximately 0.9. Typical cruise points for a snowmobile are below 50 horsepower or 6500 rpm. Engine speeds higher than this will occur mainly under transient conditions accelerating to or decelerating from a WOT condition; therefore steady-state testing was not performed at these points.

## 6.0 RESULTS, CONCLUSIONS, AND RECOMMENDATIONS

### 6.1 FUEL CHOICE AND CALIBRATION VALUES

During testing, excessive knock was observed with the stock compression head and 87 octane fuel, therefore, the decision was made to run with a 110 octane fuel made for racing applications. Knock was only observed at WOT and above 7500 rpm, where the peak cylinder pressures are the highest. Table 5 shows the values used at WOT with the 110 octane fuel.

Engine Speed (RPM)	Torque (ft-lb)	Ignition (°BTDC)	Fuel (mm <sup>3</sup> )	SOI (°BTDC)	$\lambda$
5000	50	25	34	220	1.03
6000	74	22	64	230	1.0
7000	94	19	62	245	0.9
8000	100	13	62	245	0.9

**Table 5:** Wide Open Throttle Parameters Chart.

With a two-stroke engine as the engine speed increases, the ignition timing is reduced because of the increase in charge density from the tuned pipe plugging pulse and higher flame speeds. The SOI has to be increased to maintain the atomization time of the fuel with the air. As the engine speed increases the early injection angle does allow some of the fuel to short-circuit but will be returned into the cylinder from the plugging pulse of the pipe at designed speeds. This is shown in the simple two-stroke description in section 1.3, figure 5. If the plugging pulse is not effective, the BSFC would decrease and UHC emissions would increase. At part-load the engine will handle leaner values of lambda (greater than 1) for better BSFC, but for the higher engine speeds the engine must be run richer to keep internal temperatures down, which reduces the chance of pre-ignition and detonation. Lambda values at part-load can be greater than one because of the lower peak pressures, internal temperatures, and heat release. As discussed in the previous testing section, these values were changed based on boost pressure, lambda, and EGTs.

At the part-load points, low injection quantities, less required mixing time from the lower engine speed, and less intense scavenging flows than WOT allow the SOI to be

reduced even further. This is a major benefit compared with a carbureted or SDI engine, when the time available for fuel to enter the cylinder is fixed based on the geometry of the transfer ports. Introducing fuel through the ports limits the amount of fuel that can be trapped in the cylinder. This highlights one benefit of GDI, where fuel can continue to be introduced after transfer ports have closed. SOI values were determined based on the runability of the engine and avoiding afterburn. The ignition timing was also increased to improve torque, BSFC, and part-load runability. Table 6 shows the part-load chart, load-line points with 110 octane fuel.

Engine Speed (RPM)	Torque (ft-lb)	Ignition ( $^{\circ}$ BTDC)	Fuel ( $\text{mm}^3$ )	SOI ( $^{\circ}$ BTDC)	% Throttle
5000	20	30	9	160	16.5
6000	31	25	12	180	20
7000	44	17	18	200	21
8000	100	12	62	245	100

**Table 6:** Turbocharged Load-line Parameters.

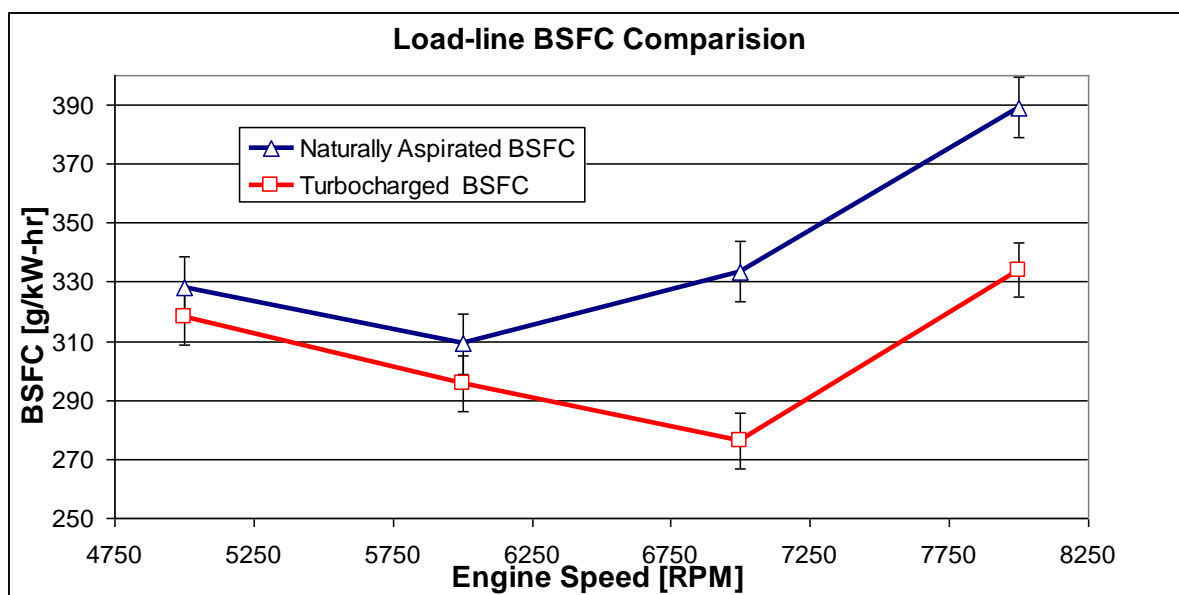
At the 5000 rpm and 6000 rpm points the turbocharged engine's torque values are slightly higher than the naturally aspirated. This is from the boost pressure, higher throttle opening, and helps improve the BSFC. The SOI for the turbocharged engine is increased to provide additional fuel mixing time, creating a more homogeneous mixture for faster burn rate and more complete combustion, reducing the afterburn effects. With the similar torque values at the 7000 rpm point, the BSFC improvement is seen by the smaller injected fuel quantity required by the turbocharged engine. Table 7 shows how the naturally aspirated engine compares in calibration parameters along the load-line.

Engine Speed (RPM)	Torque (ft-lb)	Ignition ( $^{\circ}$ BTDC)	Fuel ( $\text{mm}^3$ )	SOI ( $^{\circ}$ BTDC)	% Throttle
5000	15	30	7.5	140	9.5
6000	24	28	11	160	18
7000	43	17	20.5	195	33
8000	70	12	43	245	100

**Table 7:** Naturally Aspirated Load-line Parameters.

## 6.2 BRAKE SPECIFIC FUEL CONSUMPTION COMPARISON

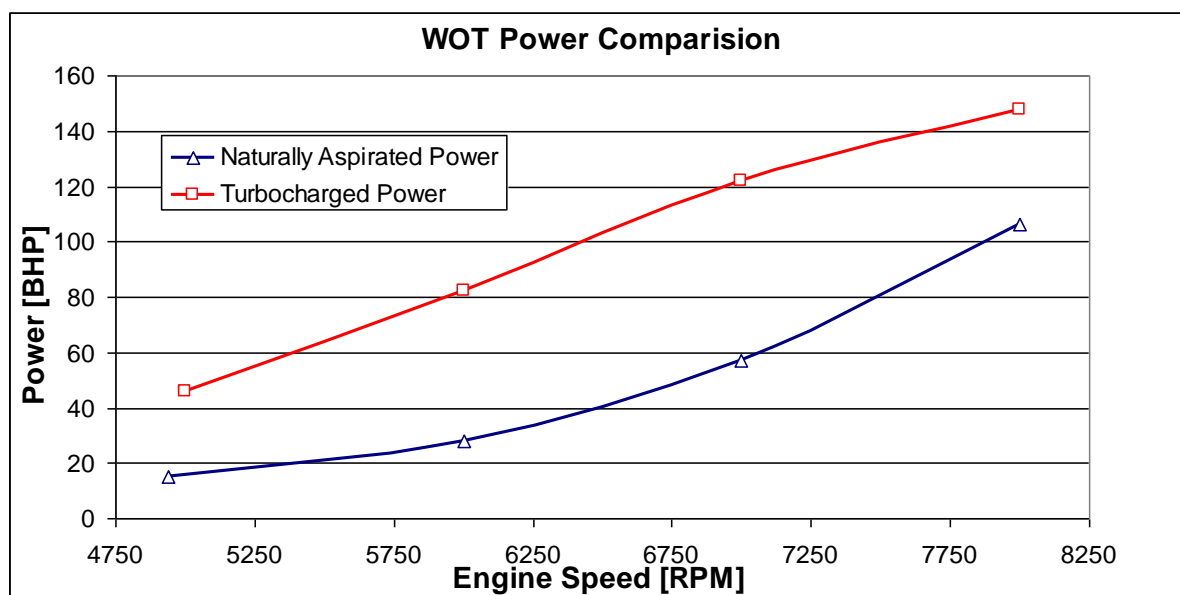
The BSFC of the turbocharged engine was improved over the entire engine operating range compared to the naturally aspirated engine. The largest gain was at 7000 rpm, with a 20% improvement. This operating point of 30% throttle and 50% peak power is very efficient because the engine has a higher scavenging efficiency due to less throttling and the benefit of tuning from the pipe compared to the lower load-line points. The improved BSFC shows the turbocharged engine is utilizing the fuel more efficiently and capturing some of the energy normally lost out of the exhaust. Scavenging flows at the throttled part-load points are improved with the higher intake pressures. Data for BSFC were collected along the load-line of the engine, including cruise points and WOT at the peak engine speed of 8000 rpm. Figure 25 shows the BSFC improvements with engine speed on the x-axis and the BSFC in g/kW-hr on the y-axis. A Root-Sum-Squared analysis shows the BSFC calculation is within 10 g/kW-hr or  $\pm 3\%$ . At the lower engine speeds, the BSFC is within the statistical error range, but a definite improvement is seen at higher engine speeds.



**Figure 25:** Load-line BSFC Improvements for Turbocharged and Naturally Aspirated Engines.

### 6.3 FULL THROTTLE PERFORMANCE

Performance is a high priority for recreational enthusiasts. The boost pressure was limited to 5 psi due to the amount of fuel the injector could deliver. Despite the mild boost pressure level, there was a significant power increase of 40 bhp at the peak engine speed and over 60 bhp at the lower engine speeds. Four psi of boost pressure was developed at 5000 rpm and 5 psi of boost pressure was developed before 6000 rpm. This highlights the ability to produce boost at a low engine speed for snowmobile engines when clutch engagement is typically around 4000 rpm. Future vehicle testing is needed to find if there is any boost pressure lag time. Figure 26 shows the power improvements with engine speed on the x-axis and power on the y-axis. The engine power was still increasing at 8000 rpm, indicating that the airflow through the engine was not a limiting factor in power output. This trend is also seen in carbureted and SDI engines that can handle higher boost pressure levels in excess of 12 psi [21].



**Figure 26:** WOT Sweep and Power Improvements for Turbocharged and Naturally Aspirated Engines.

## 6.4 FUTURE WORK AND RECOMMENDATIONS

Future work with the turbocharged engine will provide better understanding along with new challenges and complications. Before more testing is done, an upgrade of in-house equipment is needed. Suggestions are offered of what the next steps in testing could be.

- Higher Engine Speeds:
  - Find when power decreases or the torque curve becomes flat. The testing was only run to 8000 rpm and the power curve was still increasing.
  
- Higher Boost Pressure Levels:
  - Move the spark plug to the exhaust side for later SOI, thereby reducing fuel flow requirements and allowing higher boost pressures. The engine will handle more boost pressure if the injectors can deliver the required fuel flow.
  
- Improve Understanding of the Effects of Afterburn:
  - Additional exhaust system temperature and in-cylinder pressure measurements would help in understanding how afterburn affects the scavenging pressure ratio, turbocharger response, and engine dynamics.
  
- Emission Measurement and Modeling:
  - Will the engine still pass the five-mode 2012 emission standard?
  - Emission measurements will aid in the tuning process and provide the necessary information to determine air trapping efficiency, scavenging ratio, volumetric efficiency, heat release, mass fraction of fuel burned, and turbocharger efficiency.
  
- Lower Port Configurations:
  - Change the goal to only maintain power output and try to improve emissions and BSFC further.

- Chassis Installation:
  - This will require the automatic control of fuel, injection timing, and ignition timing based on boost pressure.
  - Vehicle measurements will validate BSFC and power improvements found during dynamometer testing.
  - Find any ride-ability issues in the calibration. What works on the dynamometer calibration may not work in the vehicle.
  - Optimize the intercooler and turbocharger placement for temperature reduction and packaging constraints.
  - Experiment with clutch calibrations to cruise at a lower rpm.
  
- Garrett GT 2860RS or GT2854 with a 0.64 Housing:
  - This turbocharger utilizes liquid/oil cooling and would provide a good comparison to the Aerocharger 53 series. Compare the inlet temperatures, boost pressure levels, and turbocharger lag time.
  - The aftermarket users of the Garrett GT-28 series turbocharger on 600 cc and 700 cc carbureted and throttle body fuel injected engines achieve upwards of 300 hp/liter and boost pressures of 10-12 psi with 110 octane fuels. The focus of these engines is on peak power and not part load, run quality, or BSFC.
  
- The common theme to future work is better equipment in-house to enhance the quality of the research:
  - An eddy current dynamometer.
  - Fuel flow measurement with EPA certified accuracy of  $\pm 2\%$ .
  - In-cylinder pressure measurement and data acquisition for combustion data.
  - Emissions equipment for additional combustion data and validating the EPA emission standards.



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