Performance Analysis of Turbocharger in Diesel Engine

by

Mohamad Muaamar Gadafi b. Abd Malik

Dissertation submitted in partial fulfillment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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TRONOH, PERAK

January 2009

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the referenced and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

(MOHAMAD MUAAMAR GADAFI B. ABD. MALIK)

ABSTRACT

This report explains the project entitled 'Performance Analysis of Turbocharger in Diesel Engine' where the performance of the turbocharger that operated on diesel engine is studied, Each possible aspect of the engine performance will be investigated thoroughly and the feature that enhanced the operating system of the engine with turbocharger will be determined.

The goal is to find out the best pressure ratio and the best efficiency for the selected turbocharger with the selected diesel engine. The pressure ratio and the efficiency will be determined using calculation based on Turbocharger Equation Calculation Model. All calculation results will be characterized in the turbocharger performance graph evaluation for the selected turbocharger in term of best pressure ratio and efficiency.

During the study, it was discovered that there best efficiency achieved at the pressure ration of 1.7. Prior to that, during the data gathering process, because of the cost factor, the real time test with the engine was not conducted. Recommendations were given to close the gaps that were found in this study.

ACKNOWLEDGEMENT

I would like to praise Allah the Almighty, who gives me strengths to complete my Final Year Project, entitled – Performance Analysis of Turbocharger in Diesel Engine. My deepest gratitude goes to Ir Hj Idris Ibrahim as kindly being my great supervisor for his guiding, sharing and teaching me to undertake this project.

I would like to express my warmest and greatest gratitude, appreciation to all parties who have contributed towards the success of this project. I was indebted to many individuals who contributed to the development of my project. Special thanks to all the staff at Mechanical Automotive Department for helping me throughout the project and my friend Ahmad Nadiy b. Ghazali who help provide the information related to compressor working principle.

Next, to all lecturers and staffs of Universiti Teknologi PETRONAS, especially Mechanical Engineering Department where I was trained with essential skills and knowledge in order to be a well rounded student.

Last but not least, to my beloved parents, fellow friends and community of University Teknologi PETRONAS who directly or indirectly involves in completing the project.

Thank You.

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CHAPTER 1

INTRODUCTION

1.1 Background

Force induction describes the mean to force air into the engine by either a turbocharger system or a supercharger system. For the current project, turbocharger is selected because materials required are readily available off the shelf, hence less cost in materials procurement. Garret turbine, a turbine manufacturer was chosen to be used for this project since most of the turbine properties and dimensions are available for the analysis.

The purpose of the force induction system is to raise the density of the charge, be it air or air and fuel mixture, before it is delivered to the cylinders. Thus, the increase mass of charge trapped and then compressed in each cylinder during each induction and compression stroke make more oxygen available for combustion than the conventional method of drawing the fresh charge into the cylinder. Consequently, more air and fuel mixture per cycle will be cramped into the cylinder, and this can be efficiently burnt during the combustion process to raise the engine power output to higher than would otherwise be possible.

1.2 Problem Statement

The engine power output is limited by the amount of air in the cylinder. Therefore, by increasing the mass of air in each cylinder would give higher engine output. With turbocharged engine, the torque should be higher or at least equal to the naturally aspirated engine. The challenge is to determine the turbocharger suited for the engine selected. The turbocharger user has to decide the best pressure ratio for the selected engine is to maximize its usage and operation of the engine. The bad selection of pressure ratio will also affect the compressor performance; chocked air flow enters the compressor where it will reduce the efficiency of the compressor.

This project attempts to investigate the best pressure ratio and improved the efficiency of the turbocharger for the selected engine by using calculation and simulation method.

1.3 Objectives

The objectives of this project are:

1. Evaluate the performance of the turbocharger with respect to the pressure ratio and efficiency.
2. Develop the Turbocharger Equation Calculation Model database using Excel software on the selected turbocharger (Garret GT14 maps).
3. Build up the simulation on the turbocharger to show the best efficiency of the compressor.

1.4 Scope of Study

The scopes of studies involved would to determine the best pressure ratio and the efficiency of the turbocharger. All equation is to be calculated using relevant engineering principles. Formulate relevant calculation and the result will be present in the database using Microsoft Excel spreadsheet.

CHAPTER 2

LITERATURE REVIEW

2.1 Diesel Engine

A diesel engine is an internal combustion engine which operates based on the Diesel cycle shown in **Figure 2.1**; it was based on the hot bulb engine design [11]. Diesel engines use compression ignition, a process by which fuel is injected after the air is compressed in the combustion chamber causing the fuel to self ignites as shown in **Figure 2.2**. Most diesel engines have large pistons, therefore drawing more air and fuel which results in a bigger and more powerful combustion. This is effective in large vehicles such as trucks, diesel locomotives and SUV's. To increase the power produced by diesel engine, turbocharger was mounted to the diesel engine to increase the air capacity into the combustion chamber.

In the diesel engine, air is compressed adiabatically with a compression ratio typically between 15 and 20 [9]. This compression raises the temperature to the ignition temperature of the fuel mixture. Fuel mixture is formed by injecting fuel once the air is compressed.

The ideal air-standard cycle is modelled as a reversible adiabatic compression followed by a constant pressure combustion process, an adiabatic expansion as a power stroke and a volumetric exhaust [11].

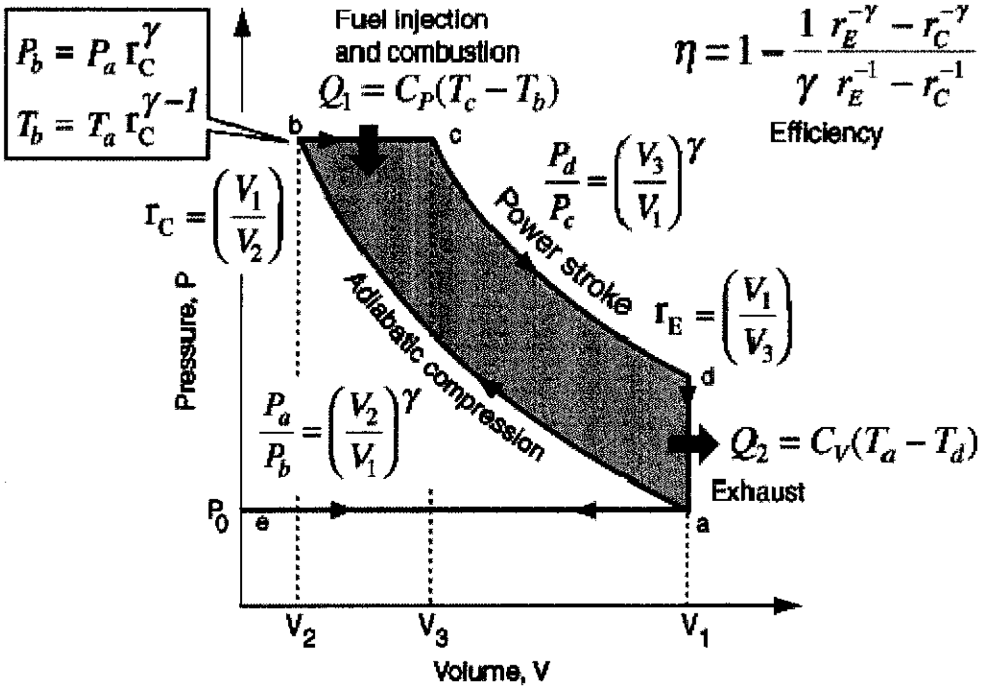

Figure 2.1: Standard Diesel Engine Cycle [11]

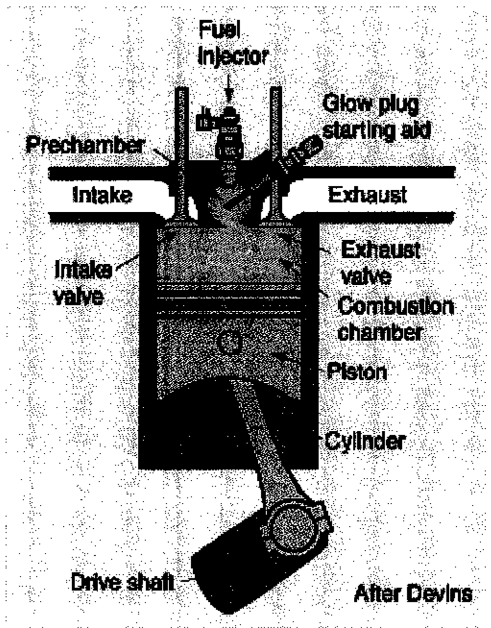

Figure 2.2: Diesel Engine Schematic Diagram [11]

2.2 Turbocharger

In an internal combustion engine, a turbocharger is a forced-induction compressor powered by the engine's exhaust gas. The purpose of a turbocharger is to increase the mass of air entering the engine to create more power. A turbocharger consists of a turbine and a compressor linked by a shared axle shown in **Figure 2.3**. The turbine inlet receives exhaust gases from the engine causing the turbine wheel to rotate. This rotation drives the compressor, compressing ambient air and delivering it to the air intake manifold of the engine at higher pressure, resulting in a greater amount of the air entering the cylinder. In some instances, compressed air is routed through an intercooler before introducing it to the intake manifold.

A typical diesel engine may harness up to 30% of the energy contained in the fuel supplied to do useful work under optimum conditions but the remaining 70% of this energy is lost in the following way [6]:

- 7% heat energy to friction, pumping and dynamic movement
- 9% heat energy to surrounding air
- 16% heat energy to engine's coolant system
- 38% heat energy to outgoing exhaust gases

A turbocharger utilizes a portion of the energy contained in the exhaust gas to drive a turbine wheel which simultaneously turns a centrifugal compressor wheel. The compressor will produce high pressure (boost) air that enters the cylinders as shown in **Figure 2.3**.

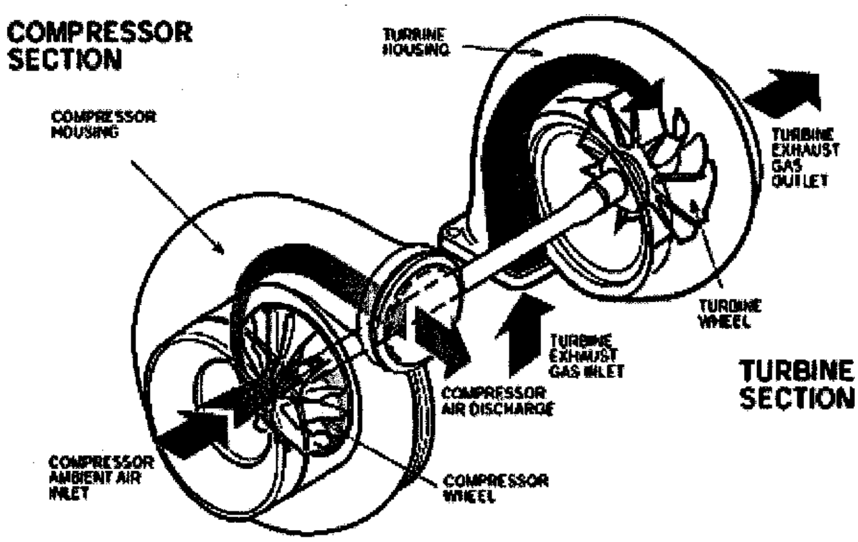

Figure 2.3: Turbocharger Operation Diagram [9]

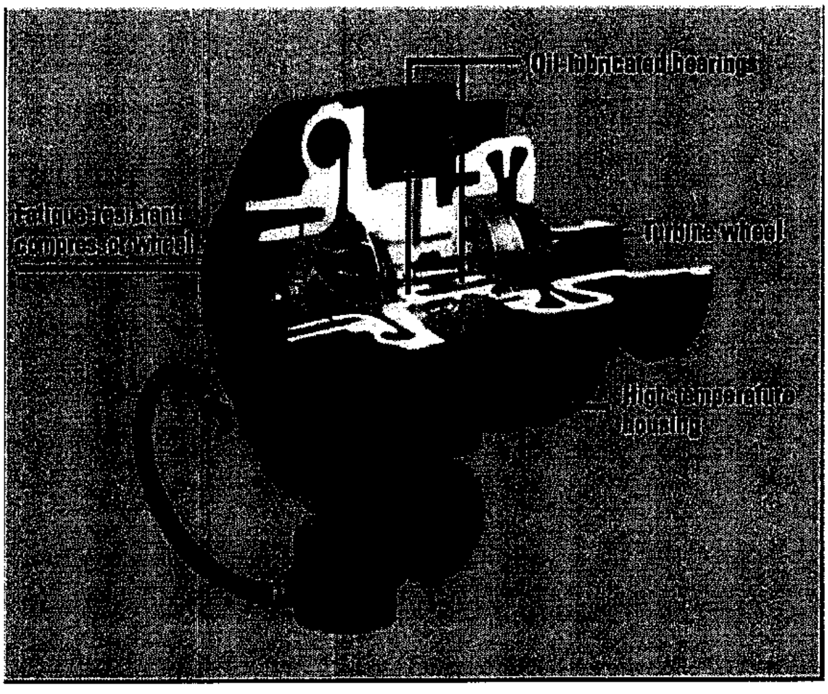

Figure 2.4: Turbocharger Basic [5]

The objective of a turbocharger is to improve the size-to-output efficiency of an engine by solving one of its cardinal limitations. A naturally aspirated automobile engine uses only the downward stroke of a piston to create an area of low pressure in order to draw air into the cylinder through the intake valves. Because the pressure in the atmosphere is no more than approximately 14.7 PSI, there ultimately will be a limit to the pressure difference across the intake valves and thus the amount of airflow entering the combustion chamber. The ability to fill the cylinder with air is its volumetric efficiency. Because the turbocharger increases the pressure at the point where air is entering the cylinder, a greater mass of air will be forced in as the inlet manifold pressure increases. The additional air makes it possible to add more fuel, and hence increasing the power and torque output of the engine.

Because the pressure in the cylinder must not go too high to avoid detonation and physical damage, the intake pressure must be controlled by controlling the rotational speed of the turbocharger. The control function is performed by a wastewater, which routes some of the exhaust flow away from the exhaust turbine. This controls shaft speed and regulates air pressure in the intake manifold.

Graph below shows the engine's performance with a restrictor in the intake manifold from a dynamo testing by two universities. The engine that University of Queensland used was without turbocharging as shown in **Figure 2.5** while Cornell University turbocharged their engine using Garrett GT12 turbine as shown in **Figure 2.6**.

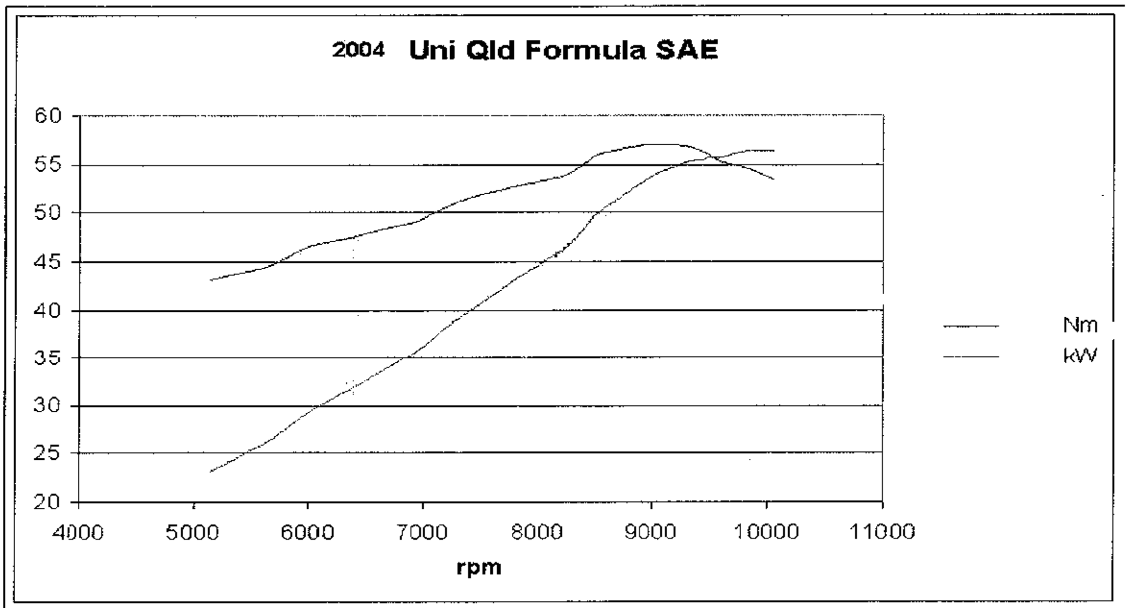

Figure 2.5: UQ Naturally Aspirated Engine Performance [3]

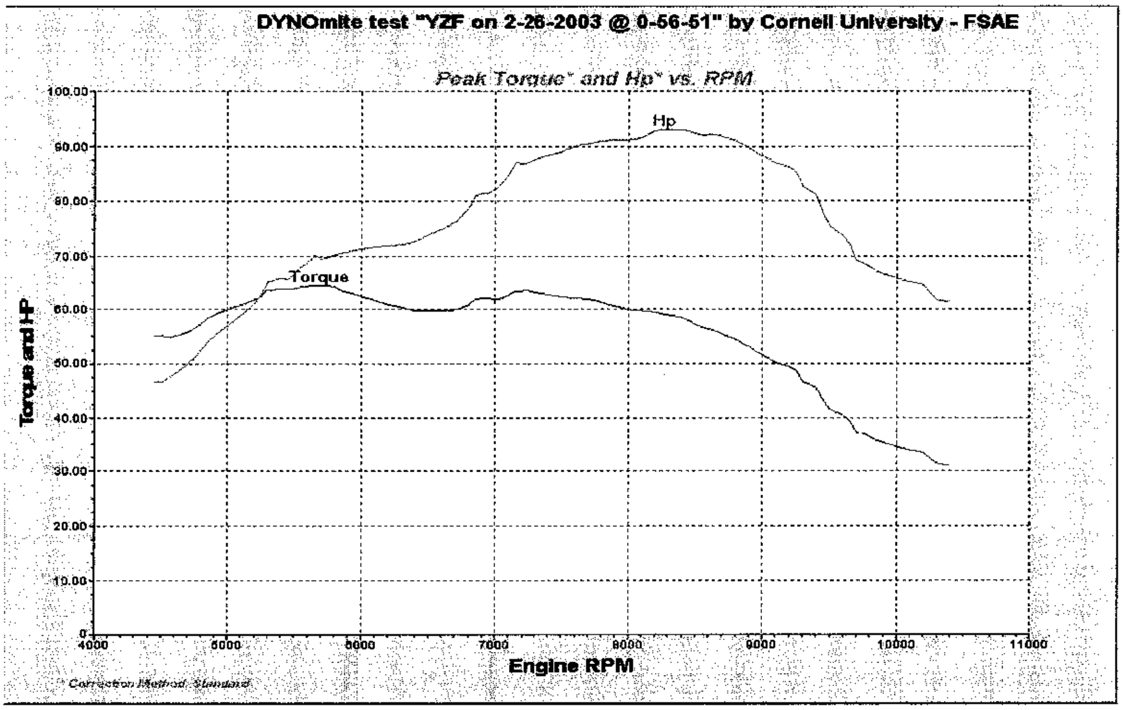

Figure 2.6 Cornell University 2003 Engine Output with a Turbocharger [3]

Comparing the natural aspirated engine graph with the turbocharger engine graph will yield the difference of performance shown in **Table 2.1**.

Table 2.1 Comparison of Naturally Aspirated and Turbocharged engine [3]

	UQ Naturally Aspirated Engine	Cornell Turbocharged Engine
Peak Torque	57 Nm @ 9000rpm	87 Nm @ 6000rpm
Peak Power	56.5 kW @ 11000rpm	69.35 kW @ 10000rpm

It can be seen from **Table 2.1** that the overall engine power increases by 22.7% by turbo charging the engine. This is because the compressor has increased the mass flow rate of air entering the cylinder thus increasing the peak cylinder pressure during the combustion. More air will mix with more fuel and will create more power for the engine. A turbocharger comprises of three major components, an exhaust-gas driven turbine and housing, a centrifugal compressor wheel and housing and an interconnecting support spindle mounted on a pair of fully floating plain bearings housing made from nickel cast iron as show in **Figure 2.7**.

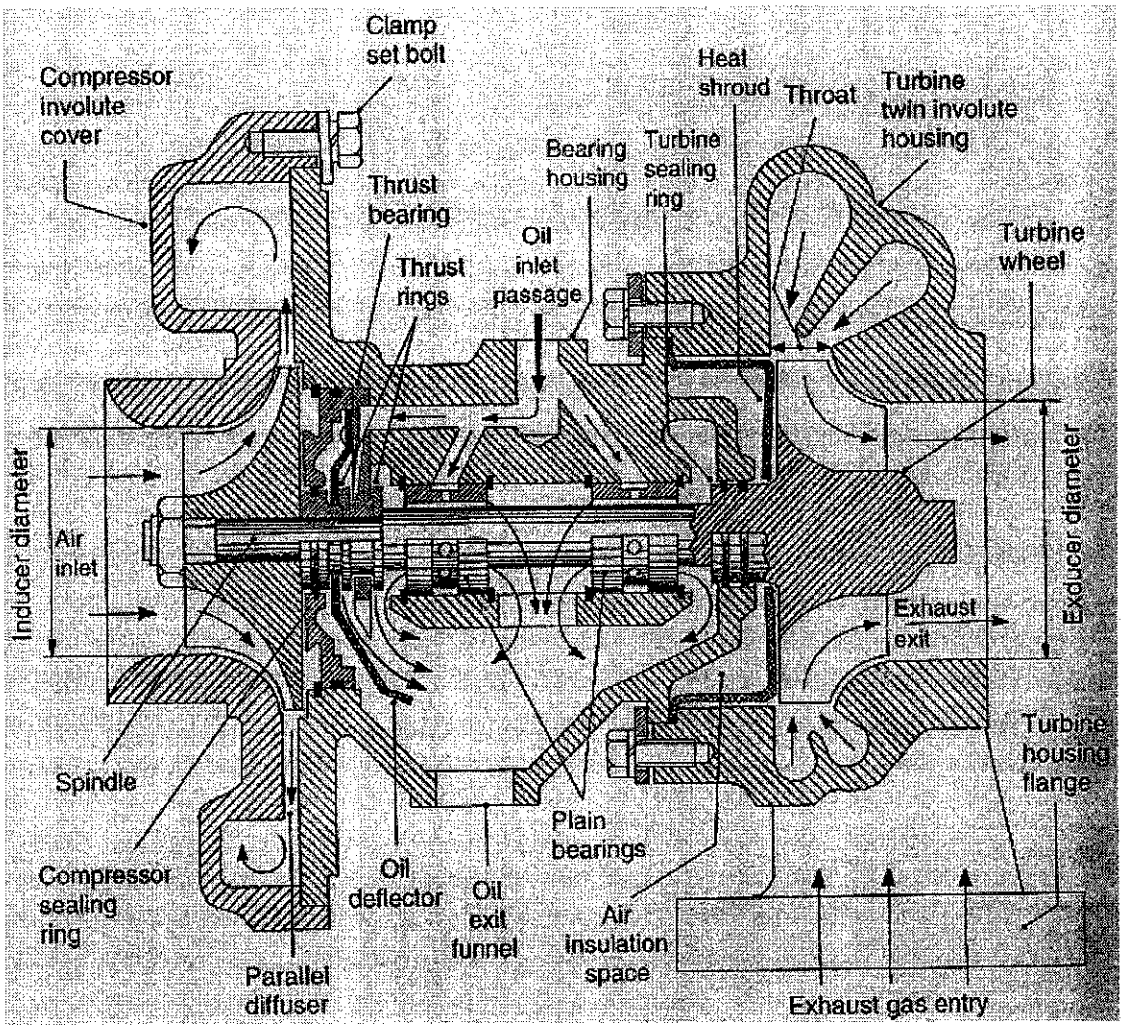

Figure 2.7 Turbocharger construction [4]

2.2.1 The Operating Principle of Turbine

Exhaust gas from the engine's cylinders is expelled via the exhaust manifold into the turbine volute circular decreasing cross-section passageway, at a very high velocity, where it is directed tangentially inwards through the throat of the turbine housing. The released gas kinetic energy impinges on the turbine-blades, thereby imparting energy to the turbine-wheel as it passes through the cells formed between adjacent blades with a corresponding decrease in both gas velocity and pressure as shown in **Figure 2.8**. The exhaust gas with its rapidly decreasing energy moves radially inwards and, at the same time, its flow path moves through a right-angle so that it passes axially along the hub before leaving the turbine housing. The expansion of the gas ejected from the turbine-wheel then produces a sudden drop in its velocity and pressure as it enters the silencer pipe system. Turbine speed and boost pressure are largely dependent upon the amount of energy contained in the hot, highly mobile exhaust gases and on the rate of energy transference from the gas to the turbine-blades. This, at idle speed fuel supplied to the engine and therefore, the energy content in the outgoing exhaust gas will also be very low whereas, with increased engine speed and load conditions, considerably more fuel is consumed by the engine, which in turn releases proportionally more energy to the escaping exhaust gases. Hence, at light load and low speed, the turbine assembly speed can be around 30000 to 50000 rev/min, whereas at high speed and high load operating conditions the spindle and wheel assembly can revolve at speeds up to 120000 and 150000 rev/min, depending upon design and application.

2.2.2 The Operating Principles of Compressor

With the spindle assembly rotating, the air cells formed between adjacent blades sweep. The entrapped air around the compressor housing curved walk the air mass is therefore subjected to centrifugal force. This force produces a radial outward motion to the air, with its velocity and, to some extent, its pressure becoming greater the further the air moves out from the center of rotation as shown in **Figure 2.8**. The air thus moves through the diverging passages of the cells to the periphery where it is flung out with a high velocity. More air will, at the same time, be drawn into the inducer due to the forward curved blades at the entrance, and this tends to generate a slight depression. Hence, it encourages a continuous supply of fresh charge to enter the eye of the **impeller**. Once leaving the outer rim of the impeller the tangential air movement relative to the impeller has its maximum kinetic energy, but, as it is pressure energy that is required, the air is expanded in the parallel diffuser so that its velocity sharply falls while, simultaneously, its pressure rises. In other words, the kinetic energy at the entrance to the parallel diffuser is partially converted into pressure energy by the time it arrives at the outer edge of the parallel annular-shaped passageway. The air then leaving the diffuser is progressively collected in the volute, from some starting point where the circular passageway is at its smallest, to its exit where the passage is at its largest cross-section. The volute therefore prevents the air discharged from the diffuser becoming congested and, at the same time, it continues the diffusion process further; that is, the air movement is slowed down even more whereas its pressure still rises.

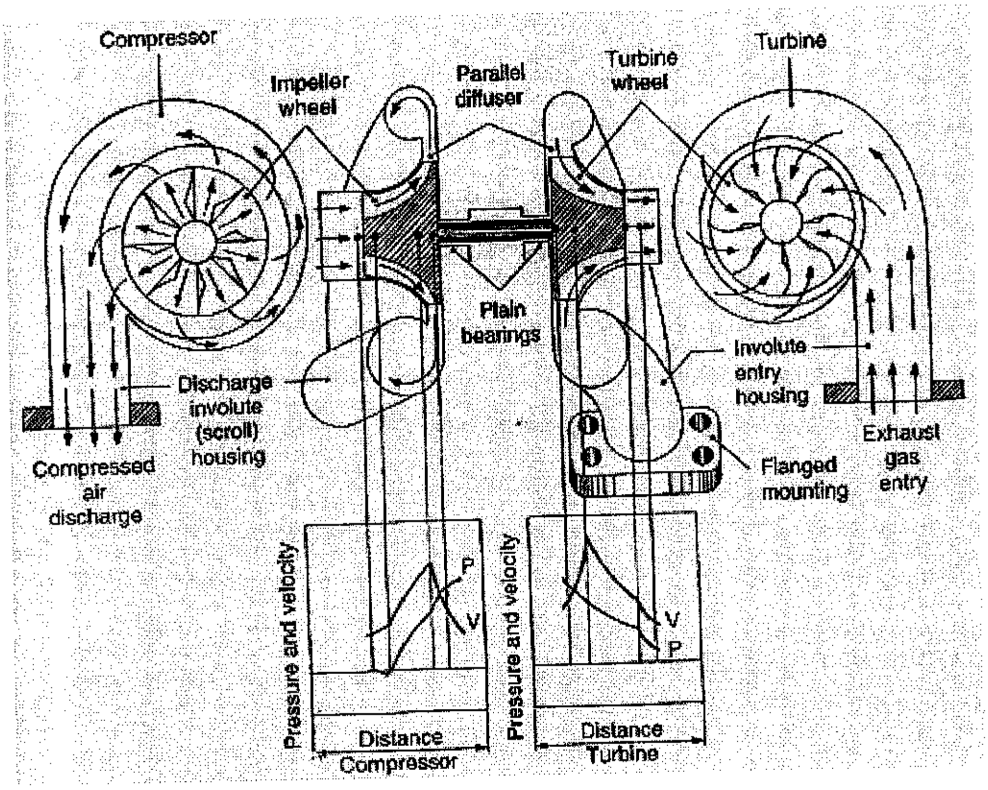

Figure 2.8 Turbocharger principle [4]

2.2.3 Turbocharger Bearing System

Turbocharger is an essential part to boost the power produced by the small scale engine. Without turbocharger, the engine will increase in size in order to produce the same output power same like the small engine with the turbocharger. Since the widely usage of the turbocharger engine in the industry, operator facing a lot of problem with the turbocharger failure at the low hour usage.

Turbocharger bearing is subjected to excessive wear during initial engine start-up. This is due to lack of lubricating fluid at the point of contact between the stationary and rotating components of the bearing system prior to full lubrication pressure being built up in the engine after start up. An aggravating factor for the turbocharger bearing system is a fact that the oil is typically siphoned away from the turbocharger as soon as the engine shut down. Since the turbocharger may continue to coast down approximately 30 minutes, the bearing are left with extremely dry condition for the next engine start up.

In typical turbocharger installation, the turbocharger is mounted at relatively high position on the engine so that it can receive exhaust gas directly from exhaust manifold and supply compressed air directly to the intake manifold or aftercooler [5]. The oil supply however typically arrives at the turbocharger bearing housing from the location farther down in the engine block where pressurized oil is present. Due to relative points of attachment, a siphoning action occurs immediately after the engine stopped and this siphoning action drained oil from the turbocharger supply line back into the engine.
Available data, reports and failure analysis indicate that thrust or journal bearing failure may comprise as much as 50% of turbocharger warranty failure. Much of this damage due to delay in oil delivery to the turbocharger after high speed engine start at the cold ambient condition.

2.3 Turbocharger Selection Requirement

2.3.1 Adiabatic Efficiency

A 100% adiabatic efficiency means that there is no gain or loss of heat during compression. Most turbochargers will have a 65-75% adiabatic efficiency. Some narrow range turbochargers can get higher; these types of turbochargers work well in engines that operate over a narrow rpm range. In general the wide range turbochargers do not have as good peak efficiency, but have better average efficiency and work better on engine that operate over a wide rpm range.

2.3.2 Pressure Ratio

This is the inlet pressure compared to the outlet pressure of the turbocharger's compressor. For single stage turbo, the inlet pressure will usually be at atmospheric (14.7 psi) and the outlet will be at atmospheric plus boost pressure. For staged turbo the inlet pressure will be the outlet pressure of the turbo before it plus the atmospheric pressure, and the outlet will be the inlet pressure plus additional boost from that turbo.

2.3.3 Density Ratio

Turbocharger compresses the air to make it denser and hence allows more oxygen into the engine and gives the potential to produce more power. The ratio of the density of the inlet air to the density of the outlet air is known as the density ratio.

2.3.4 Turbo Lag

Turbo lag is the delay felt between the opening of the throttle valve and the turbocharger providing additional power. The lag is the time the idling turbine needs to reach boost speed. Plus, the time needed for the intercooler and tubing to fill as the change is made from a vacuum to a pressure. Total lag time may be a half second or more. This 'turbo lag' is noticeable and objectionable to many drivers. One partial solution is to make the rotating parts (the compressor and turbine wheel) as light as possible. Lighter parts pick up speed more quickly. Another solution is to use two smaller turbochargers instead of a larger single unit. The smaller and lighter rotating parts reduce lag time.

The time taken for the turbine and compressor assembly to attain the maximum operating speed is dominated by the overall efficiency of the turbocharger and the polar moment of inertia of the rotating assembly. The polar moment of inertia I, is the reluctance of the rotating body to change speed, which may be represented by:

$$\mathbf{I\ (kgm2) = mk2} \qquad (1)$$

k = radius of gyration in meters, m
m = mass of rotating assembly, kg

Torque T, required to accelerate the rotating body is given by:

$$\mathbf{T = I\alpha\ (Nm)} \qquad (2)$$

I = Polar Moment of inertia, kgm2
α = Angular acceleration of the shaft, rad/s2

Hence, the heavier a rotating body, the higher the inertia of the body would be which requires a larger torque to accelerate the body.

2.3.5 Turbo Boost

Usually measured in pounds per square inch, it is the pressure the turbocharger makes in the intake manifold. One of the ways to increase airflow through a passage is to increase the pressure differential across the passage. By boosting the intake manifold pressure, airflow into the engine will increase, making more power potential. Boost is also measured in Bar.

2.3.6 Boost Threshold

Unlike turbo lag, which is the delay of boost, boost threshold is the lowest possible rpm at which there can be noticeable boost. A low boost threshold is important when accelerating from very low rpm.

2.3.7 Wastegate

The turbocharger can raise boost pressure so high that detonation and engine damage can occur. To limit boost pressure and prevent over boost, most turbochargers have a wastegate as shown in **Figure 2.9**. The wastegate is a valve that allows the exhaust gasses to bypass the turbine. The waste gate relies on boost pressure to open it. Spliced into the wastegate pressure feed there must be some form of pressure bleed. By bleeding pressure to the wastegate, it is possible to control the amount of boost by reducing the pressure at the wastegate. Then part of the exhaust gas by passes the turbine and flows through the wastegate. This exhaust gas is “wasted” because is does not help spin the turbine.

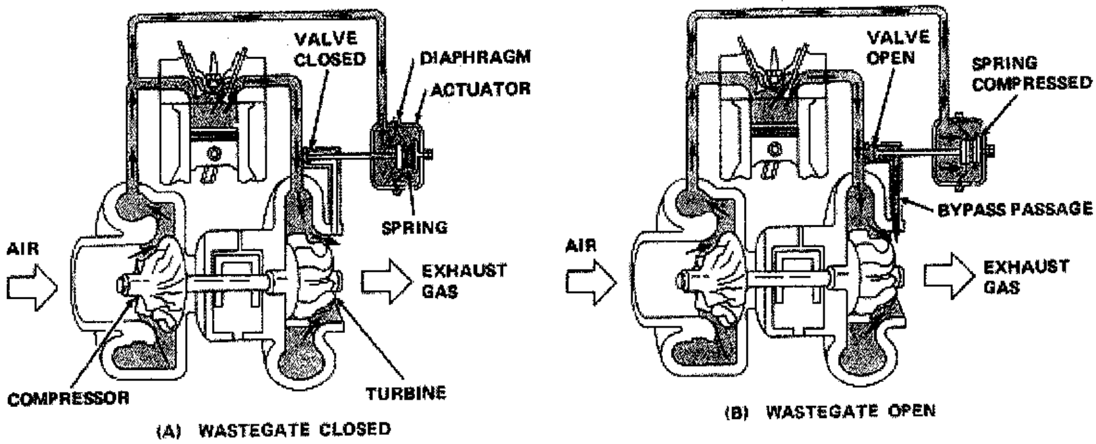

Figure 2.9 Operation of the turbocharger wastegate [4]

The wastegate may be pneumatically or computer controlled. **Figure 2.9** shows the operation of the pneumatic wastegate. Pneumatic means operated by compressed air. The actuator-diaphragm spring compresses when the boost pressure exceeds the spring force shown in **Figure 2.9**. This opens the bypass passage through the wastegate. No further increase in turbine speed is possible which limits boost pressure. Engines with a computer-controlled waste gate have a pressure sensor in the intake manifold. The sensors send the signals to the computer system known as ECM when the boost pressure goes too high. The ECM then signals a solenoid valve that controls the wastegate actuator. It opens the wastegate.

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2.3.8 Turbo Cool Down

A turbocharger is cooled by the engine oil, and in many cases, engine coolant as well. Turbochargers get very hot when boosting pressure as well as when the engine is shutdown the oil and coolant stop flowing. If you shut the engine down when the turbocharger is hot, the oil can burn and build up in the unit known as coking and eventually cause oil to leak. It is a good idea to let the engine idle for at least 2 minutes after any time you ran under boost. This will cool the turbo down and help prevent coking.

2.3.9 Area to Radius Ratio

The speed and acceleration of the turbine and compressor wheel assembly was influenced by a number of factors, but once the most critical and important controlling parameters is the A/R ratio. The A/R ratio shown in **Figure 2.10** is the smallest cross-sectional area of the intake passages in the turbine housing before the flow path spreads around the circumferential throat leading to the turbine wheel to the centroid of area A.

$$\text{A/R ratio} = \frac{\text{Smallest cross-sectional area of passage leading to volute}}{\text{Distance between cross-sectional area centroid and center of shaft}}$$

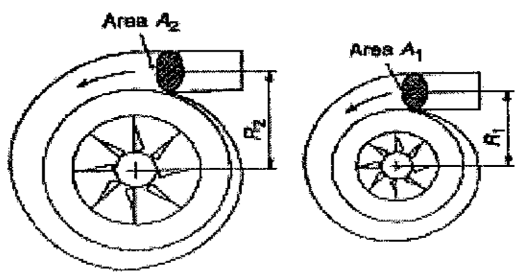


Figure 2.10 Comparative A/R ratios for vane less turbine housings [2]

A large A/R ratio reduces the turbine spin speed for a given exhaust gas flow, conversely a small A/R ratio raises the turbine-wheel speed for a similar exhaust gas delivery. A/R ratio values tend to range between 0.3 and 1.

2.3.10 Compressor Flow Map

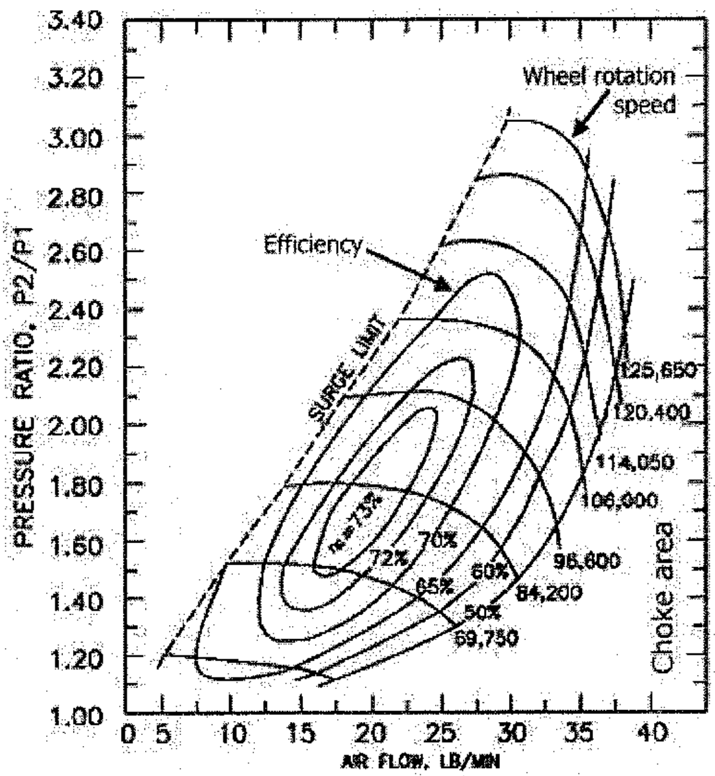


Figure 2.11 Compressor Flow Map [7]

The performance of an impeller in a particular housing is quantified using a "compressor flow map", an example of which is shown above. On the compressor flow map shown in **Figure 2.11**, the horizontal axis represents the amount of uncompressed air entering one turbo, either as volume flow or as mass flow. The vertical axis represents the amount of air compression that occurs inside the compressor; it is the ratio of the air pressure at the discharge (P2) to the air pressure at the inlet (P1). The curved lines with labels at one end such as 106,000 are the rotational speed, in revolution per minute (rpm) of the compressor wheel. The elliptical curves with labels such as 60% represent the efficiency of the compressor, or how well the compressor achieves pure adiabatic heating of the air (higher numbers are better and mean less extra heating of the air).

To the left of the surge limit line show in **Figure 2.11** of the flow map is the surge area where compressor operation can be unstable. Typically, surge occurs after the throttle plate is closed while the turbocharger is spinning rapidly and the by-pass valve does not release the sudden increase in pressure due to the backed-up air. During surge, the back-pressure build-up at the discharge opening of the compressor reduces the air flow. If the air flow falls below certain point, the compressor wheel (the impeller) will loose its "grip" on the air. Consequently, the air in the compressor stops being propelled forward by the impeller and is simply spinning around with the wheel, which is still being rotated by the exhaust gas passing through the turbine section. When this happens, the pressure build-up at the discharge opening forces air back through the impeller causing a reversal of air flow through the compressor. As the back pressure eventually decreases, the impeller again begins to function properly and air flows out of the compressor in the correct direction. Surge should be prevented at all costs because it not only slows the turbocharger wheels so that they must be spooled back up again but because it can be very damaging to the bushings or bearings and seals in the center section.

On the flow map above in **Figure 2.11**, the air flow regime to the right of the dotted line marking maximum wheel speed is called the choke area. The choke area is almost never noted on a flow map. To determine the choke area, drop a vertical line from about where the fastest wheel speed curve ends on the right side of the map. This vertical line is the approximate maximum air flow the compressor is capable of, regardless of efficiency or pressure ratio.

CHAPTER 3

METHODOLOGY

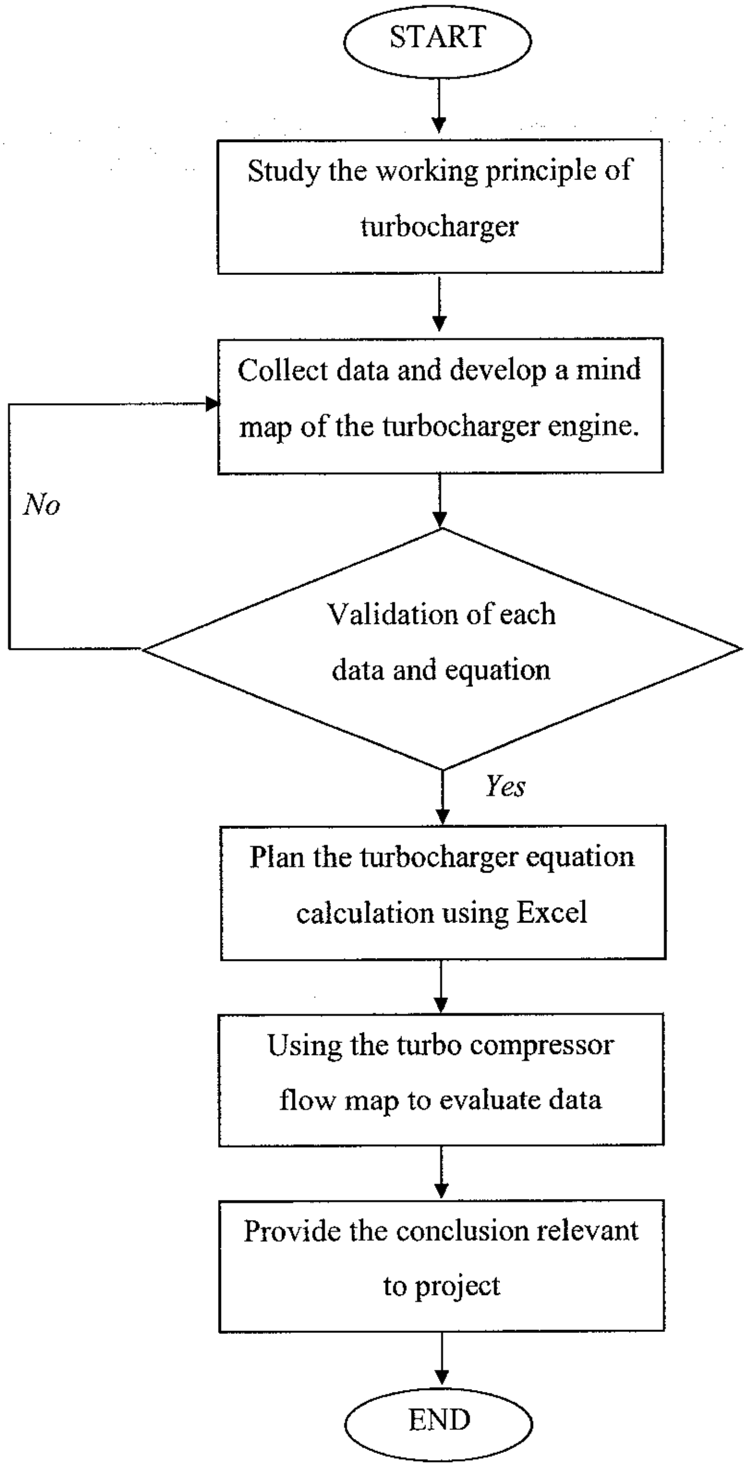


Figure 3.1: Work Flow and Methodology

The methodology is as follow:

Before the project begins, basic knowledge on the project is learned and understood. This is done through reading text books, journal and thesis in the library or in the internet on the subject matter. The methodology starts with theory understanding and parameters acquisition as shown in **Figure 3.1**.

Thermodynamics and basic principles of supercharger and turbocharger are than analyzed. Take note the advantages and also the disadvantages of turbocharger. Engine compression ratio should be changed to support the robust combustion due to force induction. Compressor flow map is than analyzed to suit the engine and also the target of turbocharging. Basically there are 2 targets of force induction, which are to emphasize on acceleration or top speed.

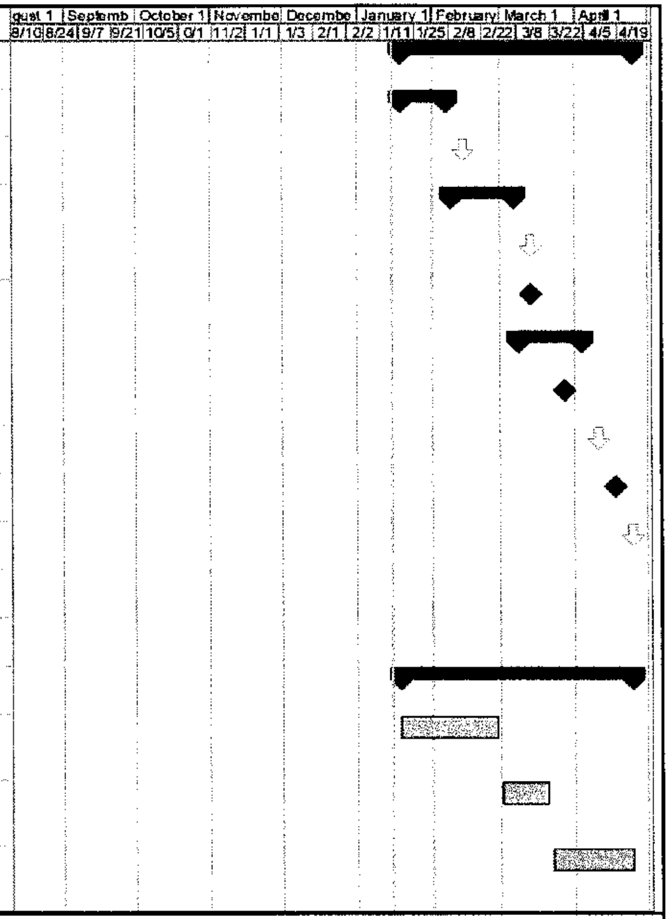
Once compressor flow map is selected, the properties and specifications of the compressor are keyed in to GT Power format. The turbocharger pressure ratio and efficiency will be determined.

The tools and equipment which are required in this Final Year Project are a GT Power Software and Windows based PC together with the programs such as Microsoft Office, especially Microsoft Excel; equipment needed basically would be data from on site results as well as from the internet and other references.

3.3 Gantt Chart for FYP 2

Table 3.1 : Project Semester Gantt Chart

ID	Task Name	Duration	Start	Finish	Predecessors
1	Final Year Project 2	14 wks	Mon 1/19/09	Fri 4/24/09	
2	Project Work Continue	3 wks	Mon 1/19/09	Fri 2/6/09	
3	Submission of Progress Report 1	1 wk?	Mon 2/9/09	Fri 2/13/09	
4	Project Work Continue	4 wks	Mon 2/9/09	Fri 3/6/09	
5	Submission of Progress Report 2	1 wk	Mon 3/9/09	Fri 3/13/09	
6	Seminar (compulsory)	1 wk	Mon 3/9/09	Fri 3/13/09	
7	Project Work Continue	4 wks	Mon 3/9/09	Fri 4/3/09	
8	Poster Exhibition	1 wk	Mon 3/23/09	Fri 3/27/09	
9	Submission of Dissertation (soft bound)	1 wk	Mon 4/6/09	Fri 4/10/09	
10	Oral Presentation	1 wk	Mon 4/13/09	Fri 4/17/09	
11	Submission of Project Dissertation (Hard Bound)	1 wk	Mon 4/20/09	Fri 4/24/09	
12					
13					
14	Final Year Project 2	14 wks	Mon 1/19/09	Fri 4/24/09	
15	Develop or design the corrective action for each problem selected.	6 wks	Mon 1/19/09	Fri 2/27/09	
16	Confirm each corrective action comply with safety aspect of the engine and operation.	3 wks?	Mon 3/2/09	Fri 3/20/09	
17	Develop the database for turbocharger failure analysis	5 wks	Mon 3/23/09	Fri 4/24/09	



Project: Gantt Chart FYP2
Date: Thu 1/15/09

Task | Milestone | External Tasks
Split | Summary | External Milestone
Progress | Project Summary | Deadline

Page 1

CHAPTER 4

RESULT AND DISCUSSION

4.1 Data Gathering and Analysis

4.1.1 Compression Ratio

The turbocharger is generally robust to higher engine compression ratios. However, peak cylinder pressures are more easily reached with forced induction, and the risk of blowing a head gasket, or even fatiguing the head bolts increases. As pressure in the cylinder increases rapidly because of force induction, compression ratio of the engine needs to be decreased to avoid detonation or pre-ignition. Once the compression ratio is increase more than 10, detonation will occur causing damage to the engine.

In order to reduce the compression ratio to 10, the cylinder head gasket thickness shown in **Figure 4.1** needs to be increased by the following calculations:

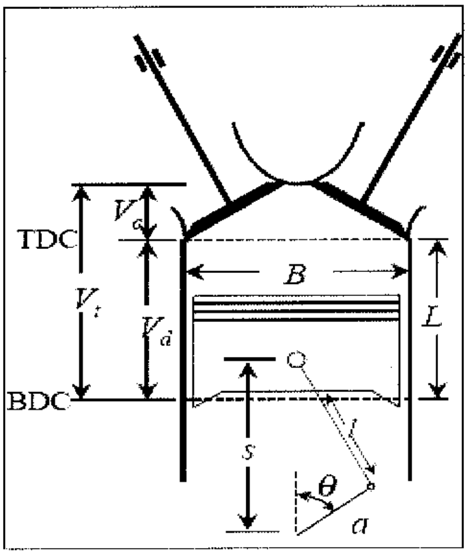


Figure 4.1: Compression Ratio calculations [8]

Compression Ratio,	CR = (Vd +Vc) / Vc	(3)
Volume displacement	Vd (cm3) = π x r2 x L x 10-3	(4)

r = radius of bore (mm) = B/2

L = Stroke (mm)

Volume clearance (cm^3) = Vc Bore(B) = 67mm, Stroke(L) = 42.5mm

Vd = 149.84cm3

Original CR = 12,

So, Vc = 13.621 cm3

Modified CR = 10,

So, Vc' = 16.65 cm3

Vc'- Vc = 16.65-13.621 = 3.028 cm3

So, the volume of the cylinder needs to increase by 3.028 cm3

Imagine a cylinder with a diameter of 67mm (bore) with unknown height (L) with a volume of 3.028 cm3.

$$\pi \times r2 \times L = 3.028 \times 103 \qquad (5)$$

L = 0.86mm (increase in gasket thickness to achieve CR = 10)

4.1.2 Selecting a Compressor

In order to select a compressor, first we need to assume the boost pressure to get the pressure ratio.

Pressure Ratio: $$Pr = P2 / P1 \qquad (6)$$

Where by,

P1 = Air pressure entering the compressor (kPa)

P2 = P1 + Boost Pressure = Air pressure exiting the compressor (kPa)

Assume Pr is 1.7 and P1 = 90kPa. Now the pressure, density and temperature of air exiting the compressor is determine:

Pr = P2 / P1 Pr = 1.7, P1 = 90kPa

So, $P2 = 153kPa$ (pressure exiting the compressor, intake pressure)

The engine cannot be boosting with high pressure for example more than 160kpa since this is a naturally aspirated engine that is designed by its manufacturer to withstand peak cylinder pressure when intake pressure is at 101kpa.

Now to determine temperature exiting the compressor, T2:

$$T2 = [T1(Pr0.2857 - 1) + \eta cT1] / \eta c \qquad (7)$$

Where by,

$T1$ = Temperature of air entering the compressor (K) = 300K (room temperature)

$T2$ = Temperature of air exiting the compressor

ηc = Compressor adiabatic efficiency = 75% (assumption)

Assumption is based on normal Compressor adiabatic efficiency used by the manufacturer.

So, $T2 = 365.48$ K

Now to determine the density of air exiting the compressor, $\rho 2$:

$\rho 2 = P2 / (T2 \times R)$

Where by,

$\rho 2$ = Air density exiting compressor (kg/m3)

R = specific gas constant for dry air = 287.05 J kg-1 K-1

So, $\rho 2 = 1.46$ kg/m3

In order to calculate the mass flow rate of air entering the cylinders, the volume flow rate of air need to be calculated first:

$$Q \text{ (volume flow rate, m3/s)} = (C \times RPM \times \eta v \times 1 \times 10\text{-}6) / 120 \quad (8)$$

Where by,

C = Engine displacement (cm3) = 600 cm3

RPM = Engine speed (rev/min)

ηv = Volumetric Efficiency = 80% (assumption)

Assumption is based on normal volumetric efficiency of the engine.

The calculations for mass flow rate of air entering the compressor are:

$$m \text{ (mass flow rate, kg/s)} = Q \times \rho 2 \qquad (9)$$

In order to use the compressor flow mapping, the mass flow rate must be in the corrected lbs/min, so:

$$m' \text{ (lbs/min)} = m \text{ (kg/s)} \times 132.24 \qquad (10)$$

$$\text{Corrected } m \text{ (lbs/min)} = m'/\sqrt{(T1'/545)} \qquad (11)$$

Where by,

T1'= Temperature of air entering the compressor (R) = 540 R

It is also known that the restrictor in the air intake system have chocked the flow at 10 lbs/min meaning the maximum mass flow rate through the restrictor and enters the compressor is at 10 lbs/min.

Using equation above, the data has been shown in the equation model database using Excel application below (**Figure 4.2**)

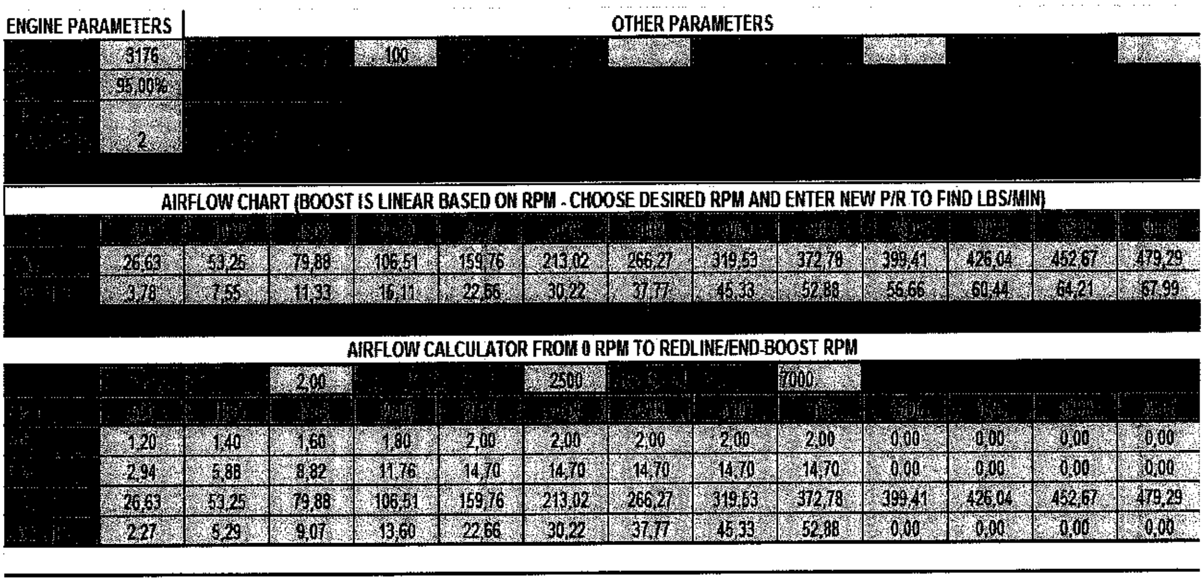


Figure 4.2: Equation Model Database

Table 4.1: Result for calculation at PR = 1.7

RPM	Air Flow Rate(m^3/s)	Mass Flow Rate (kg/s)	Mass Flow Rate (lbs/min)	Corrected Mass Flow Rate (lbs/min)
1000	0.004	0.006	0.772	0.775
2000	0.008	0.012	1.543	1.550
3000	0.012	0.018	2.315	2.326
4000	0.016	0.023	3.087	3.101
5000	0.020	0.029	3.858	3.876
6000	0.024	0.035	4.630	4.651
7000	0.028	0.041	5.402	5.426
8000	0.032	0.047	6.173	6.202
9000	0.036	0.053	6.945	6.977
10000	0.040	0.058	7.716	7.752
11000	0.044	0.064	8.488	8.527
12000	0.048	0.070	9.260	9.303
13000	0.052	0.076	10.031	10.078
14000	0.056	0.082	10.803	10.853

From the model above, the result of PR = 1.7 shown in the table above.

From **Table 4.1**, the value of calculated mass flow rate according to various engine rpm at PR =1.7 is plotted on the Garret GT 14 compressor flow map as shown in **Figure 4.3**:

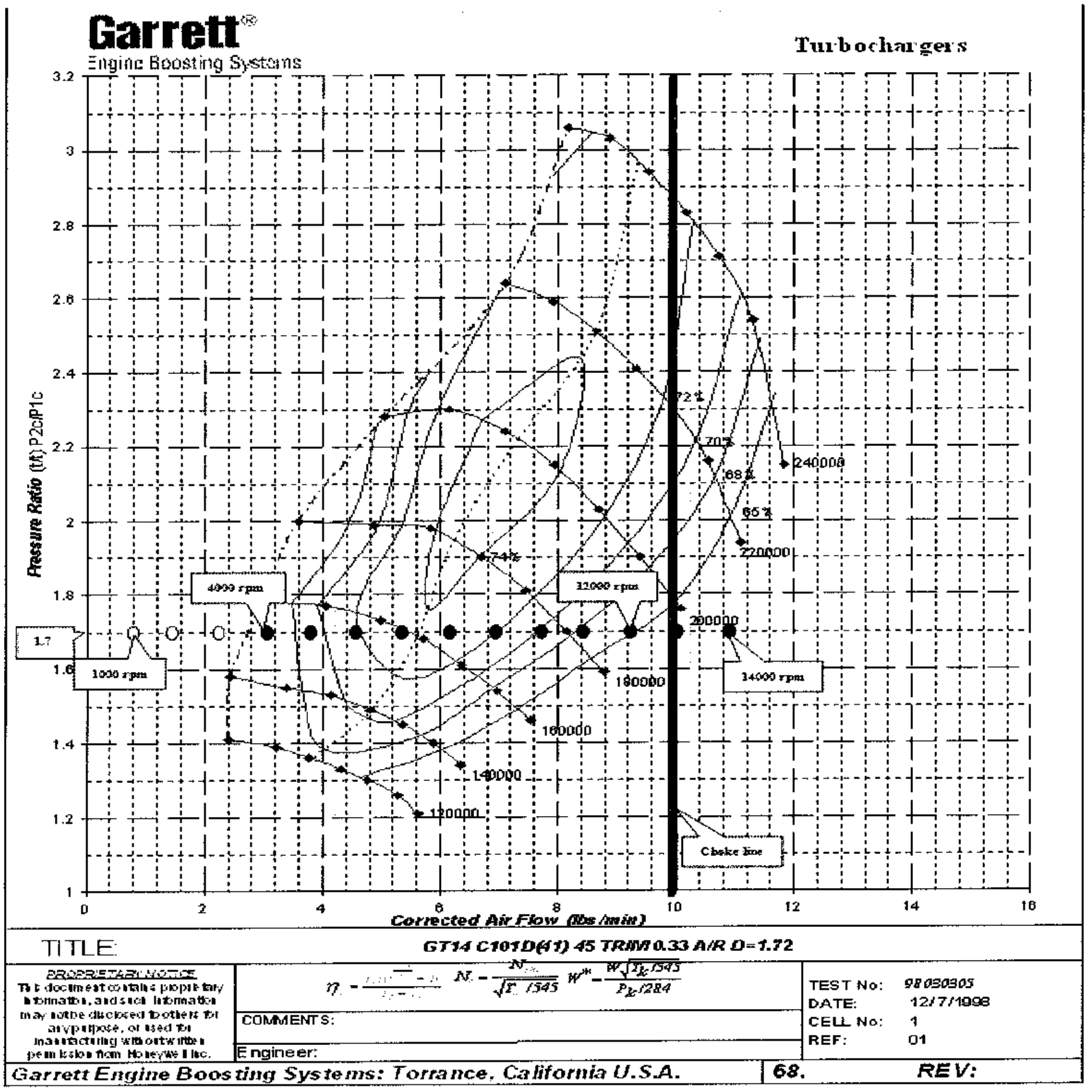


Figure 4.3: GT14 compressor flow map with Pr = 1.7

As shown in **Figure 4.3**, the compressor works steadily at point 4000 rpm to 12000 rpm. The red line indicates choking area. At point 3000 rpm, the compressor starts to be become unstable and stall. This is because the point is located out of the flow map and it is in surge region.

Now, let's vary the Pr at 1.6, 1.7 and 1.9 and observe the changes as shown in **Figure 4.4.**

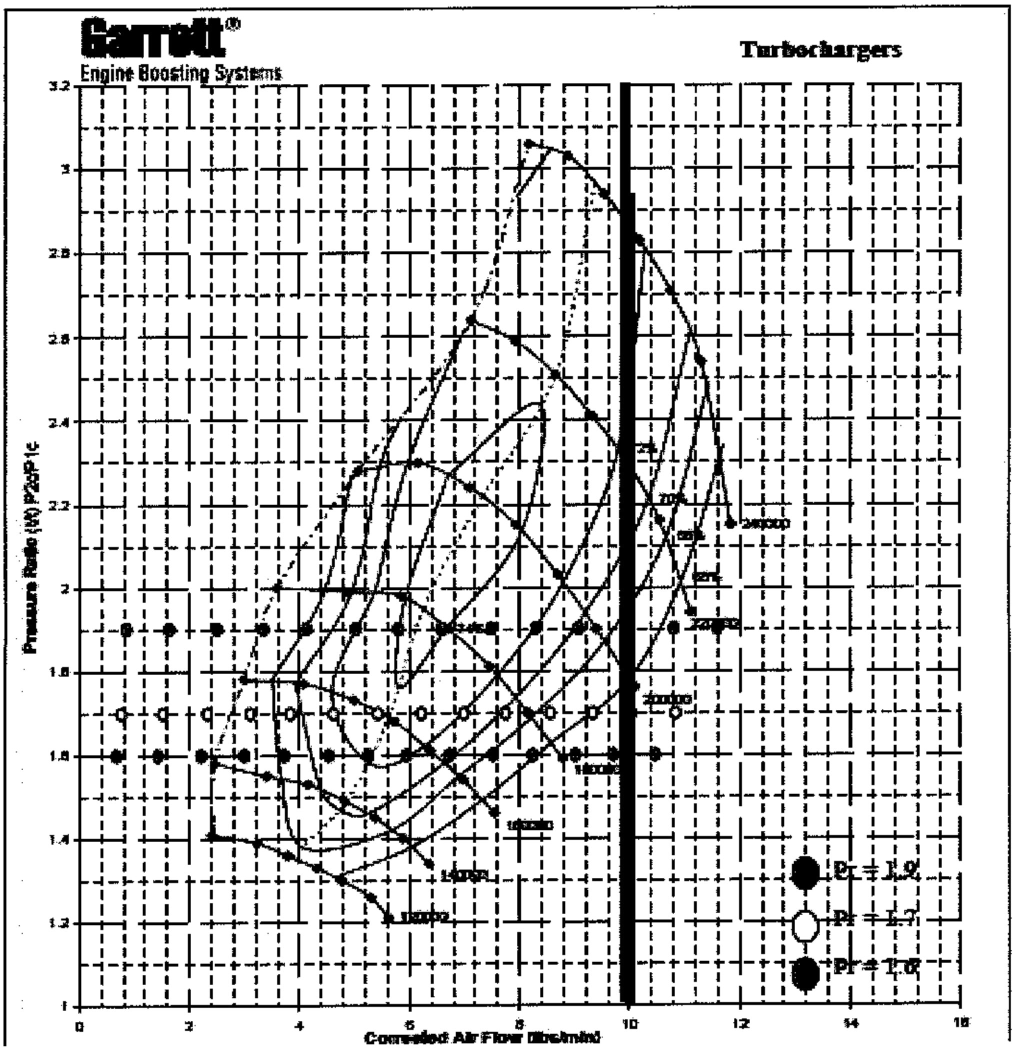


Figure 4.4: GT14 compressor flow map with variable Pr

When Pr =1.6, the compressor is starting to work when the engine speed is at 3000rpm and the compressor is running out of air at 11500rpm and when Pr = 1.9, pressure starting to built up when engine speed is at 6000rpm and end at 15000rpm but because of the restrictor limiting the air flow, it will work up to 12000rpm, it cannot go beyond the choked line.

From this observation, it can be concluded that the idle Pr for the compressor GT14 to work is at 1.7 since the desired engine speed for this project is around 4000rpm to 12000rpm.

4.2 Validation of efficiency using GT Power Simulation

In order to interpret the best efficiency of the compressor based on GT14 compressor map, the compressor flow map is converted to GT power file. **Figure 4.5** shows the mappings that are converted to GT Power format.

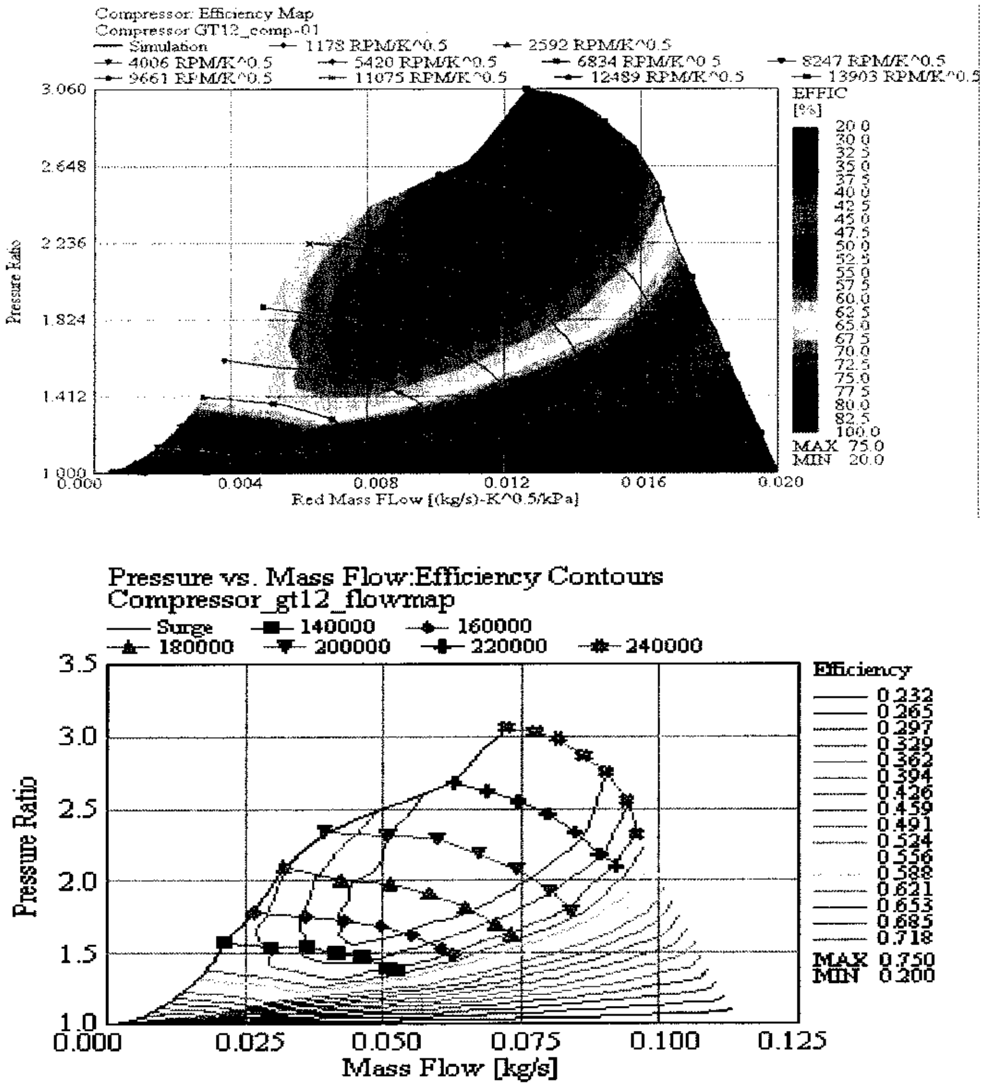


Figure 4.5: Converted GT14 compressor flow map in GT Power

From the observation of both graphs after converted to GT power software, it has been found that the red region of the compressor map represent the turbocharger best operating efficiency.

CHAPTER 5

CONCLUSION AND RECOMENDATION

5.1 Conclusion

The conclusions derived from this project work are below:

1. Compressor Mapping is important in selecting which compressor works best at desired pressure ratio and engine speed.
2. Using Turbocharger Equation Calculation Model database and Excel software for GT14 turbocharger, the flow rate of the turbocharger can be easily determined.
3. At the pressure ratio 1.7, the best efficiency of the compressor operations for turbocharger GT14 at 4000 rpm to 12000 rpm speed can be achieved.

5.2 Recommendation

1. If cost is not a factor, additional engine and also GT14 turbine are needed specially to do research and testing on it.
2. For future development of this project, engine testing on dynamometer is required to verify the simulation results.

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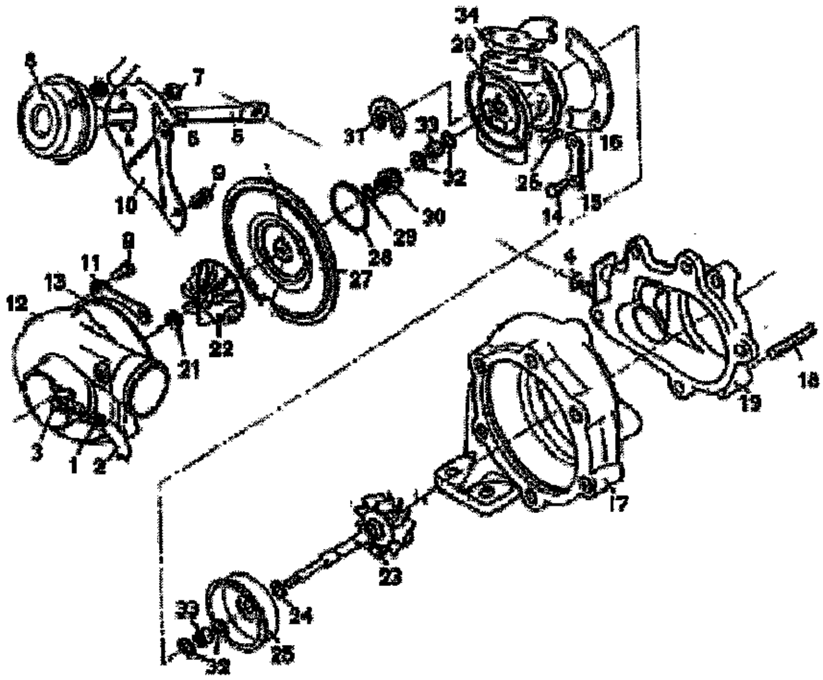
APPENDIX A



1. Clamp
2. Hose (waste gate pressure bleed)
3. Fitting
4. Clip (waste gate lever)
5. Rod (waste gate)
6. Adjusting nut
7. Nut
8. Control diaphragm (waste gate)
9. Bolt
10. Bracket (waste gate control diaphragm)
11. Locking plate (compressor housing)
12. Compressor housing
13. O-ring
14. Bolt
15. Locking plate (turbine housing)
16. Clamp plate (turbine Housing)
17. Turbine Housing
18. Exhaust Stud
19. Waste gate housing
20. Bearing housing
21. Nut (turbine shaft)
22. Compressor
23. Turbine shaft
24. Piston ring seal
25. Heat shield
26. Bolt
27. Compressor housing backing plate
28. O-ring
29. Piston ring seal
30. Trust collar
31. Thrust bearing
32. Snap ring
33. Journal bearing
34. Oil drain gasket

Figure A-1: Turbocharger Component View [1]

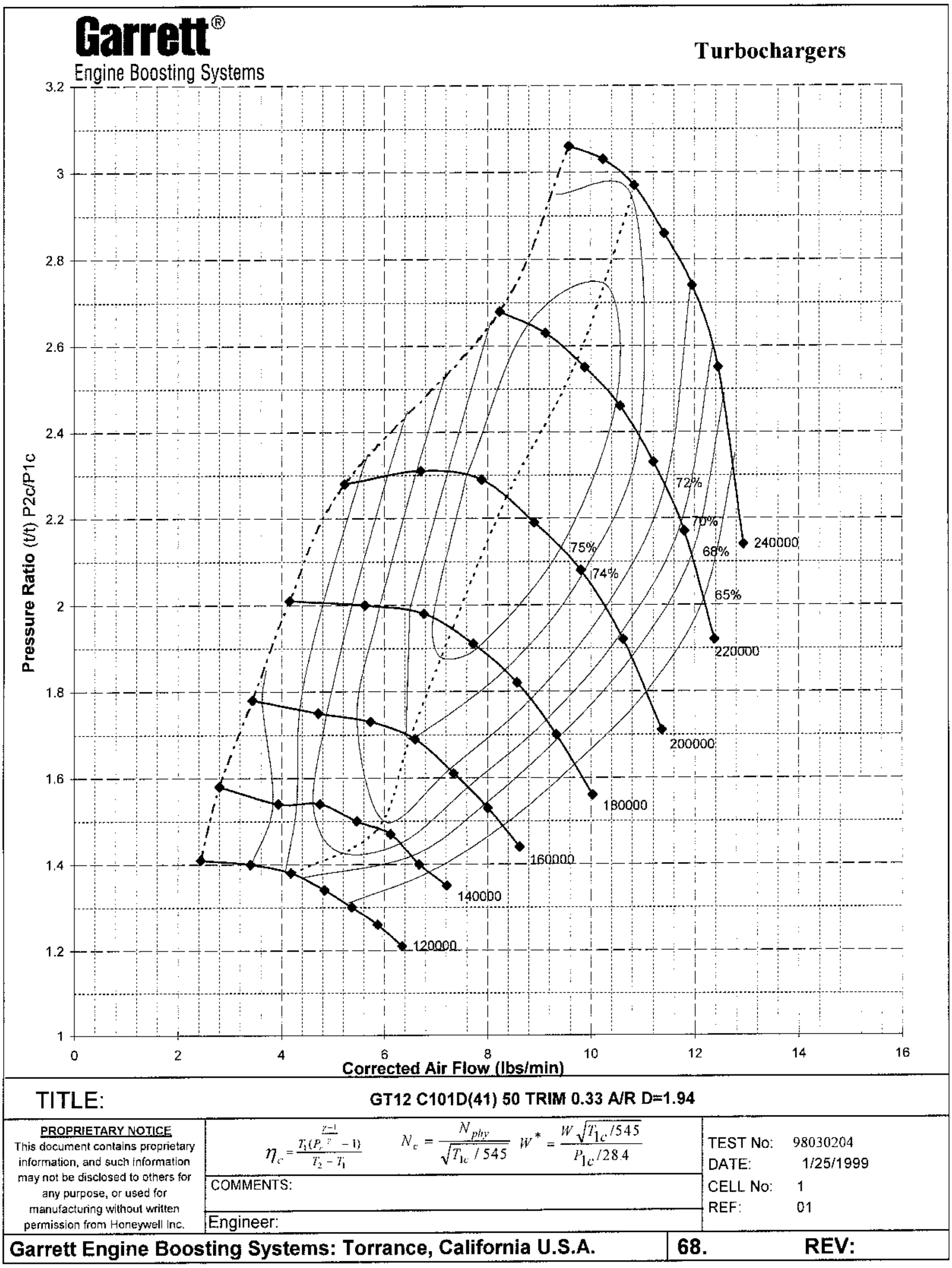


Figure A-2: GT14 compressor flow map [1]

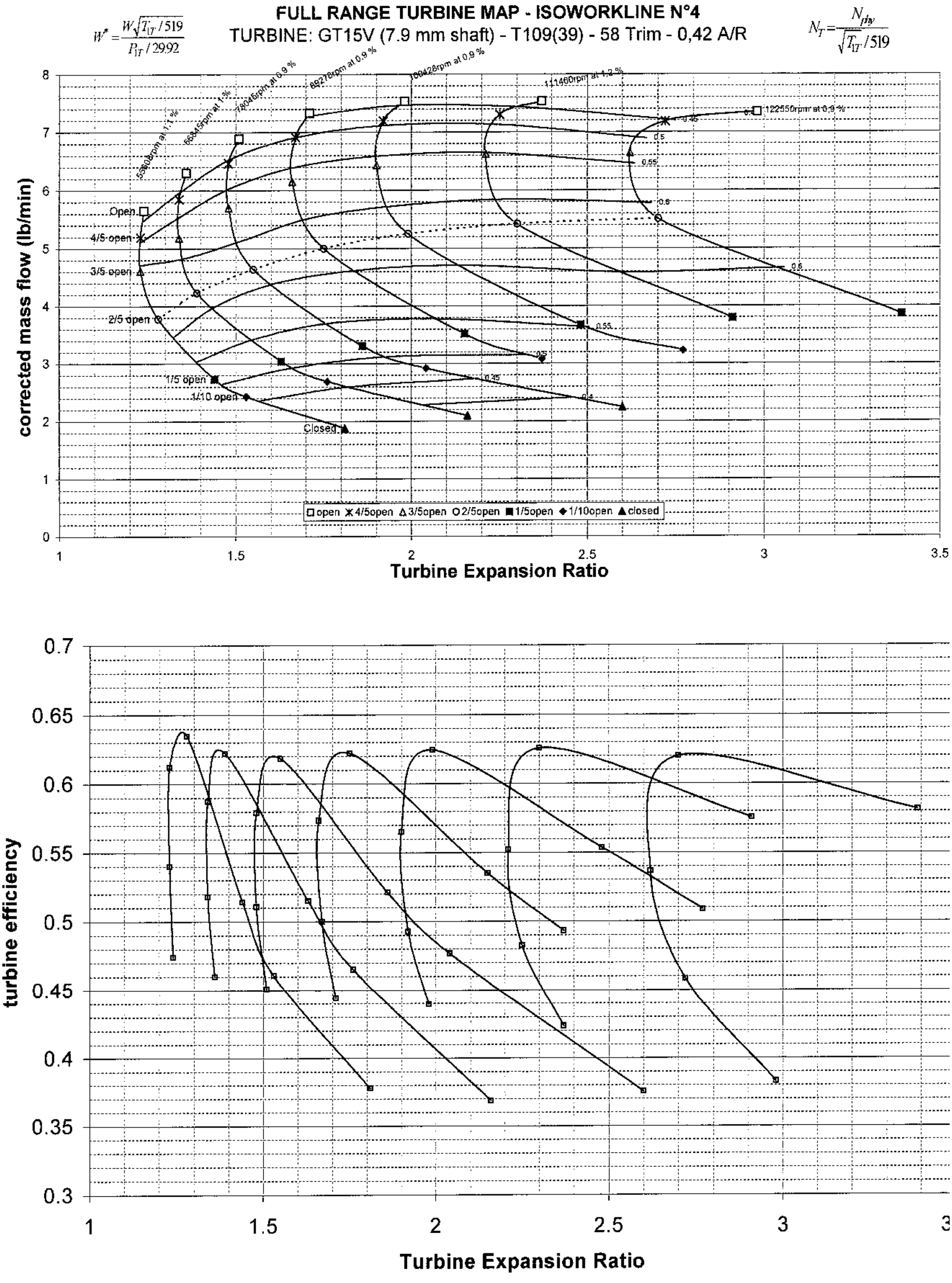
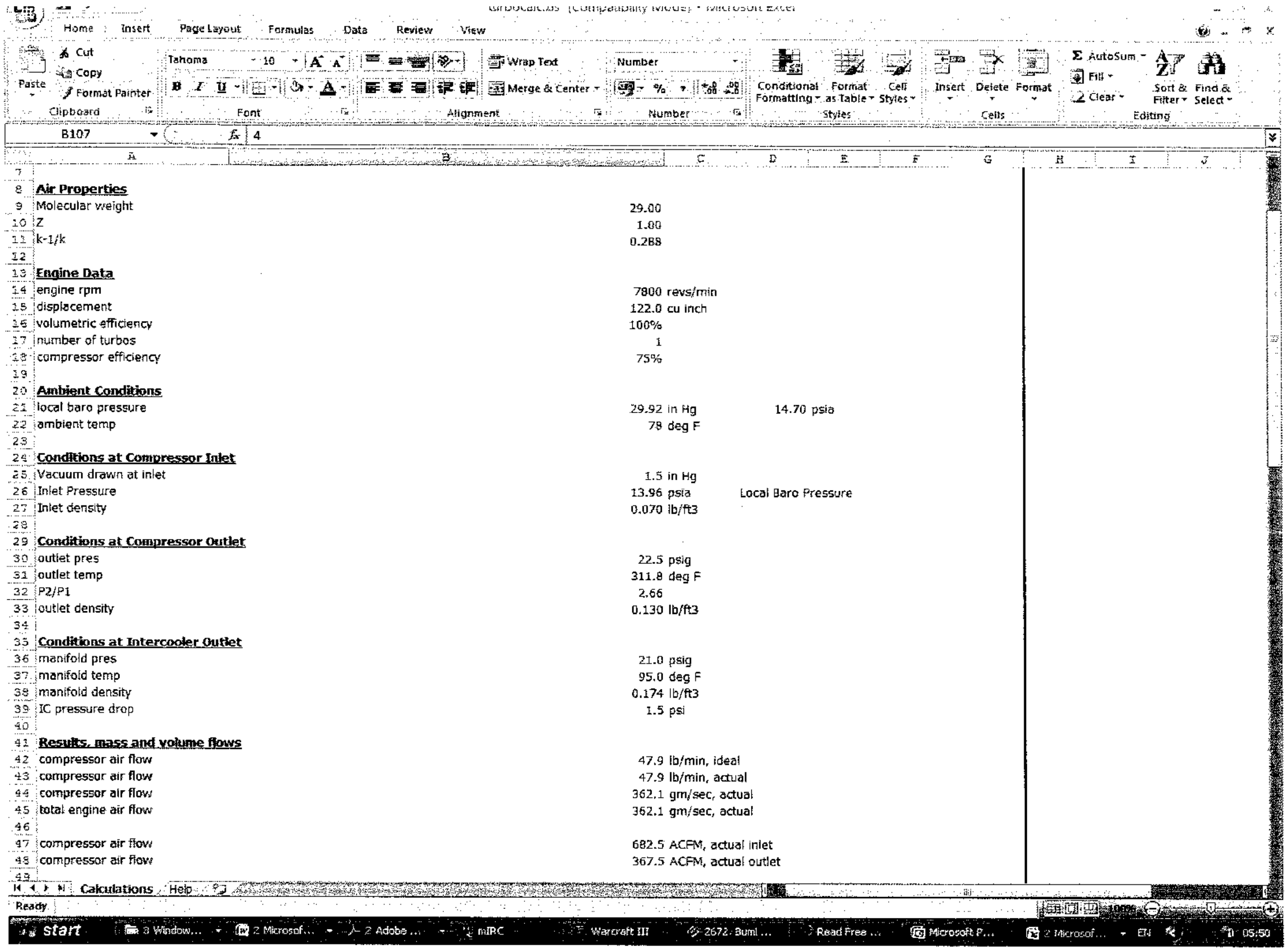


Figure A-3: GT14 turbine flow map [1]



Air Properties		
Molecular weight	29.00	
Z	1.00	
k-1/k	0.288	
Engine Data		
engine rpm	7800 revs/min	
displacement	122.0 cu inch	
volumetric efficiency	100%	
number of turbos	1	
compressor efficiency	75%	
Ambient Conditions		
local baro pressure	29.92 in Hg	14.70 psia
ambient temp	78 deg F	
Conditions at Compressor Inlet		
Vacuum drawn at inlet	1.5 in Hg	
Inlet Pressure	13.96 psia	Local Baro Pressure
Inlet density	0.070 lb/ft3	
Conditions at Compressor Outlet		
outlet pres	22.5 psig	
outlet temp	311.8 deg F	
P2/P1	2.66	
outlet density	0.130 lb/ft3	
Conditions at Intercooler Outlet		
manifold pres	21.0 psig	
manifold temp	95.0 deg F	
manifold density	0.174 lb/ft3	
IC pressure drop	1.5 psi	
Results, mass and volume flows		
compressor air flow	47.9 lb/min, ideal	
compressor air flow	47.9 lb/min, actual	
compressor air flow	362.1 gm/sec, actual	
total engine air flow	362.1 gm/sec, actual	
compressor air flow	682.5 ACFM, actual inlet	
compressor air flow	367.5 ACFM, actual outlet	

Figure A-4: Turbocharger Calculation Database