EML 4905 Senior Design Project

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BACHELOR OF SCIENCE
IN
MECHANICAL ENGINEERING

Race Engine Optimization
25% Final Report

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Ethics Statement and Signatures

The work submitted in this B.S. thesis is solely prepared by a team consisting of Alex George, David Ducassi, and Felipe Guajardo, and it is original. Excerpts from others' work have been clearly identified, their work acknowledged within the text and listed in the list of references. All of the engineering drawings, computer programs, formulations, design work, prototype development and testing reported in this document are also original and prepared by the same team of students.

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Team Leader	Team Member	Team Member

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Abstract

Formula SAE (Society of Automotive Engineers) is a competition in which university teams compete with formula-style, open-wheel race cars that are designed and constructed by students. The competition rules strictly regulate the design for homologation purposes. Among these restrictions are the requirements that all engines displace less than 610cc per cycle and that all intake air pass through a 20mm restrictor for gasoline-fueled engines. Our design objective is to adapt a motorcycle engine to optimally function under the competition operating conditions. [1]

The competition rules allow for 100-octane fuel to be used. This fuel allows for a higher compression ratio to be reached before mixture detonation occurs. [10] The street-legal motorcycle engines typically adapted for this competition are designed to run on widely-available “pump” fuels rated as low as 87-octane. [11] The pressure drop induced by the 20mm restrictor further lowers the effective compression ratio, and the effect is that combustion conditions are far from optimized to extract maximum power from the fuel available.

In unrestricted race classes, it is popular to employ forced induction for the purpose of increasing power output. Because the intake restrictor effectively limits the air flow into the cylinders, the engine will be modified to more efficiently use the oxygen – and by extension, fuel - that is available. The theoretical efficiency of an air-breathing Otto cycle engine is a function of compression ratio: [5]

$$\eta = 1 - \left(\frac{1}{r^{(\gamma-1)}}\right)$$

Where γ is the specific heat ratio and r is the compression ratio.

Any increase in the compression ratio will result in an increase in thermodynamic efficiency, with a theoretical limit of 100% efficiency achieved at infinite compression.

1 Introduction

1.1 Problem Statement

The Race Engine Optimization project's goal is to increase the compression of ratio and displacement in each cylinder to improve the performance of the Honda CB550 engine under the Formula SAE competition conditions. The cylinders are to be redesigned to raise the compression ratio from 9:1 to 13.5:1, a modification that is expected to improve power output by 10%. The total engine displacement is expected to increase to 600cc, 10.3% more than its original displacement of 544cc. [1]

1.2 Motivation

The previous version of SAE's formula car utilized an unmodified Yamaha R6 engine. The power-to-weight ratio of this configuration was not optimal. Approaching the optimization of the engine through a senior design project will allow for the application of time necessary for a properly in-depth analysis of the combustion process. The team has selected this project for the opportunity to apply advanced concepts and techniques in the engineering discipline, including thermal and fluid dynamics.

2 Project Formulation

2.1 Proposed Design

Fit Honda CB 650 crank to CB 550 block.

- 5.2mm stroke increase yields 600cc displacement, 16:1 compression ratio.
- Manipulate connecting rod length to fine tune compression ratio, quench, squish.

The team intends to acquire a complete engine/transmission unit from a 1978 Honda CB 550 sport motorcycle. All calculations have been performed with the specifications of this engine in mind.

The Honda CB 550 is a 544 cubic centimeter, four-stroke, four-cylinder, air-cooled, gasoline-fueled sport motorcycle manufactured from the late 1970s to the early 1980s. It is designed with a bore and stroke of 58.5mm and 50.6mm, respectively, and has a compression ratio of 9.0:1. It is designed to run optimally on 87-octane fuel and produces 50 horsepower at 8,500 rpm. [7]

Table 1. Honda CB550 Specifications

General Information	
Model:	Honda CB 550 K 3
Year:	1978
Category:	Sport touring
Engine and Transmission	
Displacement:	544.00 ccm (33.19 cubic inches)
Engine type:	In-line four, four-stroke
Power:	50.00 HP (36.5 kW)) @ 8500 RPM
Top speed:	175.0 km/h (108.7 mph)
Compression:	9.0:1
Bore x stroke:	58.5 x 50.6 mm (2.3 x 2.0 inches)
Valves per cylinder:	2
Fuel control:	OHC
Cooling system:	Air
Gearbox:	5-speed
final drive:	Chain

For the objective of optimizing the engine’s compression ratio for 100-octane fuel, as well as to increase the displacement of the engine closer to the displacement limit of 610cc, the team will acquire the crankshaft of a Honda CB 650. The CB 550 and CB 650 are from the same family of engines in Honda’s lineup, and the journal sizes match.

The CB 650 crank has a stroke length of 55.8mm. [8] This, with the CB550’s 58.5mm bore, produces a displacement of 600cc. The combined volume of the combustion chambers is reduced from 68.0cc to 40.0cc, consequently increasing compression ratio from 9.0:1 to an unacceptably high 16:1. The final compression ratio will be adjusted through the use of custom length connecting rods, which will control the piston’s final crown height. [7]

Calculations indicate that a 1.0mm reduction in the connecting rod length will produce a combined combustion chamber volume of 50.8cc. Preliminary estimates indicate that the resultant compression ratio of 12.8:1 will be acceptable and produce measureable power and efficiency gains. Further calculations are pending acquisition of detailed valve timing specifications.

Table 2. Compression Ratio Estimates

Engine Specifications		
Original Displacement	544	cc
	0.000544	m^3
Original CR	9.0	:1
# of Cylinders	4	
Total Head Volume	68.0	cc
Bore Diameter	5.85	cm
Stroke increase	0.52	cm
Shave Head by	0	cm
Rod Length Change	-0.075	cm
TDC Piston Height Change	0.185	cm
New Displacement	600	cc
New Head Volume	48.1	cc
New CR	13.5	**:1**

The cycle displacement increase of 56cc will increase demanded intake airflow by 10.3% at maximum engine speed. This is of concern as the Formula SAE rules stipulate the use of a 20mm diameter intake restrictor plate. Orifice plate compressible flow calculations indicate a total absolute pressure drop of 7.1 kPa. This 7% drop in pressure is accompanied by a corresponding drop in intake charge density and power. The power loss is exacerbated by the power requirements of drawing the intake charge through the plate.

Table 3. Engine Output

Outputs		
Pressure Ratio	0.93	
Manifold Vacuum	-1.03	psi
	-7.09	kPa
Manifold Absolute Pressure	13.67	psi
	94.26	kpa
Intake Air Density	1.068	kg/m^3
Power To Maintain Manifold Vacuum	-0.301	kW
	-0.404	HP
Total Power	35.0	kW
	46.9	HP

The total expected power loss is represented by the sum of the power to maintain manifold vacuum and the product of the percentage of lost intake airflow density and the unrestricted power output of the engine. [6]

$$Output\ Power =$$

$$\{[Pressure\ Ratio] \times [Unrestricted\ Output]\} - [Power\ to\ Maintain\ Manifold\ Vacuum]$$

See Appendix A for detailed calculations.

2.2 Design Alternatives

Alternative design approaches initially considered:

- Forced Induction
- Fabricate Pistons
- Shave Head
- Cylinder Bore Increase

Use of SAE's Yamaha YZ450F Engine

The most significant alteration to initial plans was the decision not to use SAE's Yamaha YZ450F engine as a starting platform. Purchasing and adapting an identical engine to the one utilized by SAE would greatly simplify the process of later using the engine in the competition vehicle.

The YZ450F is a significantly younger, more complex design. Consequently, it is significantly more expensive to acquire. Concerns over funding lead SAE to choose not to assist with funding of this project, and costs were then paid out of pocket by team members. The Honda CB550 platform was chosen for its low cost, ease of modification, and wide selection of available parts, both original and aftermarket.

At the design outset, there were several other approaches considered with the objective of increasing power.

2.2.1 Forced Induction

Forced induction methods such as supercharging and turbocharging were considered, but set aside due to the stipulated 20mm intake restriction. A plot of the ratio of absolute pressures before and after the restrictor against volume flow rate is shown below:

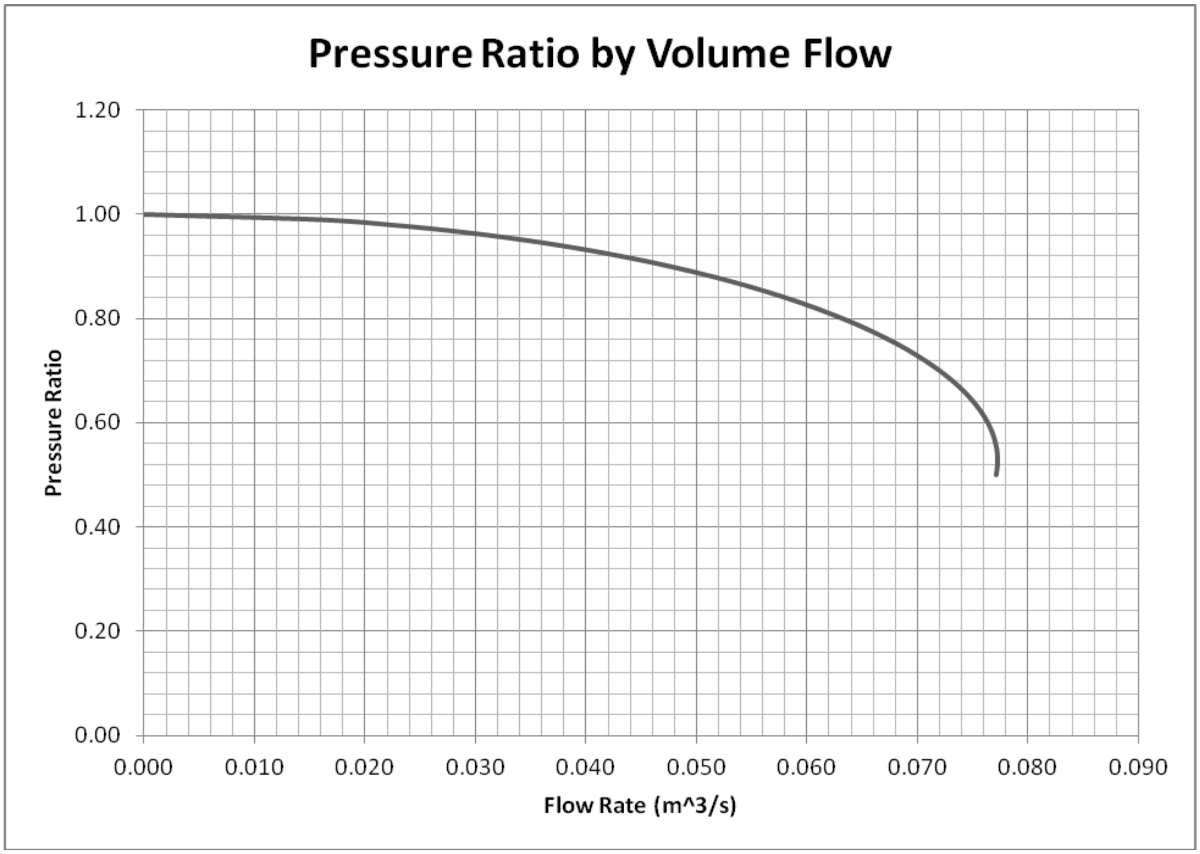

Figure 1: Pressure Ratio vs. Volume Flow [7]

Because the rate of pressure loss accelerates as flow increases, expending power to draw more air into the engine becomes less efficient as greater amounts of air are demanded. The team made the decision that the approach should be to more *efficiently* use the oxygen that is available, rather than attempt to force increasingly greater amounts of air through the same size restrictor with increasing amounts of power. [6]

2.2.2 Nitrous Oxide Injection

Fuel oxidizing additives are disallowed by the competition rules.

2.2.3 Cylinder Bore Increase

Bore increases with the objective of the maximization of engine cycle displacement were discounted for the same reason that forced induction was – the restrictor limits the gains achievable though increases in demanded airflow, and performance gains from this method alone will be modest at best.

Once the mode of optimization was selected to be increasing the compression ratio, our focus shifted to considering several methods that would accomplish this goal.

2.2.4 Piston Design & Fabrication

The first plan formulated called for the design and fabrication of custom, high compression, pistons. The additional "dome" of high compression pistons extends into the combustion chambers, reducing combustion chamber volume and thereby increasing the compression ratio. This option remains in consideration – further examination revealed that the team's goals may be better achieved through the modification of other engine parameters. By using a compatible, longer stroke crankshaft from a different engine in the Honda CB family, the team may be able to achieve the desired compression ratio without the considerable expense involved in the custom fabrication of non-standard pistons.

It may become necessary, however, to alter the piston design for valve clearance purposes. This situation will not necessitate the design of custom pistons, but should be available through the use of aftermarket pistons. A final decision is pending acquisition of the test motor and manual measurements of valve clearances, and well as camshaft specifications such as valve timing and lift.

2.2.5 Shave Head

The second plan formed involved precision machining the bottom side of the head down in order to increase the compression ratio. Assuming an "open chamber" design (i.e. no quench), this technique reduces the volume of the combustion chambers by:

$$volume = t * \pi r^2,$$

where t is the thickness of the layer removed.

This is a time-proven method, however concerns over changes induced to valve geometry by the relatively large amount of material required to be removed in order to achieve the unusually high target compression ratio caused the team to examine alternatives. [6]

3 Literature Survey

3.1 Octane Rating

All other things equal, an increase in a motor's compression ratio must be accompanied by a corresponding increase in the octane rating of the fuel utilized. Failure to supply an adequately knock-resistant fuel will result in intake charge detonation, which is harmful to engine components and detrimental to power production. [11]

The 100-octane fuel supplied by the SAE competition is rated according to the AKI (Anti-Knock Index) standard, which is the standard in universal commercial use throughout the United States. This standard represents the average of the octane ratings provided by two differing test procedures in wide use around the world. These two ratings are known as Research Octane Number (RON) and Motor Octane Number (MON). [11]

The RON test is somewhat less realistic than the MON test, typically producing octane ratings eight to ten points higher than the MON test. Both tests are run in variable-compression laboratory engines, the principle differences being engine speed and intake air temperature, RON and MON being performed at 600 rpm/125^{o}F and 900rpm/300^{o}F, respectively. [11]

3.2 Compressible Flow Through Orifice Plate

A spreadsheet was employed to handle the calculations regarding theoretical power loss caused by the addition of a restrictor plate to the engine. The spreadsheet was developed using the below system of equations:

$$Q_1 = C\ A_2 \sqrt{2\ \frac{Z\ R\ T_1}{M} \left(\frac{k}{k-1}\right) \left[(P_2/P_1)^{2/k} - (P_2/P_1)^{(k+1)/k}\right]}$$

Where,

$$C = \frac{C_d}{\sqrt{1-\beta^4}}\ ,\ \beta = d_2/d_1$$

These calculations, as well as others in the spreadsheet based on engine geometry, allow for rough predictions to be made about the flow characteristics of an engine when evaluating it as a prospective test platform, using only basic specifications easily available without purchasing and disassembling the motor.

Several assumptions were made to facilitate these calculations:

- The Discharge Coefficient (Cd) must be determined experimentally. Literature indicates that a typical value is approximately 0.6.
- The compressibility factor (Z) was taken to be constant at 1.0.
- The diameter of the intake duct interrupted by the restrictor was assumed to be 5 cm. Calculations will be updated upon the collection of measurements from a purchased motor.
- Ambient temperature was assumed to be 300 Kelvin. (80^{o}F)

The spreadsheet takes as inputs parameters regarding engine geometry, modifications made to engine geometry, and maximum engine speed and predicts the loss in intake charge density, as well as the power required to draw the intake charge through the restrictor. Together with the unrestricted output of the engine, these data can be used to predict the final output power of the engine. [3]

3.3 Survey of Selected Other Universities' Prior Designs

In order to determine the state of the competition, research was conducted on previous designs brought to the event by other, established university teams. This knowledge was used to guide the decision making process.

3.3.1 Western Washington University

WWU took the most hands-on approach observed, manufacturing a custom, purpose built V8 configuration engine for use in their Viking 30 prototype car. Utilizing the cylinder heads from two Kawasaki Ninja 250cc sport bikes, they designed and manufactured a block, and transmission casing, and re-purposed a rotating assembly from a Honda F1 motorcycle to be compatible, producing a 554cc V8. Peak power was claimed to be produced at 19,500 rpm. [12]

3.3.2 University of Maine

University of Maine selected their engine to maximize unmodified power-to-weight ratio. Utilized was an Aprilia RXV 550cc V-Twin configuration engine claiming 70 HP output at 13,000rpm, and weighing in at 72 lbs. [4]

3.3.3 University of New Hampshire

University of New Hampshire selected a 450cc Yamaha YFZ450R, single-cylinder motor. The Motor was selected to minimize weight, rather than maximize power. [2]

3.3.4 Discussion

During the survey, it was repeatedly observed that teams did not modify their chosen engines from their original, factory configuration. As these engines are taken from commercially available street motorcycles, they are not optimized for the conditions experienced at the FSAE competition. Even WWU's custom V8 – which never made it to competition – used cylinder heads and a rotating assembly from existing engines optimized for lower-grade fuels, and its Formula-One-like 19,500 rpm redline would result in severe power losses due to the specified restrictor plate.

Due to the nature of engine optimization as a typically-neglected area of Formula Car design, the team is enthusiastic over the opportunity to gain a significant competitive advantage.

4 Logistical Analysis

4.1 Timeline

Table 4. Timeline

Timeline	
Initial presentation	02/14/13
Global Learning Presentation	03/07/13
Synopsis	03/07/13
Selection of Motor	03/15/13
10% Report	03/20/13
Team Poster	03/21/13
Deadline for funding Acquisition	03/31/13
Motor Acquisition	03/31/13
Team Presentation	04/10/13
Final Design	04/10/13
25% Report	04/10/13
Initial Testing	04/31/13
Manufacturing	06/10/13
75% Report	Fall 2013
Final Testing	Fall 2013
Final Report	Fall 2013
Final Presentation	Fall 2013

4.2 Cost Estimates

Table 5. Cost Estimates

Item	Rate ($)	Qty	Est. Cost ($)	Actual Cost ($)
Honda CB 550 Engine	$600	1	$600.00	$600.00
Travel Expenses (per mile)	$0.56	250	$138.75	$138.75
Machine Shop (per hour)	$20	10	$200.00	$200.00
New Parts	-	-	$300.00	$300.00
Dynamometer (per hour)	$100	5	$500.00	$0.00
100 AKI Octane Fuel (per gallon)	$10	10	$100.00	$100.00
Poster Printing Costs	$108	1	$108.00	$108.00
Subtotal	-	-	$ 1,946.75	$ 1,446.75
Miscellaneous	15%	-	$ 292.01	$ 217.01
Total:	-	-	$ 2,238.76	$ 1,663.76

4.3 Man Hours

Table 6. Man Hours

Hours of Labor

Man Hours	Alex George	David Ducassi	Felipe Guajardo
Introduction Presentation	3	3	3
Research for Global Learning Presentation	3.5	3.5	3.5
Global Learning Presentation	2	2	2
General Research	11	7	5.5
Synopsis	3	3	3.5
Group Meeting / Planning	23.5	18	21.5
10% Report	7	4	2.5
Poster Design	0	5	2.5
Poster Finalization	2	3	2
25% Report	8	6	5
literature Survery	2.5	0	3
Timeline / Gantt Chart	0	6	0
Practice Presenation	4	4	4
Sponsor Relations	5	0	0
Miscellaneous			
Indivual Total:	74.5	64.5	58
Team Total:	197		

4.4 Gantt Chart

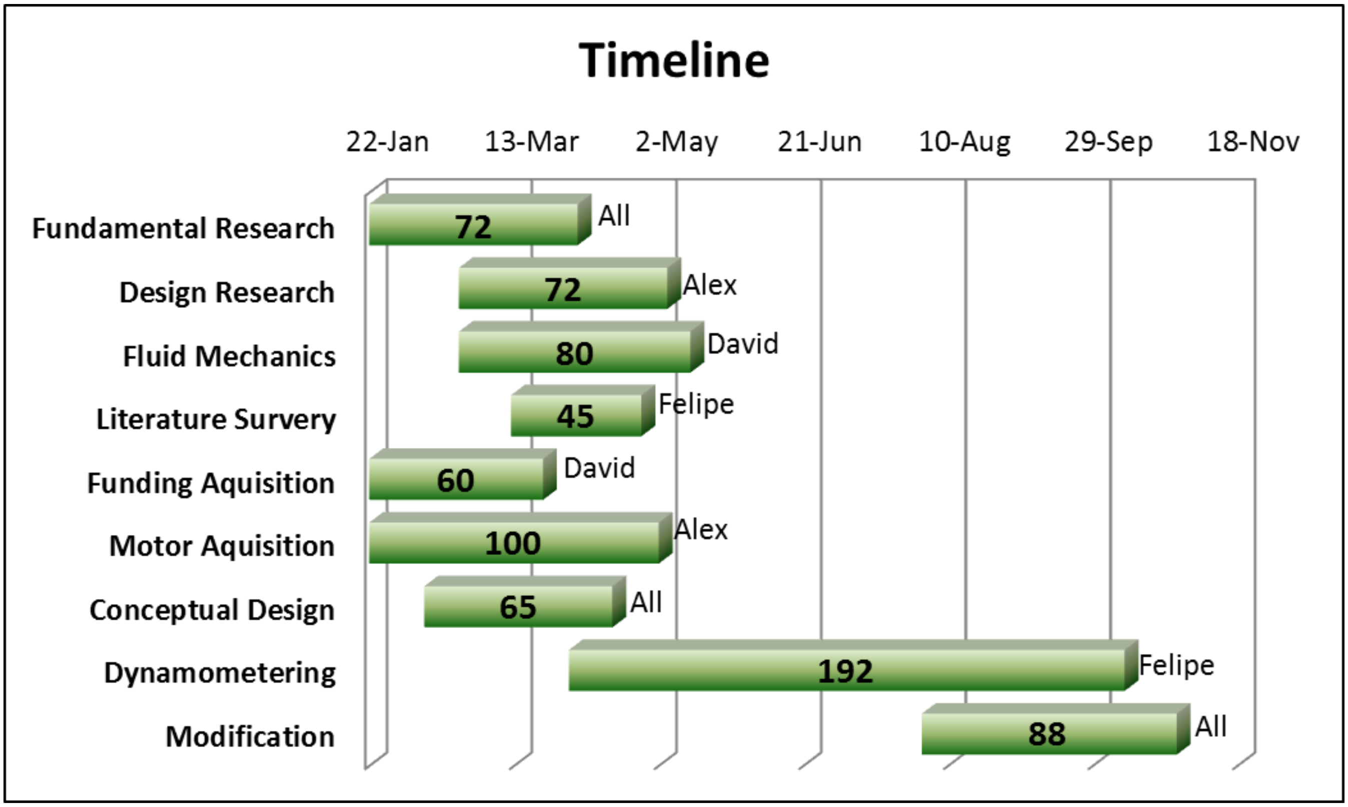

Figure 2: Gantt Chart

5 Appendices

5.1 Appendix A – Intake Airflow Through 20mm Restrictor

Flow (m3/s)	Pressure Ratio	Manifold Vacuum (psi)
0.000	1.00	0.00
0.016	0.99	-0.15
0.022	0.98	-0.29
0.027	0.97	-0.44
0.031	0.96	-0.59
0.035	0.95	-0.74
0.038	0.94	-0.88
0.041	0.93	-1.03
0.043	0.92	-1.18
0.045	0.91	-1.32
0.048	0.90	-1.47
0.050	0.89	-1.62
0.052	0.88	-1.76
0.053	0.87	-1.91
0.055	0.86	-2.06
0.057	0.85	-2.21
0.058	0.84	-2.35
0.059	0.83	-2.50
0.061	0.82	-2.65
0.062	0.81	-2.79
0.063	0.80	-2.94
0.064	0.79	-3.09
0.065	0.78	-3.23
0.066	0.77	-3.38
0.067	0.76	-3.53
0.068	0.75	-3.68
0.069	0.74	-3.82
0.070	0.73	-3.97
0.071	0.72	-4.12
0.071	0.71	-4.26
0.072	0.70	-4.41
0.073	0.69	-4.56
0.073	0.68	-4.70

Engine Specifications		
Original Displacement	544	cc
	0.000544	m^3
Original CR	9.0	:1
# of Cylinders	4	
Total Head Volume	68.0	cc
Bore Diameter	5.85	cm
Stroke increase	0.52	cm
Shave Head by	0	cm
Rod Length Change	-0.075	cm
TDC Piston Height Change	0.185	cm
New Displacement	600	cc
New Head Volume	48.1	cc
New CR	**13.5**	**:1**

Flow Demand		
New Displacement	600	cc
	0.0006	m^3/s
Max Speed	8500	rpm
Intake Flow	0.042493	m^3/s

Outputs		
Pressure Ratio	0.93	
Manifold Vacuum	-1.03	psi
	-7.09	kPa
Manifold Absolute Pressure	13.67	psi
	94.26	kpa
Intake Air Density	1.068	kg/m^3
Power To Maintain Manifold Vacuum	-0.301	kW
	-0.404	HP
Total Power	42.0	kW
	46.3	HP

0.074	0.67	-4.85
0.074	0.66	-5.00
0.075	0.65	-5.15
0.075	0.64	-5.29
0.075	0.63	-5.44
0.076	0.62	-5.59
0.076	0.61	-5.73
0.076	0.60	-5.88
0.077	0.59	-6.03
0.077	0.58	-6.17
0.077	0.57	-6.32
0.077	0.56	-6.47
0.077	0.55	-6.62
0.077	0.54	-6.76
0.077	0.53	-6.91
0.077	0.52	-7.06
0.077	0.51	-7.20
0.077	0.50	-7.35

Figure 3: Intake Airflow Through 20mm Restrictor

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