DEVELOPMENT OF ENGINE MANAGEMENT SYSTEM FOR MODIFIED DIESEL ENGINE FUELLED BY COMPRESSED NATURAL GAS (CNG)

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BORANG PENGESAHAN STATUS TESIS

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SESI PENGAJIAN: 2009/2010

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Examiner Signature

DEVELOPMENT OF ENGINE MANAGEMENT SYSTEM FOR MODIFIED DIESEL ENGINE FUELLED BY COMPRESSED NATURAL GAS (CNG)

NG ZIET HONG

Report submitted in partial fulfilment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Manufacturing Engineering.

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NOVEMBER 2009

SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Manufacturing Engineering.

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STUDENT'S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

Signature : ……………………………………

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Date : …………………………………….

To my beloved parents

NG GIM BENG
LEONG LAI PENG

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ABSTRACT

Air pollution is a serious global problem. Most country uses diesel engine, which produces substantial amount of emission, for the transportation sector. That is why development of green car is important to solve this problem. Compressed natural gas (CNG) as fuel to run diesel engine is one solution to decrease emission. This thesis deals with the finding of the best air-fuel ratio to be input into the Engine Control Unit (ECU) so that the ECU can manage the engine properly. Simulation of experiment of effect of variable AFR at variable manifold pressure for fixed RPM is run using GT-Power software. Hence a modified single cylinder diesel engine to run on CNG is studied and parameters are taken from the engine to be input into the template library of GT-Power software. In this project, the studied RPM is from 800 RPM to 3000 RPM with increment of 200 RPM. The manifold pressure range from 0.25 Bar to 1.00 Bar with increment of 0.05 Bar. As for the air-fuel ratio (AFR), the range is from 12.5 to 21. The results obtained from the simulation are viewed using GT-Post. The data are then transfer to Microsoft Excel to plot out the graph. From the graph, the best AFR is determined and input into the AFR table of Megatune software and burn into the ECU. Result shows that the engine operate well on lean mixture for all condition. Engine performance curve plotted from the result shows that the trend of curve is the same for all manifold pressure, the only difference is as manifold pressure increase the power output also increases. From published reports and books, it is stated that lean burning of air fuel mixture increases fuel conversion efficiency. Also optimum efficiency from natural gas is obtained when burnt in a lean mixture. It is concluded that the AFR table is successfully inputted with the best AFR at different RPM and manifold pressure.

ABSTRAK

Pencemaran udara merupakan masalah global yang serius. Kebanyakan negara menggunakan enjin diesel yang menjana banyak gas pencemaran untuk sektor pengangkutan mereka. Oleh sebab itu penyelidikan kereta 'green car' adalah penting untuk menyelesaikan masalah ini. Gas Semulajadi yang Dimampat (CNG) sebagai bahan bakar untuk enjin diesel merupakan satu penyelesaian untuk mengurangkan gas pencemaran. Thesis ini dibuat untuk mancari nisbah antara udara dan minyak yang terbaik untuk dimasuk ke dalam Unit Pengawalan Enjin (ECU) supaya Unit Pengawalan Enjin (ECU) dapat menguruskan enjin dengan baik. Simulasi untuk ujian kesan pemboleubah nisbah antara udara dan minyak dan pembolehubah tekanan kebuk untuk satu enjin revolusi (RPM) yang tetap dibuat dengan menggunakan perisian GT-Power. Oleh itu, modifikasi ke atas enjin diesel satu silinder untuk membakar Gas Semulajadi yang Dimampat (CNG) dikaji dan ukuran enjin yang diambil dimasukkan ke dalam perpustakaan templasi perisian GT-Power. Dalam project ini, enjin revolusi (RPM) yang dikaji ialah dari 800 RPM ke 3000 RPM dengan kenaikan 200 RPM. Tekanan kebuk pula meliputi tekanan 0.25 Bar hingga 1.00 Bar dengan kenaikan 0.05 Bar. Untuk nisbah udara kepada minyak pula, liputannya dari 12.5 ke 21. Keputusan yang diperolehi dari simulasi tersebut dilihat dengan menggnakan perisian GT-Post. Data tersebut kemudian akan dipindah ke Microsoft Excel untuk melukis graf. Dari graf, nisbah udara kepada minyak (AFR) yang terbaik ditentukan dan dimasuk ke dalam jadual nisbah udara kepada minyak (AFR) dalam perisian Megatune dan dibakar ke dalam Unit Pengawalan Enjin (ECU). Keputusan menunjukkan bahawa enjin tersebut beroperasi baik pada campuran rampinguntuk semua keadaan. Graf prestasi enjin yang diplot menunjukkan bahawa gaya semua graf pada semua tekanan kebuk yang diuji adalah sama, perbezaan yang ketara ialah bila takanan kebuk naik kuasa yang dijana juga menaik. Dari laporan dan buku yang diterbit, adalah dinyatakan bahawa pembakaran ramping bagi campuran udara minyak meningkatkan kecekapan penukaran bahan bakar. Di samping itu, kecekapan optimum dari gas semulajadi dicapai apabila campuran ramping berlaku. Jadi, dapat disimpulkan bahawa dengan berjayanya jadual nisbah udara kepada minyak (AFR) telah diisi dengan nisbah udara kepada minyak (AFR) yang terbaik pada enjin revolusi (RPM) dan tekanan kebuk yang berbeza.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

ε	Compression ratio
V_h	Swept volume
V_C	Compression volume
φ	Equivalence ratio
P	Pressure
V	Volume
m	Mass
R	Ideal gas constant
T	Absolute temperature

LIST OF ABBREVIATIONS

AFR	Air fuel ratio
BDC	Bottom Dead Centre
BHP	Brake Horse Power
CNG	Compressed Natural Gas
ECU	Engine Control Unit
MAP	Manifold Absolute Pressure
NGC	Natural Gas Car
NGV	Natural Gas Vehicle
NTC	Negative Temperature Coefficient
PTC	Positive Temperature Coefficient
RPM	Revolutions per Minute
TDC	Top Dead Centre
TPS	Throttle Position Sensor

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

This chapter gives a short description of the project background including several approaches. It then introduces objectives, scopes, problem statement of this project which is the development of engine management system for modified diesel engine fuelled by compressed natural gas (CNG).

1.2 PROJECT BACKGROUND

The problem of air pollution around the globe is real and serious. In most countries, the transportation sector uses mostly diesel-powered engines, which are heavy polluters. That is why ‘green’ cars are important. But designing, developing and marketing "green" cars is not an easy task and also to correctly convert a diesel engine to a low-polluting CNG or LPG engine is costly, technologically challenging and timely. That is why diesel-powered vehicles are still used till this day and fossil fuels still account for almost 75 percent of the world's energy consumption.

But till recent years, due to diesel price hike, concerns of increasing harmful emission and also the near depletion of crude oil reserve, vehicles that run on alternative fuel source is becoming increasingly important because they are more fuel efficient and environmentally friendly. It is noticeable that most of these countries that use diesel-powered engine vehicles import the diesel fuel they use and the fuel are expensive due to price hike. Most of them have large reserves of natural gas but no technology to use it in engines. The conversion of diesel engines to natural gas would decrease the country's dependence on imported foreign fuel and help them utilize the abundant natural gas resources.

1.3 PROBLEM STATEMENT

Modification is done to a diesel engine to run on CNG. The modification done will enabled the diesel engine to run on CNG but for the engine to run with high efficiency, hence more economic, and produce less emission, the engine will need to be managed properly. So the need of Engine Controlling Unit is the key component to manage the operation of the modified diesel engine properly. Hence, tuning the ECU is the purpose of this project so that the engine can be managed to run smoothly and efficiently.

1.4 OBJECTIVE

The objectives of this project are:

(i) To modify a diesel engine to run on CNG, that is more economical and produces less emission.
(ii) To tune the ECU so that the ECU can manage the operation of the engine to run efficiently and produce less emission.
(iii) To obtain optimum value for air fuel ratio (AFR).

1.5 PROJECT SCOPE

In order to achieve the objectives of the project, the following scopes are listed:

(i) Tuning will be done on a Engine Control Unit (ECU).

(ii) ECU used is Megasquirt II version 3.

(iii) Tuning software used is Megatune version 2.55.

(iv) Engine used is a single cylinder diesel engine with capacity of 400cc.

(v) Parameters monitored by the ECU:

a) Dependent parameters:

i. Pressure of intake manifold, the in-cylinder pressure, the exhaust pressure.

b) Independent parameters:

i. The speed of the engine, which is the RPM.

(vi) From 800 rpm to 3000 rpm, with increment of 200 rpm, variable AFR is tested to obtained best engine power at variable manifold pressure which is from 0.25 Bar to 1 Bar with increment of 0.05 Bar.

(vii) Simulation for obtaining best AFR by using GT-Power version 6.1.0.

1.6 SUMMARY

Chapter 1 discussed briefly about the project's problems statement, objectives and the scope of the project, which will be conducted to achieve the development of engine management system for modified diesel engine fuelled by compressed natural gas (CNG). This chapter is as a fundamental for this project and as a guidelines to complete this project research.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter will discuss about how a diesel engine works and also how a CNG engine works. The differences of working principle and components of both engine is compared to determine what modifications are needed to convert a diesel engine to run on CNG. The Engine Management System is also studied to recognize its working principle and also about the sensors the system uses to manage the operation of the engine. The diesel fuel and CNG fuel is compared also to recognize both fuel's properties, advantage and disadvantage.

2.2 DIESEL ENGINES

Diesel engines are internal combustion engine. They are designed to convert the chemical energy available in fuel into mechanical energy. This mechanical energy moves pistons up and down inside cylinders. The pistons are connected to a crankshaft, and the up-and-down motion of the pistons, known as linear motion, creates the rotary motion needed to turn the wheels of a car forward. In diesel engine, the air is compressed first and then the fuel is injected. Diesel engines have high compression ratio, a compression ratio of 14:1 to as high as 25:1, which produces a pressure range of about 30 bars to 50 bars. When air is compressed to this pressure range it will be heated to about 550^{o} C to 900^{o} C, which is high enough to ignite the injected fuel. (Bosch, 2005)

2.2.1 Four Stroke Cycle (Bosch, 2005)

The diesel engine uses a four-stroke combustion cycle. The four strokes are:

(i) Intake stroke - The intake valve opens up, letting in air and moving the piston down. When the piston reaches Bottom Dead Center (BDC), the cylinder capacity is at its greatest (V_h+V_c).

(ii) Compression stroke -The inlet valve and outlet valve now closed. The piston moves back up and compresses the air to the degree determined by the engine's compression ratio and the air is heated up in the process. When compression stroke is almost complete, the fuel injection system injects fuel at high pressure into the hot, compressed air.

(iii) Ignition stroke – After the ignition lag (a few degree of crank shaft rotation) has elapsed, the ignition stroke begins. The finely atomized and easily combustible diesel fuel spontaneously ignites and burns due to the heat of the air. As a result, the pressure in the cylinder rises up and the pressure force pushes the piston downwards.

(iv) Exhaust stroke –Before the piston reaches bottom dead center, the exhaust valve opens and the exhaust gas flow out of the cylinder. As the piston moves back to the top again, it pushes out the remaining exhaust gas.

2.2.2 Diesel Cycle

In the diesel engine, air is compressed adiabatically with a compression ratio typically between 14 and 25. This compression raises the temperature to the ignition temperature of the fuel mixture which is formed by injecting fuel once the air is compressed. The ideal air-standard cycle is modeled as a reversible adiabatic compression followed by a constant pressure combustion process, then an adiabatic expansion as a power stroke and an isovolumetric exhaust. A new air charge is taken in at the end of the exhaust. Figure 2.1 presents the processes as described. (HyperPhysics, Undated)

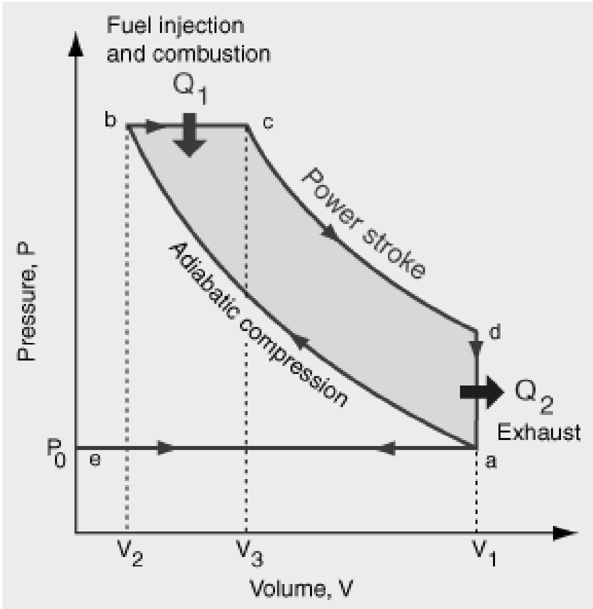


Figure 2.1: Air standard diesel engine cycle

Source: HyperPhysics (Undated)

2.2.3 Direct Injection

Diesel engines use direct fuel injection -- the diesel fuel is injected directly into the cylinder. The injector on a diesel engine is its most complex component and has been the subject of a great deal of experimentation -- in any particular engine, it may be located in a variety of places such as directly above the combustion chamber or below the intake valve. The injector has to be able to withstand the temperature and pressure inside the cylinder and still deliver the fuel in a fine mist. Getting the mist circulated in the cylinder so that it is evenly distributed is also a problem, so some diesel engines employ special induction valves, pre-combustion chambers or other devices to swirl the air in the combustion chamber. The design of piston crown such as dish-shaped, dome-shaped and intricate contour also helps to improve swirl in combustion chamber. As a result, this will improve the ignition and combustion process. (Marshall, 2009)

2.2.4 Compression

Rudolf Diesel theorized that higher compression leads to higher efficiency and more power. This happens because when the piston squeezes air with the cylinder, the air becomes concentrated. Diesel fuel has high energy content, so the likelihood of diesel reacting with the concentrated air is greater. Another way to think of it is when air molecules are packed so close together, fuel has a better chance of reacting with as many oxygen molecules as possible. (Bosch, 2005)

The compression ratio, ε, of a cylinder results from its swept volume, V_h, and its compression volume, V_c, thus: (Bosch, 2005)

$$\varepsilon = \frac{V_h + V_c}{V_c} \tag{2.1}$$

The compression ratio of an engine has a decisive effect on the following:

(i) The engine's cold-starting characteristic
(ii) The torque generated
(iii) The fuel consumption
(iv) How noisy the engine is
(v) The pollutant emissions

2.3 COMPRESSED NATURAL GAS ENGINE

The engine of a Natural Gas Car (NGC) or a Natural Gas Vehicle (NGV) works in a way that is very similar to that of a standard internal combustion engine that runs on gasoline. It uses the cylinder "sparkplug" piston concept to generate motion from controlled fuel combustion. They differ mainly on the flammability, volume, and ignitability of the fuel used. (Tech-FAQ, 2009)

2.3.1 Four Stroke Cycle

The CNG engine uses a four-stroke combustion cycle. The four strokes are:

(i) Intake stroke - The inlet valve open, the piston first descends on the intake stroke. An ignitable mixture of gasoline vapor and air is drawn into the cylinder by the partial vacuum thus created.

(ii) Compression stroke - The mixture is compressed as the piston ascends on the compression stroke with both valves closed.

(iii) Ignition stroke –As the end of the stroke is approached, the charge is ignited by an electric spark. The power stroke follows, with both valves still closed and the gas pressure, due to the expansion of the burned gas, pressing on the piston head or crown to drive the piston down.

(iv) Exhaust stroke –During the exhaust stroke the ascending piston forces the spent products of combustion through the open exhaust valve. The cycle then repeats itself.

Each cycle thus requires four strokes of the piston—intake, compression, power, and exhaust—and two revolutions of the crankshaft. (Britannica, Undated)

2.3.2 Otto Cycle

The Otto Thermodynamic Cycle is used in spark ignition internal combustion engines. Figure 2.2 shows a p-V diagram of the ideal Otto cycle. Using the engine stage numbering system, we begin at the lower left with Stage 1 being the beginning of the intake stroke of the engine. The pressure is near atmospheric pressure and the gas volume is at a minimum. Between Stage 1 and Stage 2 the piston is pulled out of the cylinder with the intake valve open. The pressure remains constant, and the gas volume increases as fuel/air mixture is drawn into the cylinder through the intake valve. Stage 2 begins the compression stroke of the engine with the closing of the intake valve. Between Stage 2 and Stage 3, the piston moves back into the cylinder, the gas volume decreases, and the pressure increases because work is done on the gas by the piston. Stage 3 is the beginning of the combustion of the fuel/air mixture. The combustion occurs very quickly and the volume remains constant. Heat is released during combustion which increases both the temperature and the pressure, according to the equation of state. Stage 4 begins the power stroke of the engine. Between Stage 4 and Stage 5, the piston is driven towards the crankshaft, the volume in increased, and the pressure falls as work is done by the gas on the piston. At Stage 5 the exhaust valve is opened and the residual heat in the gas is exchanged with the surroundings. The volume remains constant and the pressure adjusts back to atmospheric conditions. Stage 6 begins the exhaust stroke of the engine during which the piston moves back into the cylinder, the volume decreases and the pressure remains constant. At the end of the exhaust stroke, conditions have returned to Stage 1 and the process repeats itself.

During the cycle, work is done on the gas by the piston between stages 2 and 3. Work is done by the gas on the piston between stages 4 and 5. The difference between the work done by the gas and the work done on the gas is the area enclosed by the cycle curve and is the work produced by the cycle. The work times the rate of the cycle (cycles per second) is equal to the power produced by the engine. (NASA, 2008)

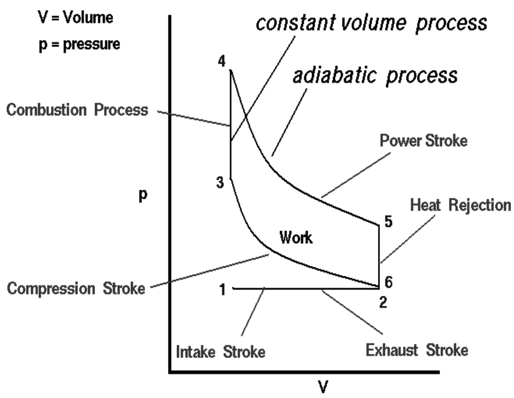


Figure 2.2: Ideal Otto cycle

Source: NASA (2008)

2.4 ENGINE MANAGEMENT SYSTEM

Engine Management System is the Engine Control Module, or the "brain", that controls the fuel supply and the ignition by combining the two separate functions into one main system. The brain controls the whole of the combustion process, making the engine more efficient and less polluting. (Motorsave, 2006)

The Engine Control Unit (ECU) uses closed-loop control, a control scheme that monitors outputs of a system to control the inputs to a system, managing the emissions and fuel economy of the engine. The system works by using sensors located on the engine. These sensors measure parameters of the engine such as the speed and the temperature of the intake air, coolant temperature, amount of oxygen in the exhaust, the speed and position of the crankshaft and etc. These information is then gathered and the ECU performs millions of calculations each second, including looking up values in tables, calculating the results of long equations to decide on the best spark timing and determining how long the fuel injector is open.. This makes the engine more efficient, economical, less polluting and more powerful. The fuel is injected into the engine at high pressure and at a calculated rate by electronically controlled injectors. The ignition is also controlled by the brain to allow for movement in ignition timing which will produce better combustion so more of the fuel entering the engine can be burnt. (Karim, 2009)

2.5 SENSORS

Sensors register operating states such as engine speed. They convert physical quantities such as pressure or chemical quantities such as exhaust gas concentration into electric signals. (Bosch, 2005)

2.5.1 Engine Speed Sensor

Engine speed sensors are used in Motronic systems for measuring the engine speed and determining the crankshaft position.

The Inductive Speed Sensor is one type of engine speed sensor. The sensor is mounted directly opposite a ferromagnetic trigger wheel where both are separated by a narrow air gap in the range of 0.5mm to 1.8mm. The sensor works through magnetic induction. The teeth of the trigger wheel will concentrate the magnet's leakage flux while the gap will weakens the magnetic flux. When the trigger wheel rotates, the magnetic-flux changes induce sinusoidal output voltage in the coil which is proportional to the rate of change of the flux and thus the engine speed. The speed range of the inductive speed sensor is 20 to 7000 rpm. (Bosch, 2005)

2.5.2 Manifold Absolute Pressure Sensors

Manifold Absolute Pressure Sensors or MAP Sensors, are used to measure inlet manifold 'pressure' to give an indication of engine load. These sensors are generally used in "Speed/Density" or "Manifold Pressure Controlled" engine management systems that do not use an Air Flow/Mass Sensor. The MAP sensor measures "Absolute" pressure not "Gauge" pressure, so normal atmospheric pressure is a value of 1 bar. In the MAP sensor there is a silicon chip mounted inside a reference chamber. On one side of the chip is a reference pressure. This reference pressure is either a perfect vacuum or a calibrated pressure, depending on the application. On the other side is the pressure to be measured. When the silicon chip flexes with the change in pressure, the electrical resistance of the chip changes. These changes in resistance alter the voltage signal. The ECU interprets the voltage signal as pressure and any change in the voltage signal means there was a change in pressure. Intake manifold pressure is directly related to engine load. Knowing what the intake manifold pressure is, the ECU can calculate how much fuel to inject, when to ignite the cylinder and other functions. The MAP sensor is located either directly on the intake manifold or it is mounted high in the engine compartment and connected to the intake manifold with vacuum hose. The MAP sensor uses a perfect vacuum as a reference pressure. The difference in pressure between the vacuum pressure and intake manifold pressure changes the voltage signal. The MAP sensor converts the intake manifold pressure into a voltage signal. Figure 2.3 shows how a MAP sensor looks like and how it operates. (Toyota Motor Sales, Undated)

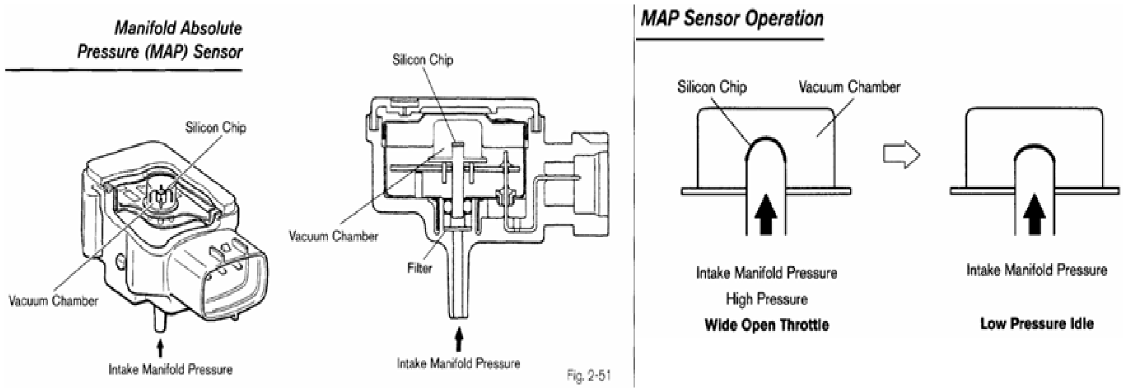


Figure 2.3: Manifold Absolute Pressure Sensor

Source: Toyota Motor Sales (Undated)

2.5.3 Intake Air Temperature Sensor

The purpose of an intake air temperature sensor is to help the computer calculate air density. A change in temperature changes the resistance in the sensor. Simply stated, the higher the air temperature gets the less dense the air becomes. As the air becomes less dense the computer knows that it needs to lessen the fuel flow. If the fuel flow was not changed the engine would become rich, possibly losing power and consuming more fuel. (Free Engine Info, Undated)

2.5.4 Coolant Temperature Sensor

Coolant temperature sensor changes resistance as the engine's coolant temperature changes. The sensor's output is monitored by the engine computer to regulate various ignition, fuel and emission control functions, variable valve timing, transmission shifting and to turn the radiator-cooling fan on and off as needed. In the PTC (Positive Temperature Coefficient) type of sensor, ohms go up with temperature. In the NTC (Negative Temperature Coefficient) type, resistance goes down as heat goes up. Figure 2.4 shows how a Coolant Temperature Sensor looks like. (Matthew, 2002)

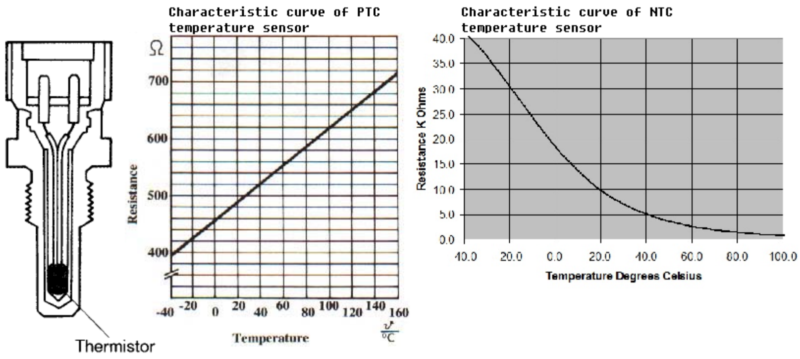


Figure 2.4: Coolant Temperature Sensor

Source: SJM Autotechnik (2007)

2.5.5 Throttle Position Sensor

A throttle position sensor (TPS) is a sensor used to monitor the position of the throttle in an internal combustion engine. The sensor is usually located on the throttle body so that it can directly monitor the position of the throttle valve and converts the throttle valve angle into electrical signal. (Toyota Motor Sales, Undated)

The ECU uses throttle valve position information to know:

(i) Engine mode: idle, part throttle, wide open throttle.
(ii) Air-fuel ratio correction.
(iii) Power increase correction.
(iv) Fuel cut control.

The sensor is a potentiometer; it provides a variable resistance dependent upon the position of the valve. Five volts are supplied to the TPS from the VC terminal of the ECU. The TPS voltage signal is supplied to the VTA terminal. A ground wire from the TPS to the E2 terminal of the ECU completes the circuit. Figure 2.5 shows the Throttle Position Sensor circuit. (Toyota Motor Sales, Undated)

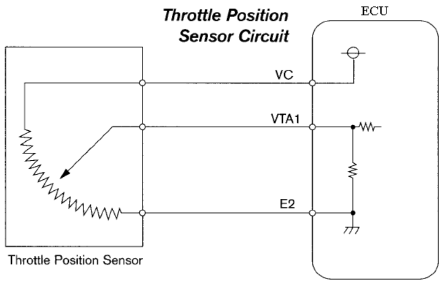


Figure 2.5: Throttle Position Sensor circuit.

Source: Toyota Motor Sales (Undated)

At idle, voltage is approximately 0.6 – 0.9 volts on the signal wire. From this voltage, the ECU knows the throttle plate is closed. At wide open throttle, signal voltage is approximately 3.5 – 4.7 volts. (Toyota Motor Sales, Undated)

2.5.6 Oxygen Sensor

The oxygen sensor is positioned in the exhaust pipe and can detect rich and lean mixtures. The mechanism involves a chemical reaction that generates a voltage. It is constantly making a comparison between the oxygen inside the exhaust manifold and air outside the engine. If this comparison shows little or no oxygen in the exhaust manifold, a voltage is generated. The engine's computer looks at the voltage to determine if the mixture is rich or lean, and adjusts the amount of fuel entering the engine accordingly (Marshall, 2009). The difference between the heated oxygen sensor and a non-heated sensor is the ability to operate during engine warm up or low load conditions. The optimum operating temperature of oxygen sensor is 400° C. For the heated sensor, it uses an internal coil to heat the ceramic element to the desired 400° Celsius in 30 or 40 seconds. This temperature is also maintained when the car is at idle for extended periods of time or is under low load conditions where the exhaust gas temperatures fall below 400° C. The non-heated sensor relies on the exhaust gas heat to keep it at its operating temperature. Under normal operating conditions the exhaust gas temperature will be much greater than 400° C. and so heating is not necessary for the sensor. This works most of the time but when the engine is operating at low load conditions or idling, the exhaust gas temperature will drop below 400° C. This will cause the sensor's temperature to drop below its desired operating temperature and hence shows a leaner than actual mixture as its output drops to zero. (Bruce and Al, 2008)

There are two types of Oxygen Sensor, which are the Wideband and Narrowband Oxygen Sensor. Narrowband oxygen sensors are extremely imprecise. They can't tell the computer the exact air/fuel ratio like wideband oxygen sensors. Technically narrowband oxygen sensors produce an exponential voltage signal, whereas wideband oxygen sensors produce a linear current signal. When the air/fuel ratio is perfectly balanced, a narrowband O2 sensor produces a signal of about 0.5 volts. When the fuel mixture goes rich, even just a little bit, the O2 sensor's voltage output shoots up quickly to its maximum output of close to 0.9 volts. Conversely, when the fuel mixture goes lean, the sensor's output voltage drops to 0.1 volts (Water Fuel LLC, 2009). With a narrow band sensor, it only tells whether the air-fuel mixtures are rich or lean, but not by how much. Figure 2.6 shows that for a narrow band sensor, the 12.5:1 AFR required for maximum power can give O2 voltage from 0.8 to 0.95, yet this same range of O2 voltages can indicate mixtures from 10:1 to 14.5:1. So its reliability cannot be used to set mixtures for full power. With a wide-band sensor, 12.5:1 corresponds to 2.08 volts, and 2.08 volts means 12.5:1. Thus there is no ambiguity over air-fuel ratio and voltages. Hence, any mixture in the range that is likely to be used can be measured, from full power through to maximum economy. (Bruce and Al, 2008)

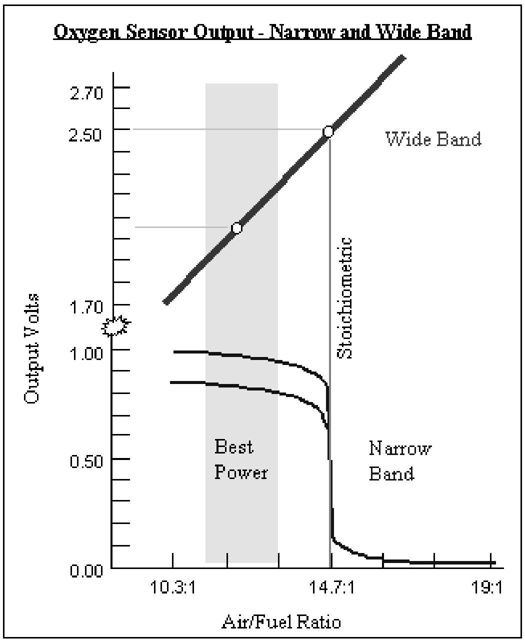


Figure 2.6: Graphs of output volts of Wideband and Narrowband Oxygen Sensor

Source: Bruce and Al (2008)

2.5.7 Knock Sensor

A knock sensor assures that the engine is producing as much power and fuel economy as is possible. It allows the engine to run with the ignition timing as far advanced as possible. The computer will continue to advance the timing until the knock sensor detects pinging. At that point the computer retards the ignition timing just enough for the pinging to stop.

The knock sensor responds to spark knock caused by over advanced timing. A knock is a sudden increase in cylinder pressure caused by preignition of the air fuel mixture as the flame front moves out from the spark plug ignition point. Pressure waves in the chamber crash into the piston or cylinder walls resulting in a sound known as a knock or ping. This is caused by using a fuel with a low octane rating, overheating, or over advanced timing. Sometimes it can be caused by hot carbon deposits on the piston or cylinder head that raise compression.

Knock occurs when the air-fuel mixture doesn't burn smoothly or is ignited too soon. Knock can be caused by "hot spots" in the cylinder, such as carbon deposits or spark plugs that are too hot for the engine, or high combustion chamber temperatures. These hot spots ignite the air-fuel mixture before the spark plug fires. The result is two flame fronts, one from the hot spot and one from the spark plug, occurring at the same time. The fronts slam into each other violently as the burning gases rapidly expand. Knocking also occurs if spark timing is advanced too far. In this type of situation, there is only one flame front, but it happens while the piston is too far before Top Dead Centre (TDC). Instead of a collision between flame fronts, there is a "collision" between the rapidly expanding gases from combustion and the top of the piston as it moves up the cylinder wall. (Wells, 2000)

2.6 COMPRESSED NATURAL GAS (CNG)

CNG is odorless, colorless, and tasteless. It consists mostly of methane and is drawn from gas wells or in conjunction with crude oil production. The octane rating for CNG is high; in a dedicated engine, a CNG vehicle's power, acceleration, and cruise speed can be great. In addition, due to the cleaner burning characteristics of natural gas, CNG vehicle engines can run more efficiently, thereby extending the life of the vehicle. In heavy-duty vehicles, CNG engines are also generally less noisy than diesel engines. (EPA, Undated)

The advantages of Compressed Natural Gas are as follow: (Barbotti CNG, 2002)

(i) Compressed natural gas is the cleanest burning fuel operating today. This means less vehicle maintenance and longer engine life.
(ii) Reductions in carbon monoxide emissions of 90 to 97 percent, and reductions in carbon dioxide emissions of 25 percent.
(iii) Reductions in nitrogen oxide emissions of 35 to 60 percent.
(iv) Potential reductions in nonmethane hydrocarbon emissions of 50 to 75

percent.

(v) Fewer toxic and carcinogenic pollutants and little to no particulate matter produced.

(vi) Dedicated Natural Gas Vehicles (NGV) has little or no emissions during fueling. In gasoline vehicles, fueling emissions account for at least 50% of a vehicle's total hydrocarbon emissions.

(vii) Intervals between oil changes for natural gas vehicles are dramatically extended--anywhere from 10,000 to 25,000 additional miles depending on how the vehicle is used.

(viii) Natural gas does not react to metals the way gasoline does, so pipes and mufflers last much longer.

(ix) Natural gas gives the same mileage as gasoline in a converted vehicle.

(x) Dedicated CNG engines are superior in performance to gasoline engines.

(xi) Because CNG is already in a gaseous state, CNG vehicles have superior starting and driveability, even under severe hot and cold weather conditions.

(xii) CNG vehicles experience less knocking and no vapor locking.

(xiii) Natural gas is cheaper compared to gasoline or diesel.

The disadvantages of Compressed Natural Gas are as follow: (Haresh, 2008)

(i) To store CNG the vehicle need to have a pressurized fuel tank in it, which always carries with it some safety concerns.

(ii) Another disadvantage of CNG is that the fuelling process is very slow.

(iii) The energy density of natural gas and CNG is low, thus the performance of the engine reduces.

(iv) Since CNG is a gaseous fuel, it has low volumetric efficiency.

Table 2.1: Combustion related properties of CNG

Properties	CNG
Motor octane number	120
Molar mass (kg/mol)	16.04
Carbon weight fraction (mass%)	75
$(A/F)_s$ = Stoichiometric air fuel ratio	16.79
Stoichiometric mixture density (kg/m^3)	1.24
Lower heating value (MJ/kg)	47.377
Lower heating value of stoic. mixture (MJ/kg)	2.72
Flammability limits (vol% in air)	5–15
Spontaneous ignition temperature (°C)	645

Source: Aslam et al. (2005)

Table 2.2: Typical composition (volumetric %) of CNG

Component	Symbol	Volumetric %
Methane	CH_4	94.42
Ethane	C_2H_6	2.29
Propane	C_3H_8	0.03
Butane	C_4H_{10}	0.25
Carbon dioxide	CO_2	0.57
Nitrogen	N_2	0.44
Others	(H_2O+)	2.0

Source: Aslam et al. (2005)

2.7 DIESEL FUEL

Petroleum fuel, or crude oil, is naturally found in the Earth. When crude oil is refined at refineries, it can be separated into several different kinds of fuels, including gasoline, jet fuel, kerosene and, of course, diesel. Diesel fuel is heavier and oilier and because of that diesel fuel evaporates more slowly. It contains more carbon atoms in longer chains where the formula is $C_{14}H_{30}$. Diesel fuel has a higher energy density. On average, 1 gallon (3.8 L) of diesel fuel contains approximately 155×10^6 joules (147,000 BTU). (EIA, 2006)

The advantages of Diesel Fuel are as follow: (Aaron, 2004)

(i) Diesel fuel gives about 20 to 30% more fuel economy.
(ii) Diesel fuel provides significantly more torque than any other fuel.
(iii) Diesel fuel has more energy content.
(iv) Diesel fuel is safer and less flammable than other form of fuel, resulting in safer storage and handling.
(v) Diesel fuel is less corrosive than other fuel.
(vi) Diesel fuel has a higher combustion temperature than other fuel.

The Disadvantages of Diesel Fuel are as follow: (Marshall, 2009)

(i) Diesel produces more carbon dioxide and nitrogen oxide. Burning 100L of diesel emits about 270kg of carbon dioxide into the atmosphere.
(ii) Diesel cars emit more particles of soot into the air. This contributes to smog, acid rain and health issues like asthma and lung cancer.
(iii) The initial cost of buying a diesel car is more than a normal car running on petrol.
(iv) Diesel is more expensive compared to CNG.

Table 2.3: Fuel properties of diesel and CNG

Property	Diesel	CNG
Chemical formula	$C_{12}H_{26}$	CH_4
State	Liquid	Gas
Energy content	110%	25%
Octane rating		120~130
Auto ignition temp.	220°C	450°C
Stoichiometric ratio	15	16.79

Source: Kaleemuddin and Rao (2009)

2.8 SUMMARY

The working principle of diesel engine and CNG engine works is discussed. Modifications done to convert a diesel engine to run on CNG are based on the comparison of differences of working principle and components of both engines. The Engine Management System and also the sensors the system uses to manage the operation of the engine are studied and recognized. The diesel fuel and CNG fuel is compared also to recognize both fuel's properties, advantage and disadvantage.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

Chapter 3 discusses methodology of the project in general, with a specific focus on the development of engine management system for diesel engine fueled by CNG. The work is based on methodology flow chart. Chapter 3 presents current progress such as modification done on the diesel engine. Understanding prior and current research in this project provides method for the research contributions outlined in subsequent chapters.

3.2 METHODOLOGY FLOW CHART

Methodology flow chart is used as guidelines and the sequences to make this project go on schedule. As illustrated in Figure 3.1, firstly information on literature review was being studied on the field that regards to this project, for example the working principle of diesel and CNG engine, the engine management system, the sensors and others. Then, the process begins with study on the modification done and principle and method of tuning. Then the process proceeds to the simulation of the experiment of effect of variable AFR to engine power at a fixed RPM at variable manifold pressure using GT-Power to obtain the best power output at a fixed RPM. Then the best air fuel ratio is determined and input into the AFR table of Megatune software and burn into the ECU.

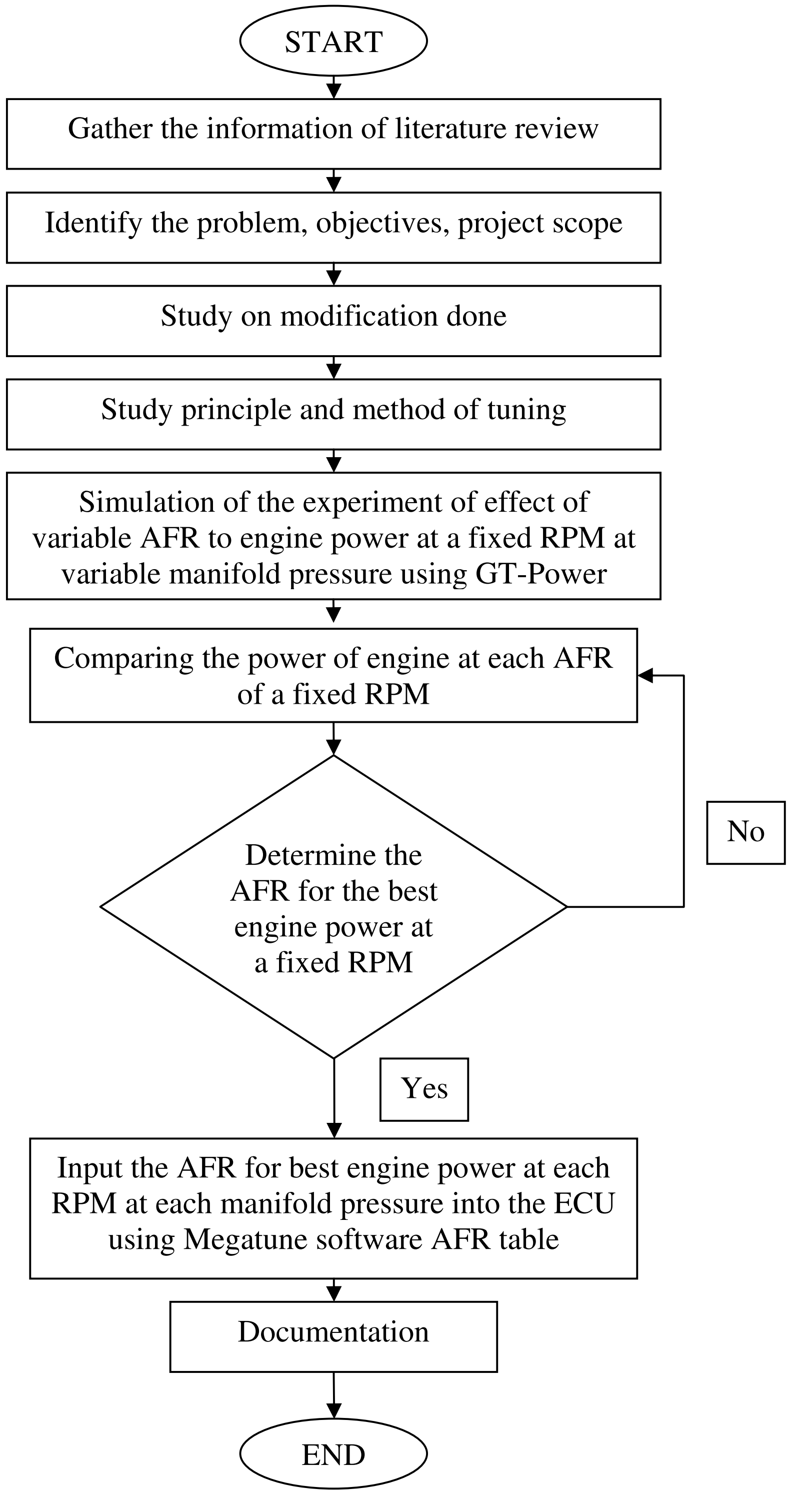


Figure 3.1: Methodology flow chart

3.3 LITERATURE STUDIES

This project is required to do preliminary study and research about project. The various resources have been searching from books, articles, journals, and internet webpage to develop a personal plan for information processing. Information like working principle of diesel engine and CNG engine, properties of diesel and CNG fuel and others generate a basic idea of how variable air fuel ratio have effect on engine power at a fixed RPM.

3.4 MODIFICATION DONE

To convert a diesel engine to a natural gas engine comprises of inserting a spark plug into a diesel fuel injector opening in a cylinder head; installing a throttle body on the diesel engine; modifying the piston, whereby the compression ratio of the piston is decreased during operation of the piston; installing a spark distributor and spark ignition coil; installing a CNG fuel pressure cylinder tank; installing a CNG fuel pressure regulator.

Table 3.1: Original specification of the diesel engine used

Model	CF186F
Type	vertical, single cylinder, 4-stroke,air-coolig diesel
Combustion system	direct injection
Bore x Stroke (mm x mm)	86 × 70 mm
Total displacement (L)	0.406 L
Compression ratio	20
Rated output	6.50 kW(8.8 HP) 3600 r/min
Max output	7.30 kW(10.0 HP) 3600 r/min
Specific fuel	273.5 g/kW.h
Cooling system consumption	Forced air cooled

Source: China Depot (Undated)

Table 3.2: Conversion specification

Engine and Intake Parameter	Diesel Engine	CNG Engine
Bore (mm)	86.0	86.0
Stroke (mm)	70.0	70.0
Displacement (cc)	407.0	407.0
Compression ratio	20.28	14.5
Intake Valve Close (CA)	496	496
Exhaust Valve Open (CA)	191	191
Intake Valve Open (CA)	361	361
Exhaust Valve Close (CA)	325	325
Ignition system	Compression	Spark
Fuel system	Direct Injection	Port Injection
Fuel	Diesel	Natural Gas
Intake port diameter in (mm)	40.69	40.69
Intake port diameter out (mm)	32.78	32.78
Intake port length (mm)	55.2	55.2
Discreatization length (mm)	34.4	34.4

Source: Semin et al. (Undated)

3.4.1 Spark Plug Assembly

To convert a diesel engine to a spark-ignited natural gas engine, the first step involves modifying the cylinder head to accept spark plugs. This modification involves removing the diesel fuel injector and requires machining of the diesel injector hole and threading the hole to accept the spark plug. Figure 3.2 shows the working spark plug.

Figure 3.2: Spark plug

3.4.2 Throttle Body

A throttle body is installed to regulate the flow of air from the atmosphere to an intake manifold that provides an air/fuel mixture to engine combustion chambers. Providing maximum airflow to the combustion chamber enables the engine to reach maximum power. The engine produces a minimum amount of power when airflow is almost entirely restricted, such as when the engine is idling. Of course, airflow must be regulated by the throttle body between maximum and minimum engine power so that the engine may perform under a variety of vehicle operating conditions. Figure 3.3 shows a working throttle body.

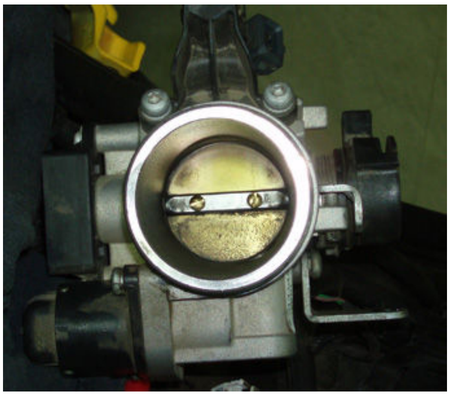


Figure 3.3: Throttle body

3.4.3 Modified Piston

To convert a diesel engine to a spark-ignited natural gas engine, the third step involves modifying the piston configuration. The diesel compression ratio is typically large, on the order of about 14:1 to 25:1. In order to convert a diesel engine to a natural gas burning engine, the ratio is lowered to about 9:1 to 12:1, and preferably between 10:1 to 11:1 or to be more precisely at a ratio of 10.5:1. Figure 3.4 shows the piston crown machined for lower compression ratio.

Figure 3.4: Modified piston

3.4.4 Spark Distributor

A distributor is a device that routes high voltage from the ignition coil to the spark plugs in the correct firing order. Since a spark plug is installed to the cylinder head, a spark distributor is needed to route high voltage from the ignition coil to the spark plug. Figure 3.5 shows a working spark distributor.

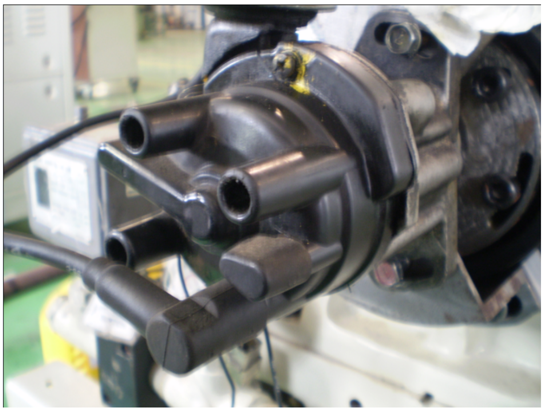


Figure 3.5: Spark distributor

3.4.5 Ignition Coil

An ignition coil is an induction coil in an automobile's ignition system which transforms the battery's 12 volts to the thousands of volts needed to spark the spark plugs. Since spark plug is installed, an ignition coil must be installed to supply the spark plug with the spark needed to ignite the fuel. Figure 3.6 shows the ignition coil.

Figure 3.6: Ignition coil

3.4.6 CNG Fuel Tank

Compressed Natural Gas is stored in a special tank made of either aluminium, steel or others and the tanks are wrapped with a composite layer on the inner side of the tank. It can withstand a pressure of 250 bar because the natural gas is stored in the tank in liquid form so that more gas can be stored without having to refill frequently. Figure 3.7 shows the CNG fuel tank used to store the CNG fuel.

Figure 3.7: CNG fuel tank

3.4.7 Pressure Regulator

Pressure regulator is used to reduce the pressure of CNG fuel in the tank. From a pressure of 250 bar, it reduces the CNG fuel pressure to 4 bar. Figure 3.8 shows the pressure regulator used.

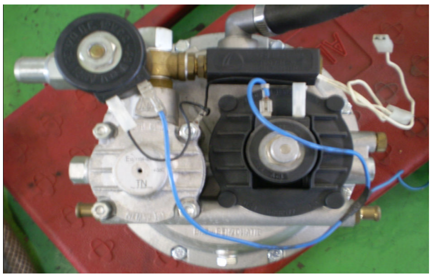


Figure 3.8: Pressure regulator

3.5 WIRING DIAGRAM

Figure 3.9 shows the wiring diagram of the engine management system. Connection of sensors, injectors and other components to their respective port is showed clearly in the wiring diagram. A relay board is used to protect the engine control unit (ECU) in case any short circuits were to happen.

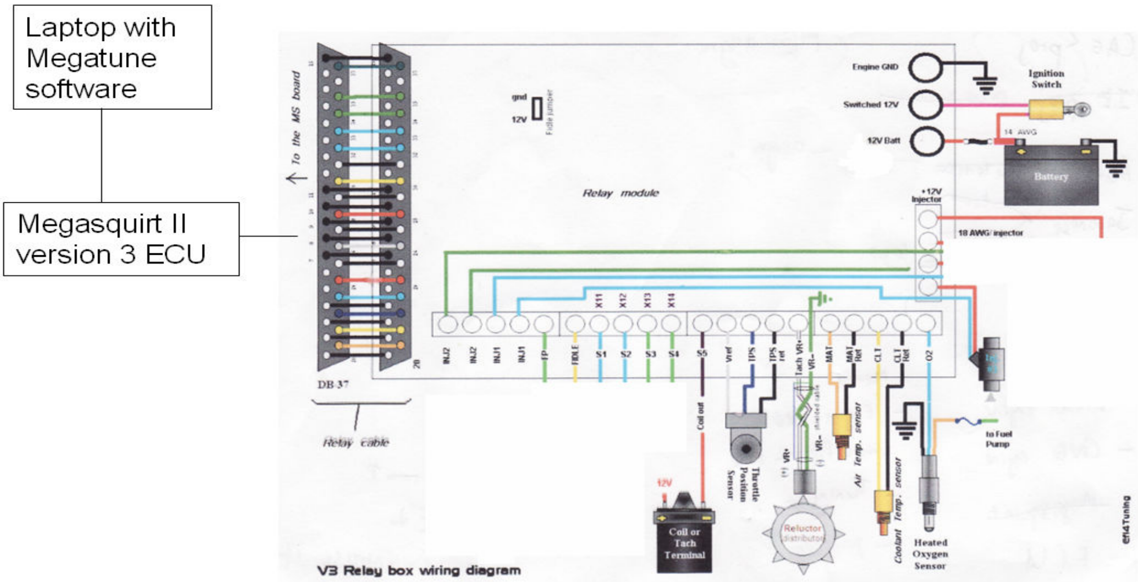
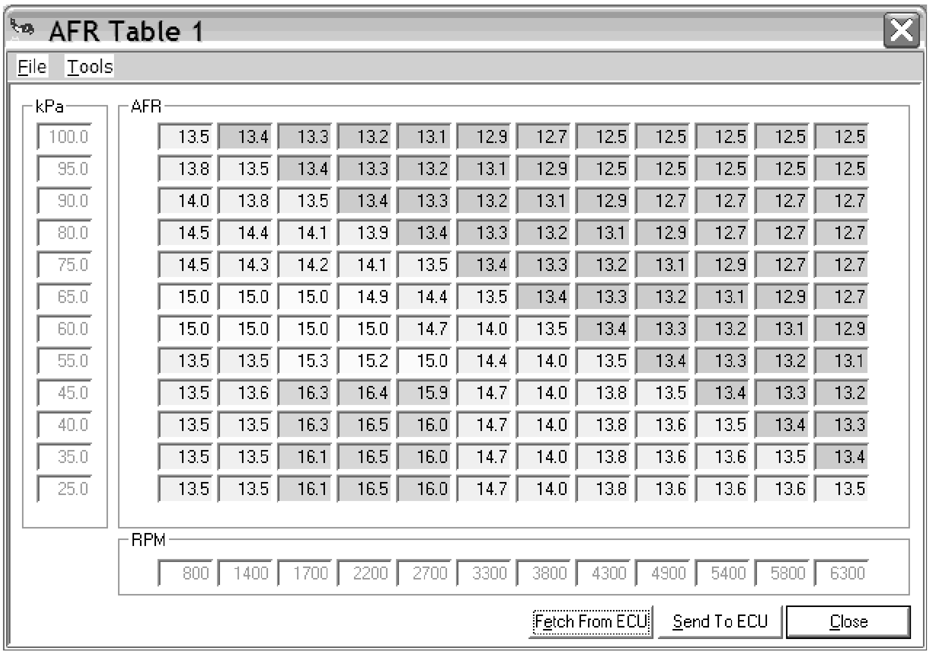


Figure 3.9: Wiring diagram

Source: Bruce and Al (2008)

3.6 MEGATUNE SOFTWARE

MegaTune is MegaSquirt configuration editor software for Windows 95 (and later). It is written by Eric Fahlgren, and it used for the MegaSquirt and MegaSquirt-II EFI controller. Figure 3.10 shows the air fuel ratio (AFR) table in the software that will be used to input the best AFR, which is obtained from the experiment conducted, for best engine power at a fixed RPM. The software then burns the input data into the Megasquirt-II ECU memory database.



kPa	800	1400	1700	2200	2700	3300	3800	4300	4900	5400	5800	6300
100.0	13.5	13.4	13.3	13.2	13.1	12.9	12.7	12.5	12.5	12.5	12.5	12.5
95.0	13.8	13.5	13.4	13.3	13.2	13.1	12.9	12.5	12.5	12.5	12.5	12.5
90.0	14.0	13.8	13.5	13.4	13.3	13.2	13.1	12.9	12.7	12.7	12.7	12.7
80.0	14.5	14.4	14.1	13.9	13.4	13.3	13.2	13.1	12.9	12.7	12.7	12.7
75.0	14.5	14.3	14.2	14.1	13.5	13.4	13.3	13.2	13.1	12.9	12.7	12.7
65.0	15.0	15.0	15.0	14.9	14.4	13.5	13.4	13.3	13.2	13.1	12.9	12.7
60.0	15.0	15.0	15.0	15.0	14.7	14.0	13.5	13.4	13.3	13.2	13.1	12.9
55.0	13.5	13.5	15.3	15.2	15.0	14.4	14.0	13.5	13.4	13.3	13.2	13.1
45.0	13.5	13.6	16.3	16.4	15.9	14.7	14.0	13.8	13.5	13.4	13.3	13.2
40.0	13.5	13.5	16.3	16.5	16.0	14.7	14.0	13.8	13.6	13.5	13.4	13.3
35.0	13.5	13.5	16.1	16.5	16.0	14.7	14.0	13.8	13.6	13.6	13.5	13.4
25.0	13.5	13.5	16.1	16.5	16.0	14.7	14.0	13.8	13.6	13.6	13.6	13.5

Figure 3.10: Megatune software AFR table

Source: Eric (2009)

3.7 SIMULATION

A simulation of experiment to determine effect of variable AFR to engine power at a fixed RPM is conducted using GT-Power software. From 800 RPM to 3000 RPM, with increment of 200 RPM, variable air-fuel ratio is tested at the variable manifold pressure. the stoichiometric air-fuel ratio of CNG is 16.79, hence by extending from 16.79 with ± 4 and increment of 0.5 the range of 12.5 to 21 is to be tested. As for the manifold pressure, the range is from 0.25 Bar to 1.00 Bar with increment of 0.05 Bar. All of the components data from the CNG engine is input to the GT-Power engine model window libraries of the computational modeling menu. The parameters of the engine are shown in Table 3.2. The model is start from intake environment and finish in exhaust environment. There are three sub-system in the computational model of the sequential injection dedicated compressed natural gas (CNG) engine spark ignition. The first is design and development of intake system, the second is design and development of engine cylinder and engine crank train and the third is design and development of exhaust system of sequential injection compressed natural gas (CNG) engine spark ignition. Then input all of the CNG engine components for all of the three sub-system CNG engine and connect all of the engine components to develop the compressed natural gas (CNG) spark ignition engine.

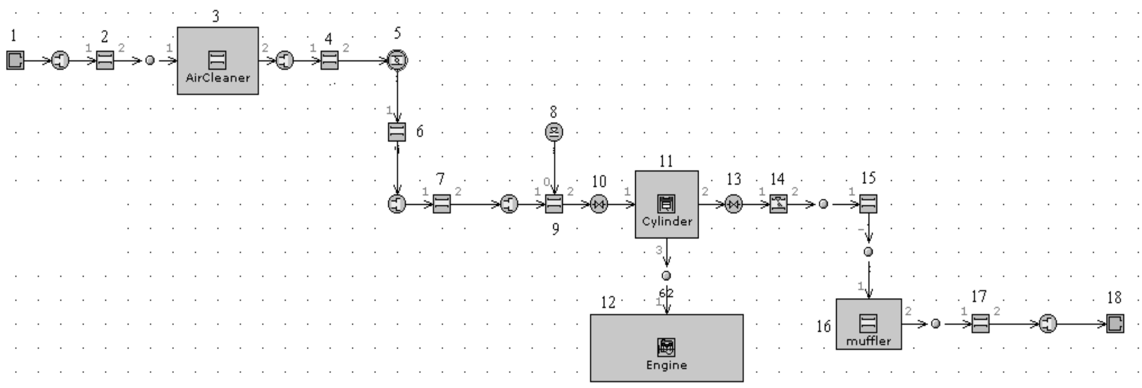


Figure 3.11: CNG engine modelling using GT-Power

Figure 3.11 shows the CNG engine is modeling using GT-Power where, 1 is intake environment, 2 is intake pipe1, 3 is air cleaner, 4 is intake pipe2, 5 is throttle, 6 is intake pipe3, 7 is intake runner, 8 is fuel injector, 9 is intake port, 10 is intake valve, 11 is engine cylinder, 12 is engine crank train, 13 is exhaust valve, 14 is exhaust port, 15 is exhaust runner, 16 is muffler, 17 is exhaust pipe and 18 is exhaust environment. After the computational modeling is completed, the model is run to obtain the results.

3.8 COMPARING AND DETERMINE THE BEST AFR

The results from the simulation can be obtained from GT-Post. Data from the GT-Post is transferred to Microsoft Excel to plot the graph. From the plotted graph of engine power versus RPM, the engine power of each AFR with respect to each fixed RPM at each manifold pressure condition is compared. The best AFR is determined from the comparison and will be input into the AFR table of the Megatune software to be burn into ECU later..

3.8 DOCUMENTATION

All the literature studies will be documented. The results obtained and the discussion on the results made will also be documented. A thesis report is written and will be submitted at later time.

3.9 SUMMARY

Methodology of the project in general, with a specific focus on the development of engine management system for diesel engine fueled by CNG is discussed. The work done is based on methodology flow chart. Progress such as modification done on the diesel engine is discussed in this chapter. Understanding prior and current research in this project provides method for the research contributions outlined in subsequent chapters.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

Chapter 4 will show graphs obtained from GT-Power simulation. Readings are taken from these graphs to determine the air-fuel-ratio (AFR) that produces highest brake horse power for a fixed RPM at different intake manifold pressure which ranges from 0.25 Bar to 1.00 Bar with increment of 0.05 Bar.

4.2 RESULTS

Following are graphs showing the brake horse power produced by variable air-fuel-ratio (AFR) at variable manifold pressure.

4.2.1 Results for manifold pressure of 0.25 Bar

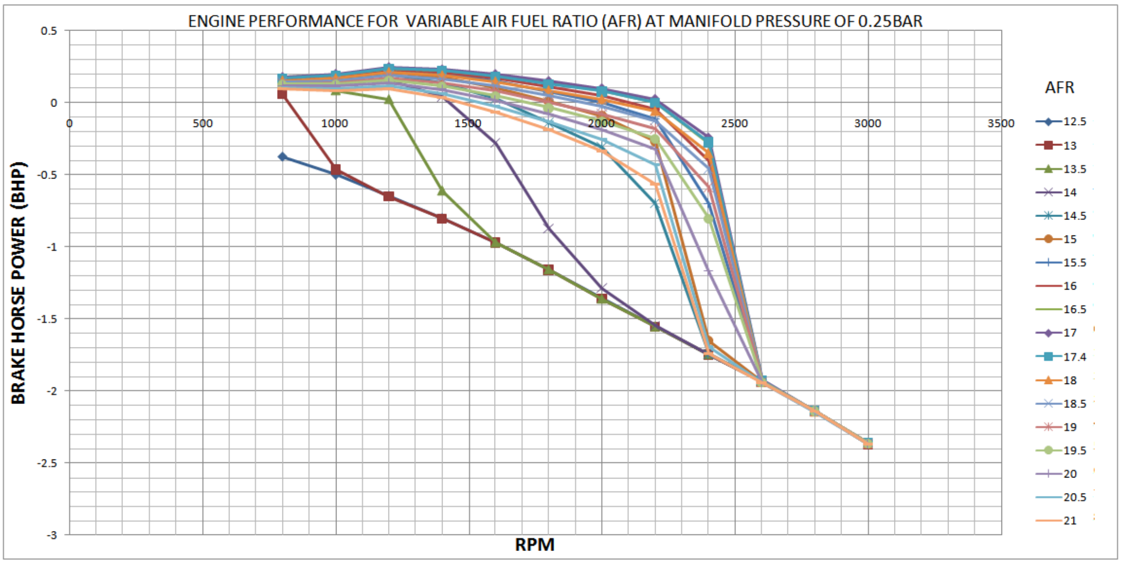


Figure 4.1: Engine performance for variable AFR at manifold pressure of 0.25 Bar

From Figure 4.1, it is observed that the maximum power produced occurred in the range of 1000-1500 RPM. Due to the low pressure in the intake manifold, mass of air also decreased, which is concluded by the ideal gas equation *PV=mRT*. So, even though enough fuel is supplied, there is not enough air to expand to produce the pressure needed to produce higher power. The power produced is not enough to overcome the friction losses; hence we can see that after the RPM range of 1000-1500 RPM the power drops down. For the negative value of horse power, it means that the engine has already shut down and instead of giving power, it is a load it self. These negative values are put in so that the graphs are consistent with each other for graphs of other manifold pressure. Next, effect of variable AFR at each RPM is looked into.

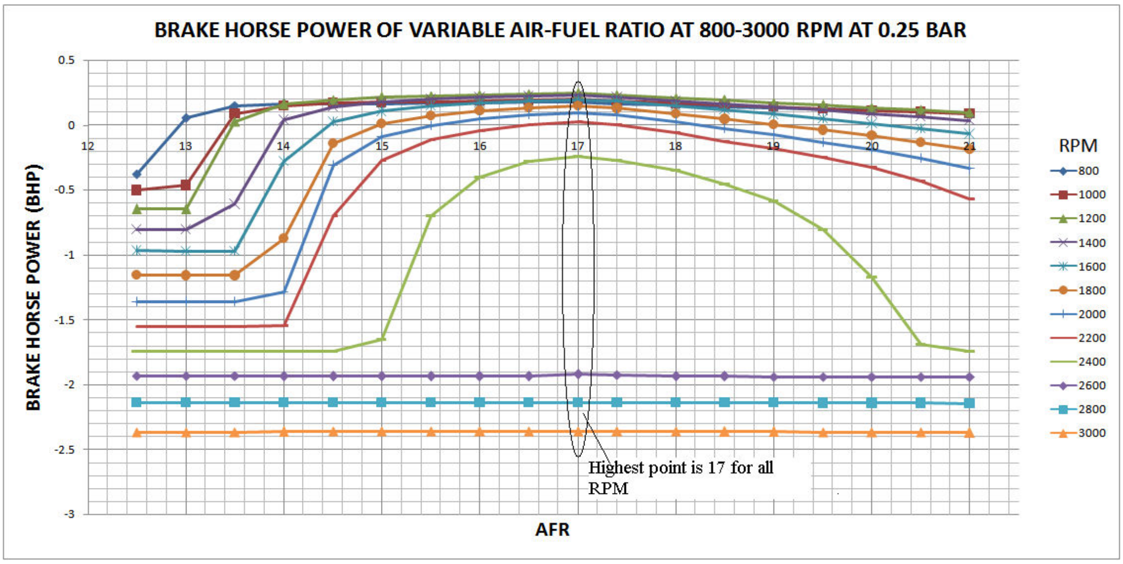
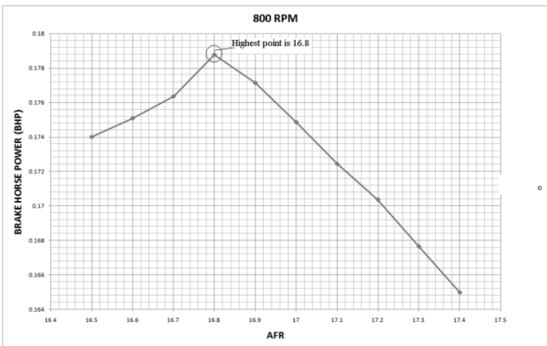
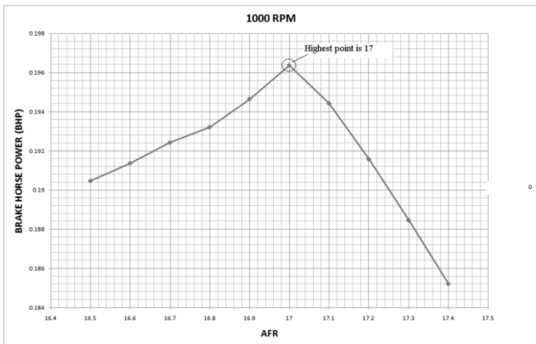


Figure 4.2: Overview of effect of variable AFR at each RPM at 0.25 Bar

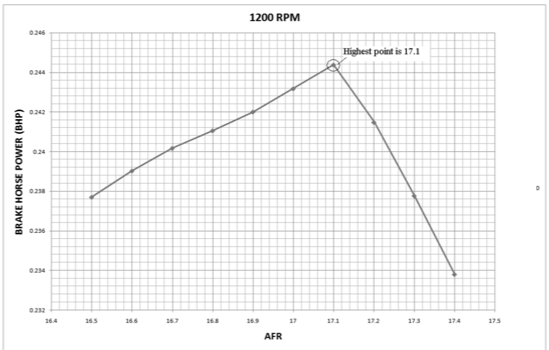
Figure 4.2 shows an overview of effect of variable AFR to brake horse power at each RPM. The tested AFR range is from 12.5 to 21 where the stoichiometric AFR for CNG is 16.79; hence by extending from 16.79 with ± 4 and increment of 0.5 the range of 12.5 to 21 is obtained. In the overview of figure above it is clear that highest AFR for all RPM is 17. From there a more refined view of each RPM is done in the range of 16.5 to 17.4 with increment of 0.1 to get a more accurate AFR.



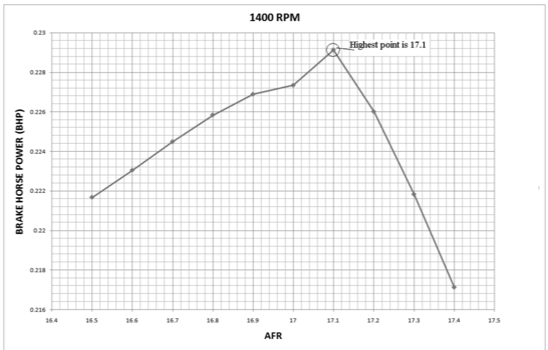
(a) 800 RPM



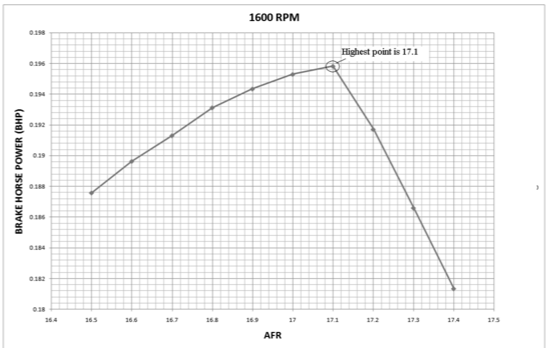
(b) 1000 RPM



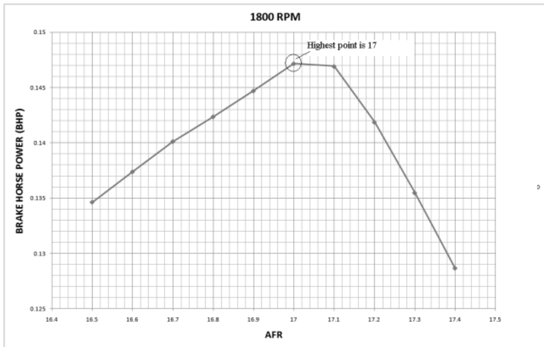
(c) 1200 RPM



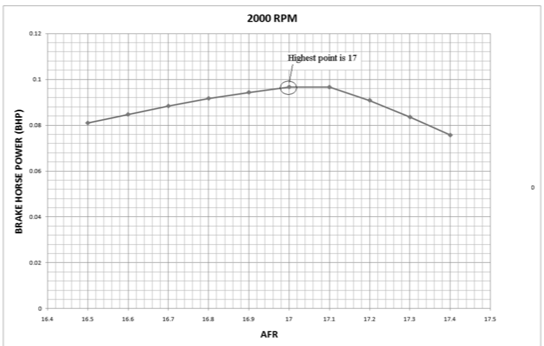
(d) 1400 RPM



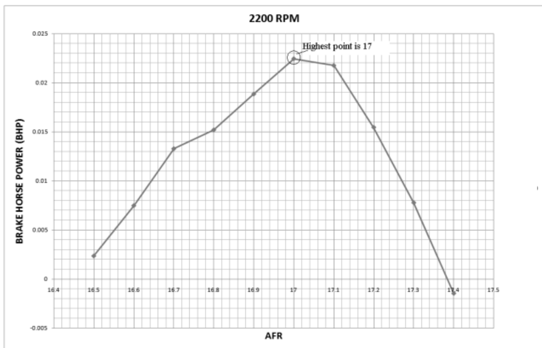
(e) 1600 RPM



(f) 1800 RPM



(g) 2000 RPM



(h) 2200 RPM

Figure 4.3: Refined view of effect of variable AFR at each RPM at 0.25 Bar

Figure 4.3 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (h) the refined view of the RPM range from 800-2200 RPM is observed. RPM range of 2400-3000 RPM is not showed because as can be seen in Figure 4.2 the power is in negative range, meaning the engine has already stop running so no point in considering the AFR of this RPM range. The table below shows the best AFR for each RPM read from the refined view graphs.

Table 4.1: Best AFR for each RPM at manifold pressure of 0.25 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	16.8	17	17.1	17.1	17.1	17
RPM	2000	2200				
AFR	17	17				

4.2.2 Results for manifold pressure of 0.30 Bar

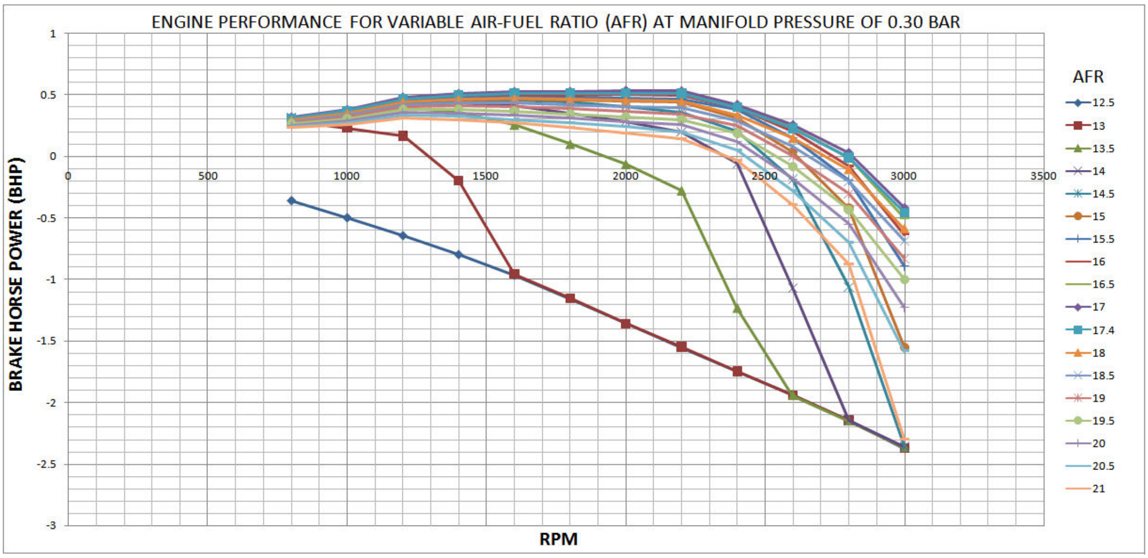


Figure 4.4: Engine performance for variable AFR at manifold pressure of 0.30 Bar

Figure 4.4 shows the brake horse power obtained with variable AFR at each RPM with manifold pressure of 0.30 Bar. Highest power occurred in the range of 1500-2000 RPM. Increasing the manifold pressure increases the mass of air hence much pressure can be produced during combustion; hence higher RPM can be reached to produce more power. After the peak power range, the power drop again due to friction build up and losses. Again there are negative values which indicate the engine shutting down for the rich AFR. Too rich causes the combustion chamber to be wet due to the excessive unburned hydrocarbons. This causes the flame to be terminated early hence no power is produced. Next the overview of effect of variable AFR is looked into.

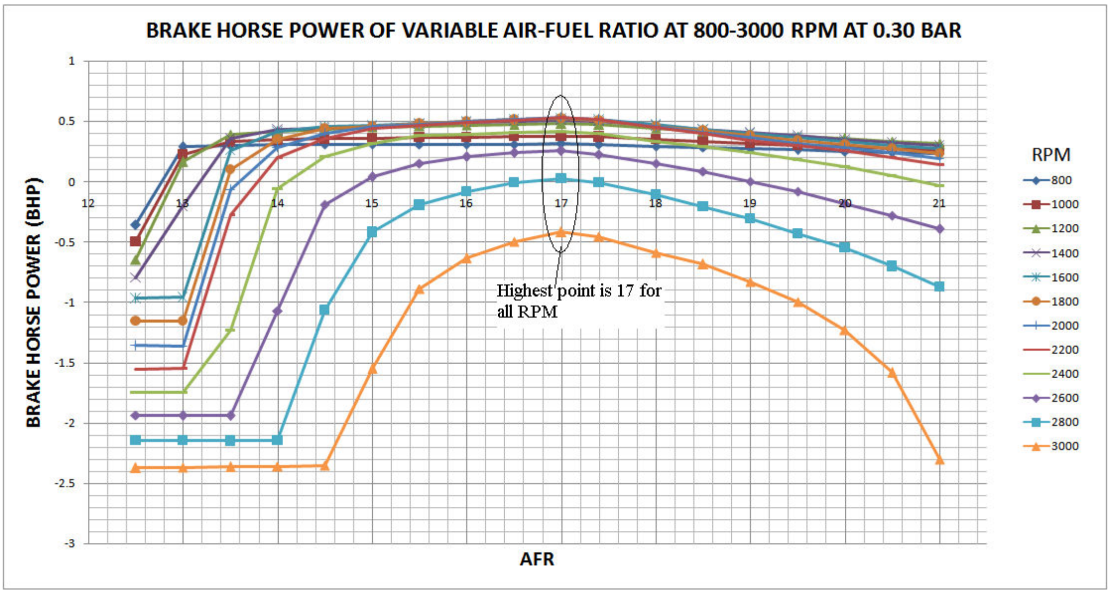
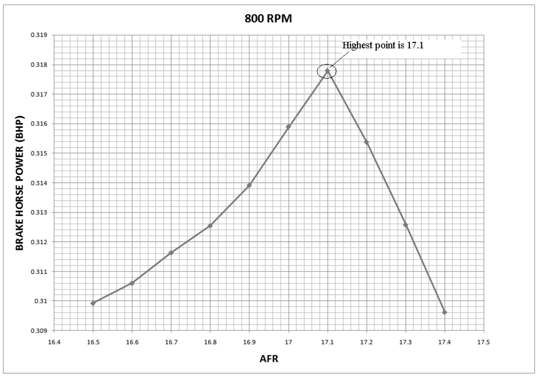
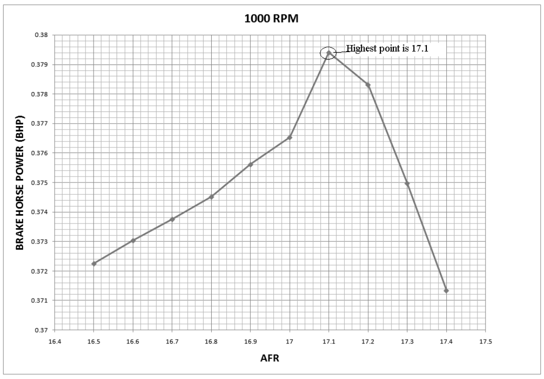


Figure 4.5: Overview of effect of variable AFR at each RPM at 0.30 Bar

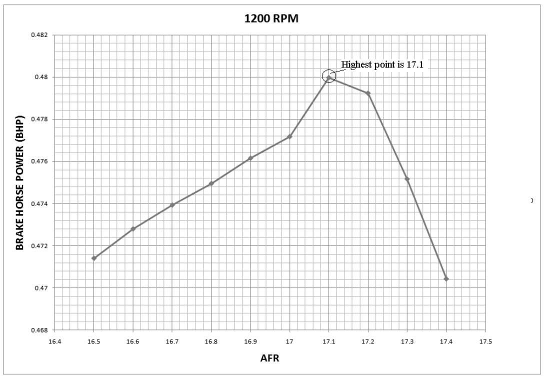
Figure 4.5 shows the overview of the brake horse power produced by variable AFR at manifold pressure of 0.30 Bar. As can be seen the highest point is 17 for all RPM. Hence a refined view in the range of 16.5 to 17.4 is done to find the best AFR for each RPM. It can be observed that for 3000 RPM the brake horse power is in the negative range, meaning the engine has shut down.



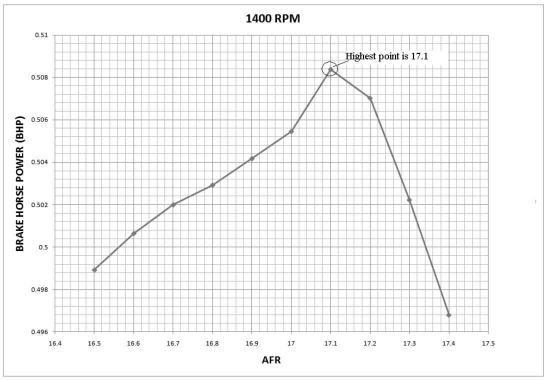
(a) 800 RPM



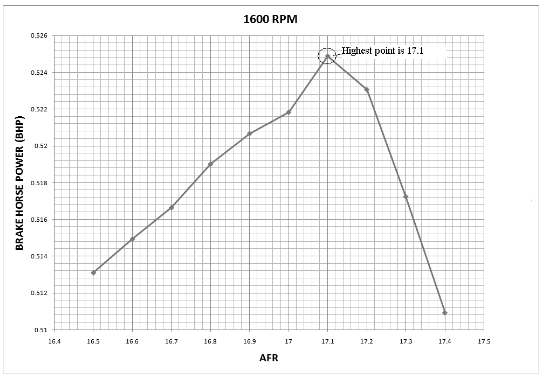
(b) 1000 RPM



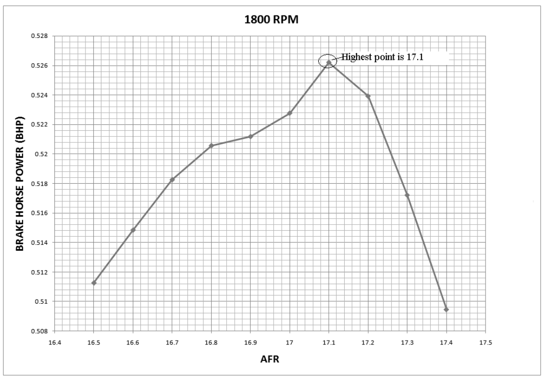
(c) 1200 RPM



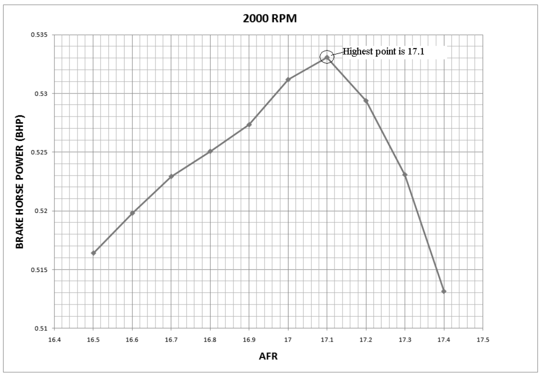
(d) 1400 RPM



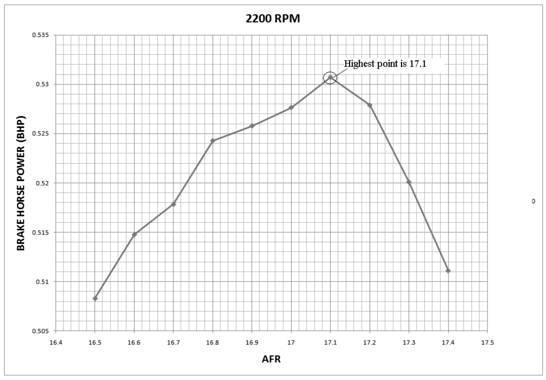
(e) 1600 RPM



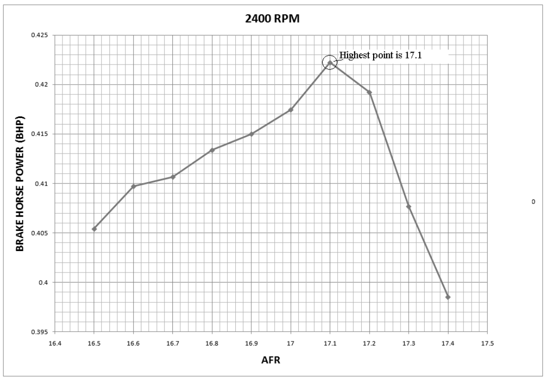
(f) 1800 RPM



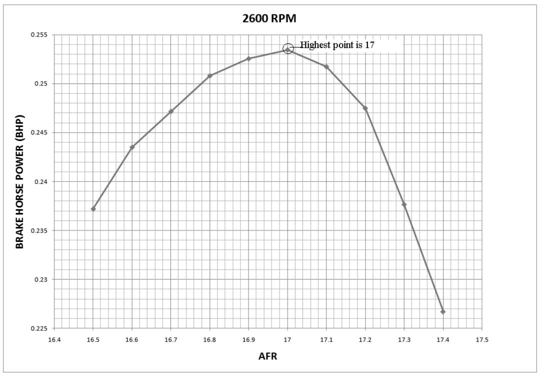
(g) 2000 RPM



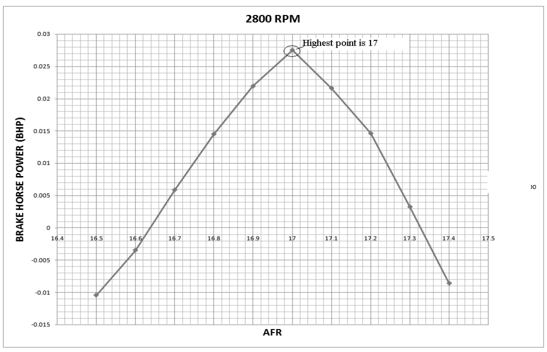
(h) 2200 RPM



(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM

Figure 4.6: Refined view of effect of variable AFR at each RPM at 0.30 Bar

Figure 4.6 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (k) the refined view of the RPM range from 800-2800 RPM is observed. 3000 RPM is not showed because as can be seen in Figure 4.5 the power is in negative range, meaning the engine has already stop running so no point in considering the AFR of this RPM. The table below shows the best AFR for each RPM read from the refined view graphs.

Table 4.2: Best AFR for each RPM at manifold pressure of 0.30 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	17.1	17.1	17.1	17.1	17.1	17.1
RPM	2000	2200	2400	2600	2800	
AFR	17.1	17.1	17.1	17	17	

4.2.3 Results for manifold pressure of 0.35 Bar

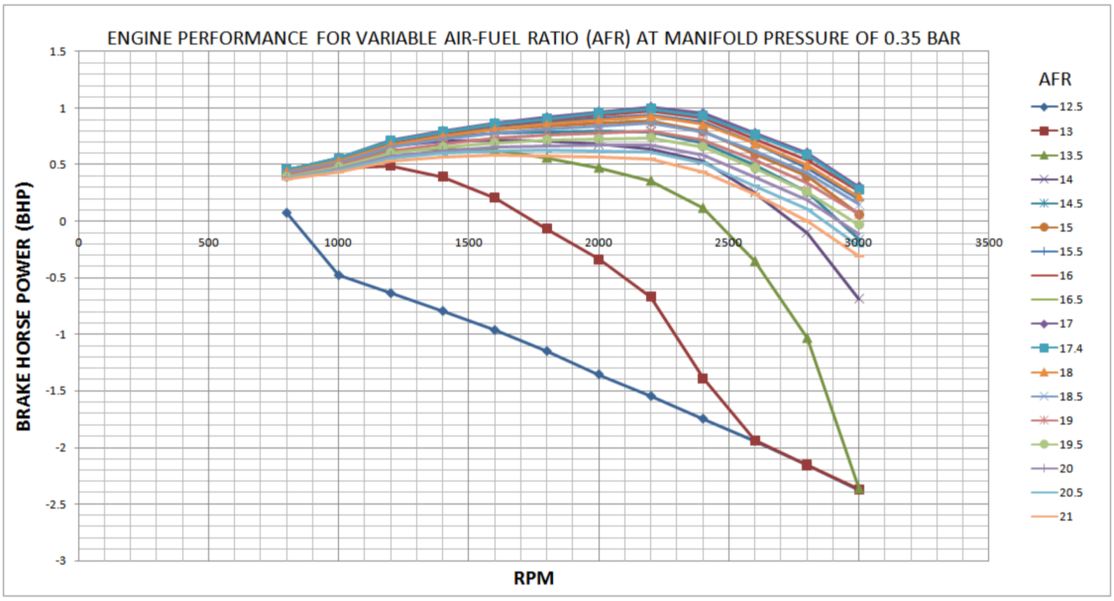


Figure 4.7: Engine performance for variable AFR at manifold pressure of 0.35 Bar

Figure 4.7 shows the performance curve at manifold pressure of 0.35 Bar. As can be observed increasing the pressure increases the peak value of horse power produced. The peak value occurs in the 2000-2500 RPM range for most of the AFR except for the rich ones. Too rich causes the combustion chamber to be wet and the unburned fuel will cool down the combustion chamber causing optimum temperature cannot be attained hence causing the engine to shut down eventually as can be seen the negative value of horse power in the graph. After the peak value the horse power drop again due to losses and friction build up. Next the overview of effect of variable AFR is looked into.

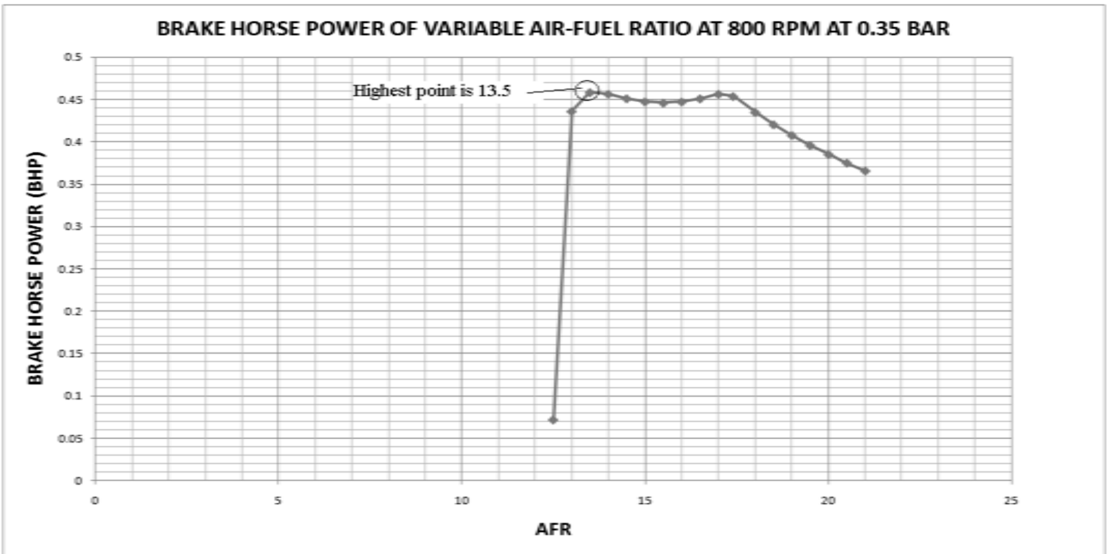


Figure 4.8: Overview of effect of variable AFR at 800 RPM at 0.35 Bar

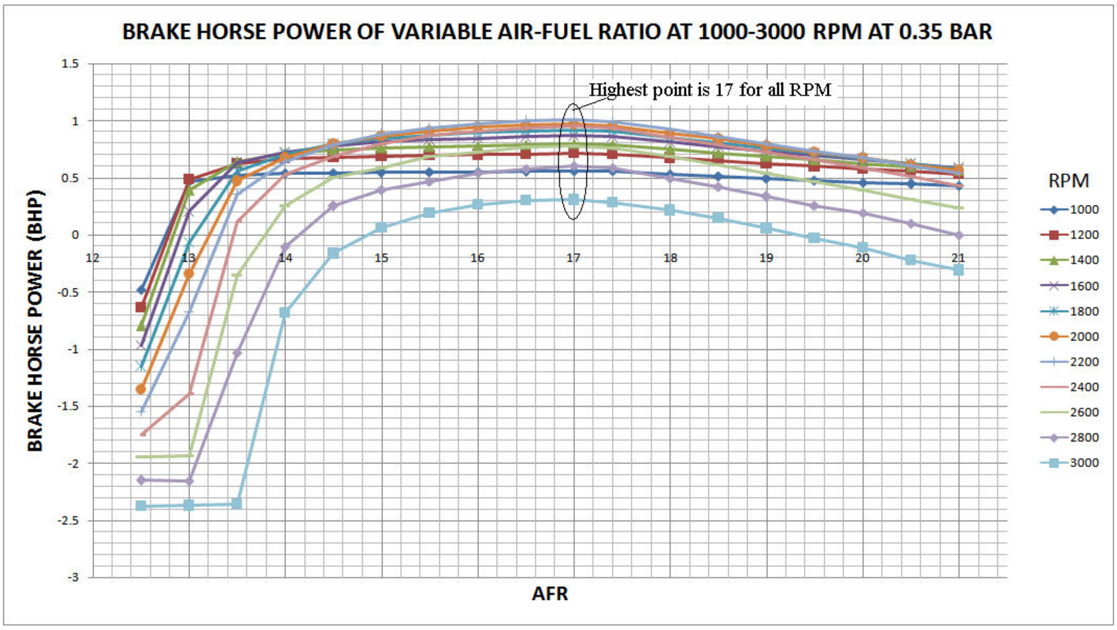
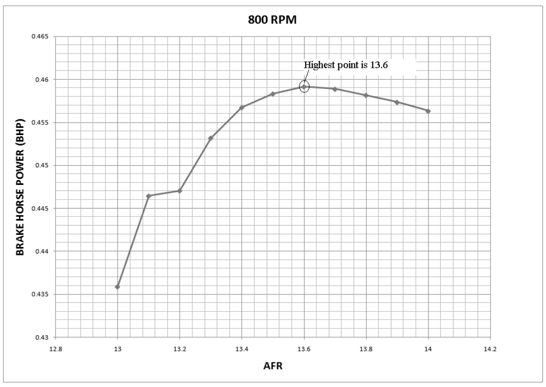
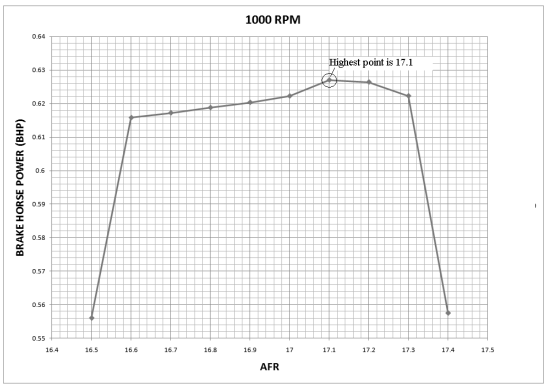


Figure 4.9: Overview of effect of variable AFR at 1000-3000 RPM at 0.35 Bar

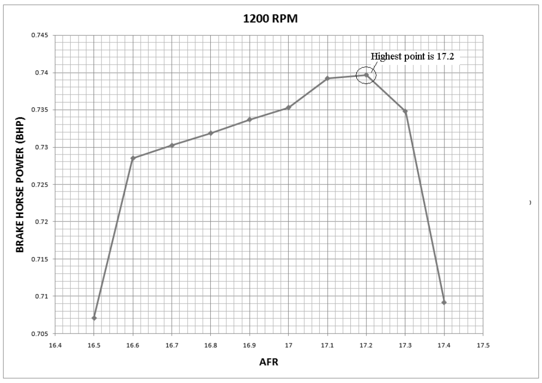
Figure 4.8 and Figure 4.9 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.8 the highest point for 800 RPM is 13.5 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13.5 for 800 RPM, the refined view is ranged from 13 to 14 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



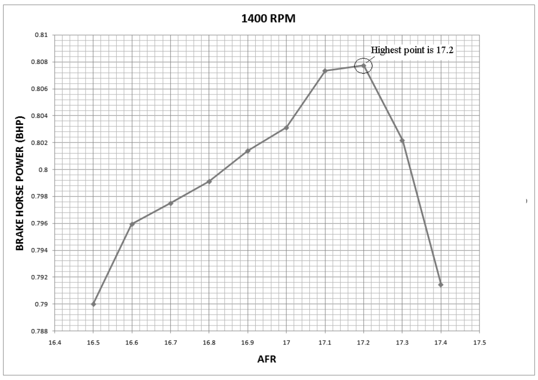
(a) 800 RPM



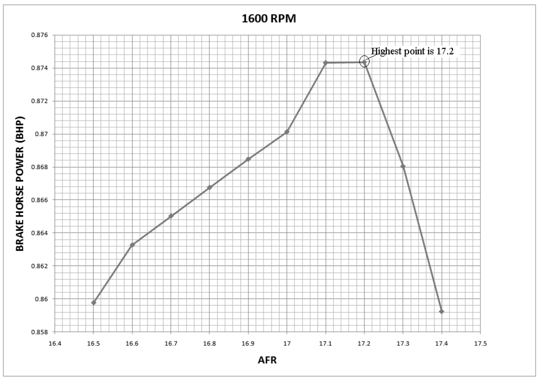
(b) 1000 RPM



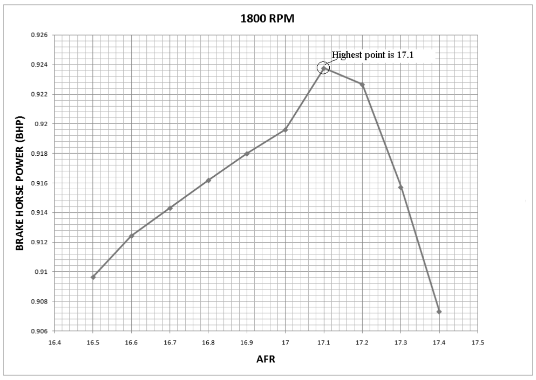
(c) 1200 RPM



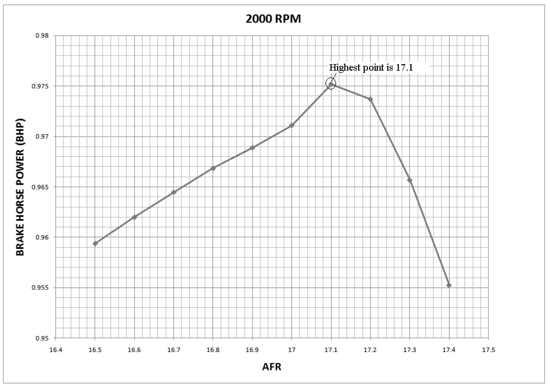
(d) 1400 RPM



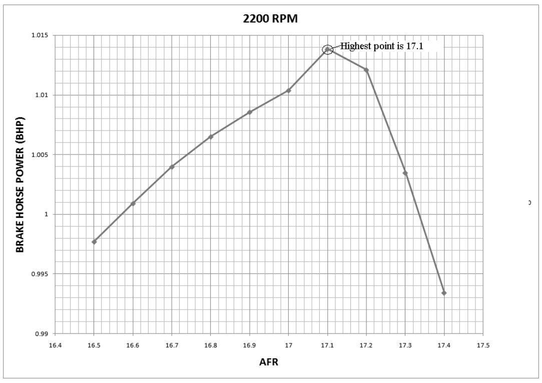
(e) 1600 RPM



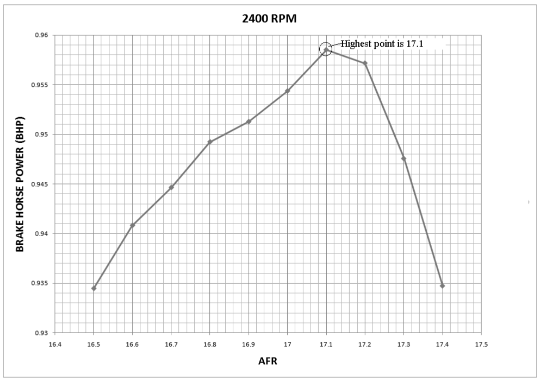
(f) 1800 RPM



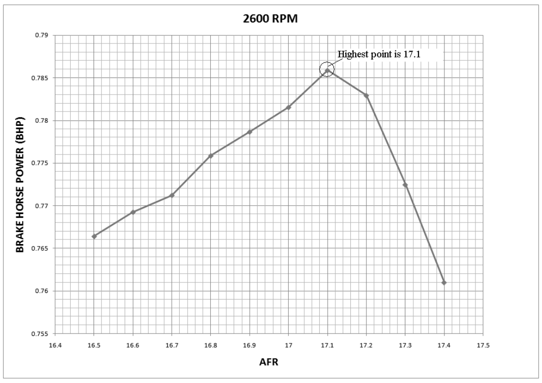
(g) 2000 RPM



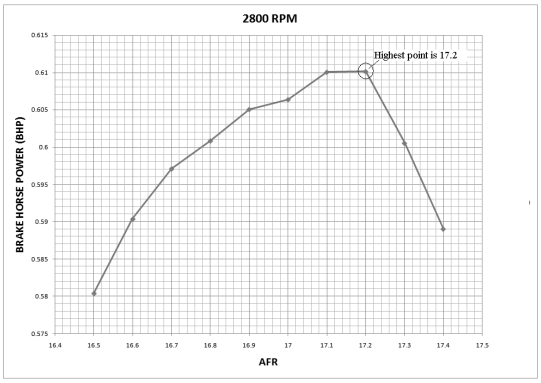
(h) 2200 RPM



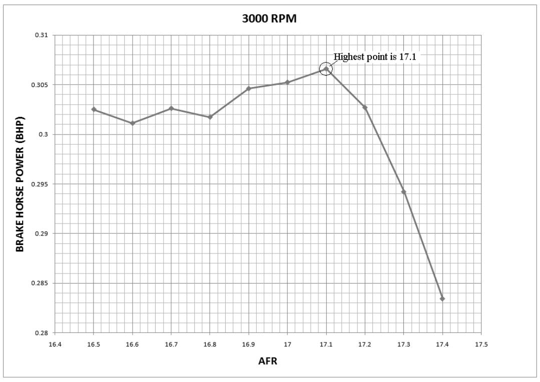
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.10: Refined view of effect of variable AFR at each RPM at 0.35 Bar

Figure 4.10 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.3 shows the best AFR for each RPM read from the refined view graphs.

Table 4.3: Best AFR for each RPM at manifold pressure of 0.35 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.6	17.1	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	17.2	17.1

4.2.4 Results for manifold pressure of 0.40 Bar

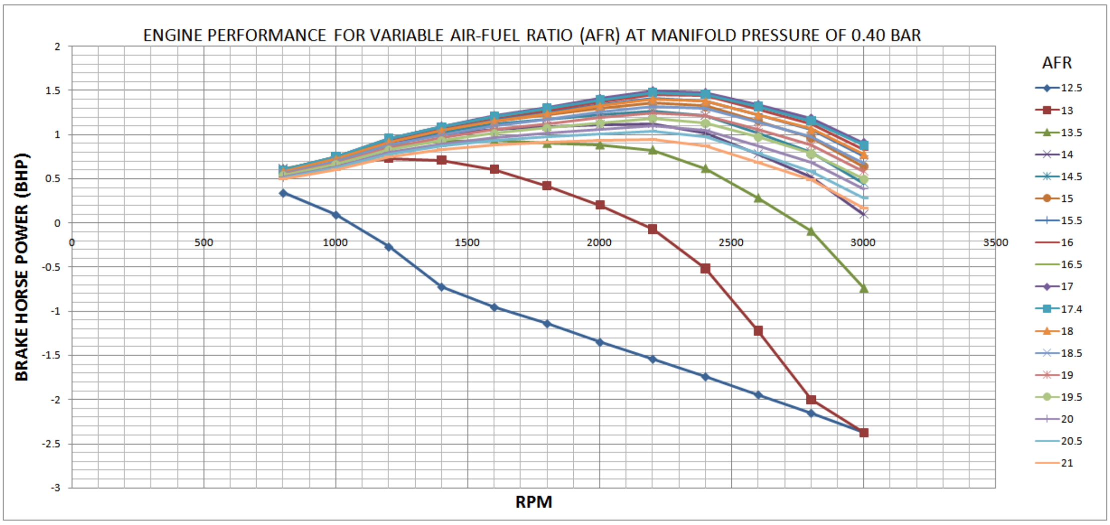


Figure 4.11: Engine performance for variable AFR at manifold pressure of 0.40 Bar

Figure 4.11 shows the engine performance at manifold pressure of 0.40 Bar. Again the peak occurs in the range of 2000-2500 RPM which is same as the performance at 0.35 Bar. The only difference is the increase of brake horse power. For 0.40 Bar the highest horse power attained is 1.5 BHP while for 0.35 Bar the highest point attained is 1 BHP. Hence increasing the pressure increases the mass of air intake in to cylinder. More air expands to produce more pressure to increase the horse power.

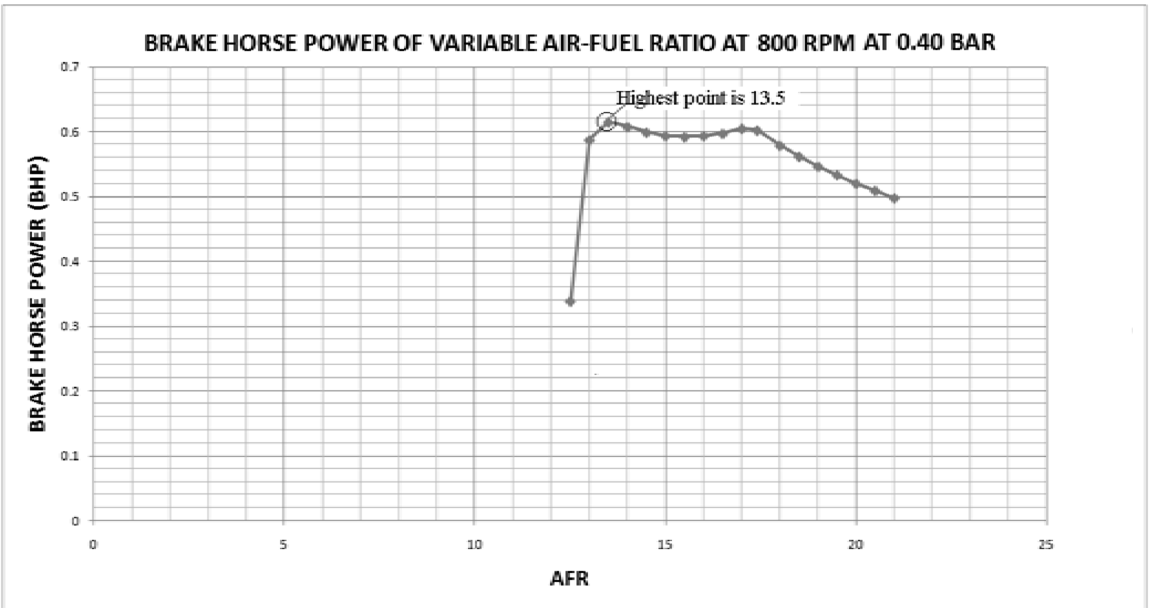


Figure 4.12: Overview of effect of variable AFR at 800 RPM at 0.40 Bar

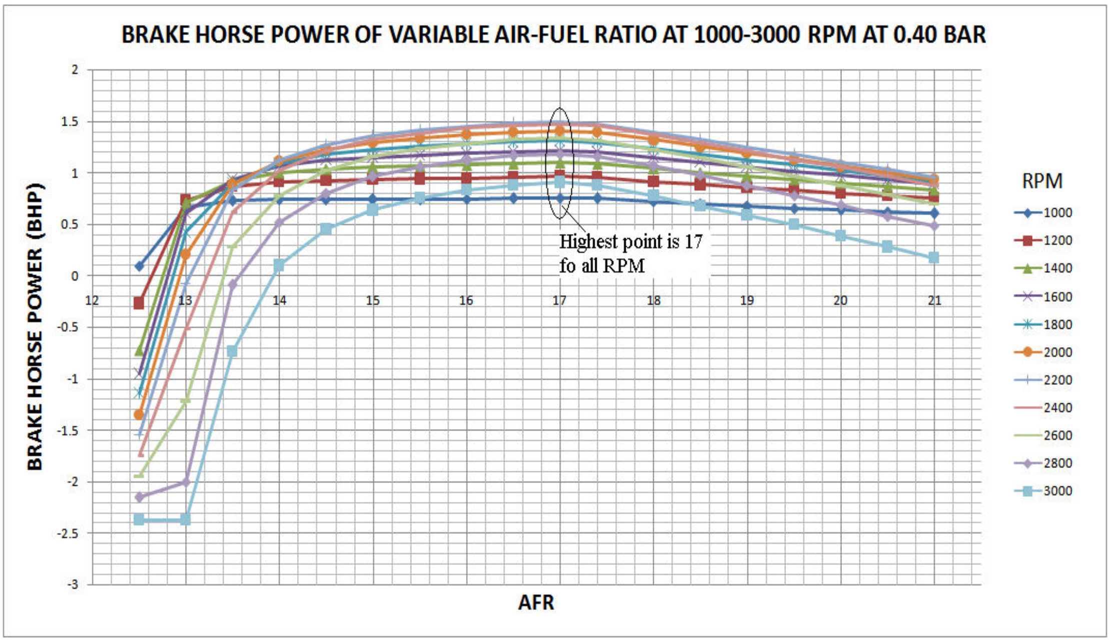
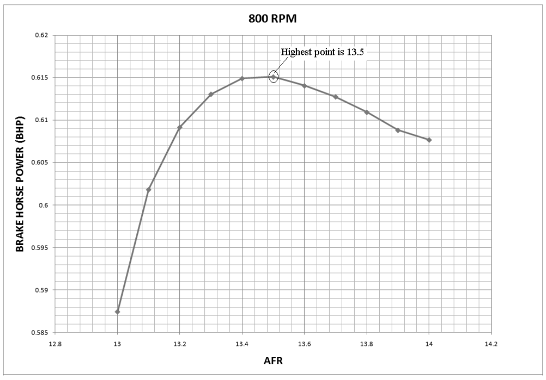
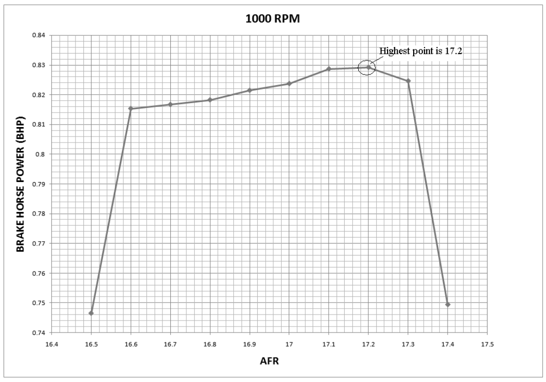


Figure 4.13: Overview of effect of variable AFR at 1000-3000 RPM at 0.40 Bar

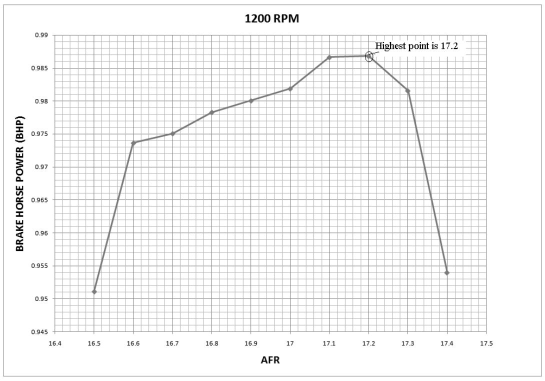
Figure 4.12 and Figure 4.13 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.12 the highest point for 800 RPM is 13.5 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13.5 for 800 RPM, the refined view is ranged from 13 to 14 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



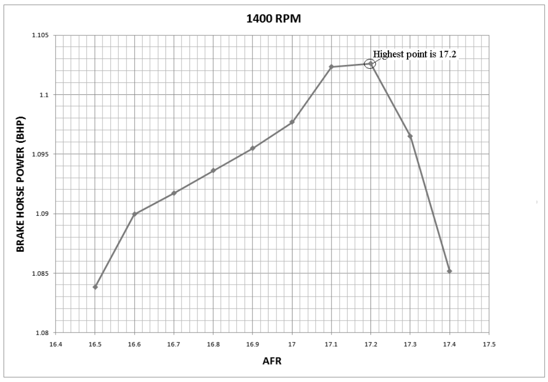
(a) 800 RPM



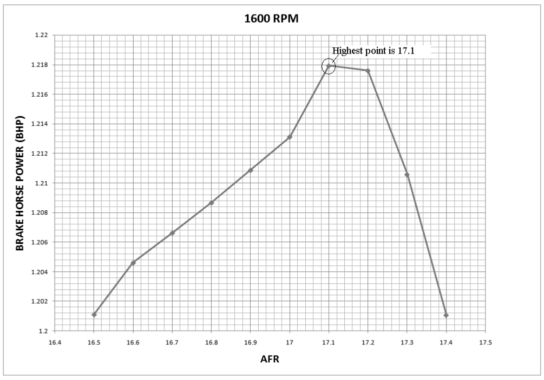
(b) 1000 RPM



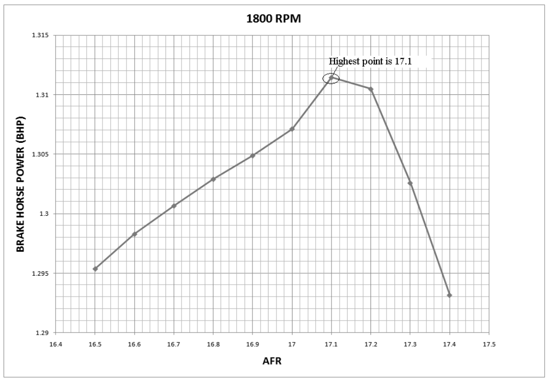
(c) 1200 RPM



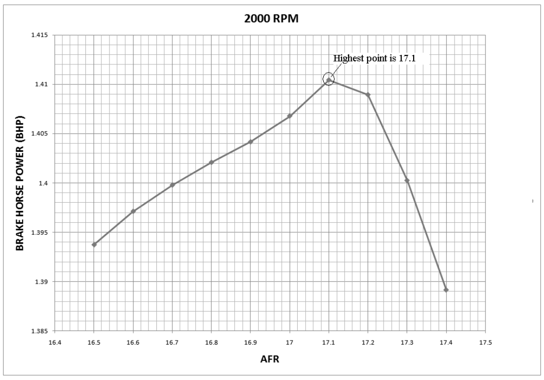
(d) 1400 RPM



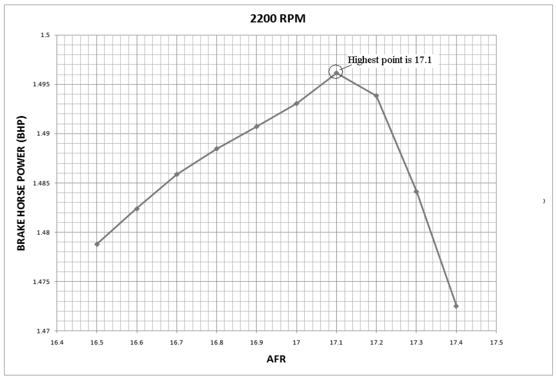
(e) 1600 RPM



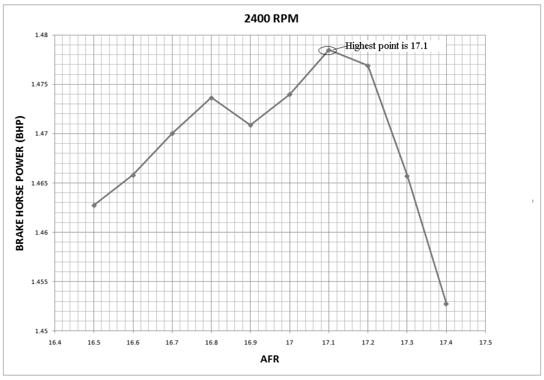
(f) 1800 RPM



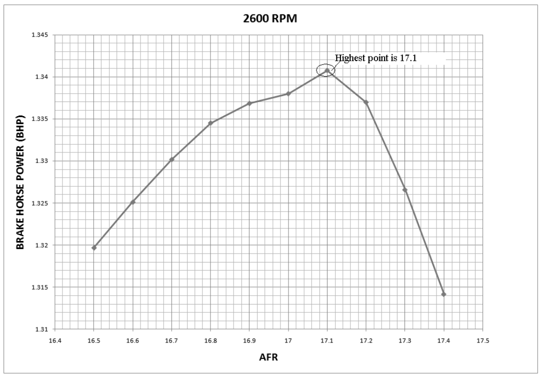
(g) 2000 RPM



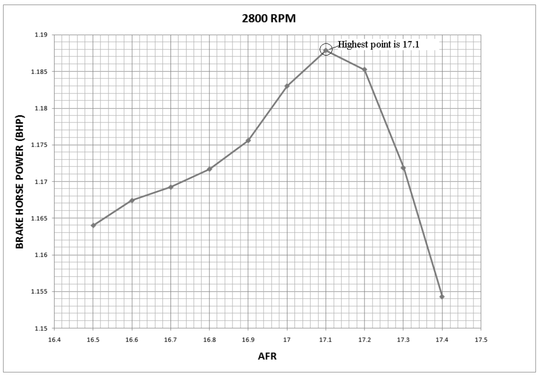
(h) 2200 RPM



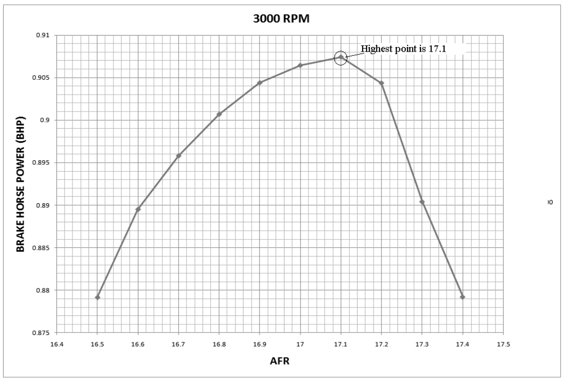
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.14: Refined view of effect of variable AFR at each RPM at 0.40 Bar

Figure 4.14 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.4 shows the best AFR for each RPM read from the refined view graphs.

Table 4.4: Best AFR for each RPM at manifold pressure of 0.40 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.5	17.2	17.2	17.2	17.1	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	17.1	17.1

4.2.5 Results for manifold pressure of 0.45 Bar

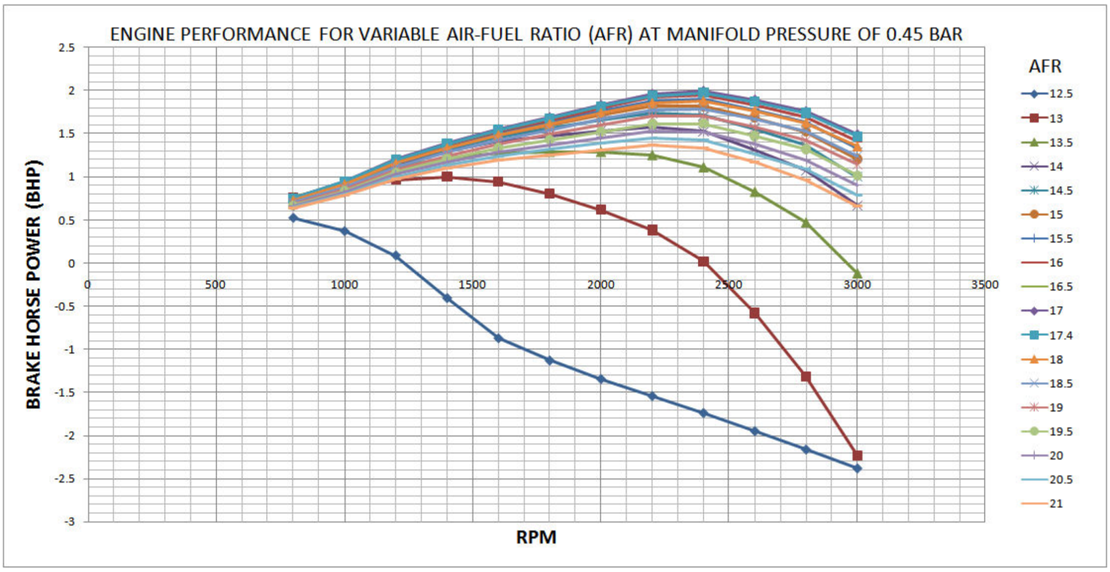


Figure 4.15: Engine performance for variable AFR at manifold pressure of 0.45 Bar

Figure 4.15 shows the engine performance at manifold pressure of 0.45 Bar. The peak horse power for most AFR occurs at 2400 RPM. As can be seen, the maximum horse power attained is 2 BHP. As pressure increase more air mass can be inducted into the cylinder hence more air expands during combustion creating higher pressure to produce more power. As for the rich AFR the trend is the same as at lower manifold pressure.

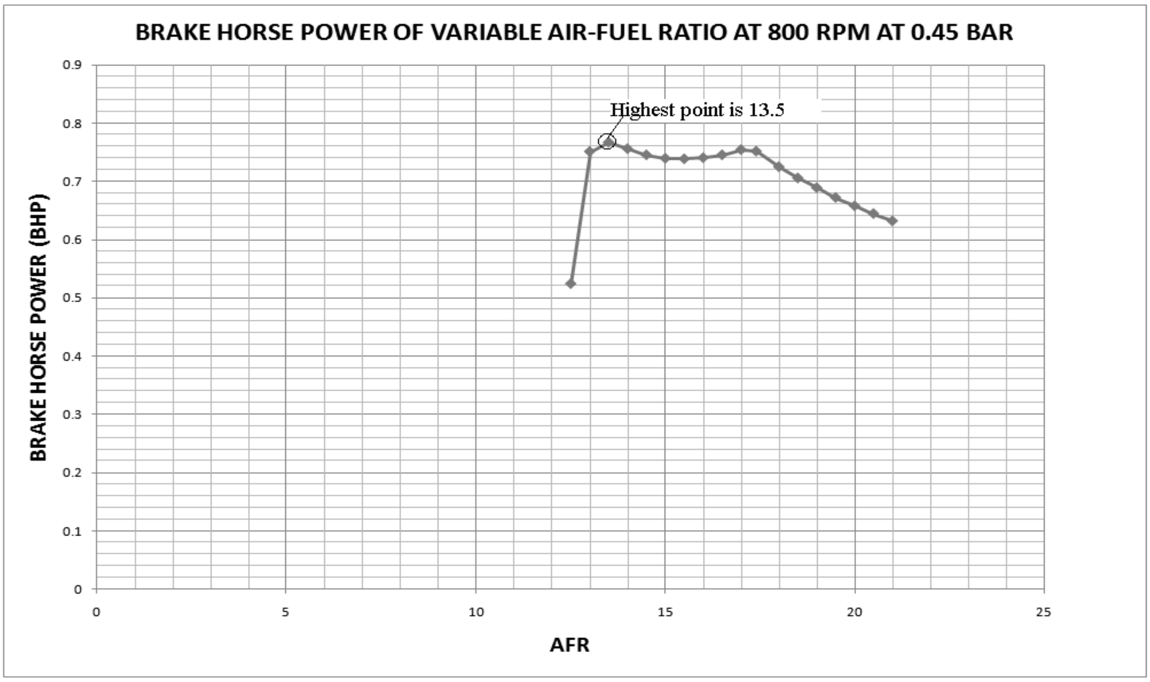


Figure 4.16: Overview of effect of variable AFR at 800 RPM at 0.45 Bar

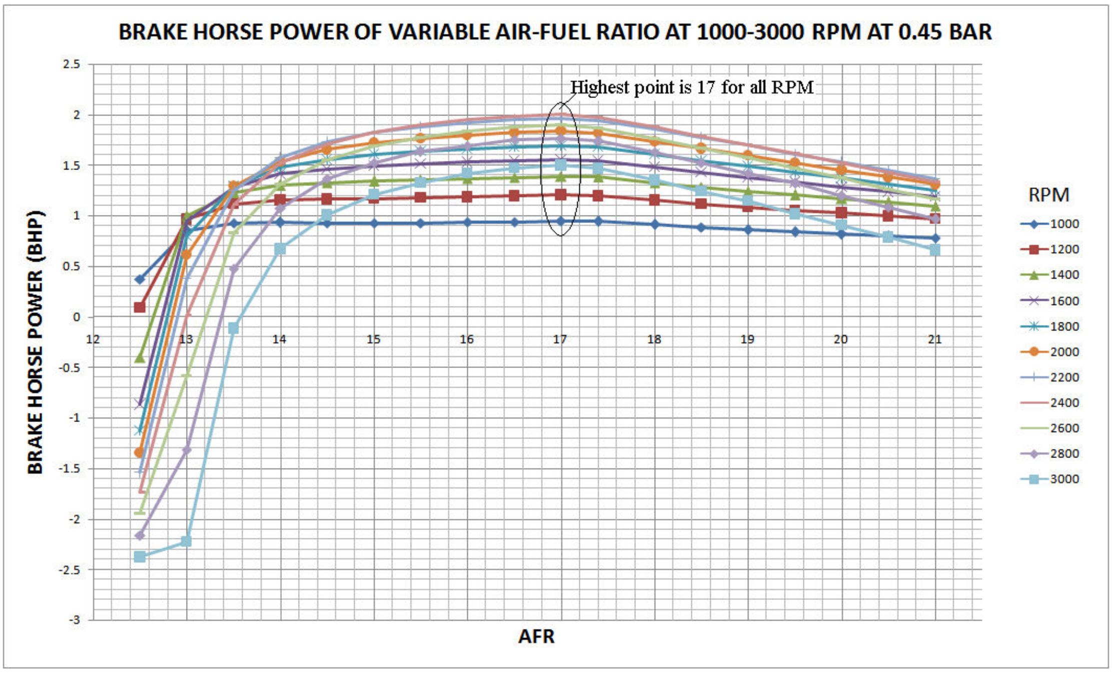
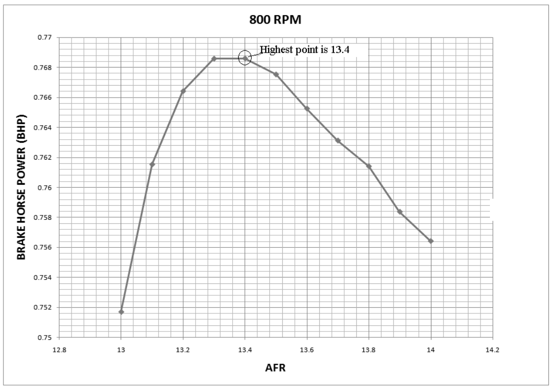
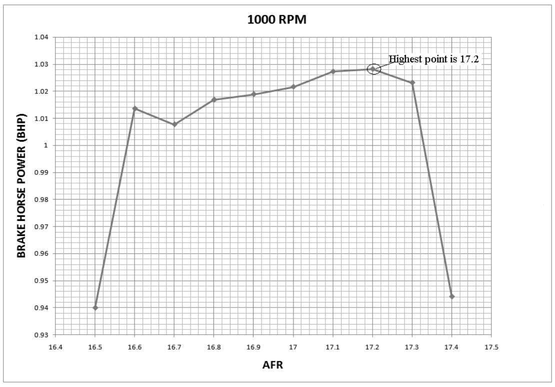


Figure 4.17: Overview of effect of variable AFR at 1000-3000 RPM at 0.45 Bar

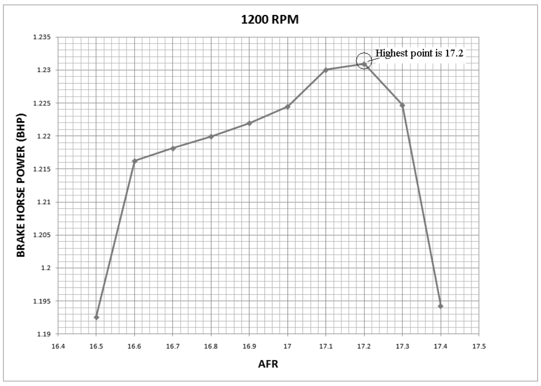
Figure 4.16 and Figure 4.17 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.16 the highest point for 800 RPM is 13.5 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13.5 for 800 RPM, the refined view is ranged from 13 to 14 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



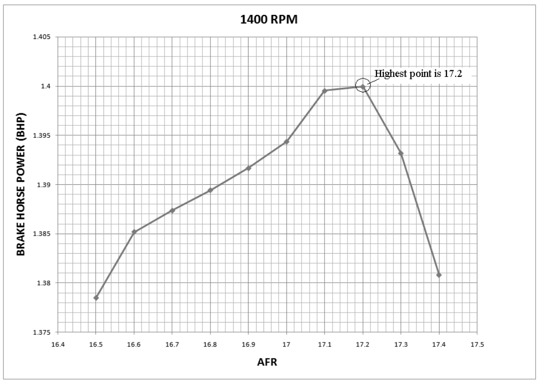
(a) 800 RPM



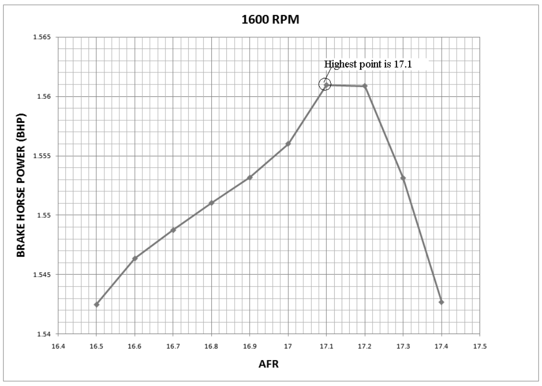
(b) 1000 RPM



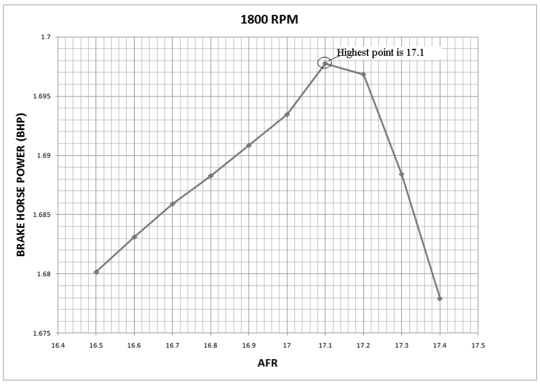
(c) 1200 RPM



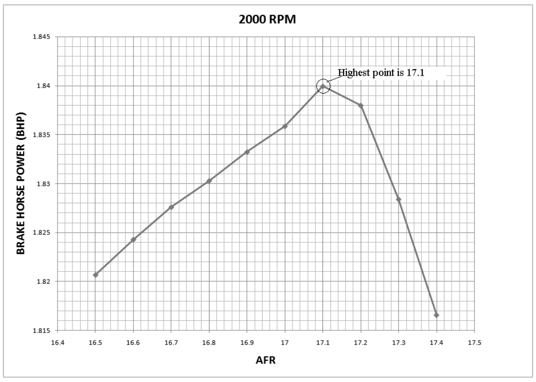
(d) 1400 RPM



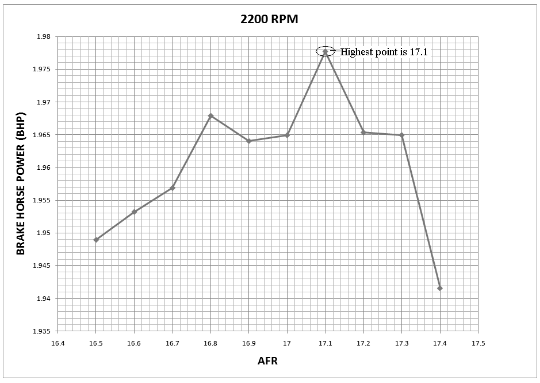
(e) 1600 RPM



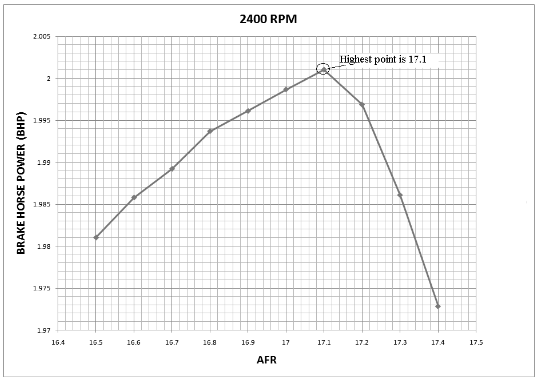
(f) 1800 RPM



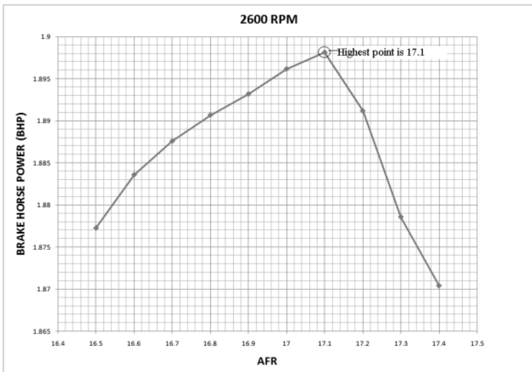
(g) 2000 RPM



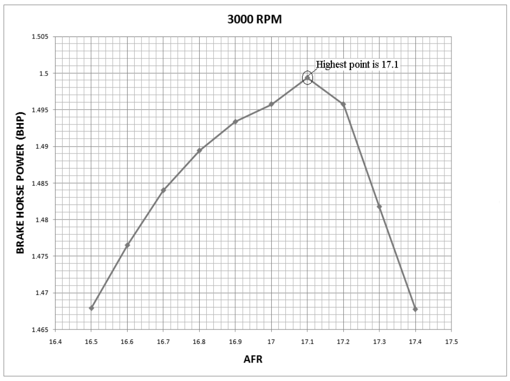
(h) 2200 RPM



(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM

(l) 3000 RPM

Figure 4.18: Refined view of effect of variable AFR at each RPM at 0.45 Bar

Figure 4.18 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.5 shows the best AFR for each RPM read from the refined view graphs.

Table 4.5: Best AFR for each RPM at manifold pressure of 0.45 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.4	17.2	17.2	17.2	17.1	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	17.1	17.1

4.2.6 Results for manifold pressure of 0.50 Bar

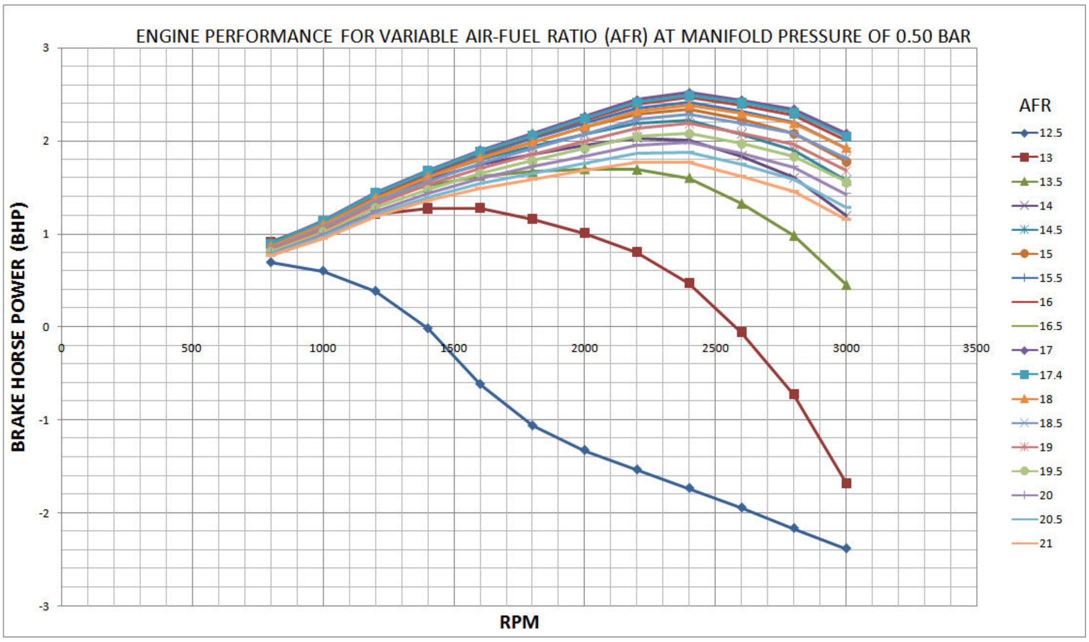


Figure 4.19: Engine performance for variable AFR at manifold pressure of 0.50 Bar

Figure 4.19 shows the engine performance at manifold pressure of 0.50 Bar. Again the peak power occurs at 2400 RPM for most AFR. The trend is the same as performance at 0.45 Bar. The only difference is the maximum horse power attained is 2.5 BHP, which has increased again with the increased manifold pressure.

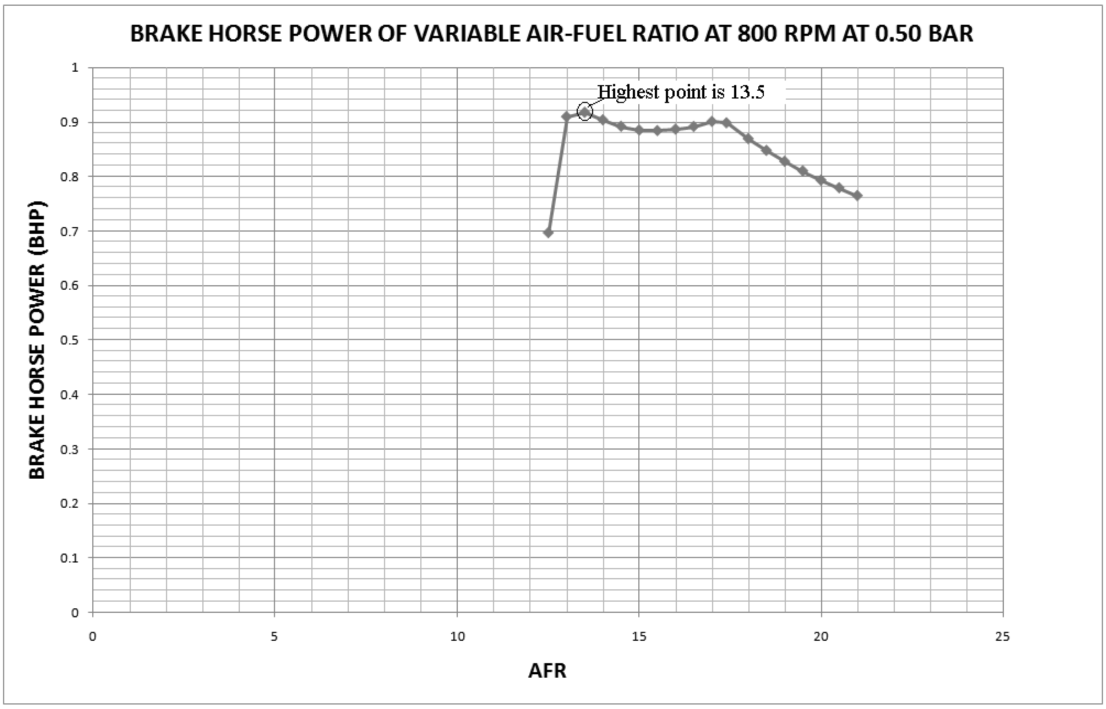


Figure 4.20: Overview of effect of variable AFR at 800 RPM at 0.50 Bar

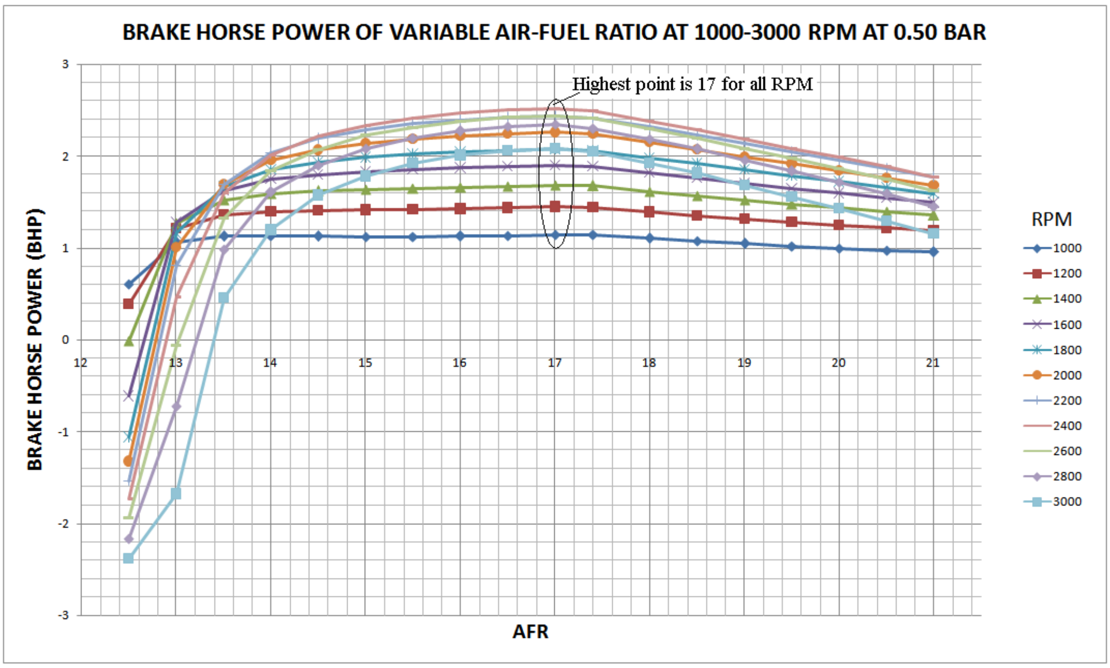
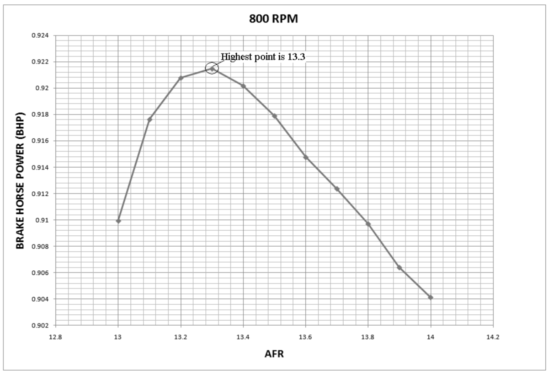
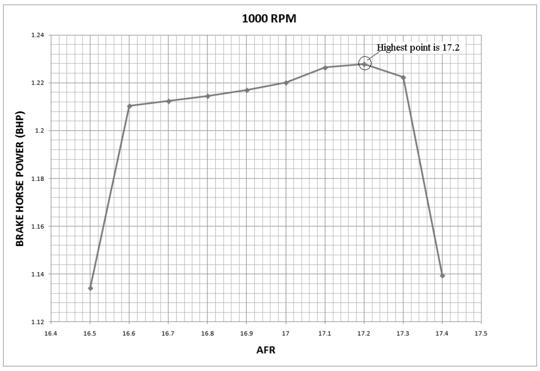


Figure 4.21: Overview of effect of variable AFR at 1000-3000 RPM at 0.50 Bar

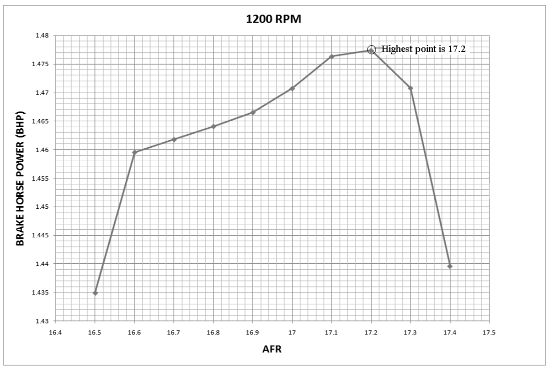
Figure 4.20 and Figure 4.21 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.20 the highest point for 800 RPM is 13.5 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13.5 for 800 RPM, the refined view is ranged from 13 to 14 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



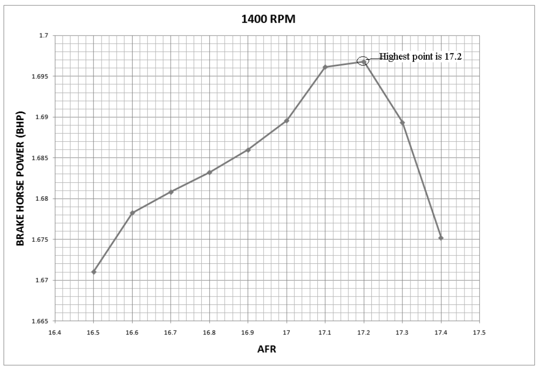
(a) 800 RPM



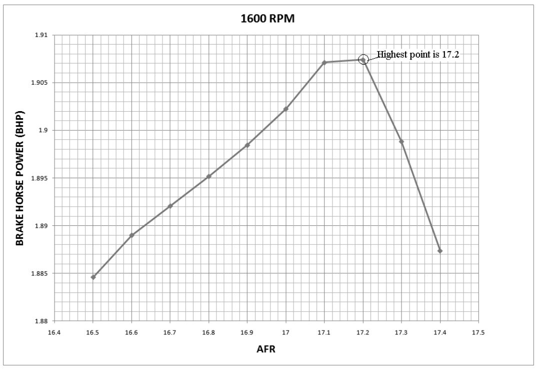
(b) 1000 RPM



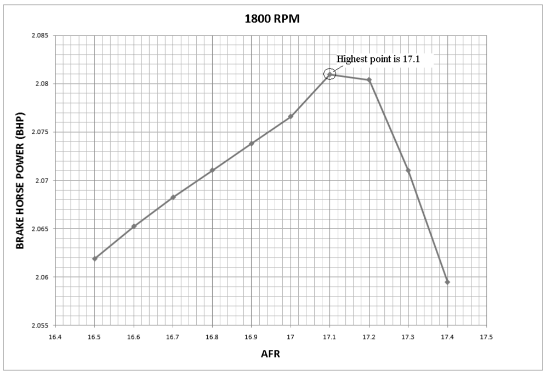
(c) 1200 RPM



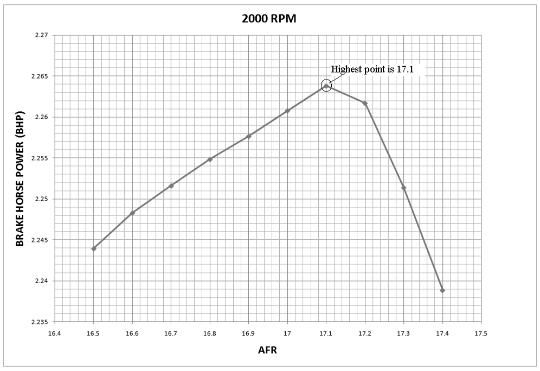
(d) 1400 RPM



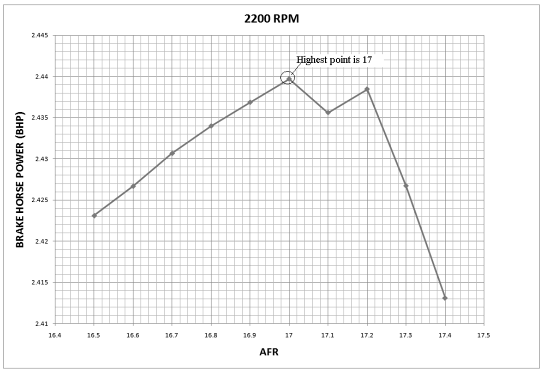
(e) 1600 RPM



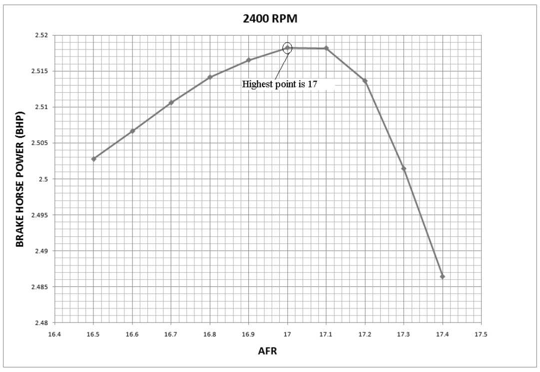
(f) 1800 RPM



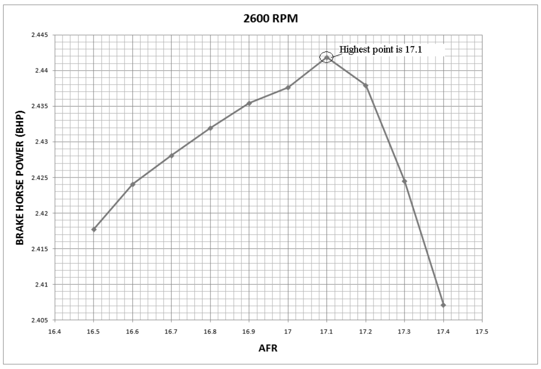
(g) 2000 RPM



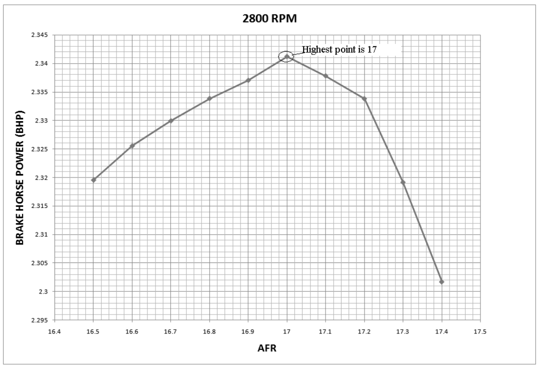
(h) 2200 RPM



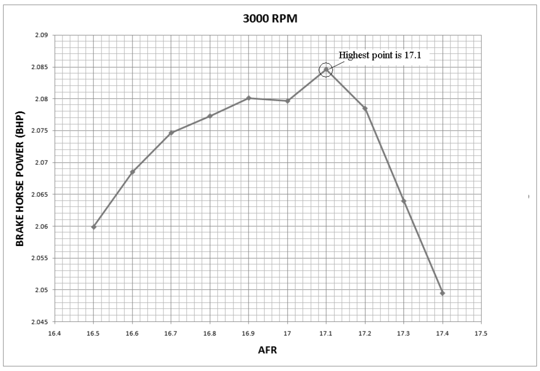
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.22: Refined view of effect of variable AFR at each RPM at 0.50 Bar

Figure 4.22 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.6 shows the best AFR for each RPM read from the refined view graphs.

Table 4.6: Best AFR for each RPM at manifold pressure of 0.50 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.3	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17	17	17.1	17	17.1

4.2.7 Results for manifold pressure of 0.55 Bar

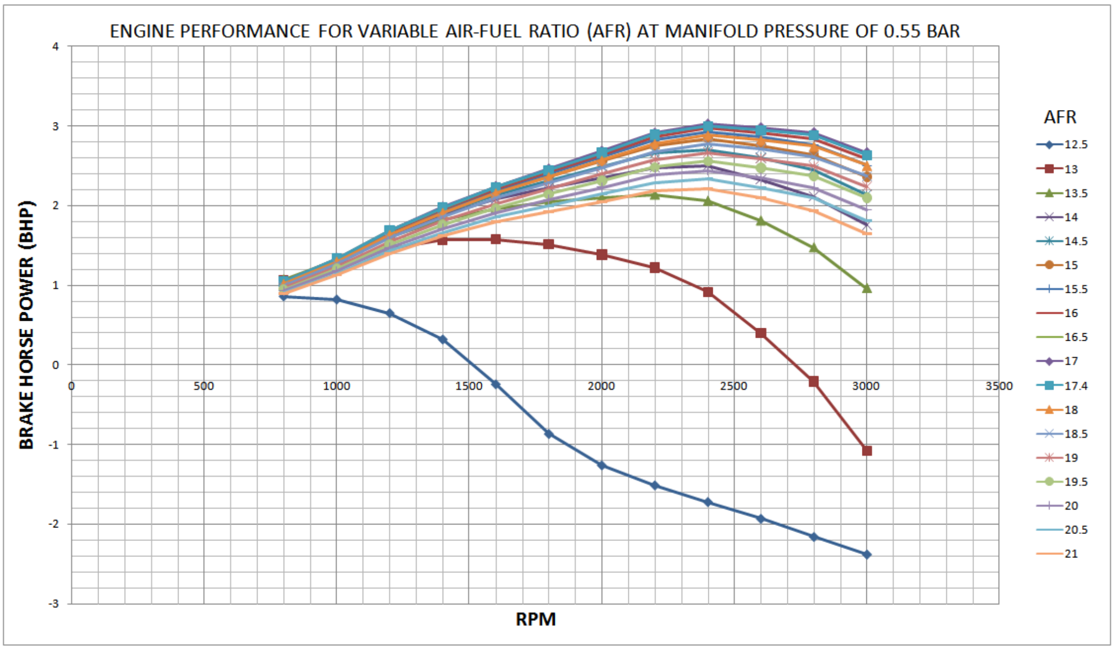


Figure 4.23: Engine performance for variable AFR at manifold pressure of 0.55 Bar

Figure 4.23 shows the engine performance at manifold pressure of 0.55 Bar. Again the peak power occurs at 2400 RPM for most AFR. The trend is the same as performance at 0.50 Bar. The only difference is the maximum horse power attained is 3 BHP, which has increased again with the increased manifold pressure.

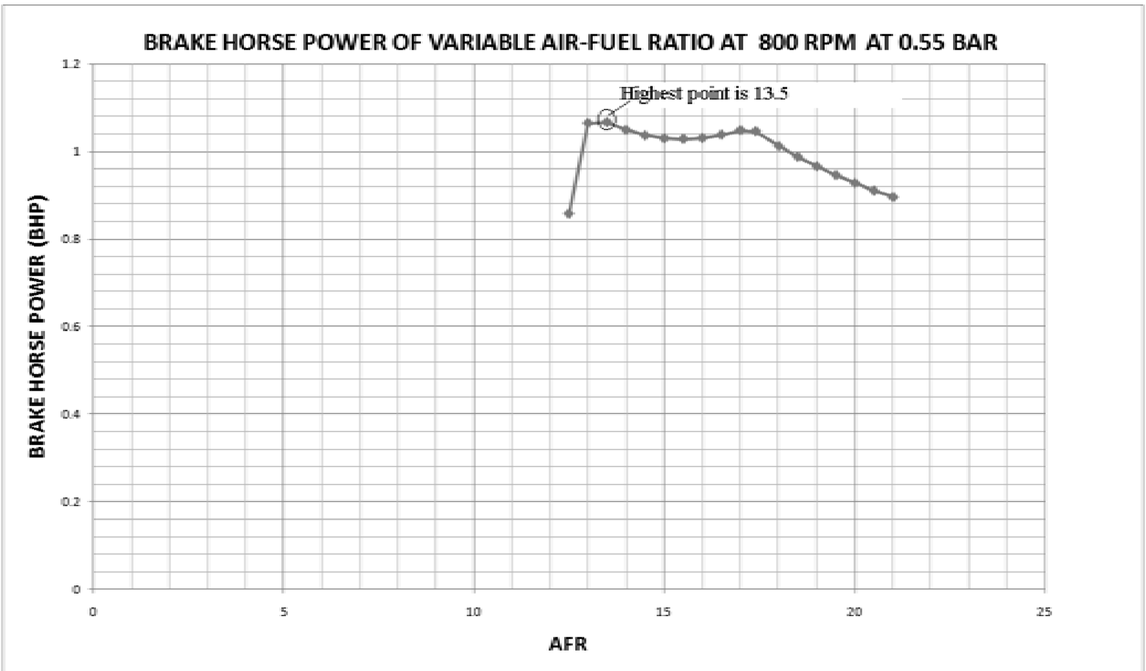


Figure 4.24: Overview of effect of variable AFR at 800 RPM at 0.55 Bar

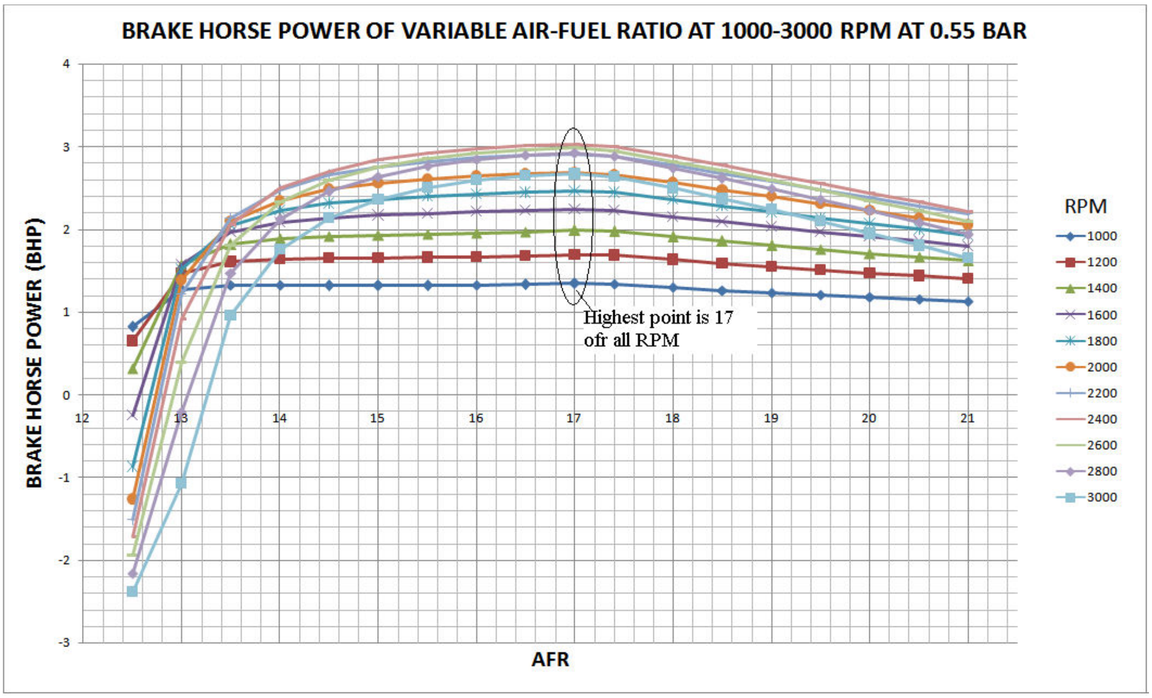
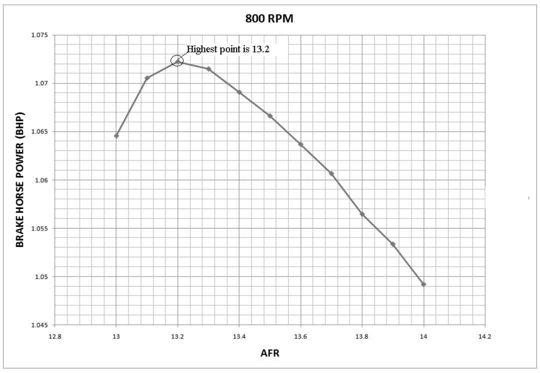
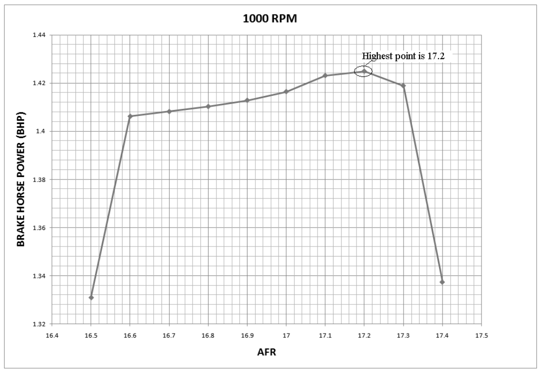


Figure 4.25: Overview of effect of variable AFR at 1000-3000 RPM at 0.55 Bar

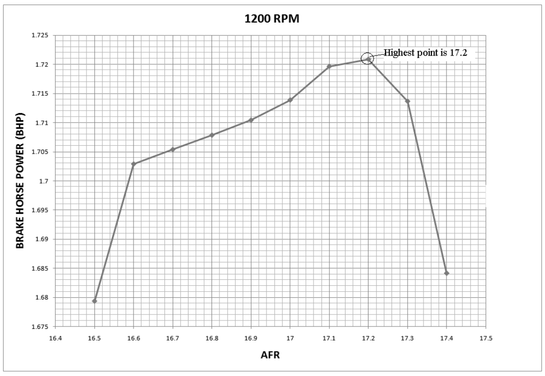
Figure 4.24 and Figure 4.25 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.24 the highest point for 800 RPM is 13.5 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13.5 for 800 RPM, the refined view is ranged from 13 to 14 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



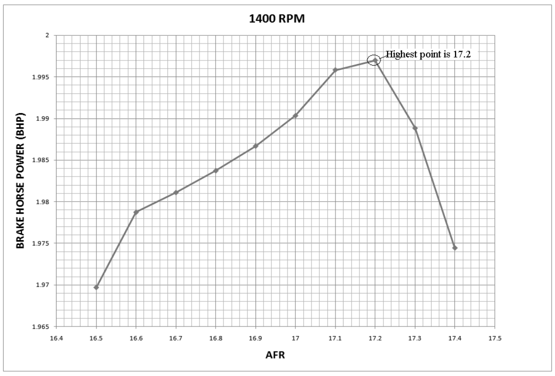
(a) 800 RPM



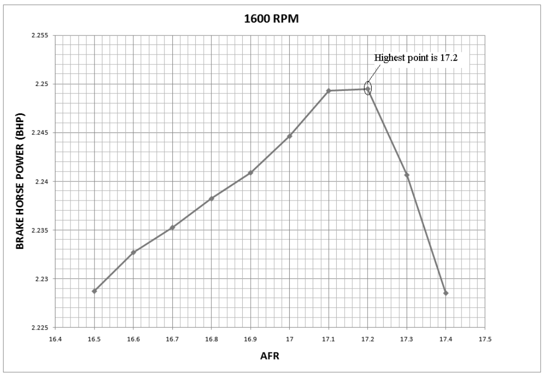
(b) 1000 RPM



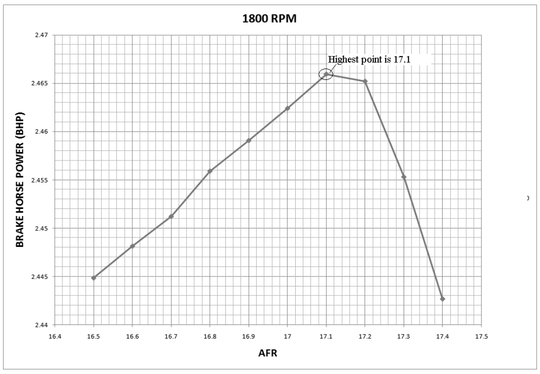
(c) 1200 RPM



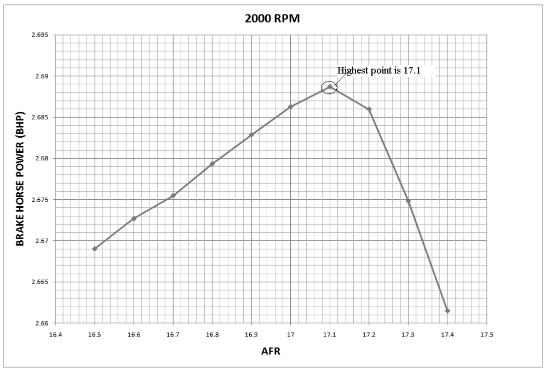
(d) 1400 RPM



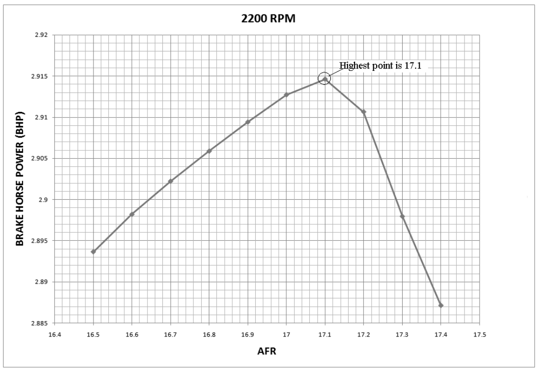
(e) 1600 RPM



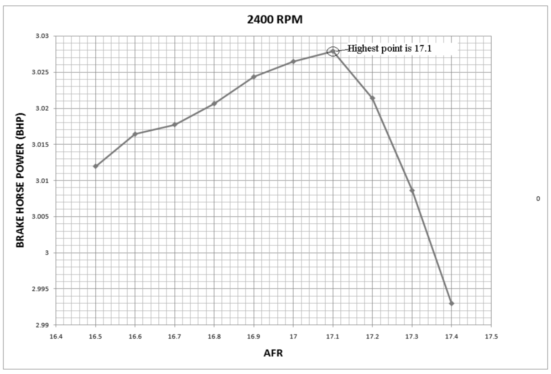
(f) 1800 RPM



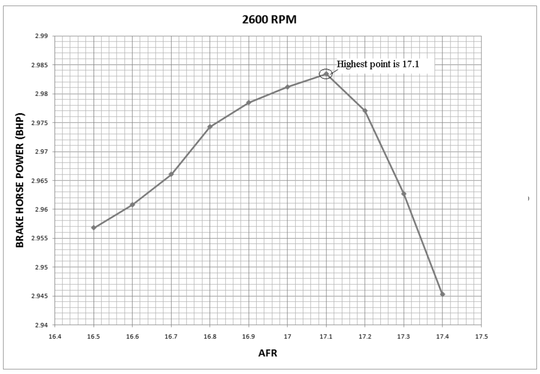
(g) 2000 RPM



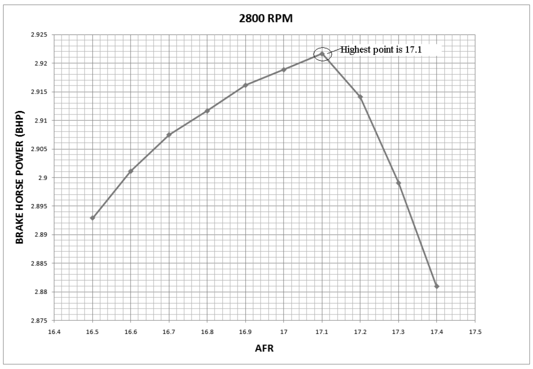
(h) 2200 RPM

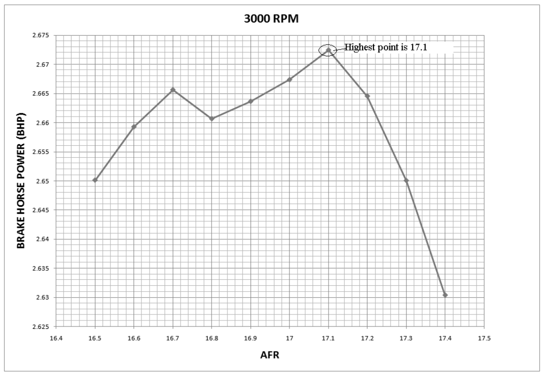


(i) 2400 RPM



(j) 2600 RPM





(k) 2800 RPM

(l) 3000 RPM

Figure 4.26: Refined view of effect of variable AFR at each RPM at 0.55 Bar

Figure 4.26 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.7 shows the best AFR for each RPM read from the refined view graphs.

Table 4.7: Best AFR for each RPM at manifold pressure of 0.55 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.2	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	17.1	17.1

4.2.8 Results for manifold pressure of 0.60 Bar

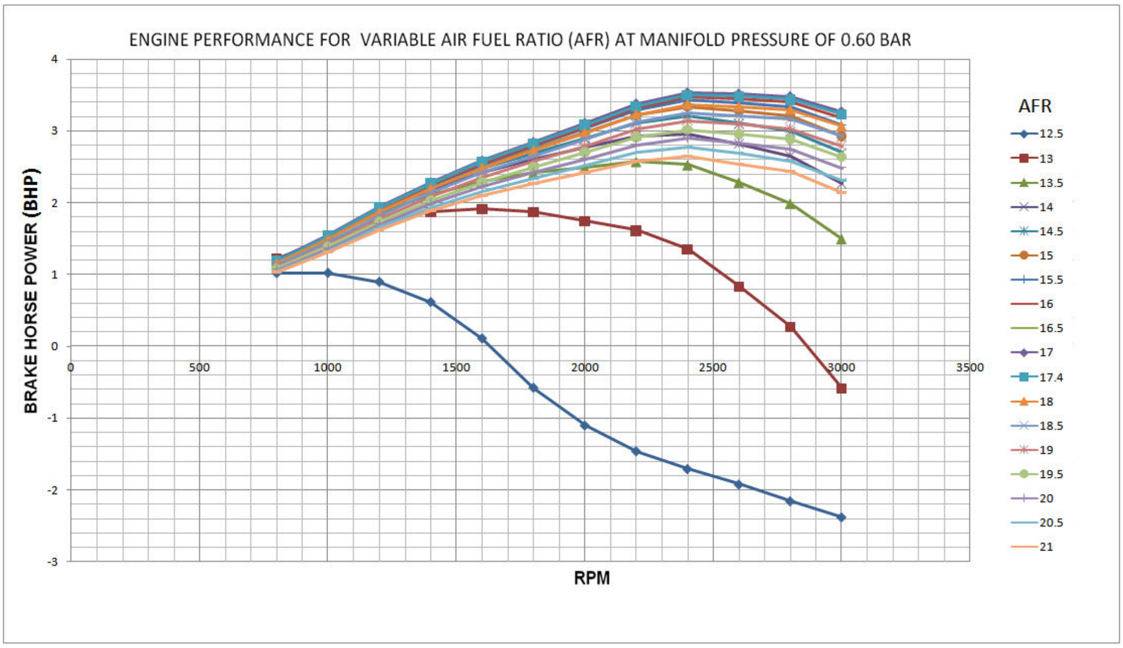


Figure 4.27: Engine performance for variable AFR at manifold pressure of 0.60 Bar

Figure 4.27 shows the engine performance at manifold pressure of 0.60 Bar. Again the peak power occurs at 2400 RPM for most AFR. The trend is the same as performance at 0.55 Bar. The only difference is the maximum horse power attained is 3.5 BHP, which has increased again with the increased manifold pressure.

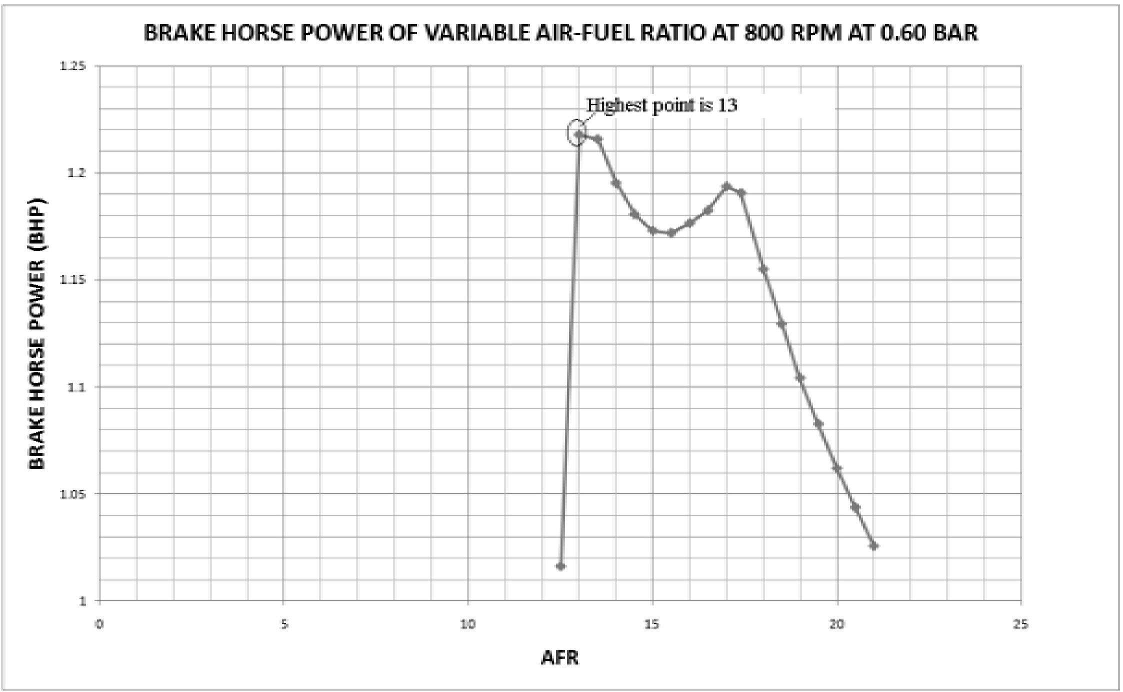


Figure 4.28: Overview of effect of variable AFR at 800 RPM at 0.60 Bar

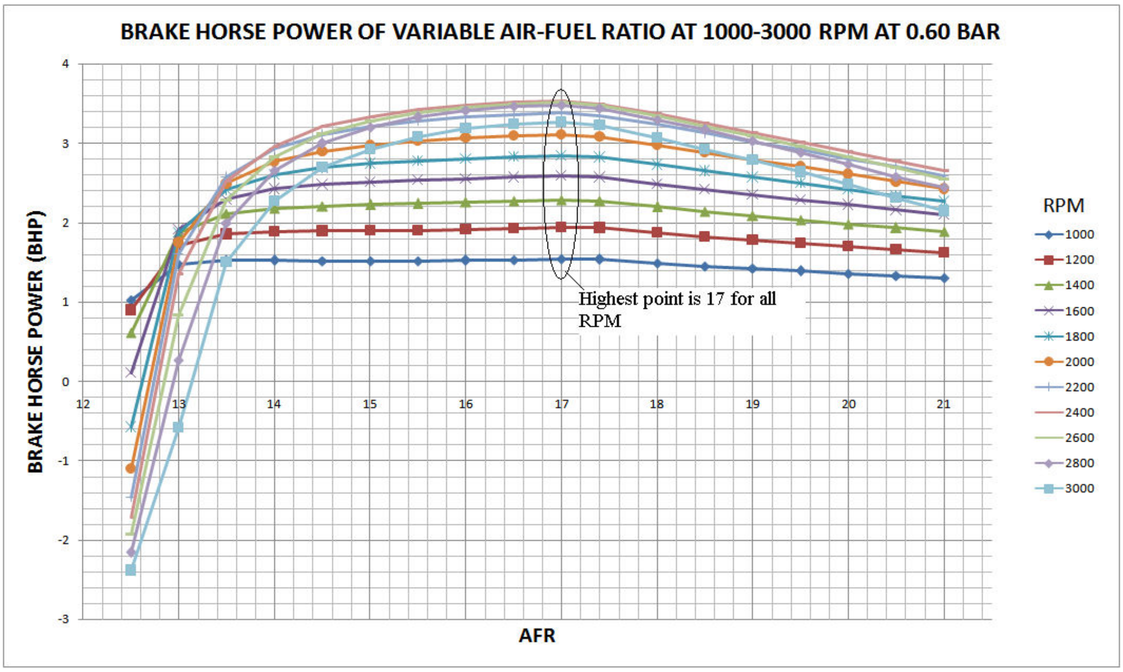
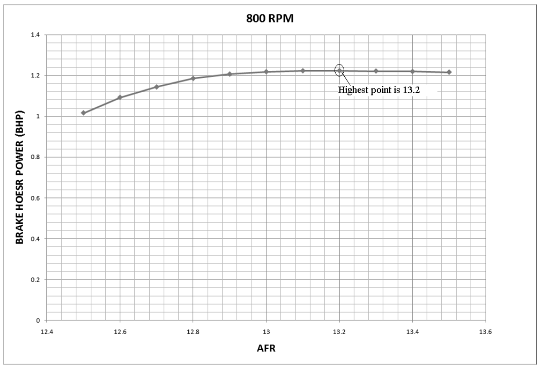
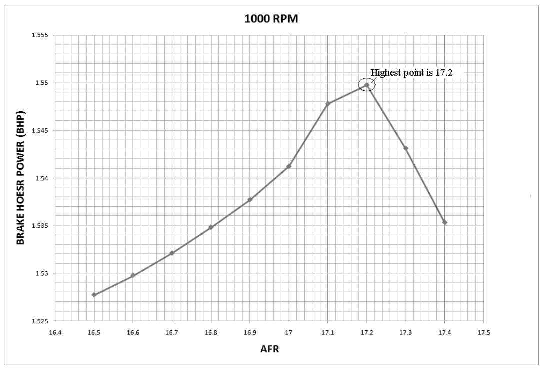


Figure 4.29: Overview of effect of variable AFR at 1000-3000 RPM at 0.60 Bar

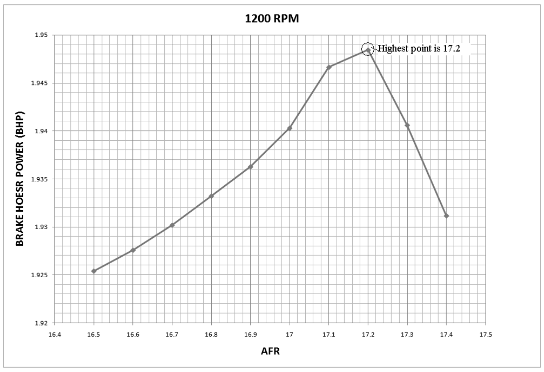
Figure 4.28 and Figure 4.29 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.28 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



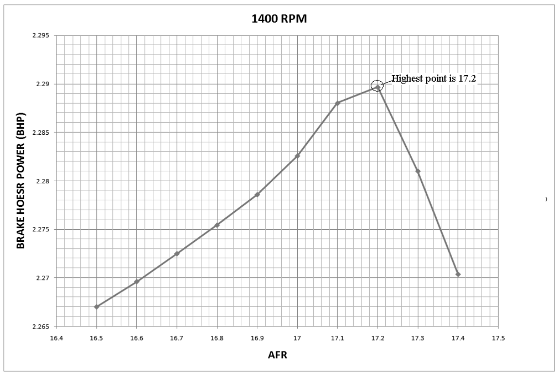
(a) 800 RPM



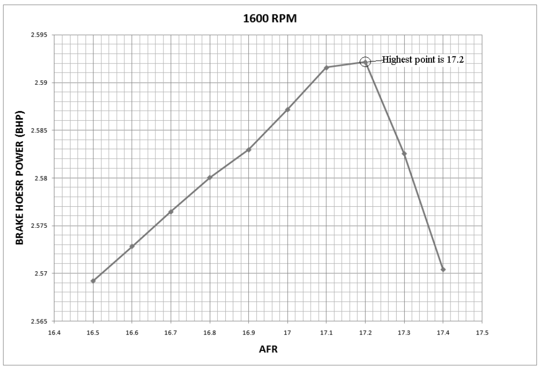
(b) 1000 RPM



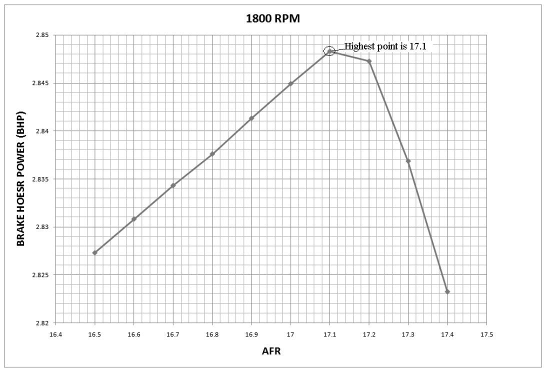
(c) 1200 RPM



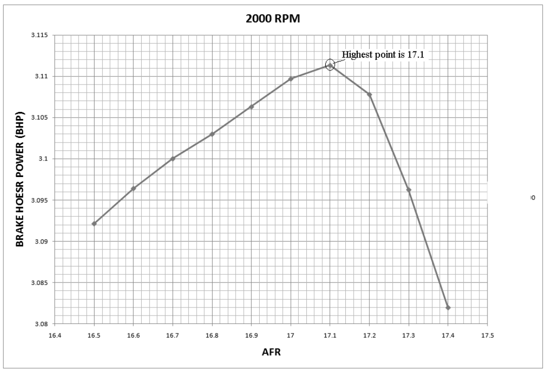
(d) 1400 RPM



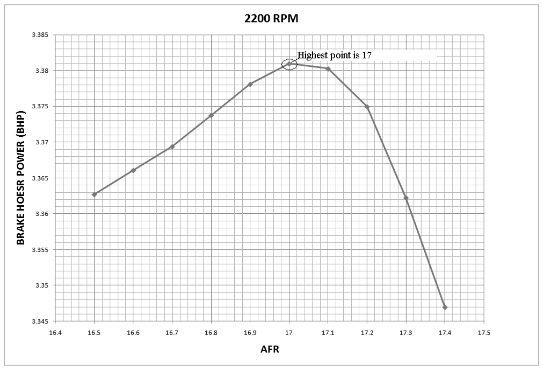
(e) 1600 RPM



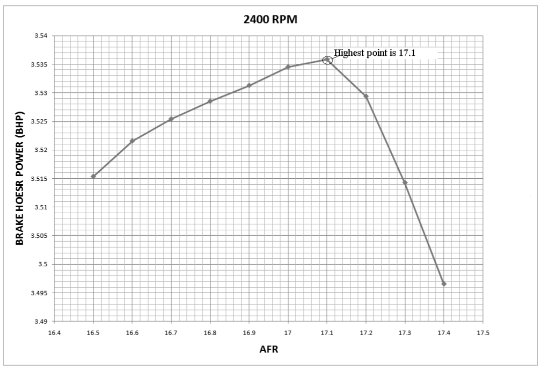
(f) 1800 RPM



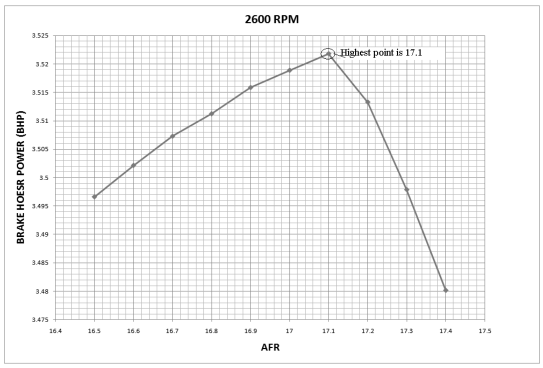
(g) 2000 RPM



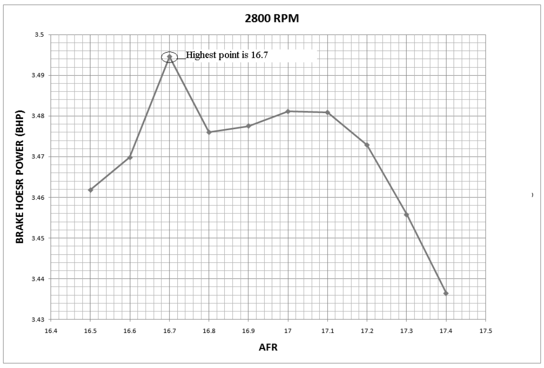
(h) 2200 RPM



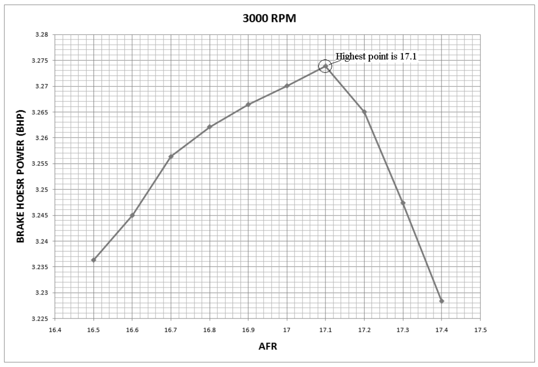
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.30: Refined view of effect of variable AFR at each RPM at 0.60 Bar

Figure 4.30 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.8 shows the best AFR for each RPM read from the refined view graphs.

Table 4.8: Best AFR for each RPM at manifold pressure of 0.60 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.2	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17	17.1	17.1	16.7	17.1

4.2.9 Results for manifold pressure of 0.65 Bar

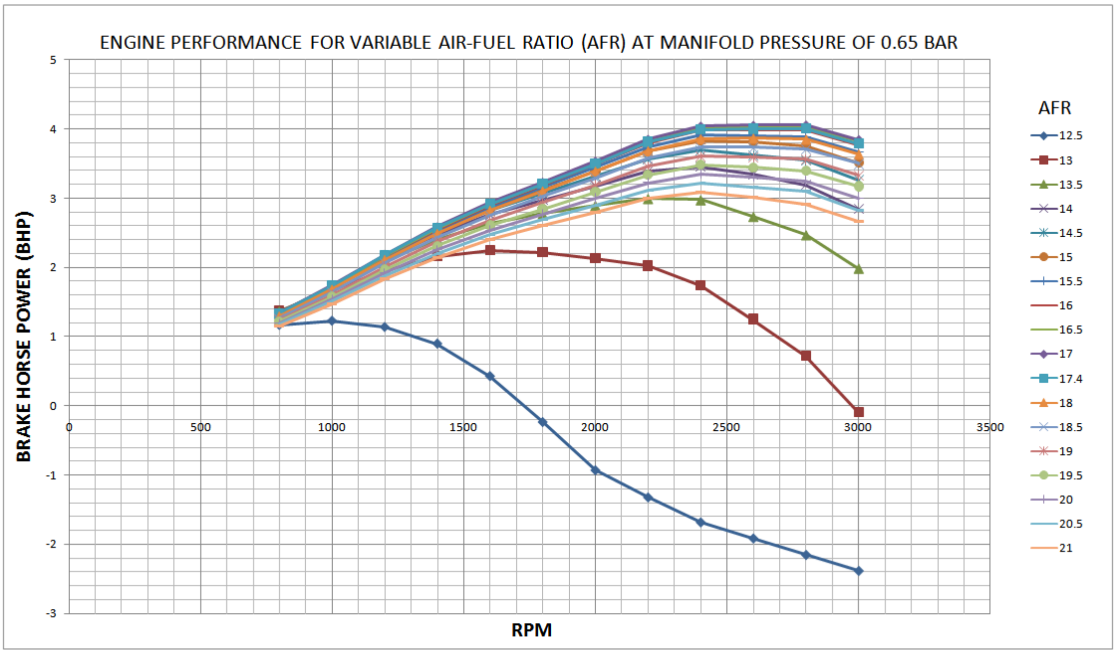


Figure 4.31: Engine performance for variable AFR at manifold pressure of 0.65 Bar

Figure 4.31 shows the engine performance at manifold pressure of 0.65 Bar. The peak power occurs in the range of 2400 RPM to 2800 RPM for most AFR. The highest power attained is about 4 BHP. The rich AFR such as 13 and 13.5 is not in the negative horse power range anymore as in before. But their power still lack for high RPM range. This is due to the mixture is too wet to have adequate time to burn. At high RPM the combustion time is lesser because the time for a complete cycle is lesser due to the high engine speed. There is not enough time to achieve complete burning of the mixture which is hard to burn due to wet as a consequence of too rich. Also the ignition timing in this simulation is fixed; it is not advanced in the high RPM range causing the ignition to be late during the high RPM cycle.

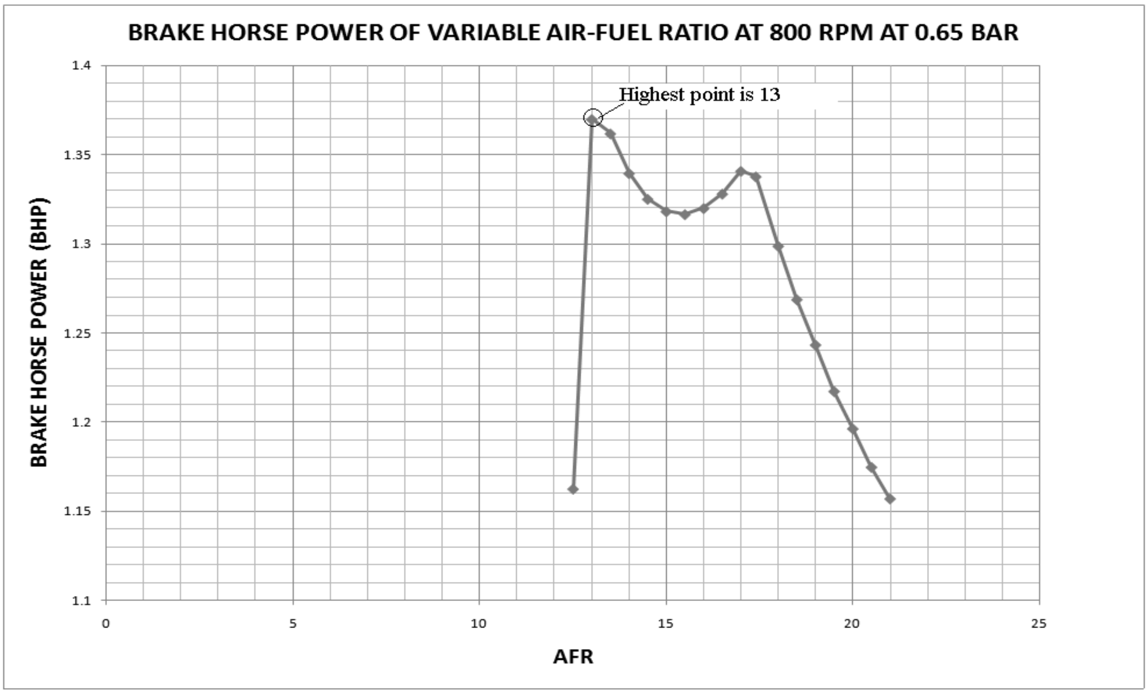


Figure 4.32: Overview of effect of variable AFR at 800 RPM at 0.65 Bar

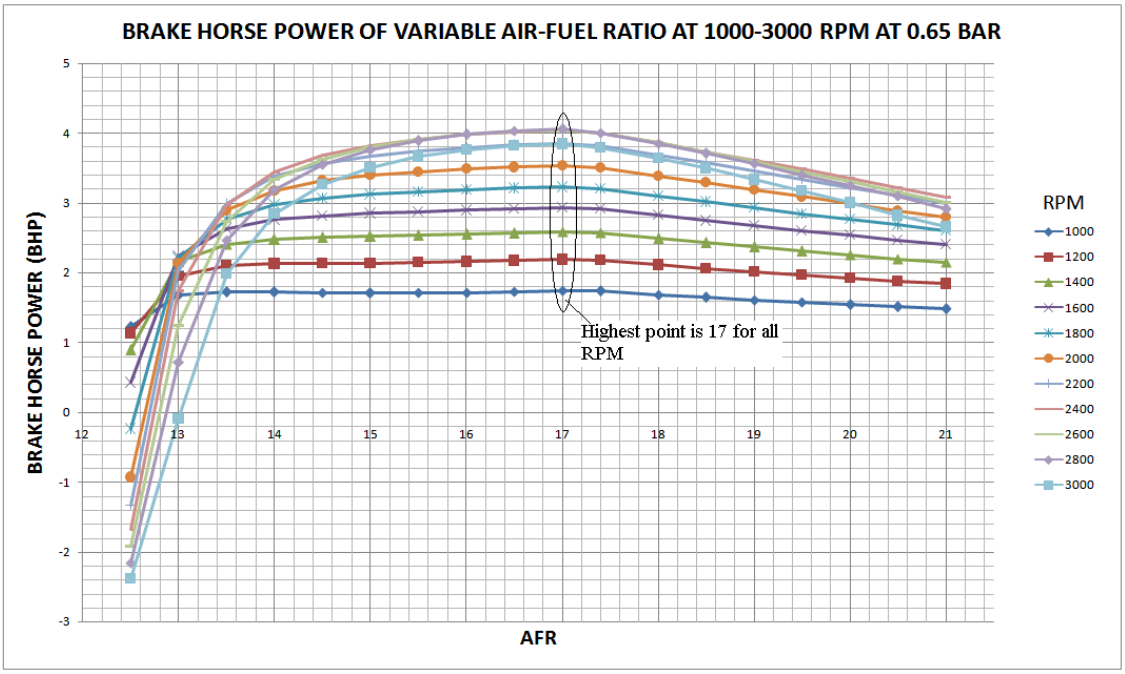
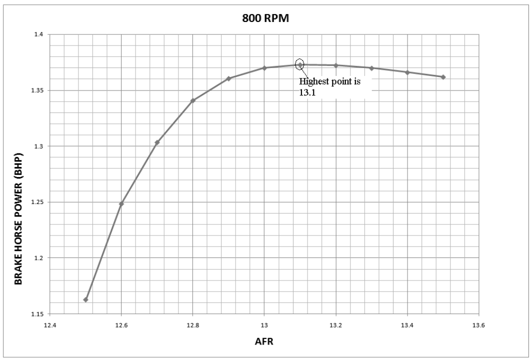
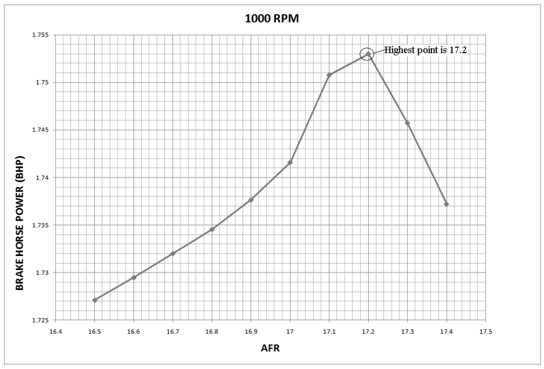


Figure 4.33: Overview of effect of variable AFR at 1000-3000 RPM at 0.65 Bar

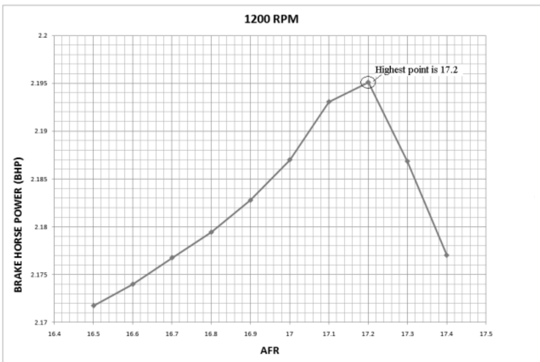
Figure 4.32 and Figure 4.33 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.32 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



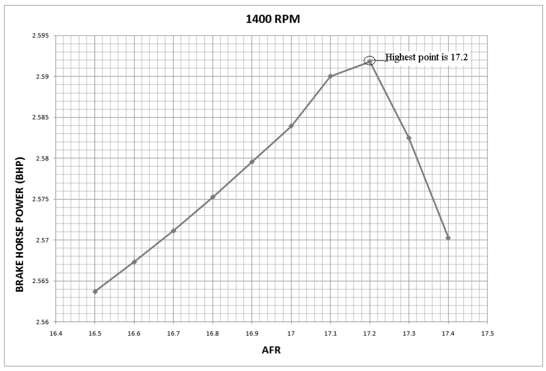
(a) 800 RPM



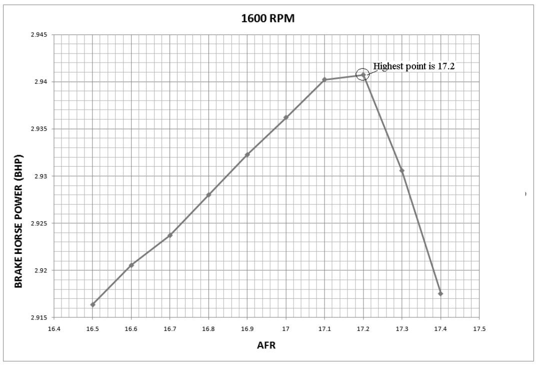
(b) 1000 RPM



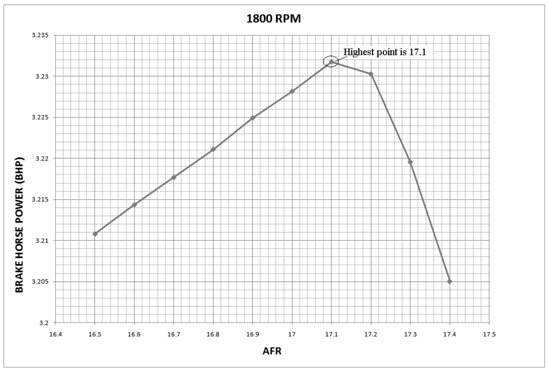
(c) 1200 RPM



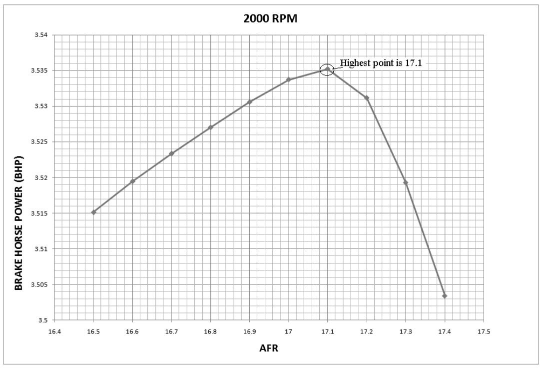
(d) 1400 RPM



(e) 1600 RPM



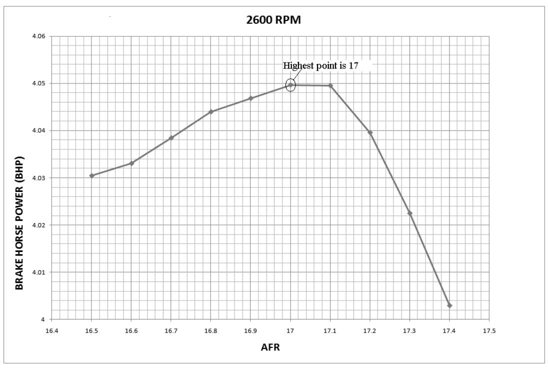
(f) 1800 RPM



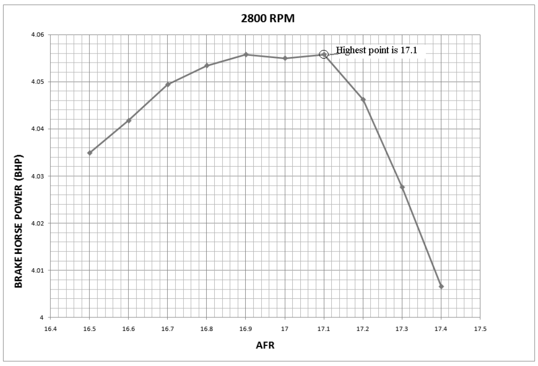
(g) 2000 RPM

(h) 2200 RPM

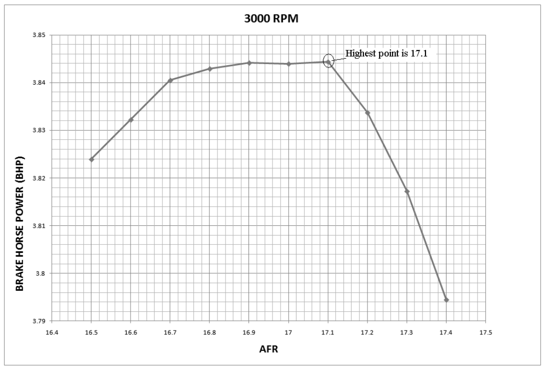
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.34: Refined view of effect of variable AFR at each RPM at 0.65 Bar

Figure 4.34 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.9 shows the best AFR for each RPM read from the refined view graphs.

Table 4.9: Best AFR for each RPM at manifold pressure of 0.65 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.1	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17	17.1	17.1

4.2.10 Results for manifold pressure of 0.70 Bar

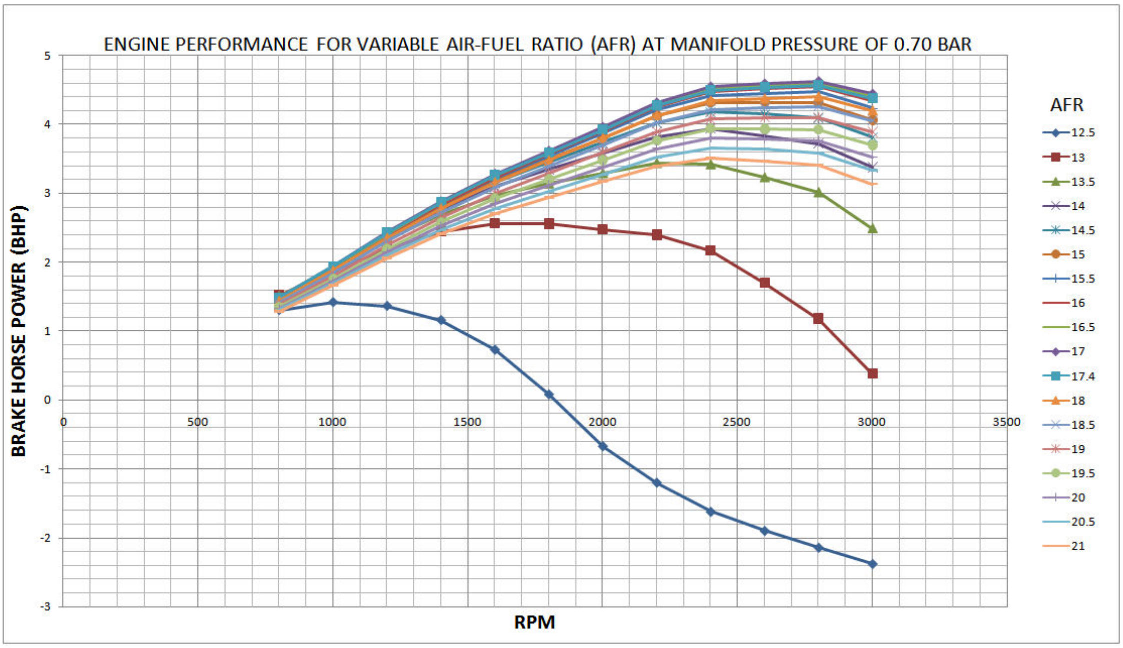


Figure 4.35: Engine performance for variable AFR at manifold pressure of 0.70 Bar

Figure 4.31 shows the engine performance at manifold pressure of 0.70 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM for most AFR. The highest power attained is about 4.5 BHP which increase with the increased manifold pressure. The performance trend is the same as for manifold pressure of 0.65 Bar. The power is produced at a higher RPM for this manifold pressure. More air mass is inducted at higher pressure; hence higher pressure can be produced due to expansion of air during combustion.

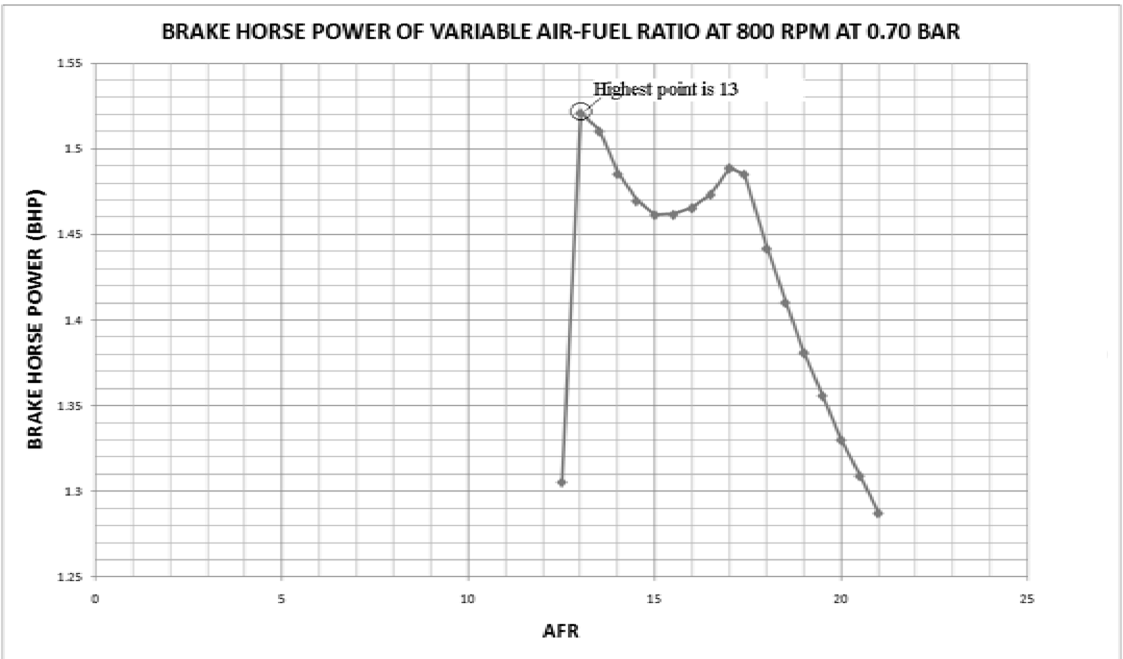


Figure 4.36: Overview of effect of variable AFR at 800 RPM at 0.70 Bar

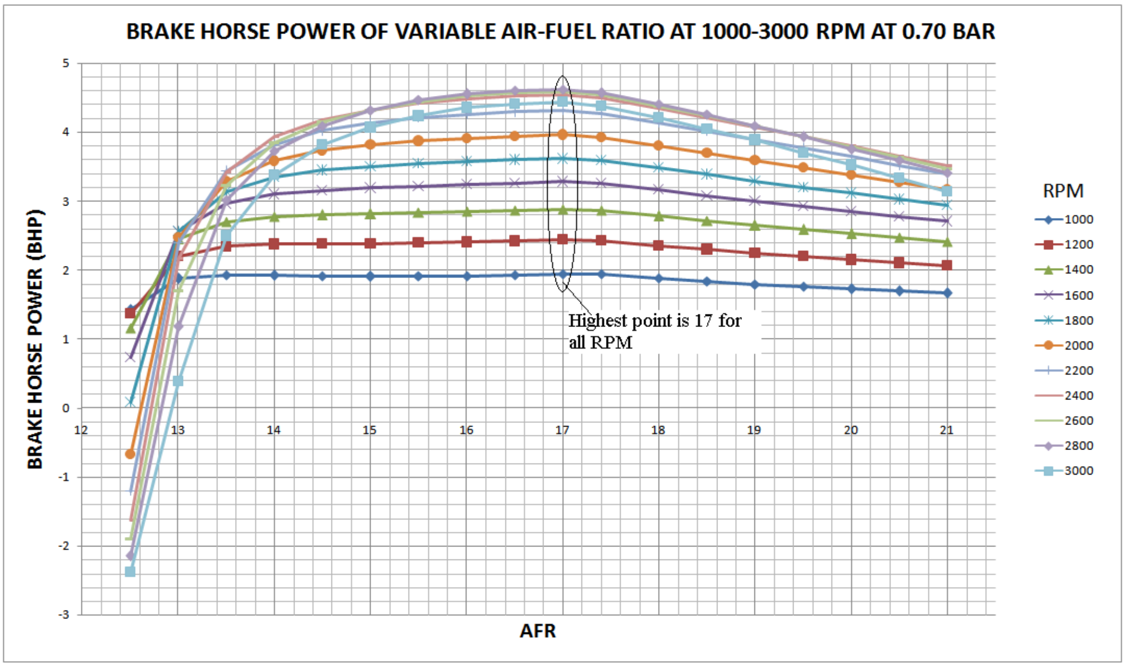
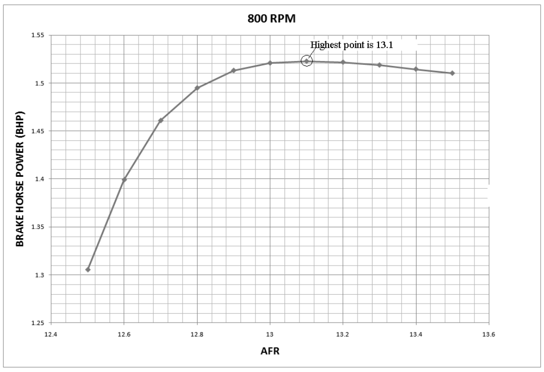
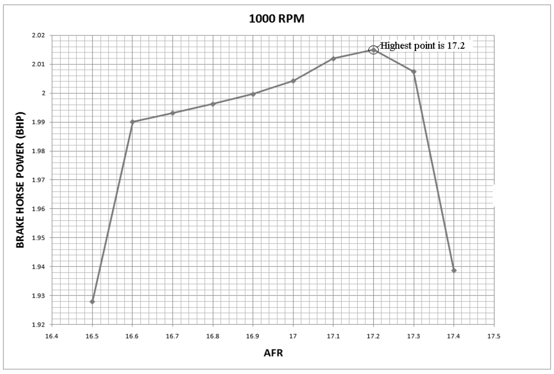


Figure 4.37: Overview of effect of variable AFR at 1000-3000 RPM at 0.70 Bar

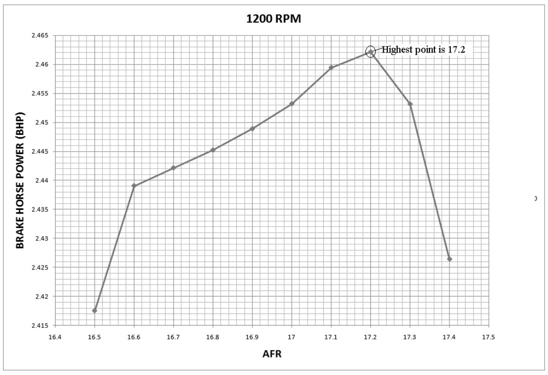
Figure 4.36 and Figure 4.37 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.36 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



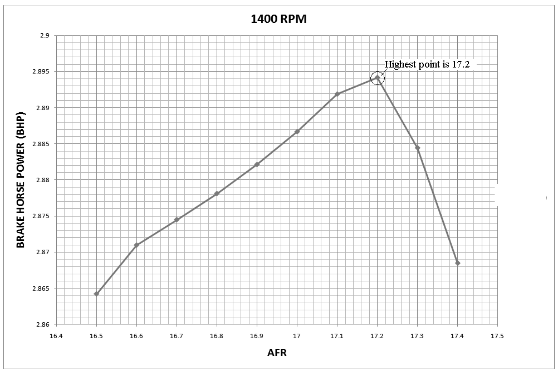
(a) 800 RPM



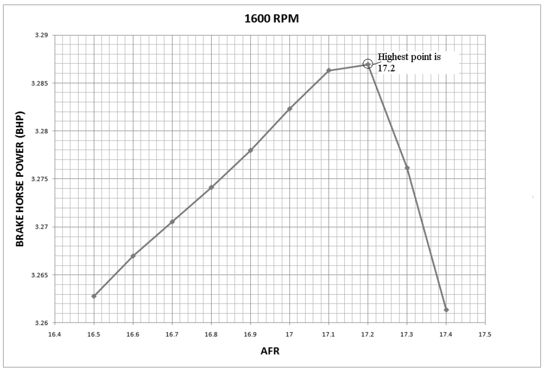
(b) 1000 RPM



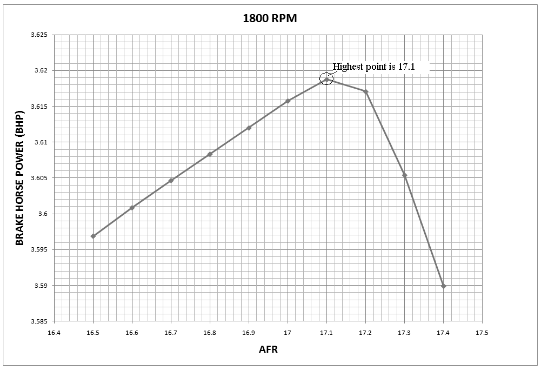
(c) 1200 RPM



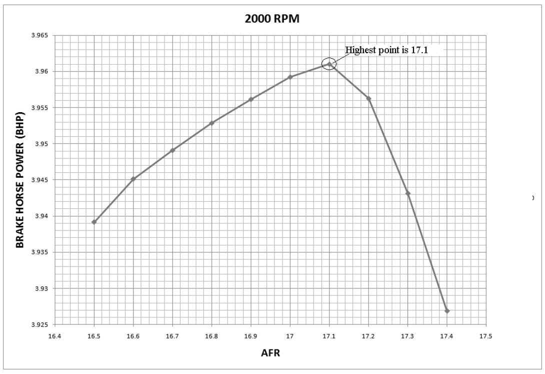
(d) 1400 RPM



(e) 1600 RPM



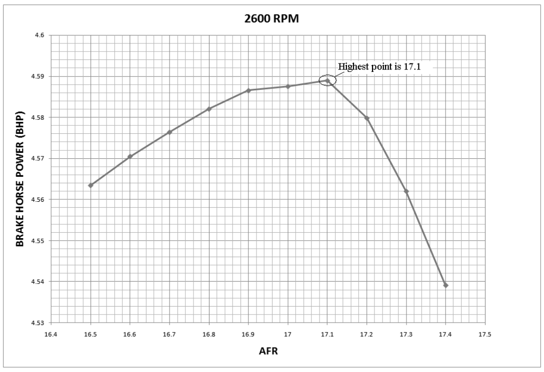
(f) 1800 RPM



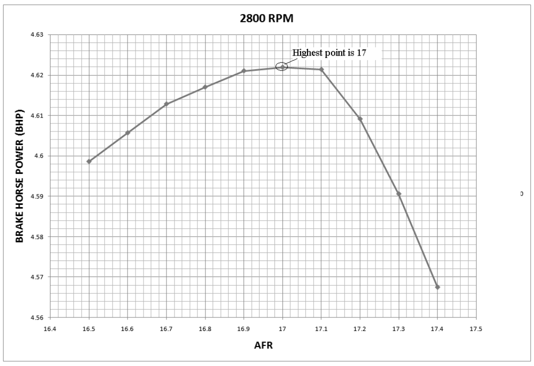
(g) 2000 RPM

(h) 2200 RPM

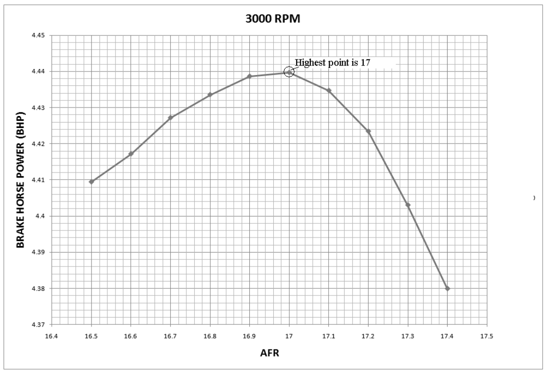
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.38: Refined view of effect of variable AFR at each RPM at 0.70 Bar

Figure 4.38 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.10 shows the best AFR for each RPM read from the refined view graphs.

Table 4.10: Best AFR for each RPM at manifold pressure of 0.70 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.1	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	17	17

4.2.11 Results for manifold pressure of 0.75 Bar

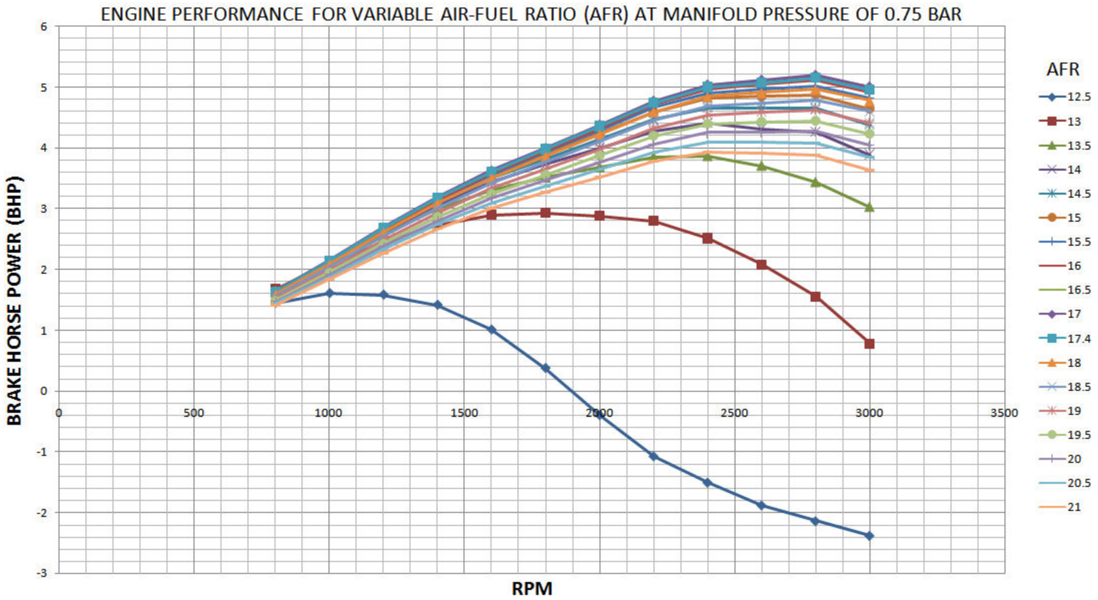


Figure 4.39: Engine performance for variable AFR at manifold pressure of 0.75 Bar

Figure 4.31 shows the engine performance at manifold pressure of 0.75 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.70 Bar. The only difference is the maximum power attained, which is about 5 BHP compared to about 4.5 BHP at manifold pressure of 0.70 Bar.

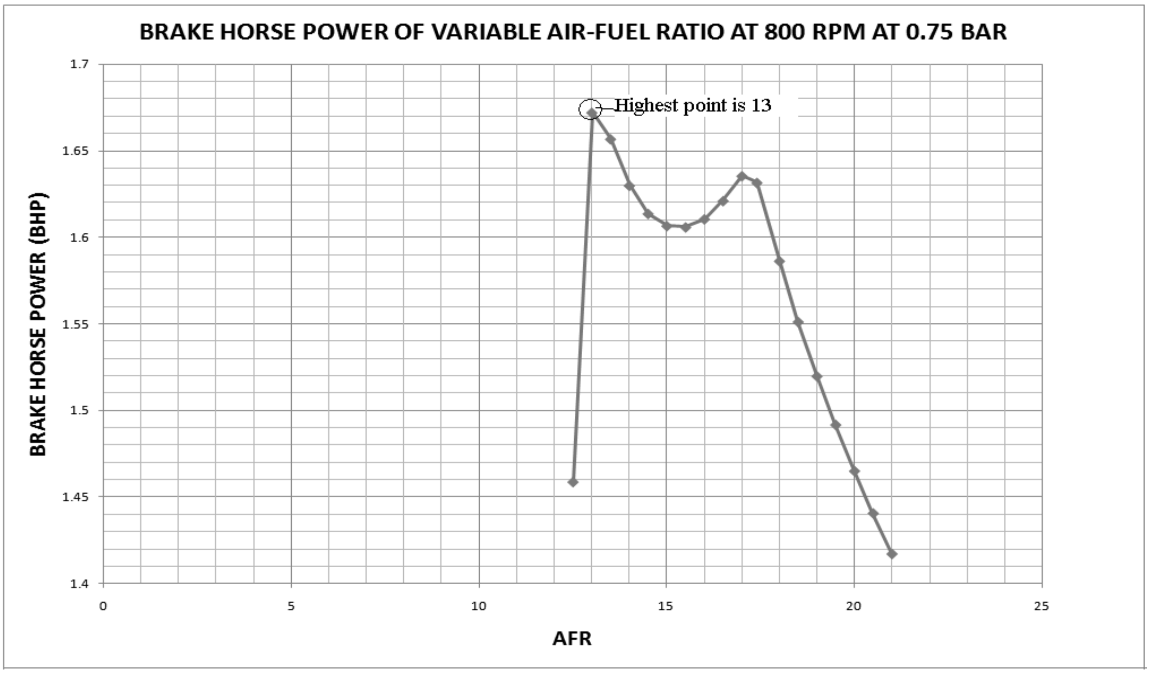


Figure 4.40: Overview of effect of variable AFR at 800 RPM at 0.75 Bar

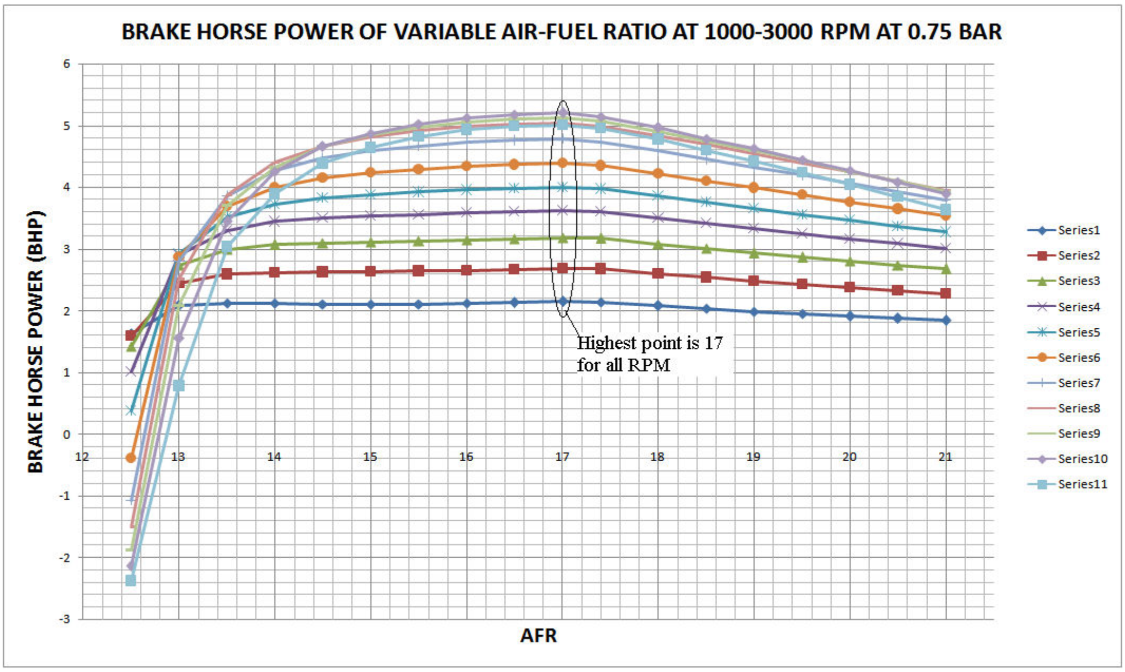
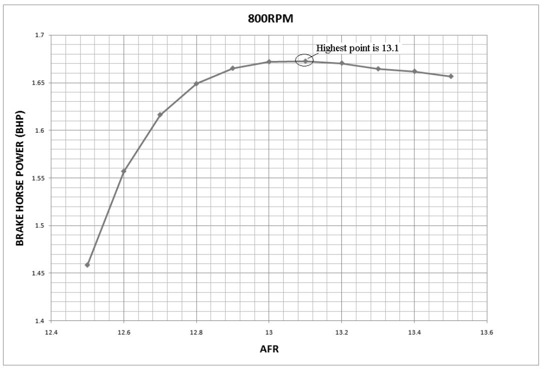
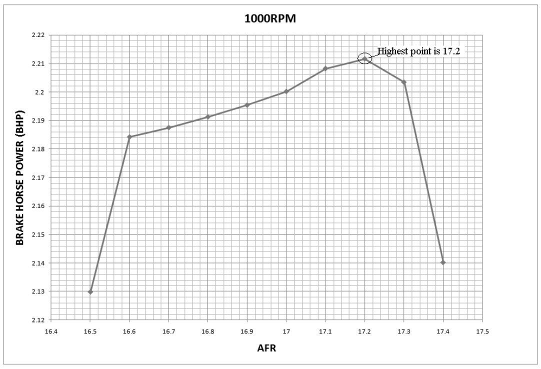


Figure 4.41: Overview of effect of variable AFR at 1000-3000 RPM at 0.75 Bar

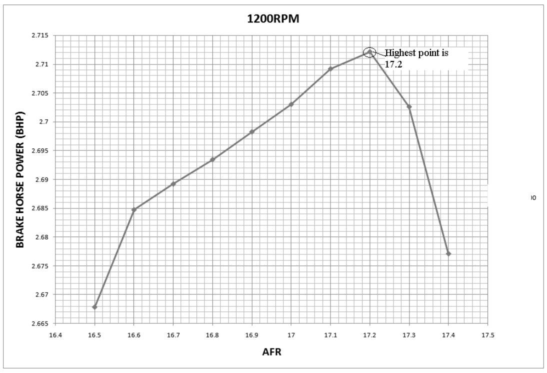
Figure 4.40 and Figure 4.41 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.40 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



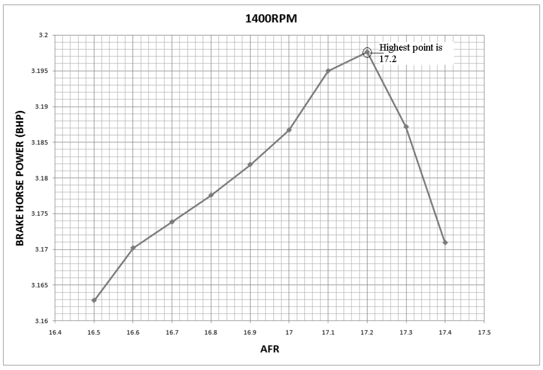
(a) 800 RPM



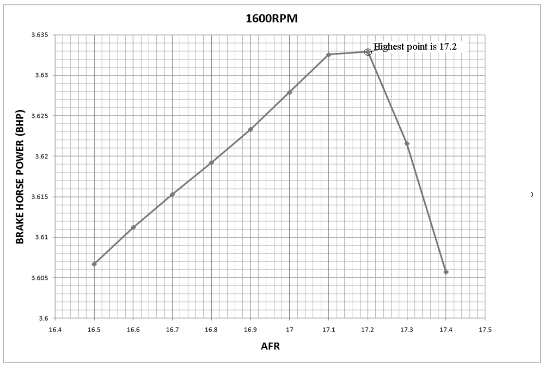
(b) 1000 RPM



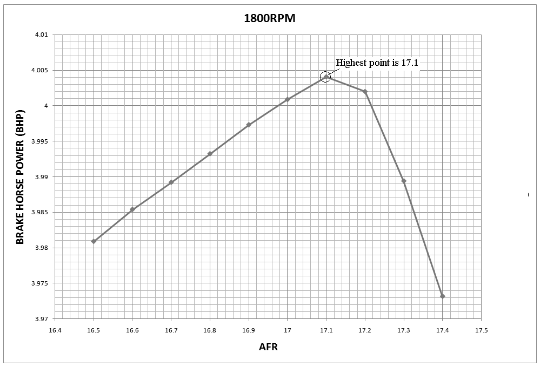
(c) 1200 RPM



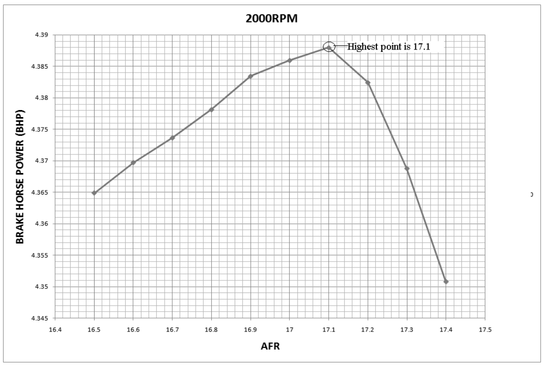
(d) 1400 RPM



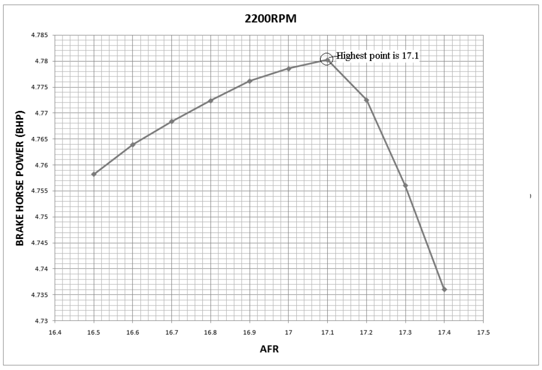
(e) 1600 RPM



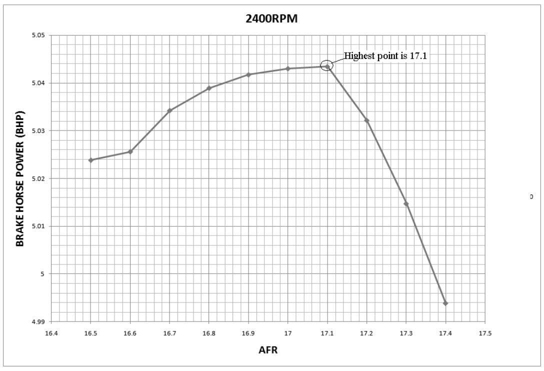
(f) 1800 RPM



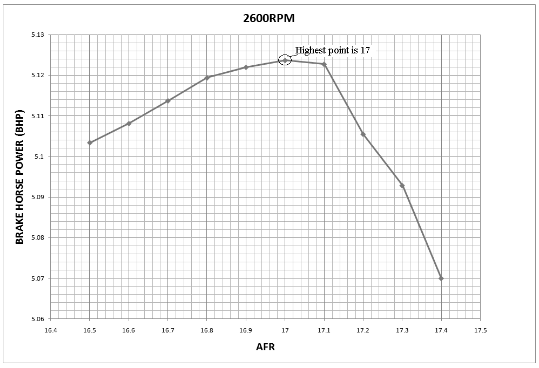
(g) 2000 RPM



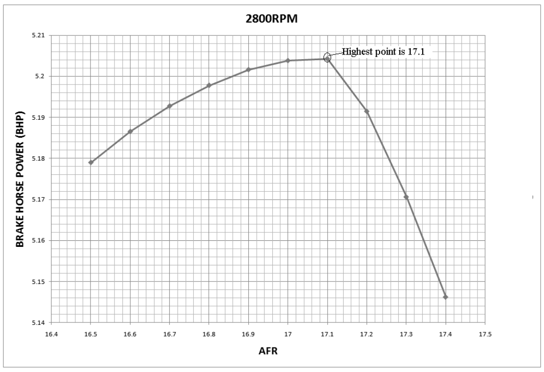
(h) 2200 RPM

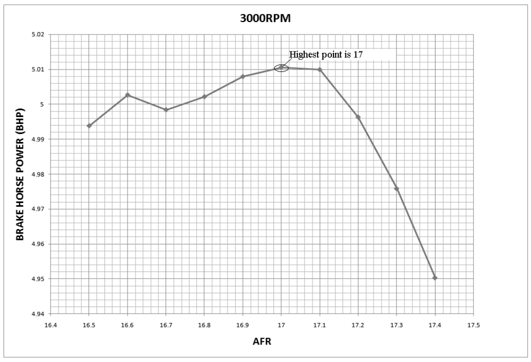


(i) 2400 RPM



(j) 2600 RPM





(k) 2800 RPM

(l) 3000 RPM

Figure 4.42: Refined view of effect of variable AFR at each RPM at 0.75 Bar

Figure 4.42 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.11 shows the best AFR for each RPM read from the refined view graphs.

Table 4.11: Best AFR for each RPM at manifold pressure of 0.75 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.1	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17	17.1	17

4.2.12 Results for manifold pressure of 0.80 Bar

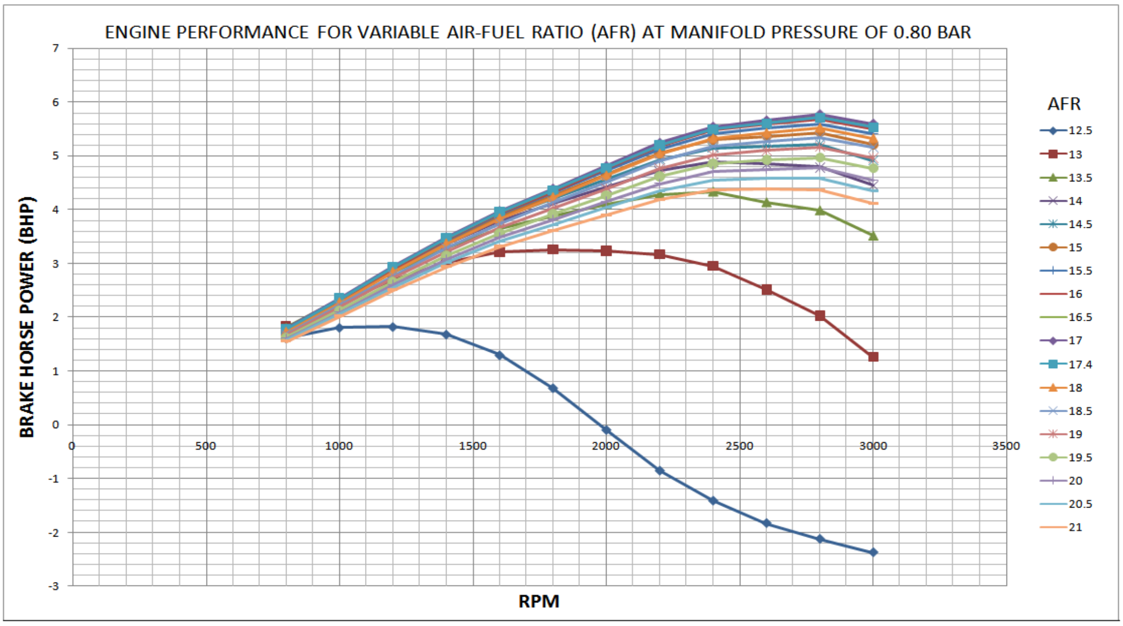


Figure 4.43: Engine performance for variable AFR at manifold pressure of 0.80 Bar

Figure 4.43 shows the engine performance at manifold pressure of 0.75 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.75 Bar. The only difference is the maximum power attained, which is about 5.8 BHP compared to about 5 BHP at manifold pressure of 0.75 Bar.

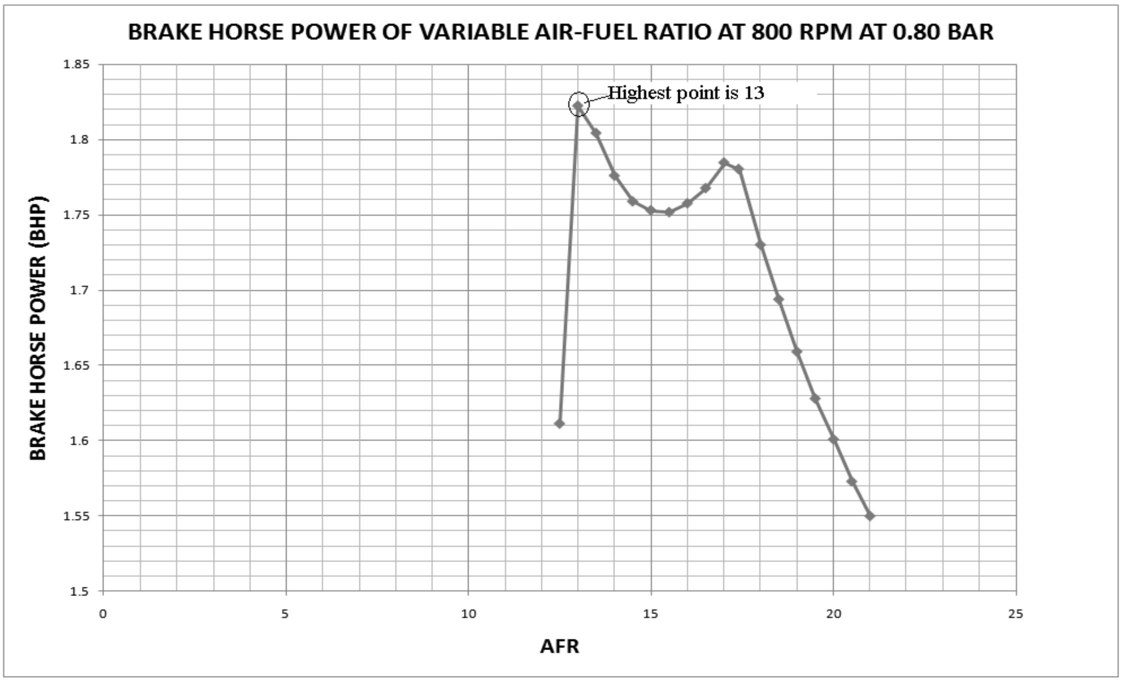


Figure 4.44: Overview of effect of variable AFR at 800 RPM at 0.80 Bar

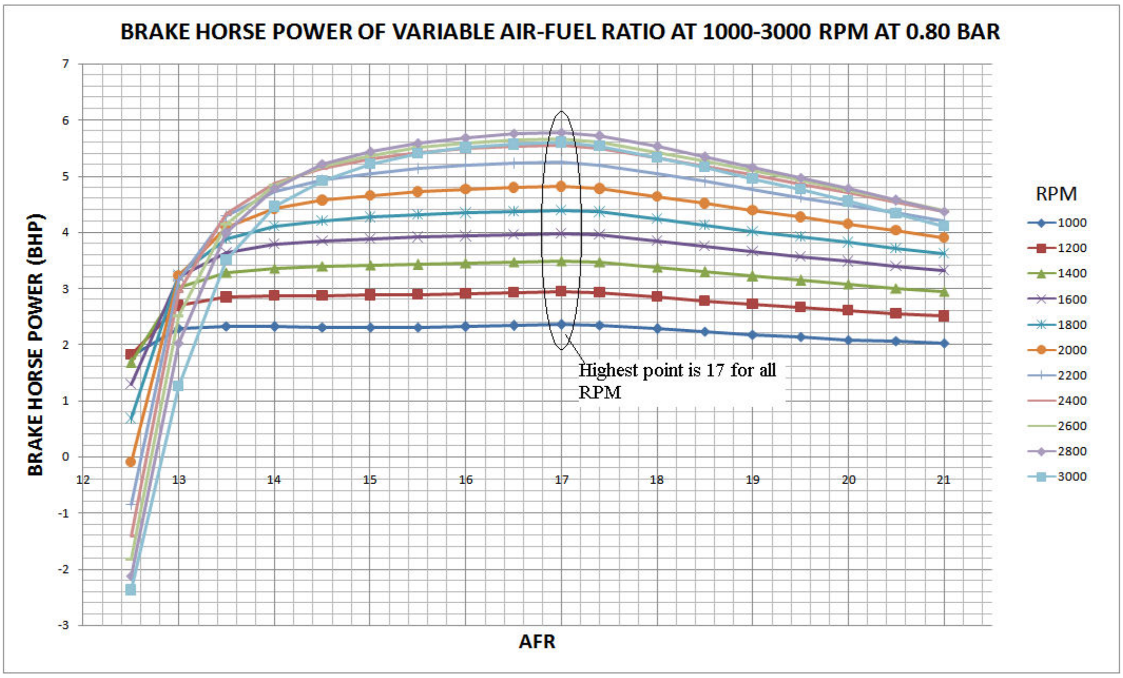
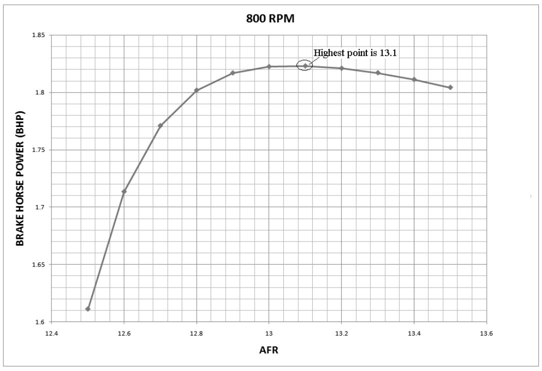
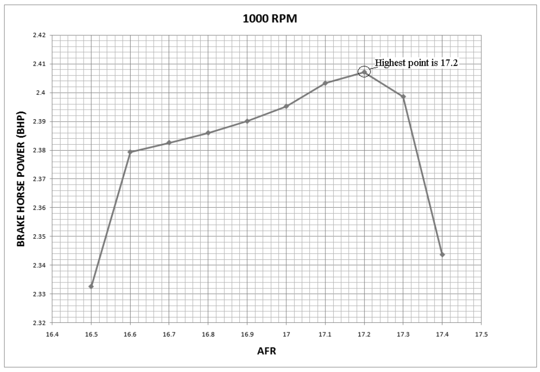


Figure 4.45: Overview of effect of variable AFR at 1000-3000 RPM at 0.80 Bar

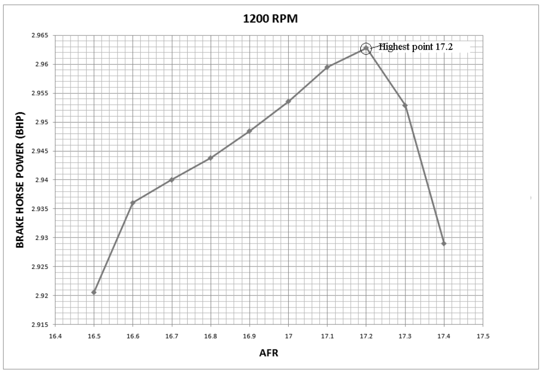
Figure 4.44 and Figure 4.45 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.44 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



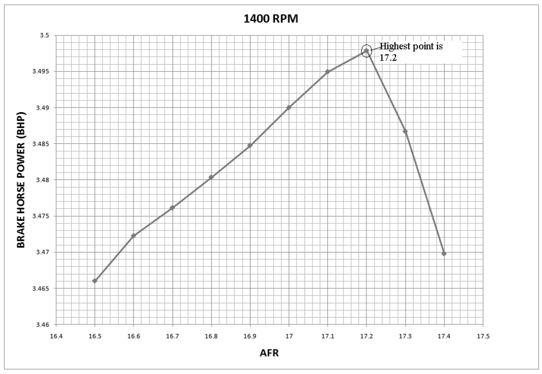
(a) 800 RPM



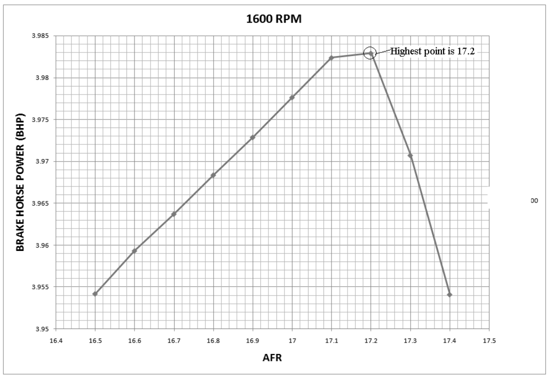
(b) 1000 RPM



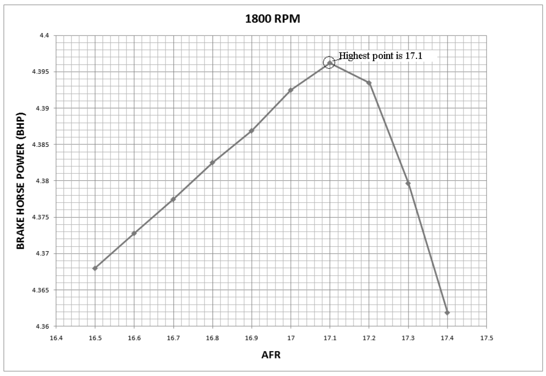
(c) 1200 RPM



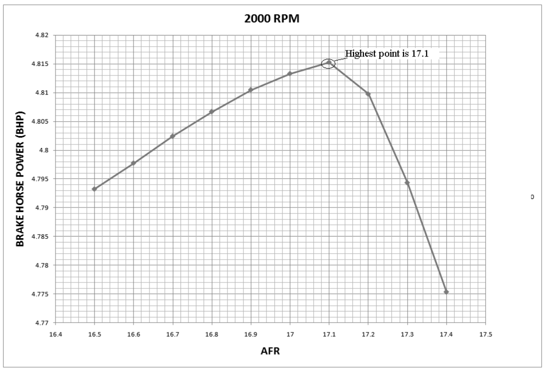
(d) 1400 RPM



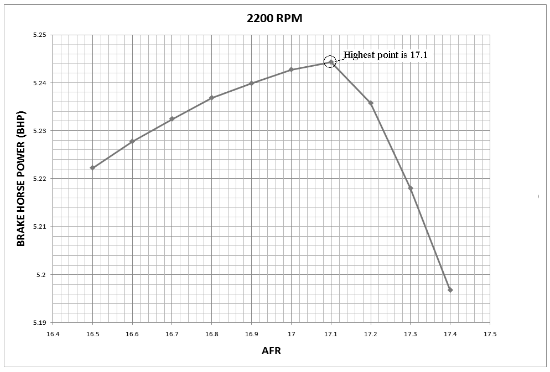
(e) 1600 RPM



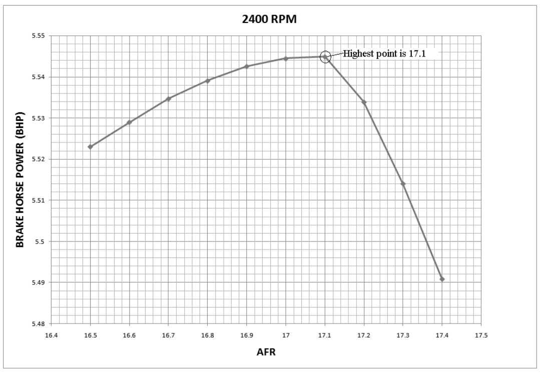
(f) 1800 RPM



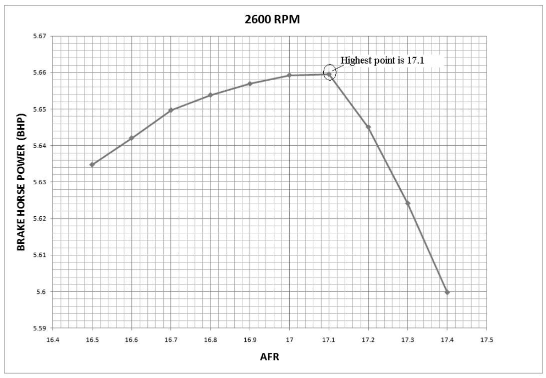
(g) 2000 RPM



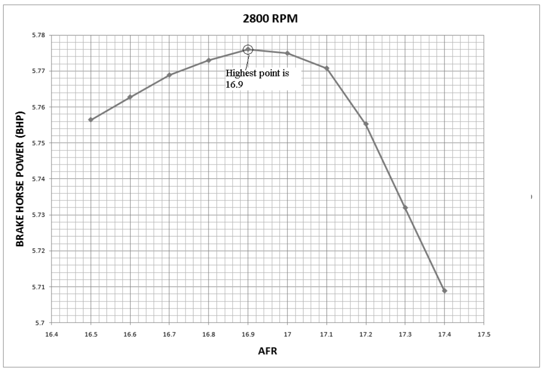
(h) 2200 RPM



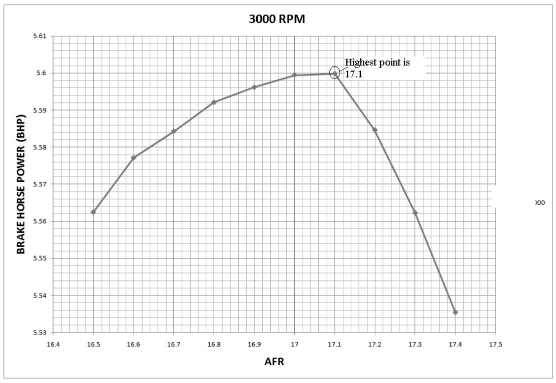
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.46: Refined view of effect of variable AFR at each RPM at 0.80 Bar

Figure 4.46 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.12 shows the best AFR for each RPM read from the refined view graphs.

Table 4.12: Best AFR for each RPM at manifold pressure of 0.80 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.1	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17.1	16.9	17.1

4.2.13 Results for manifold pressure of 0.85 Bar

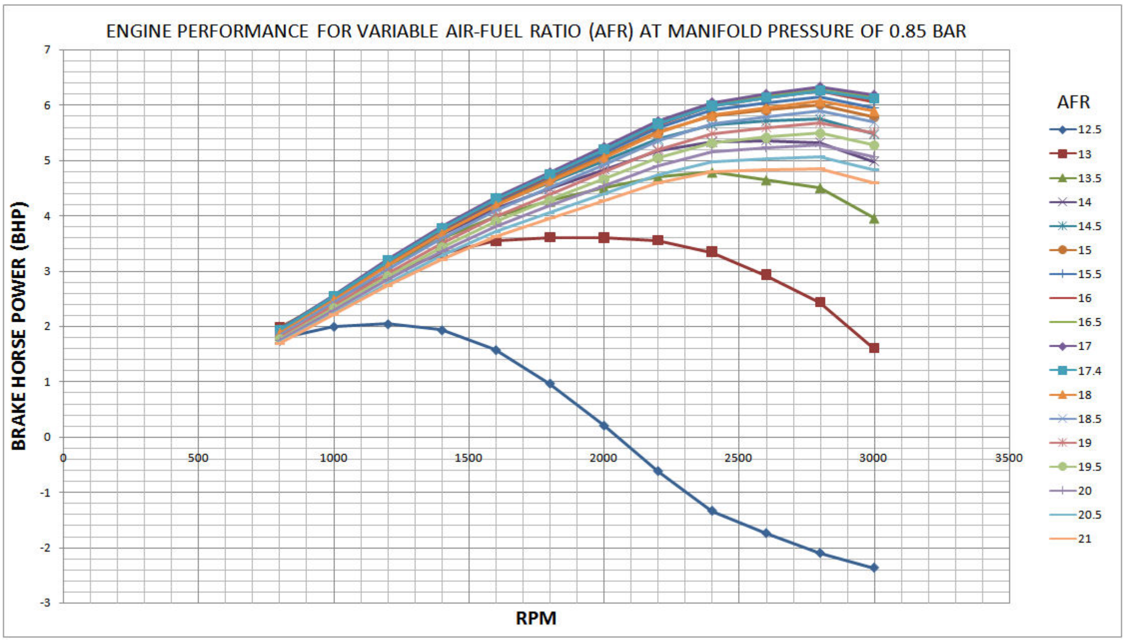


Figure 4.47: Engine performance for variable AFR at manifold pressure of 0.85 Bar

Figure 4.47 shows the engine performance at manifold pressure of 0.85 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.80 Bar. The only difference is the maximum power attained, which is about 6.4 BHP compared to about 5.8 BHP at manifold pressure of 0.80 Bar.

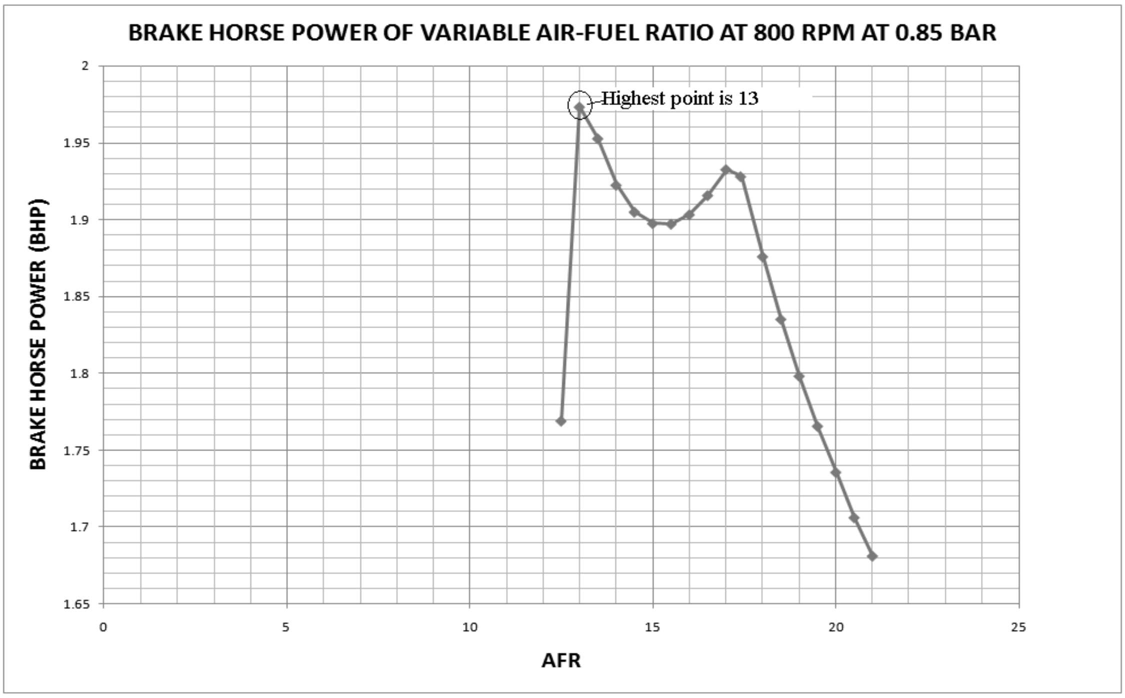


Figure 4.48: Overview of effect of variable AFR at 800 RPM at 0.85 Bar

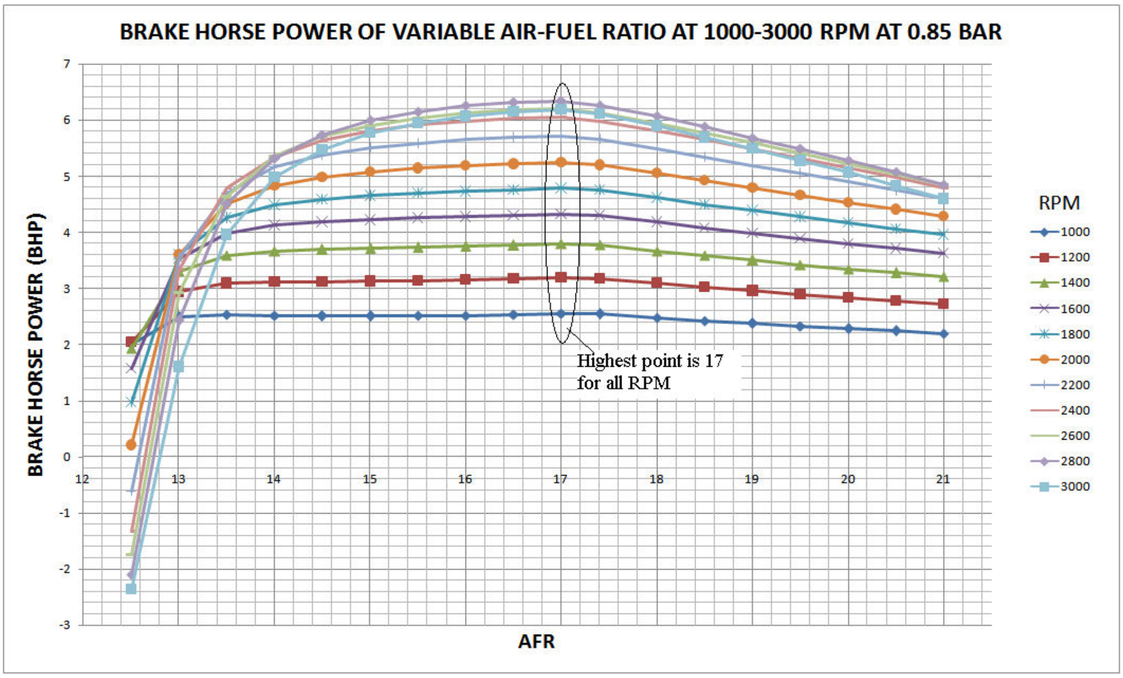
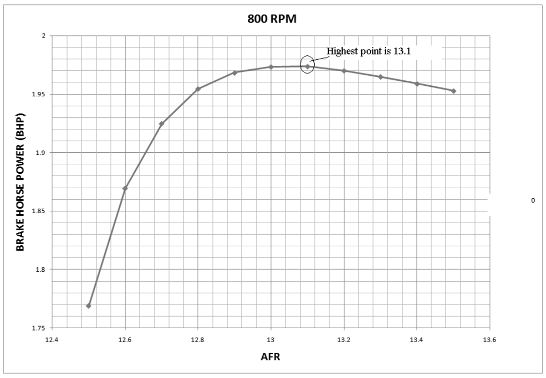
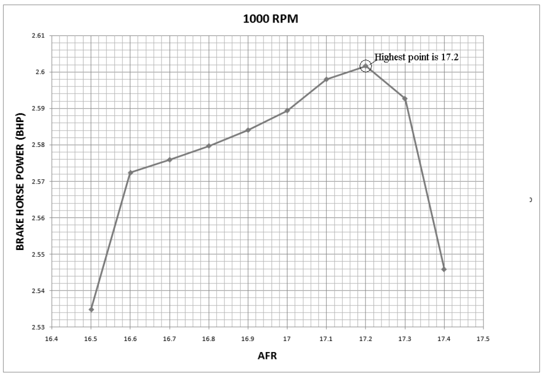


Figure 4.49: Overview of effect of variable AFR at 1000-3000 RPM at 0.85 Bar

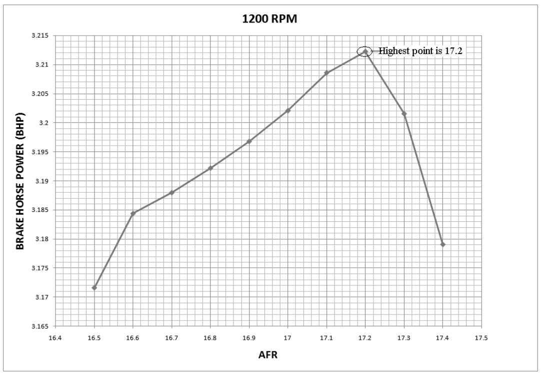
Figure 4.48 and Figure 4.49 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.48 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



(a) 800 RPM

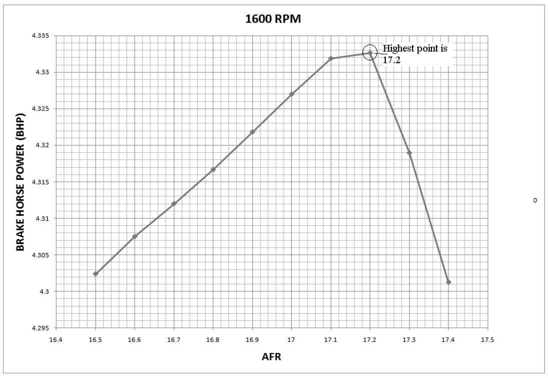


(b) 1000 RPM



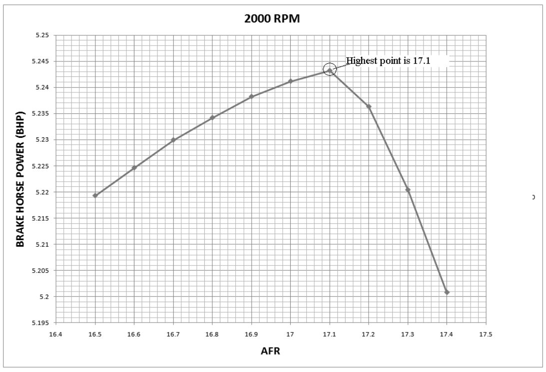
(c) 1200 RPM

(d) 1400 RPM

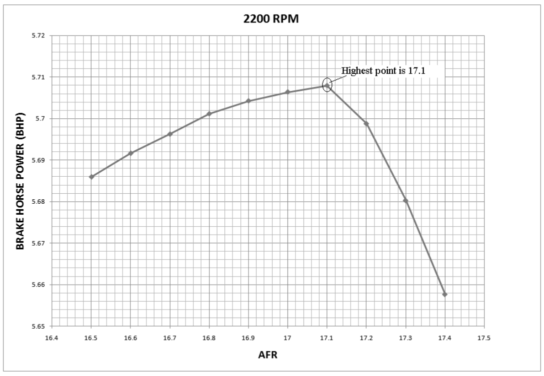


(e) 1600 RPM

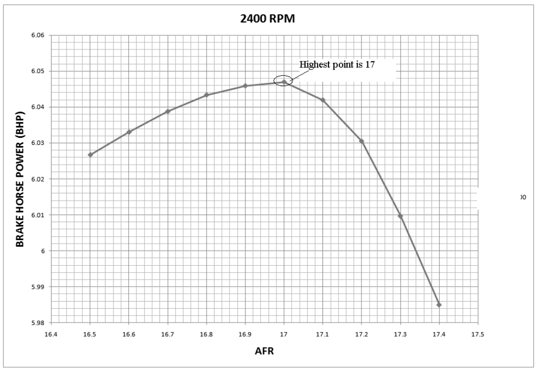
(f) 1800 RPM



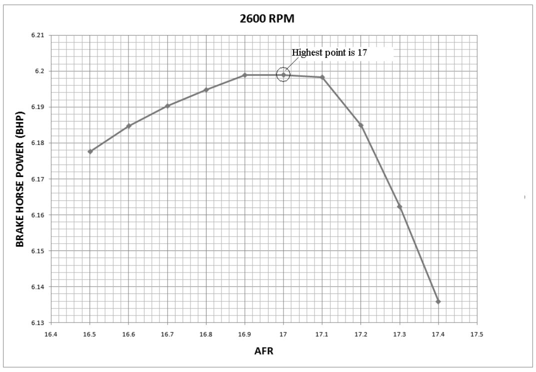
(g) 2000 RPM



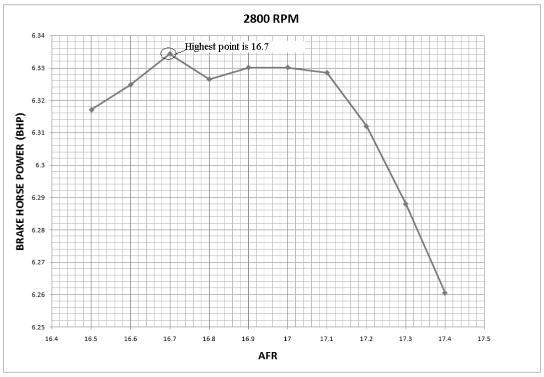
(h) 2200 RPM



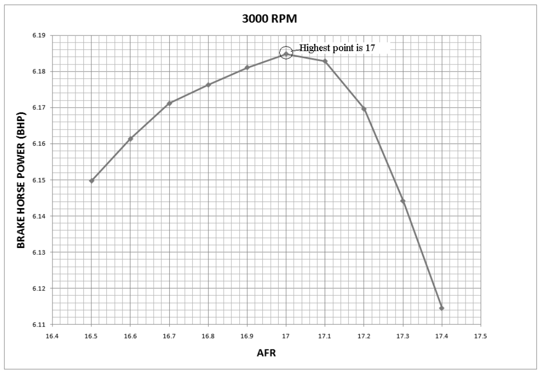
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.50: Refined view of effect of variable AFR at each RPM at 0.85 Bar

Figure 4.50 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.13 shows the best AFR for each RPM read from the refined view graphs.

Table 4.13: Best AFR for each RPM at manifold pressure of 0.85 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13.1	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17	17	16.7	17

4.2.14 Results for manifold pressure of 0.90 Bar

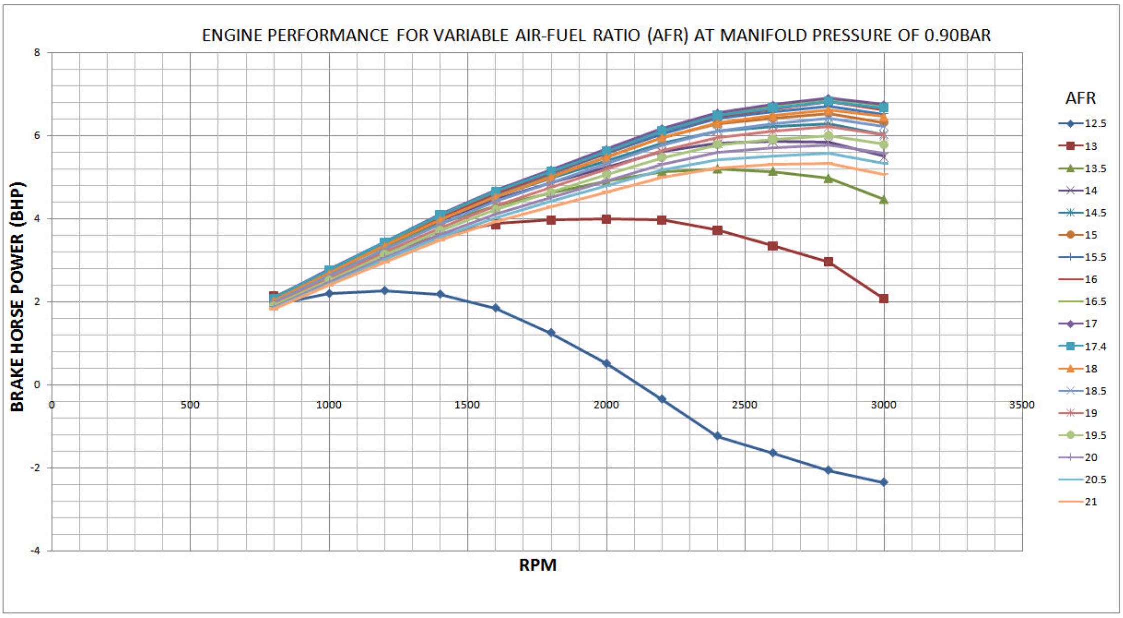


Figure 4.51: Engine performance for variable AFR at manifold pressure of 0.90 Bar

Figure 4.51 shows the engine performance at manifold pressure of 0.90 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.85 Bar. The only difference is the maximum power attained, which is about 7 BHP compared to about 6.4 BHP at manifold pressure of 0.85 Bar. As seen the power attained is increasing slowly with increasing manifold pressure. The engine is slowly reaching its limit of maximum power hence the increment is small.

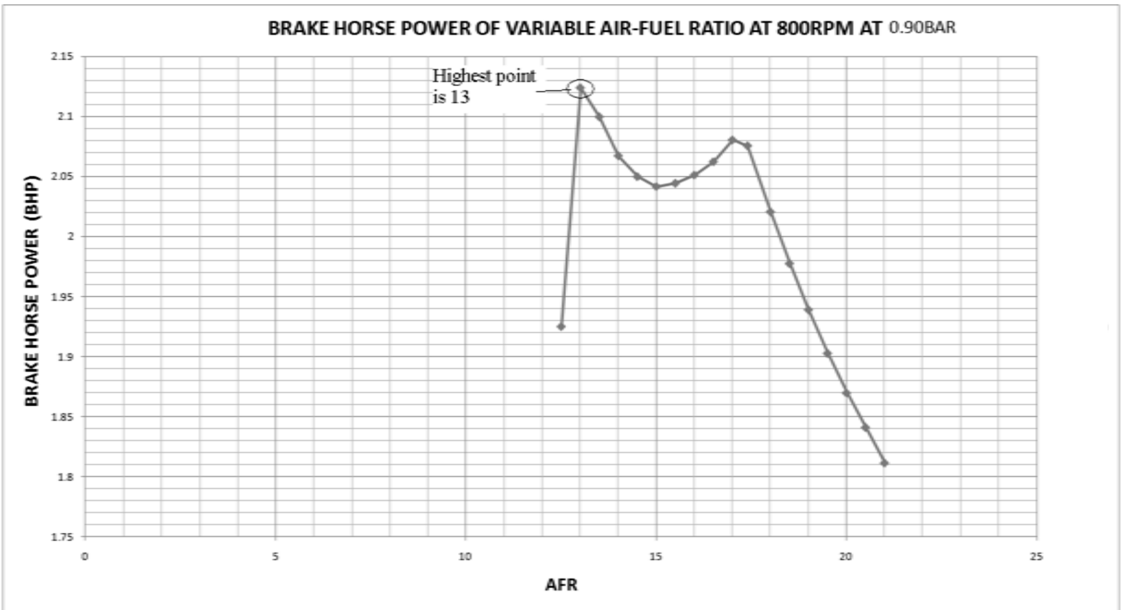


Figure 4.52: Overview of effect of variable AFR at 800 RPM at 0.90 Bar

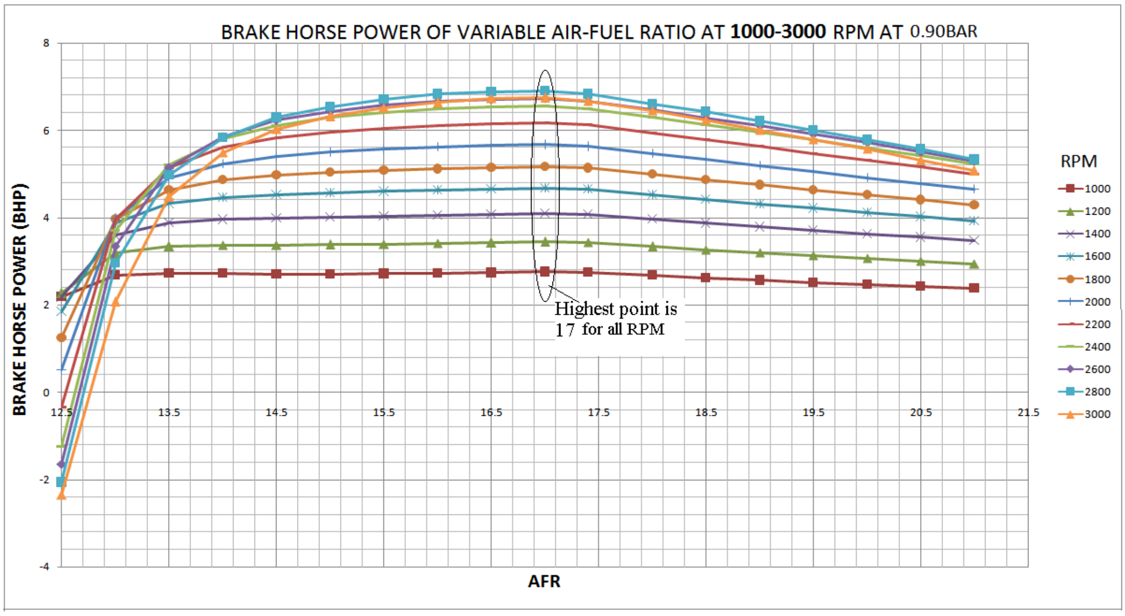
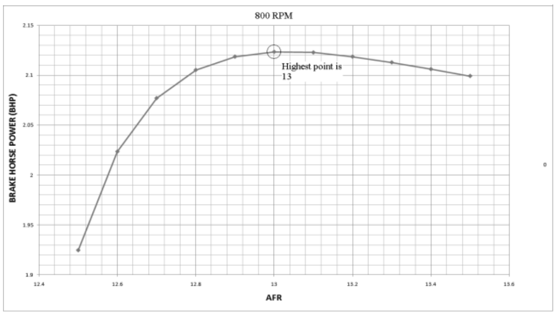
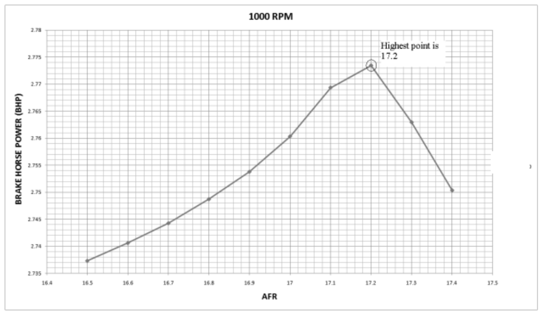


Figure 4.53: Overview of effect of variable AFR at 1000-3000 RPM at 0.90 Bar

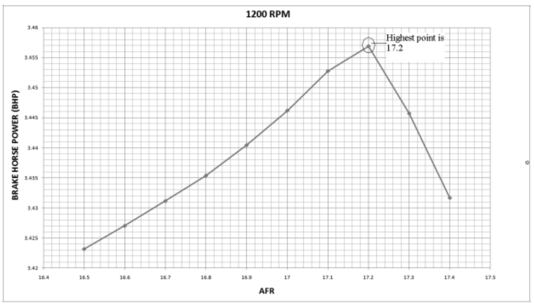
Figure 4.52 and Figure 4.53 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.52 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



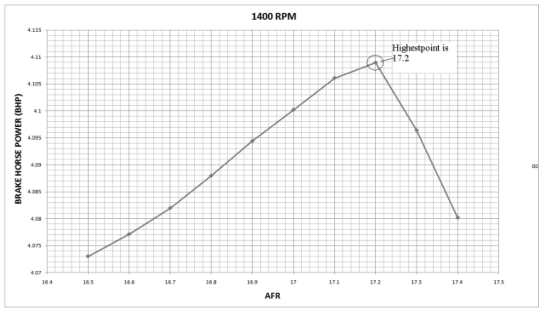
(a) 800 RPM



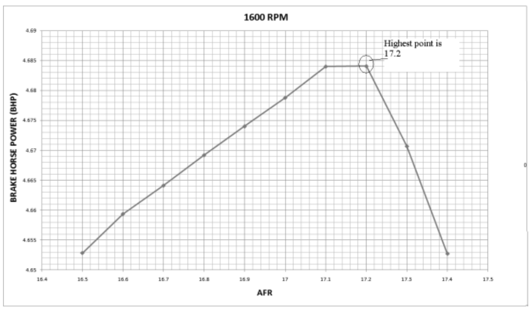
(b) 1000 RPM



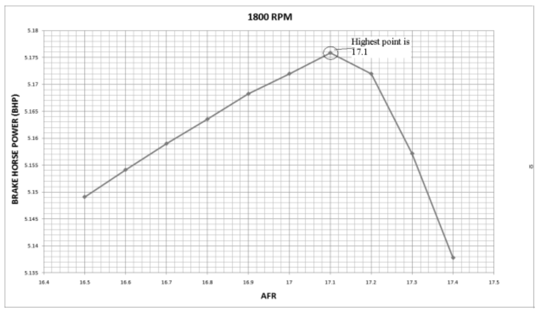
(c) 1200 RPM



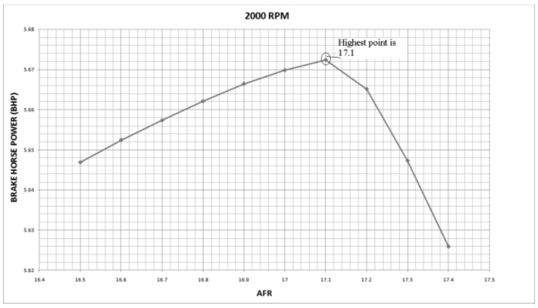
(d) 1400 RPM



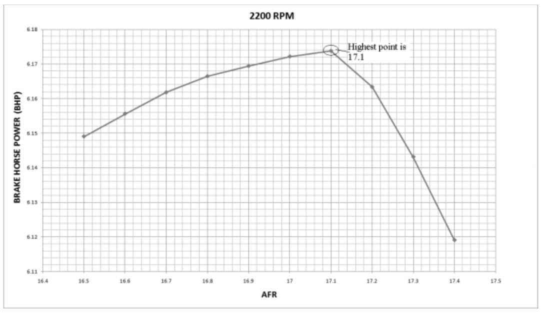
(e) 1600 RPM



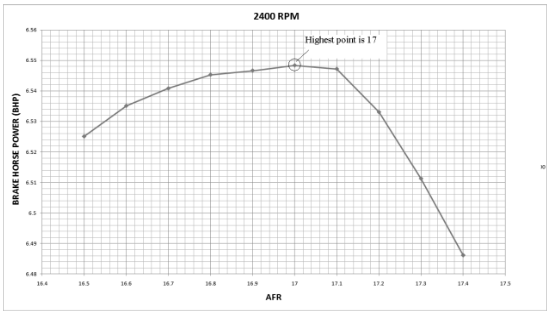
(f) 1800 RPM



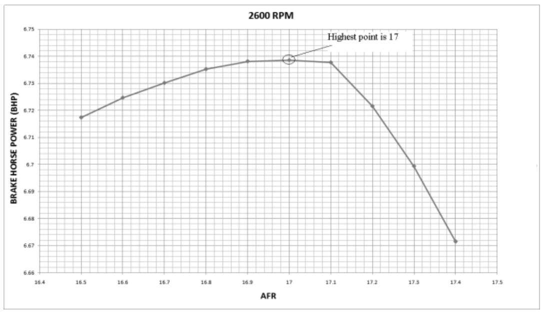
(g) 2000 RPM



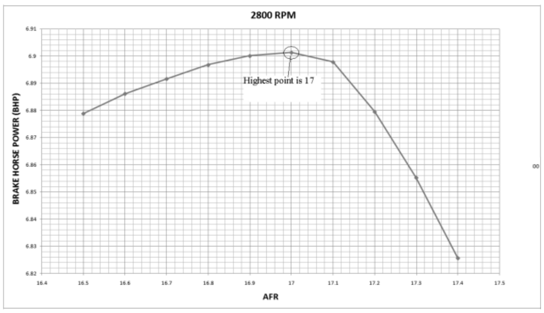
(h) 2200 RPM



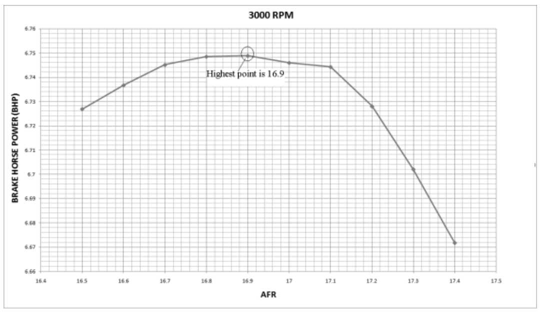
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.54: Refined view of effect of variable AFR at each RPM at 0.90 Bar

Figure 4.54 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.14 shows the best AFR for each RPM read from the refined view graphs.

Table 4.14: Best AFR for each RPM at manifold pressure of 0.90 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17	17	17	16.9

4.2.15 Results for manifold pressure of 0.95 Bar

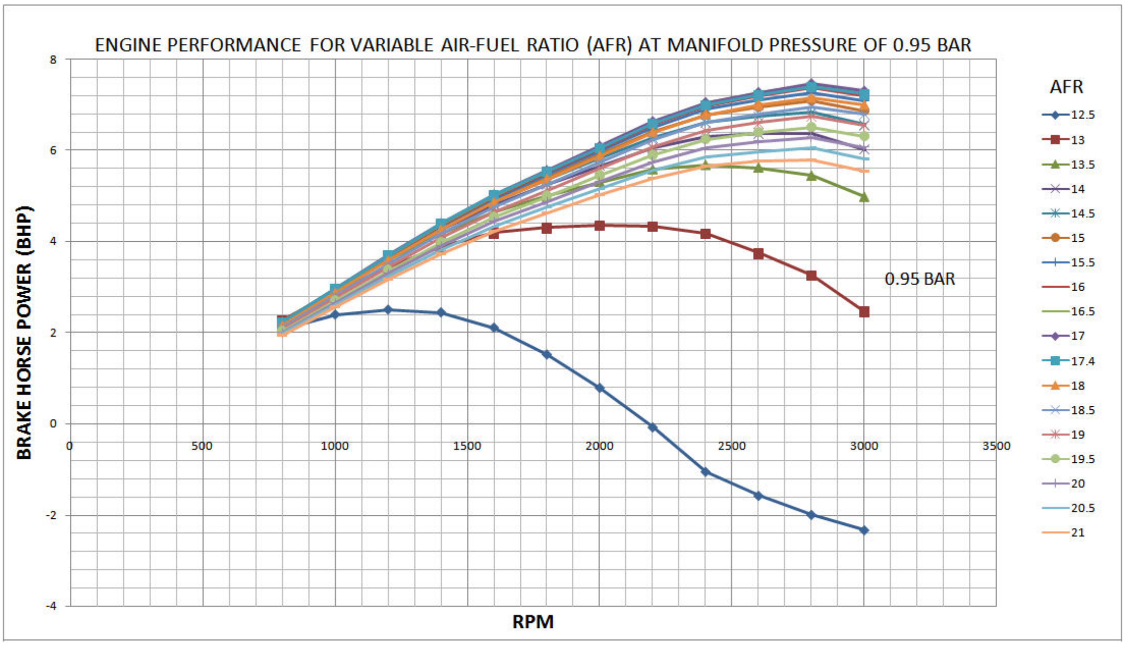


Figure 4.55: Engine performance for variable AFR at manifold pressure of 0.95 Bar

Figure 4.55 shows the engine performance at manifold pressure of 0.95 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.90 Bar. The only difference is the maximum power attained, which is about 7.6 BHP compared to about 7 BHP at manifold pressure of 0.90 Bar. As seen the power attained is increasing slowly with increasing manifold pressure. The engine is slowly reaching its limit of maximum power hence the increment is small.

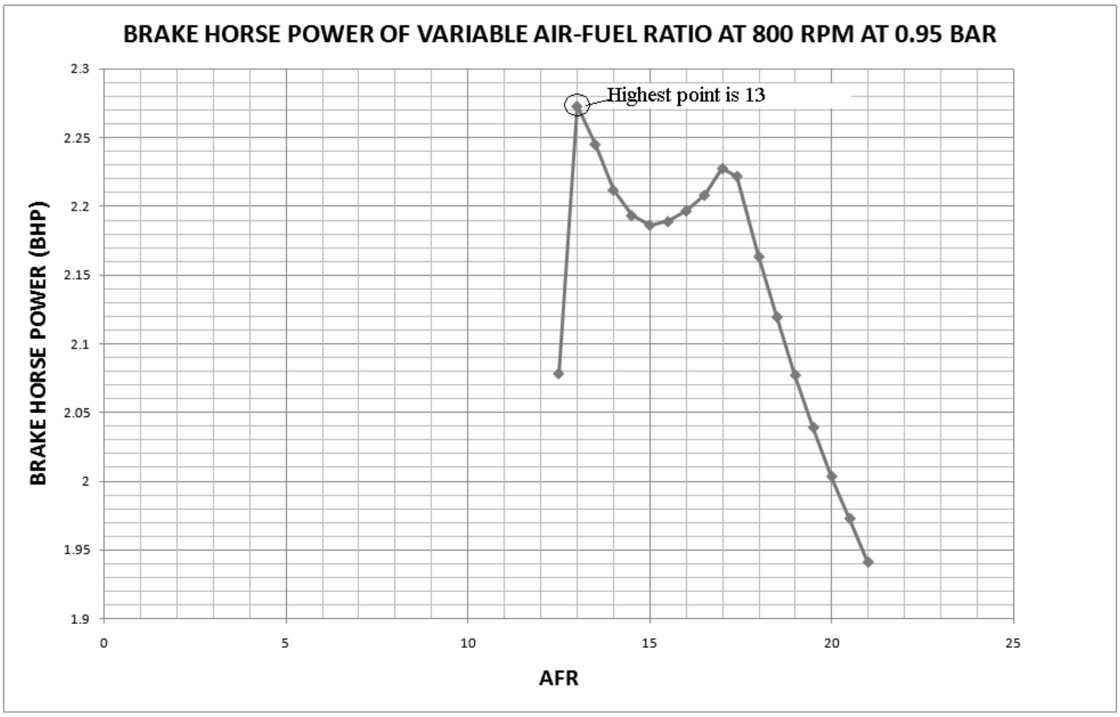


Figure 4.56: Overview of effect of variable AFR at 800 RPM at 0.95 Bar

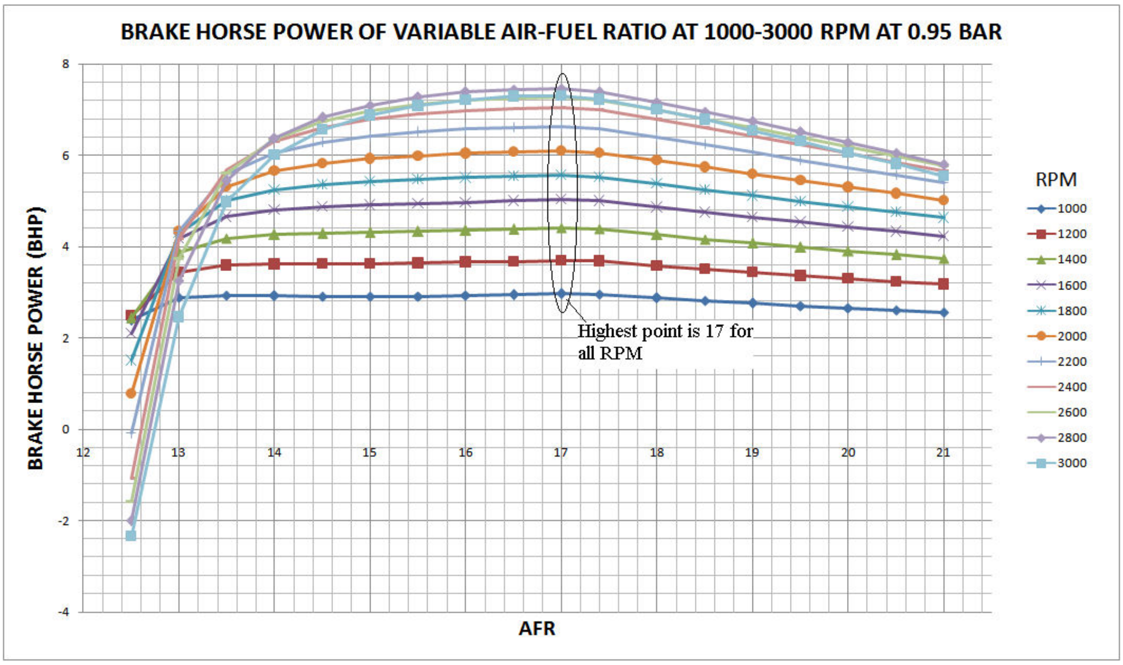
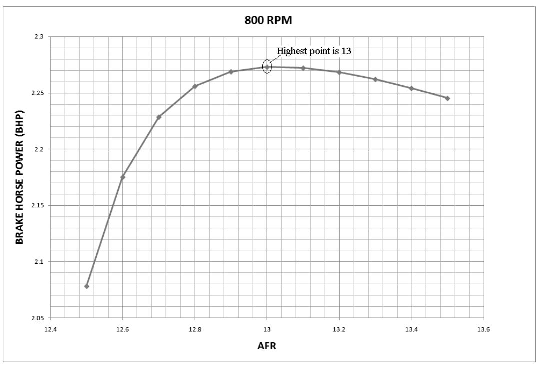
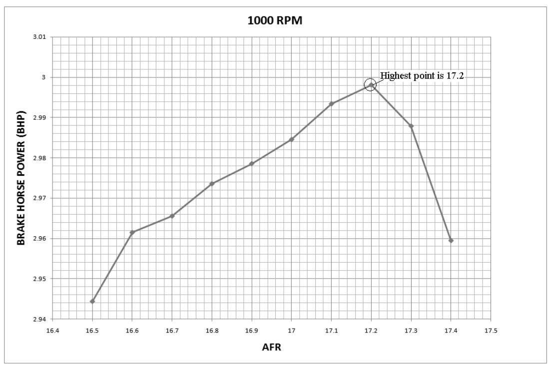


Figure 4.57: Overview of effect of variable AFR at 1000-3000 RPM at 0.95 Bar

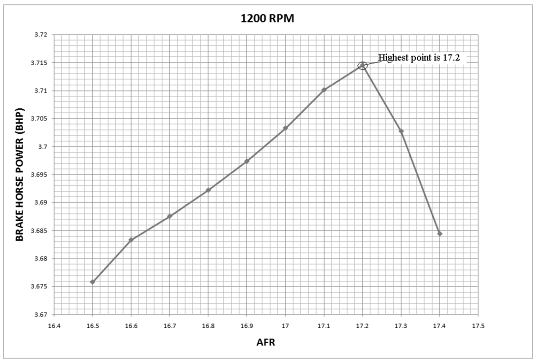
Figure 4.56 and Figure 4.57 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.56 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



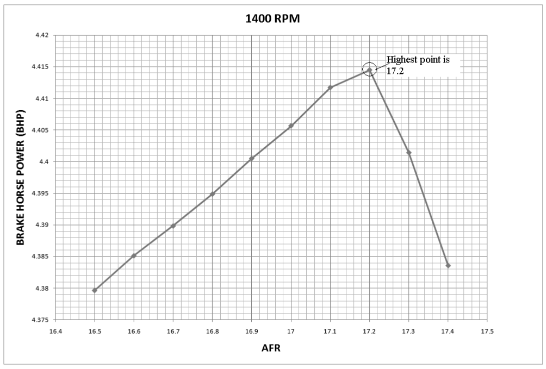
(a) 800 RPM



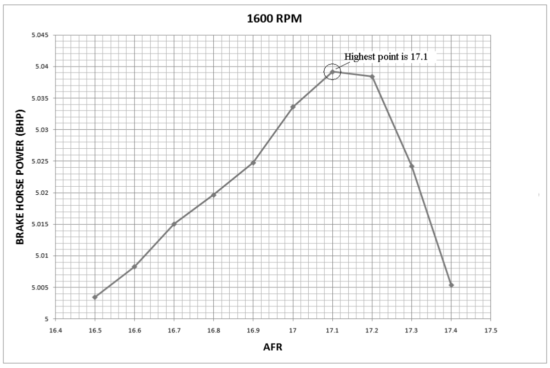
(b) 1000 RPM



(c) 1200 RPM

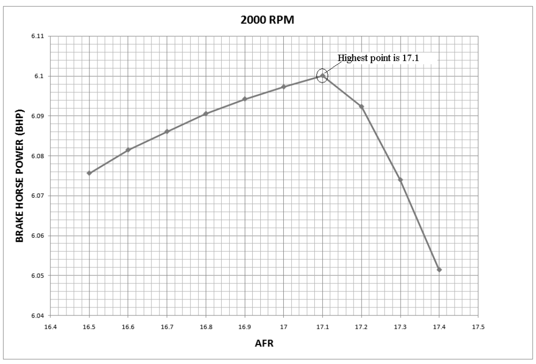


(d) 1400 RPM

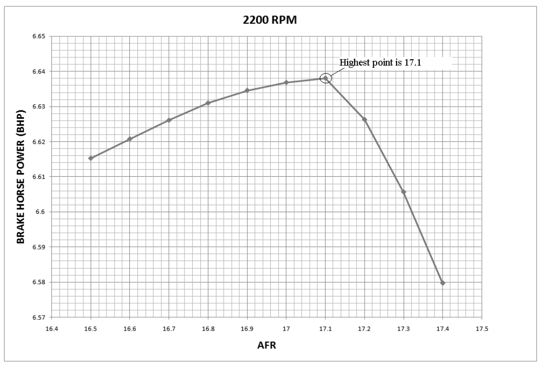


(e) 1600 RPM

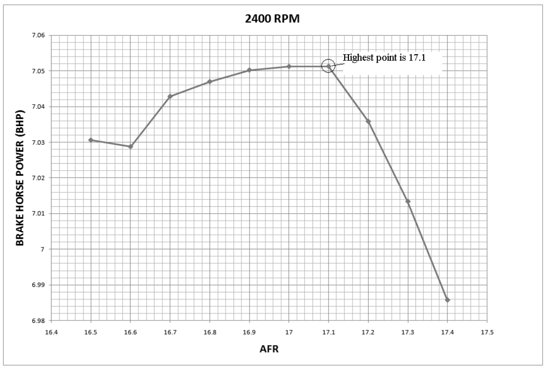
(f) 1800 RPM



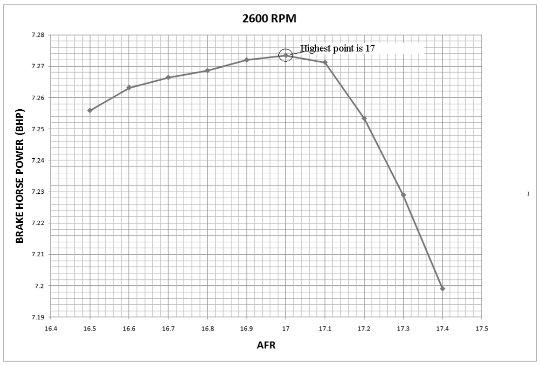
(g) 2000 RPM



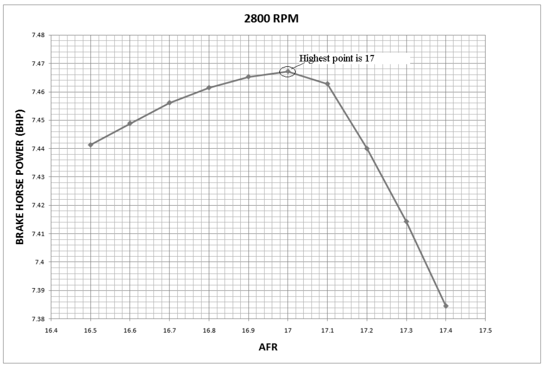
(h) 2200 RPM



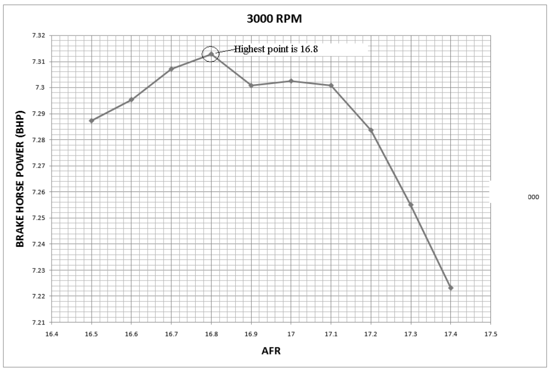
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.58: Refined view of effect of variable AFR at each RPM at 0.95 Bar

Figure 4.58 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.15 shows the best AFR for each RPM read from the refined view graphs.

Table 4.15: Best AFR for each RPM at manifold pressure of 0.95 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13	17.2	17.2	17.2	17.2	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17.1	17	17	16.8

4.2.16 Results for manifold pressure of 1.00 Bar

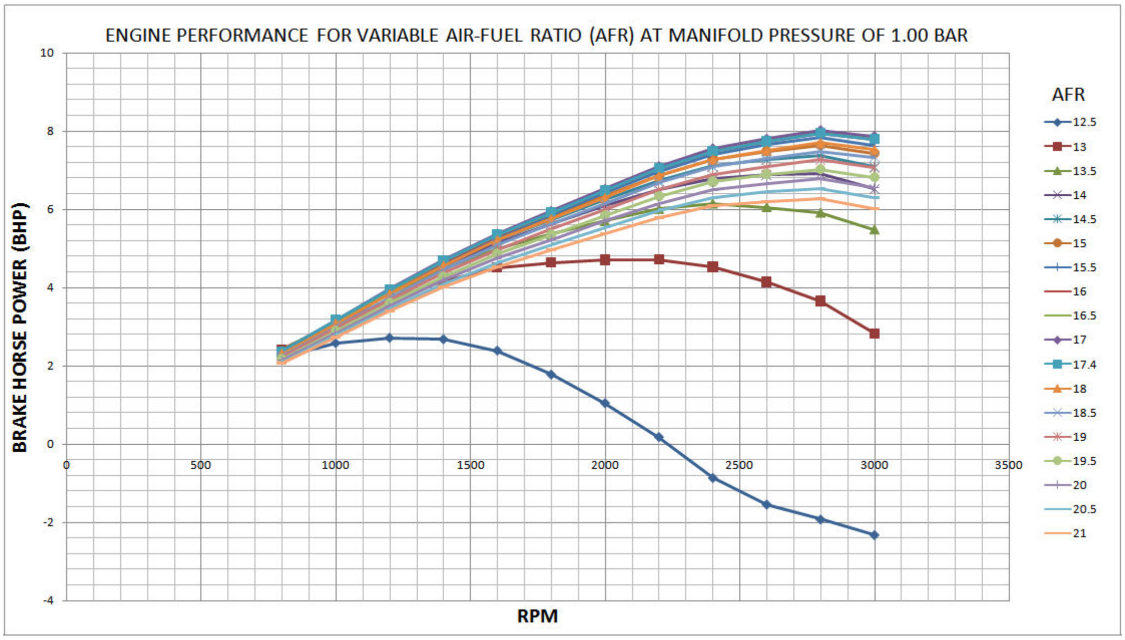


Figure 4.59: Engine performance for variable AFR at manifold pressure of 0.95 Bar

Figure 4.59 shows the engine performance at manifold pressure of 1.00 Bar. The peak power occurs in the range of 2500 RPM to 3000 RPM, mostly at 2800 RPM to be precise, for most AFR. The trend is the same as the performance at manifold pressure of 0.95 Bar. The only difference is the maximum power attained, which is about 8 BHP compared to about 7.6 BHP at manifold pressure of 0.95 Bar. As seen the power attained is increasing slowly with increasing manifold pressure. The engine is slowly reaching its limit of maximum power hence the increment is small.

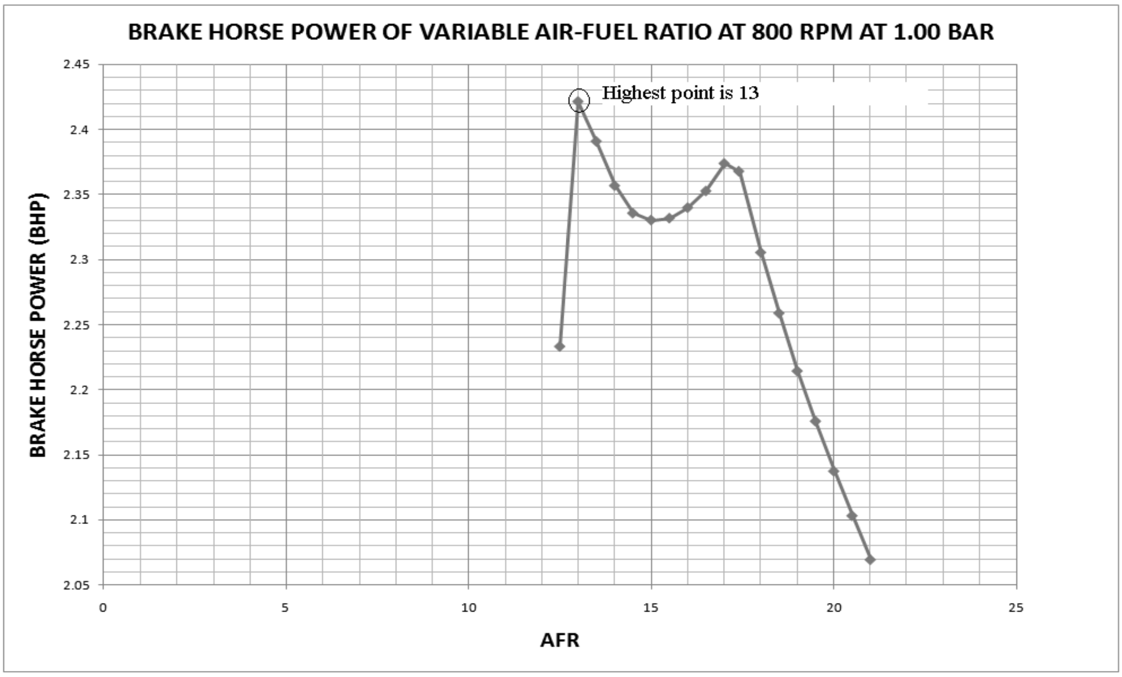


Figure 4.60: Overview of effect of variable AFR at 800 RPM at 1.00 Bar

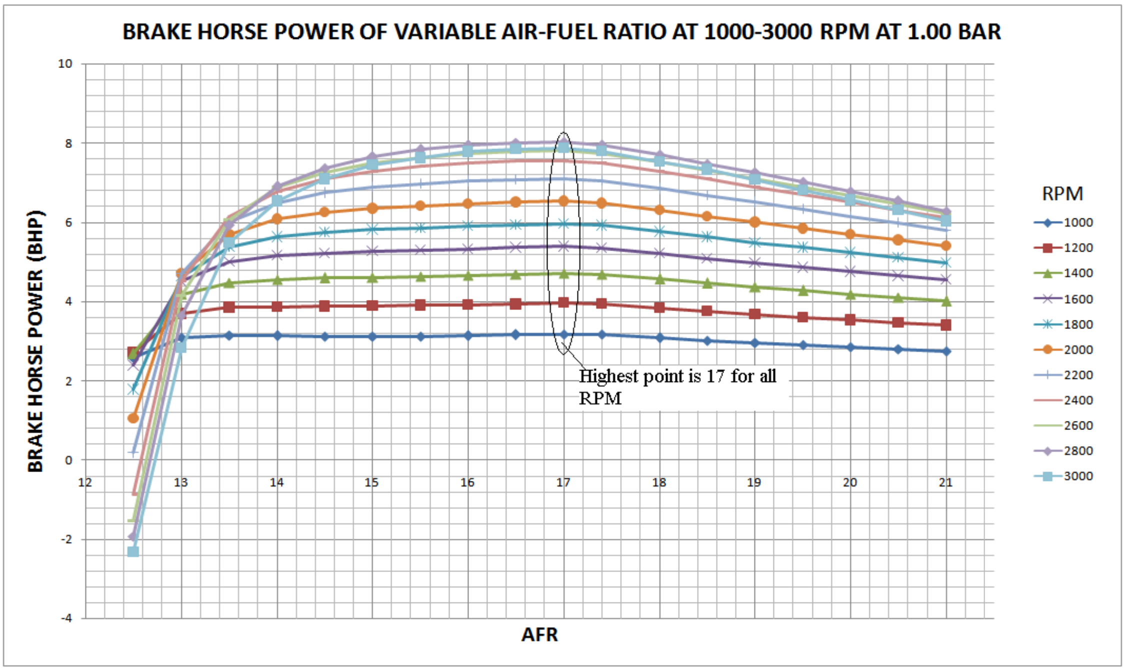
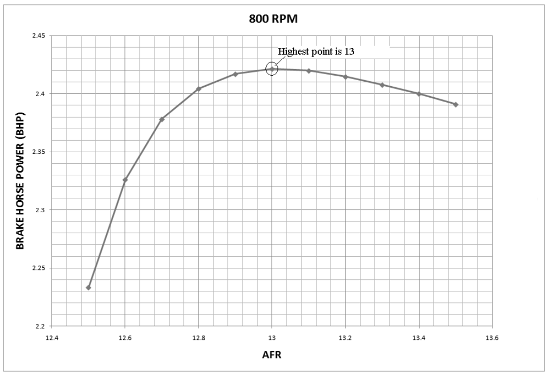
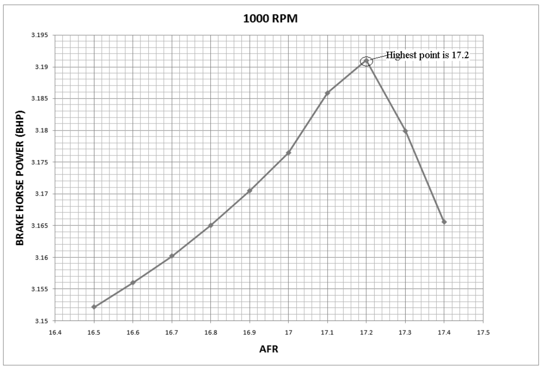


Figure 4.61: Overview of effect of variable AFR at 1000-3000 RPM at 1.00 Bar

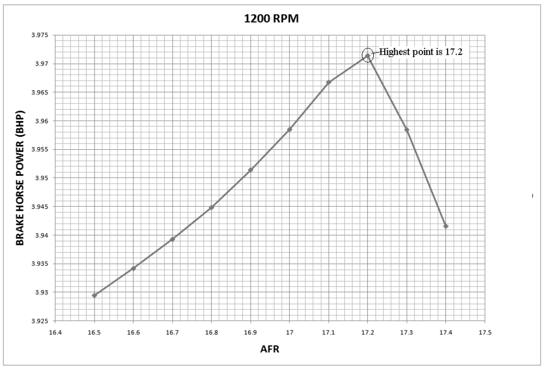
Figure 4.60 and Figure 4.61 shows the overview of effect variable AFR to horse power at 800 RPM and 1000-3000 RPM. As can be seen in Figure 4.60 the highest point for 800 RPM is 13 and as for 1000-3000 RPM the highest point is 17 for all. Since the highest point is 13 for 800 RPM, the refined view is ranged from 12.5 to 13.5 for 800 RPM. As for 1000-3000 RPM the range is still 16.5 to 17.4.



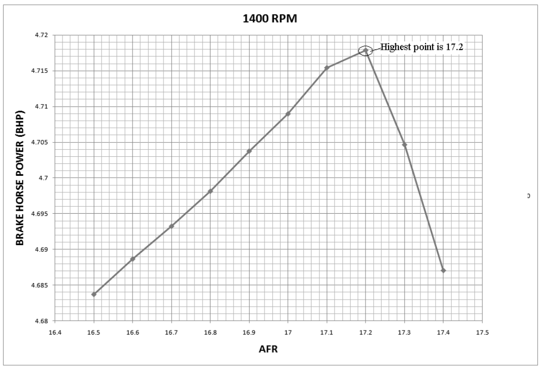
(a) 800 RPM



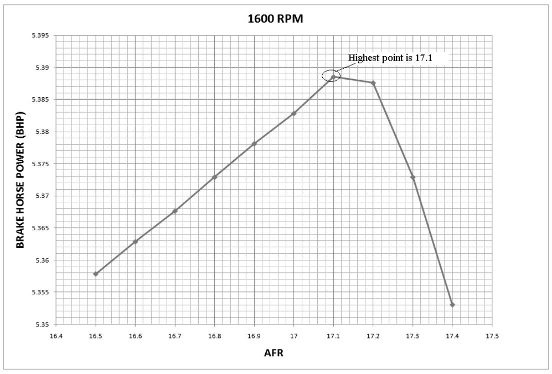
(b) 1000 RPM



(c) 1200 RPM

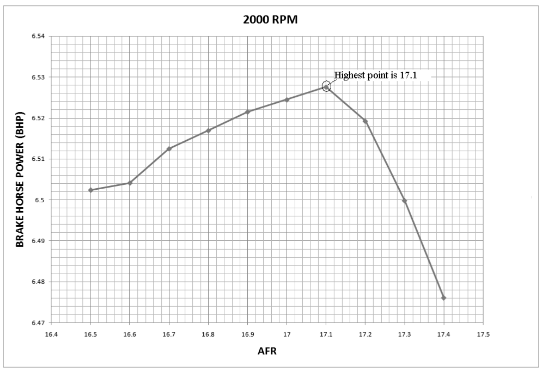


(d) 1400 RPM

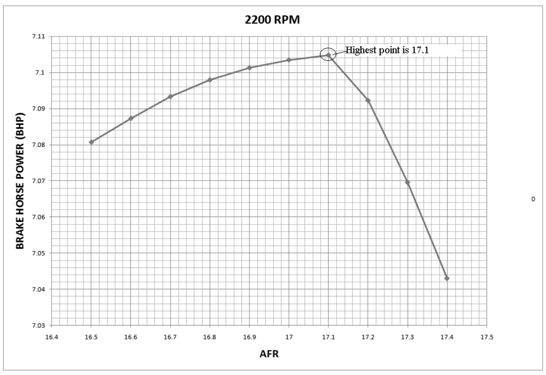


(e) 1600 RPM

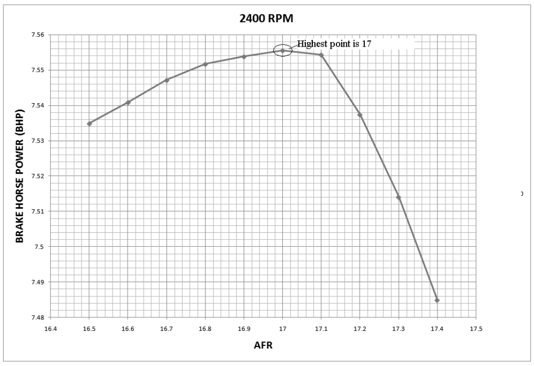
(f) 1800 RPM



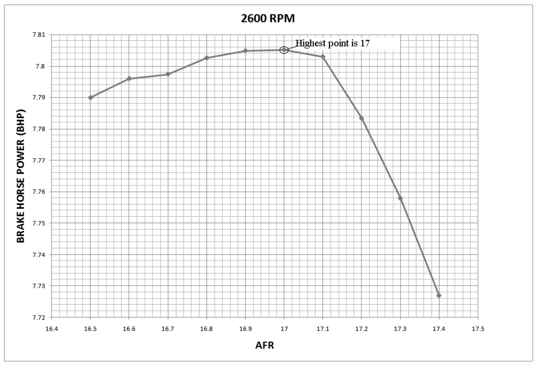
(g) 2000 RPM



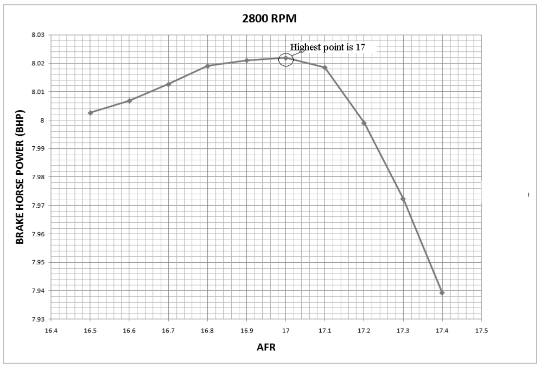
(h) 2200 RPM



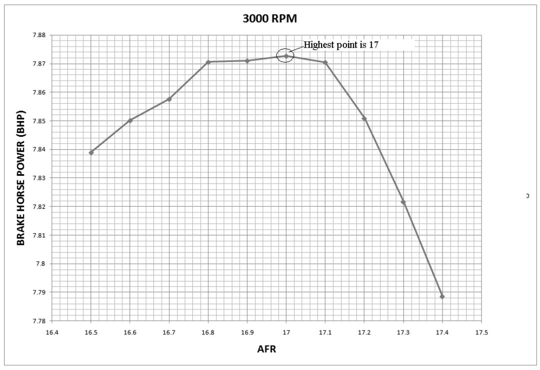
(i) 2400 RPM



(j) 2600 RPM



(k) 2800 RPM



(l) 3000 RPM

Figure 4.62: Refined view of effect of variable AFR at each RPM at 1.00 Bar

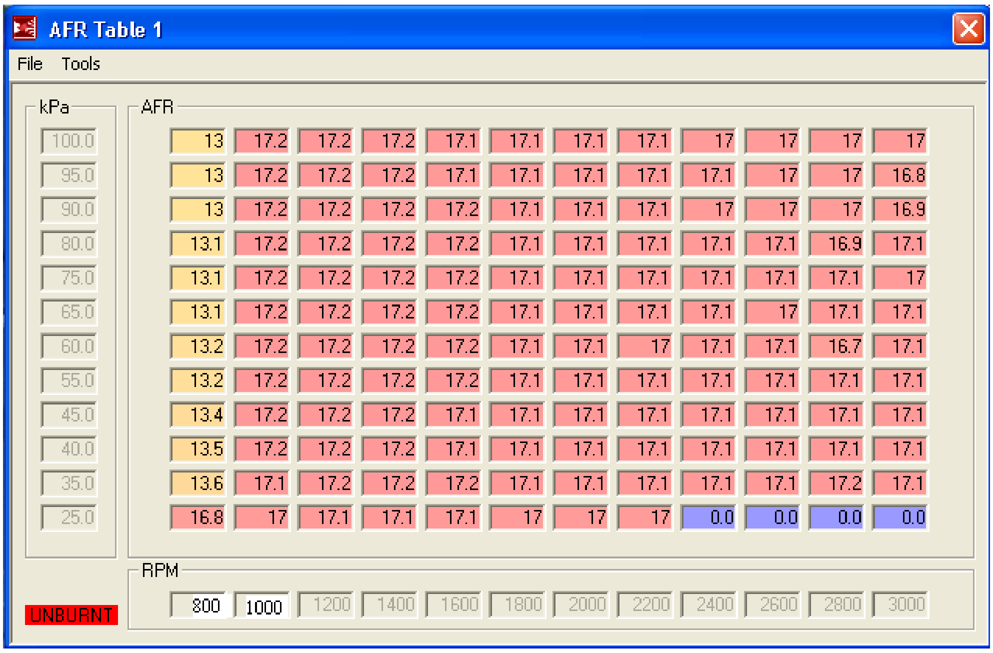
Figure 4.62 shows the refined view of effect of variable AFR to the brake horse power at each RPM. From (a) to (l) the refined view of the RPM range from 800-3000 RPM is observed. Table 4.16 shows the best AFR for each RPM read from the refined view graphs.

Table 4.16: Best AFR for each RPM at manifold pressure of 1.00 Bar

RPM	800	1000	1200	1400	1600	1800
AFR	13	17.2	17.2	17.2	17.1	17.1
RPM	2000	2200	2400	2600	2800	3000
AFR	17.1	17.1	17	17	17	17

4.3 OVERALL RESULTS

The AFR table of the Megatune software is inputted with the results obtained from the simulation.



kPa	800	1000	1200	1400	1600	1800	2000	2200	2400	2600	2800	3000
100.0	13	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17	17	17	17
95.0	13	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17	17	16.8
90.0	13	17.2	17.2	17.2	17.2	17.1	17.1	17.1	17	17	17	16.9
80.0	13.1	17.2	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	16.9	17.1
75.0	13.1	17.2	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17.1	17
65.0	13.1	17.2	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17	17.1	17.1
60.0	13.2	17.2	17.2	17.2	17.2	17.1	17.1	17	17.1	17.1	16.7	17.1
55.0	13.2	17.2	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17.1	17.1
45.0	13.4	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1
40.0	13.5	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1
35.0	13.6	17.1	17.2	17.2	17.2	17.1	17.1	17.1	17.1	17.1	17.2	17.1
25.0	16.8	17	17.1	17.1	17.1	17	17	17	0.0	0.0	0.0	0.0

Figure 4.63: AFR table with inputted data

Figure 4.63 shows the AFR table input with the best AFR for each RPM from Table 4.1 to Table 4.16 from each manifold pressure. The column on the left is the manifold pressure and the column below is the RPM.

4.4 DISCUSSION

As can be seen from Figure 4.63, at most of the RPM the engine burns lean which is in the range 17 to 17.2. As studied by Sanuri et al. (2008) using the same GT-Power simulation model, by converting the diesel engine to CNG engine, the intake pressure in intake port of the engine will increase. The increasing of gas flow pressure in intake manifold is caused by the development of gas fuel injector in the intake manifold of port injection CNG engine, where in the diesel engine is using direct injection. Higher pressure means lower flow rate of air. Hence lean fuel is injected so that the mixture won't be too rich. As mentioned by Heywood (1988) in his book, for mixture lean of stoichiometric, the theoretical fuel conversion efficiency increase as equivalence ratio, φ decrease below 1.0. Combustion of mixtures leaner than stoichiometric produces products at lower temperature and with less dissociation of the triatomic molecules CO_2 and H_2O. Thus the fraction of the chemical energy of the fuel which is released as sensible energy near Top Center (TC) is greater; hence a greater fraction of the fuel's energy is transferred as work to the piston during expansion, and the fraction of the fuel's available energy rejected to the exhaust system decreases. Also optimum efficiency from natural gas is obtained when burnt in a lean mixture in the range φ =1.3 to 1.5 (Semin and Abu Bakar, 2008). Hence even with lean mixture, high power is produced.

The maximum brake horse power (BHP) produced is 8.0219 BHP at 2800 RPM at manifold pressure of 1.00 Bar as can be seen in Figure 4.59. By referring to Table 3.1 from Chapter 3, the maximum output of the original diesel engine is 10 BHP; hence we can see a reduction of 20% of power when conversion is done. This is mainly due to premixed SI engines suffer 30% lower power output than equivalent size diesel engines due to knock limitations (Kato, 1999). In addition, SI engines suffer high pumping losses, due to the need to throttle the intake air at part load conditions. These factors result in a 15 to 30% reduction in volumetric efficiency as compared to diesel engines (Brombacher, 1997). Another reason is composition of hydrocarbon for each fuel. Diesel which has higher hydrocarbon compares to CNG will give more energy to the engine (Semin et al., Undated). In this simulation, the peak power is reached at 2800 RPM while for the original diesel engine the peak is reached at 3600 RPM as can be seen in Table 3.1 of Chapter 3. This is explained from the study case by Semin et al. (2009), which is using the same simulation model. This is due to CNG engine friction torque is higher than diesel engine. The friction torque of both engines will increase with increasing engine speed but the CNG engine give the higher friction torque. Meaning the conversion of diesel engine to CNG engine can cause the engine's friction torque to increase. This is mainly due to the properties of the natural gas, where natural gas as a gas fuel has less lubrication property compared to diesel fuel as a liquid fuel.

For the range of horse power obtained from the simulation, which is 0-8 BHP is reasonable. Comparing to the study done by Kaleemuddin and Rao (2009), the engine model they used is almost similar with the engine model used in this project simulation. The power they obtained at 2800 RPM is around 6 BHP which is lower than this project of 8 BHP but the compression ratio they used is 9:1 while this project uses 14.5:1. 14.5:1 is applicable due to one significant characteristic of natural gas which is the flammability limit. Natural gas has high flammability limit (Hassan et al., 2009). Also the octane rating of natural gas is about 130, meaning that engines could operate at compression ratio of up to 16:1 without "knock" or detonation (Semin and Abu Bakar, 2008). Thus it can withstand the higher temperature produced due to higher compression ratio without knocking occur. Also comparing to the specification of the original diesel engine from Table 3.1 from Chapter 3, the maximum output power is 10 BHP hence the power range obtained is reasonable. As can be seen from the results, varying the manifold pressure doesn't affect the air-fuel ratio used to obtain best performance. The only effect of varying the manifold pressure is the output power increment of the engine.

4.5 SUMMARY

Graphs obtained from GT-Power simulation are shown. Readings are taken from these graphs to determine the air-fuel-ratio (AFR) that produces highest brake horse power for a fixed RPM at different intake manifold pressure. Discussion is done on the results obtained.

CHAPTER 5

CONCLUSION

5.1 INTRODUCTION

Chapter 5 summarizes all the main research points and analysis of this project. It concludes that all the important information and observation resulting from the project for the future research.

5.2 CONCLUSION

The study of best air-fuel ratio at a fixed RPM at variable manifold pressure is investigated by performing the simulation of the experiment using GT-Power software. It is found that the best air-fuel ratio is in the lean range. Engine performance curves were plotted in Microsoft Excel and the result shows that when the manifold pressure increase, the horse power also increases. The best AFR, which is determined from the results by comparison of which produces the highest horse power at each RPM, is inputted into the AFR table of the Megatune software. The data is then to be burn into the Engine Control Unit (ECU) so that the ECU can determine which AFR is the best at each RPM during different manifold pressure to improve efficiency of the engine and also to reduce emission.

5.3 RECOMMENDATION

For further improvement the Spark Timing table and Volumetric Efficiency table for the ECU can be tuned since this project only focuses on the AFR table of the ECU. As mentioned by some journal the compressed natural gas has long ignition delay and low volumetric efficiency due to its property as a gas. Experiment for spark timing and volumetric efficiency can be done to find the optimum spark timing, which is advance or retard and the volumetric efficiency for each parameter condition as studied above.

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