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BORANG PENGESAHAN
LAPORAN AKHIR PENYELIDIKAN

TAJUK PROJEK : PERFORMANCE ANALYSIS OF TURBOCHARGER EFFECT ON ENGINE IN LOCAL CARS

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ABSTRACT

The performance of a gasoline-fueled internal combustion engines can be increased with the use of a turbocharger. However, the amount of performance increment for a particular engine should be studied so that the advantages and drawbacks of turbocharging will be clarified. This study is mainly concerned on the suitable turbocharger unit selection, engine conversions required and guidelines for testing a Proton 4G92 SOHC 1.6-litre naturally aspirated gasoline engine. The engine is tested under its stock naturally aspirated condition and after been converted to turbocharged condition. The effect of inter cooled turbocharged condition is also been tested. Boost pressure is the main parameter in comparing the performance in different conditions as it influences the engine torque, power, efficiency and exhaust emissions. The use of a turbocharger on this test engine has clearly increased its performance compared to its stock naturally aspirated form. The incorporation of an intercooler to the turbocharger system increases the performance even further. With the worldwide effort towards environmental-friendly engines and fossil fuel shortage, the turbocharger can help to create engines with enhanced performance, minimum exhaust emissions and maximum fuel economy.

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LIST OF NOMENCLATURES

Roman Symbols

A	-	Cross-sectional area of piston
AF	-	Air-fuel ratio
B	-	Bore
$b.p$	-	Brake power
$bmep$	-	Brake mean effective pressure
DR	-	Density ratio
F	-	Force applied to crank
FA	-	Fuel-air ratio
$i.p$	-	Indicated power
$imep$	-	Indicated mean effective pressure
m_a	-	Mass of air
$\dot{m}_a$	-	Steady state mass airflow rate into the engine
mep	-	Mean effective pressure
m_f	-	Mass of fuel
$\dot{m}_f$	-	Rate of mass fuel flow into engine
N	-	Engine speed
n	-	Number of revolutions per cycle
n_c	-	Number of engine cylinders
p	-	Engine power
P_{atm}	-	Atmospheric pressure

PR	-	*Pressure ratio*
Q_{HV}	-	Heating value of fuel
r	-	Effective crank-arm radius
r_c	-	Compression ratio
S	-	Stroke
sfc	-	Specific fuel consumption
T	-	Engine torque
V_{BDC}	-	Cylinder volume at bottom dead center
V_{cl}	-	Clearance volume
V_d	-	Displacement volume
V_{TDC}	-	Cylinder volume at top dead center
W	-	Work of one cycle
W_b	-	Brake work of one revolution

Greek Symbols

η_m	-	Mechanical efficiency
η_v	-	Volumetric efficiency
η_t	-	Thermal efficiency
$(\eta_t)_b$	-	Brake thermal efficiency
$(\eta_t)_i$	-	Indicated thermal efficiency
η_c	-	Combustion efficiency
ϕ	-	Equivalence ratio
ρ_a	-	Air density at atmospheric condition outside the engine

CHAPTER I

INTRODUCTION

1.0 Introduction

The performance and power level from a particular engine can be improved by increasing its displacement. However, this increased displacement is normally associated with decreased fuel economy during part throttle driving. Therefore, this approach is not a desirable solution. One alternative to increase power output while maintaining the displacement is to turbo charge the engine.

A turbocharger is an exhaust driven device that utilizes exhaust gas energy (normally dissipated in the form of heat and pressure) to compress air to increase its density and consequently the total mass delivered to the engine. This increased air and fuel flow, when burned during combustion, is then realized as additional power output. Turbocharger consists of three major sections; the turbine, compressor and center housing assembly. The turbine and compressor section are mechanically connected. The center housing contains the bearing, seals and fittings necessary for the operation of the turbocharger. Turbo charging differs from other supercharging method by eliminating the mechanical connection between the compressor and the crankshaft, therefore reducing the continual power drain. This is essential to maintain fuel economy during part-throttle or low speed driving. Therefore, by using energy that is normally expelled through the exhaust system, the turbocharged engine can increase wide-open throttle power while maintaining fuel consumption at part-load of the smaller displacement engine.

1.1 OBJECTIVE

To study the performance of Proton 4G92 engine at steady state, wide-open throttle (WOT) condition, in terms of:

i) Power
ii) Torque
iii) Specific fuel consumption (sfc)
iv) Emissions
v) Volumetric efficiency
vi) Brake thermal efficiency

1.2 SCOPE

- A non turbo gasoline engine will be used for testing
- Separate turbo-charging unit will be used for testing
- Testing will be conducted with and without intercooler
- Emission analysis will be based on CO, CO_2, O_2 and HC.
- Engine performance analysis will be based on volumetric efficiency, brake horsepower, torque, fuel consumption, brake thermal efficiency and emission quality.

CHAPTER II

LITERATURE REVIEW

2.1 ` Internal Combustion Engines

The internal combustion engine (ICE) is a heat engine that converts chemical energy in a fuel into mechanical energy, usually made available on a rotating output shaft. Burning or oxidizing the fuel inside the engine releases this energy. The reciprocating type is the most common form of engine used as an automotive power source. W.W. Pulkrabek (1997) and Richard Stone (1999) discussed further on this topic.

There are two types of the internal combustion engines, classified based on the combustion system; spark-ignition (SI) and compression-ignition (CI). Because of their simplicity, ruggedness and high power to weight ratio, these two types of engine have found wide application in transportation (land, sea and air) and power generation (Heywood, 1988). The combustion process of the SI engine is initiated by the use of a spark plug (often called gasoline engine). The combustion process in the CI engine starts when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression (often called diesel engine).

However, only gasoline engine will be the studied on this project.

2.2 Combustion Process In Gasoline Engines

Gasoline engines have components quite identical to that of diesel engines. The principal difference is between the combustion systems. The gasoline engines use a carburettor or fuel injection system to mix air and fuel in the intake manifold so that a homogeneous mixture is compressed in the cylinder whereas in diesel engines, air alone is compressed in the cylinder.

A spark is used to control the initiation of combustion, which then spreads throughout the mixture. The mixture temperature during compression must be kept below the self-ignition temperature of the gasoline. Once combustion has started, it takes time for the flame front to move across the combustion chamber burning the fuel. During this time, the unburnt ‘end-gas’ (furthest from the spark plug, Figure 2.1) is heated by further compression and radiation from the flame front. If it reaches the self-ignition temperature before the flame front arrives, a large quantity of mixture may burn rapidly, producing severe pressure waves in the combustion chamber. This situation is commonly referred to as ‘knock’ or detonation and if this process continues for more than a few seconds, it will result in severe cylinder head and piston damage. Therefore the maximum compression ratio of the gasoline engines is limited by the ignition properties of the fuel. The minimum compression ratio is limited by the resulting low efficiency.

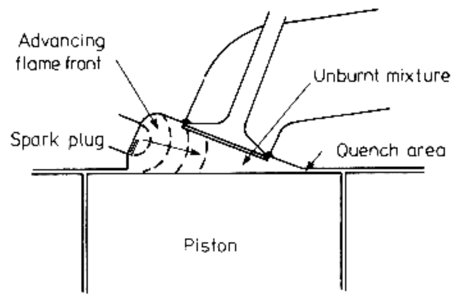


Figure 2.1: Flame propagation in a gasoline engine.

2.3 Turbocharger

As the turbocharger is not mechanically connected to the crankshaft, the turbine will not instantaneously respond to the throttle position. It takes several engine revolutions to change the exhaust flow rate and to speed-up the turbine. Figure 2.2 shows the principle of operation of a turbocharger with intercooler installed between the compressor outlet and the intake manifold. Figure 2.3 illustrates the typical arrangement of a turbocharged engine with the presence of the intercooler and the associated pressures and temperatures are shown. Figure 2.4 shows the cut-away view of a turbocharger.

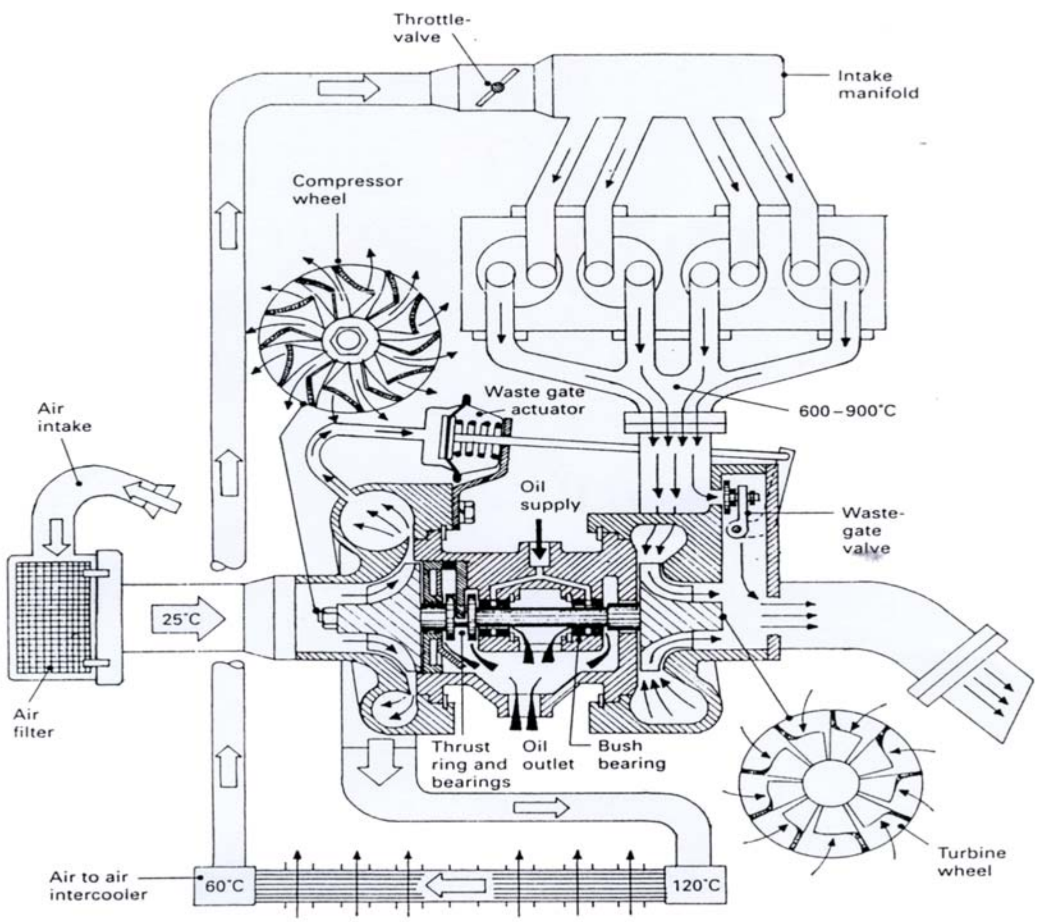


Figure 2.2: Turbocharger principle of operation.

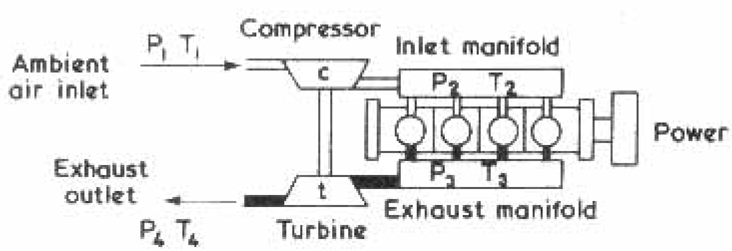


Figure 2.3: Typical arrangement of a turbocharged engine.

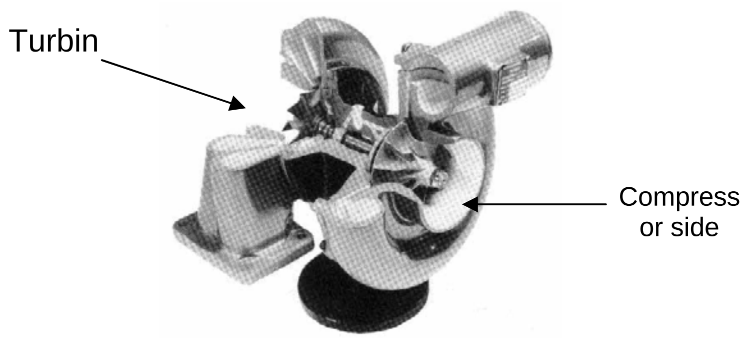


Figure 2.4: Cut-away view of a turbocharger

2.4 Turbo charging the Gasoline Engines

Figure 2.5 compares the naturally aspirated (NA) and turbocharged ideal engine cycles. The turbocharged cycle starts at a higher pressure (and density) at point 1'. Extra fuel can be burned between 2'- 4' because more air is available (the same volume, but higher density). The area inside the diagram, area 1-2-3-4-5-1, gives net power output. Two things are clear; the turbocharged engine has a greater power output (area under the diagram) and a much higher maximum pressure.

The high maximum pressure may not be acceptable unless the engine is designed to be turbocharged – the engine may not withstand the stresses involved. By reducing the compression ratio, the clearance volume (V_{cl}) is increased and maximum pressure will be reduced. If the compression ratio is suitable, the maximum pressure in the turbocharged engine can equal to that of the naturally aspirated one (Figure 2.6). The power output of the turbocharged engine remains greater than of the naturally aspirated engine.

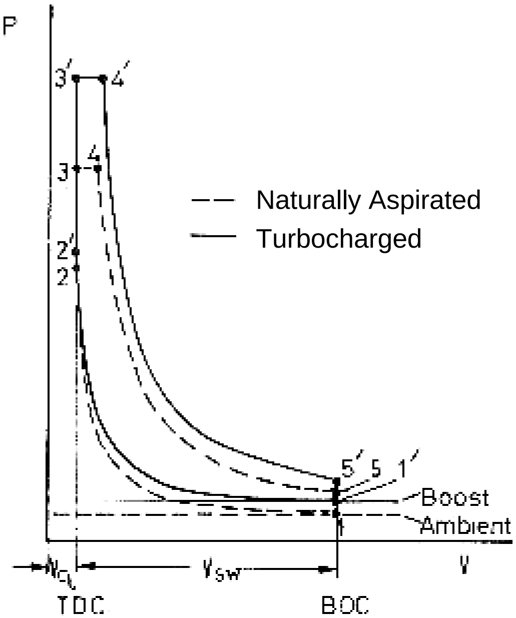


Figure 2.5: Comparison of turbocharged and NA air standard Otto cycle having the same compression ratio.

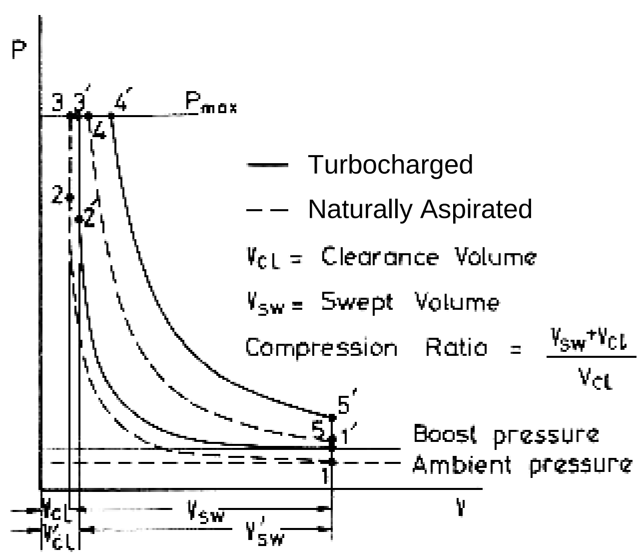


Figure 2.6: Comparison of turbocharged and NA air standard Otto cycle having the same maximum pressure but different compression ratios.

Turbo charging results in not only a higher boost pressure, but also a higher temperature. Unless the compression ratio of a gasoline engine is reduced, the temperature at the end of the compression stroke will be too high and the engine will experience detonation. Thus, the potential power output of a turbocharged gasoline engine is limited by its fuel properties. To control engine pressures, stresses and temperatures, the maximum allowable boost pressure must be controlled. Discussions on the important aspects of turbo charging the gasoline engines are on the following sections.

2.4.1 Compression Ratio

A turbocharged engine has effect with a variable compression ratio. At low engine revolutions when boost pressure is not being applied to the engine, the effective compression ratio will be the basic geometric compression ratio, but as the engine speed and load is increased, the cylinder will be subjected to increasing boost pressure with a result increase in effective compression ratio.

A reduction in geometric compression ratio from a similar naturally aspirated engine is an essential feature of a turbocharged engine. A turbocharger compresses air and thus, raises the temperature of the air induced into the engine. The effect is to increase the peak cylinder temperature and approaches the temperature at which detonation commences. The Otto cycle efficiency is largely governed by the compression ratio. By reducing the geometric compression ratio to avoid knock; the Otto cycle efficiency is also reduced and probably the overall engine efficiency will suffer.

Figure 2.7 shows the relationship between the geometric compression ratio of the engine and the overall effective compression ratio by increasing boost pressure. For example, a compression ratio of 9:1 for a naturally aspirated engine must be reduced to

6.7:1 if a boost of 0.5 bar is to be used, with no change in other knock-controlling parameters.

The most important fact is that the charge temperature should be kept as low as possible so that the geometric compression ratio can be maintained as high as possible (below the knock margin).

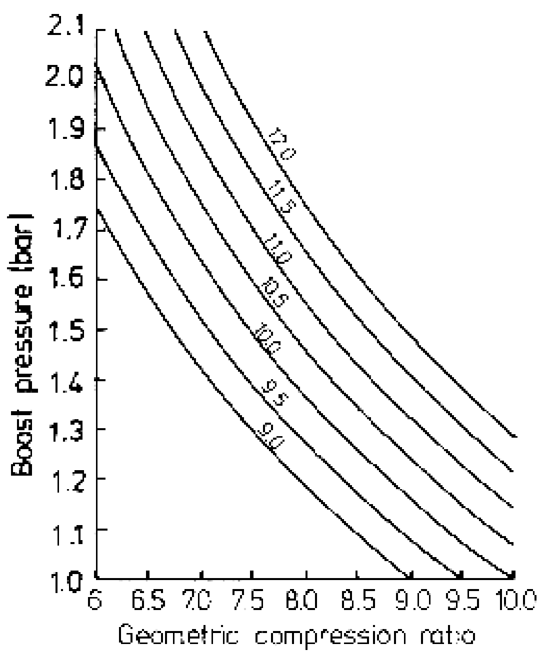


Figure 2.7: Relationship between boost, geometric compression ratio and effective compression ratio.

2.4.2 Ignition System

Because a turbocharged engine has effect on variable compression ratio as stated in previous section, it requires a different ignition advance curve from that of a naturally aspirated engine. Generally, a reduction in geometric compression ratio of an engine will require increased advance of the ignition timing, due to the slower burn rate of the fuel at lower cylinder pressures. As the boost rises and the mixture becomes denser and more turbulent, some retard may be necessary. Allard (1986) states that ignition retard increasing in step with rising boost at 1-2° per 1psi boost increase may be necessary.

Boost pressure in a cylinder increases the peak cylinder pressure and temperature. In addition, the boost pressure produces a denser, more rapidly burning mixture, which needs less ignition advance. These two factors mean that over-advanced timing can destroy a turbocharged engine by detonation or pre-ignition or both. Reducing the amount of ignition advance (retarding) at near or full load will reduce the tendency of the mixture to detonate by reducing the peak cylinder pressure and temperature. To avoid unnecessary fuel consumption with retarded timing, the technique should only be used when high boost pressure produced. Thus, at low speed and part load, conventional timing of the particular engine is retained.

One undesirable feature of retarded timing is an increase in heat rejection to the exhaust system, since the complete combustion and expansion process is delayed. Thus the turbine inlet temperature rises (Figure 2.8). Although the increase is small, very high temperature of SI engine exhaust gas (up to 1000°C) is a problem for the turbine manufacturer and can cause oxidation of the lubricating oil. Furthermore, the potential power increase obtainable by turbo charging with retarded timing alone is limited. Higher boost pressures can be used if compression ratio is also reduced. A change to a one or two range cooler than the same naturally aspirated engine spark plug specification will usually be required to hold the tip temperature to normal limits.

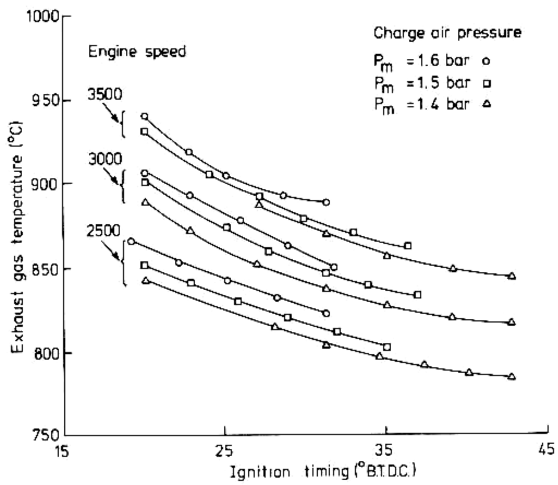


Figure 2.8: Exhaust gas temperature depended on spark timing and charge air pressure.

2.4.3 Inter cooling

The intercooler is a heat exchanger – positioned between the turbocharger and the intake manifold. Its purpose is to reduce the increased temperature of the intake charge due to compressive heating (Figure 2.2). The intake charge may be cooled by the use of ambient air, engine jacket water, iced water or some other low-temperature liquid as a cooling medium.

A perfect (100% efficient) intercooler could reduce the intake charge temperature by the cooling medium without any drop in pressure. This is not possible in actual world because there will be a pressure drop through the intercooler and it is not possible to lower the charge temperature to that of the cooling medium temperature. Of the cooling medium and intercooler design generally available, 70% to 75% efficiency is common.

Removing heat from the intake charge has two huge areas of merit. First, temperature reduction of the intake charge makes it denser. Higher charge density means more mass of air per volume per minute to flow through the engine at any given intake manifold pressure. This means more fuel can be burned to produce more power output. Second, reduced intake charge resulting in the overall temperature reduction of the remaining phase of the cycle. Therefore the engine will be operating under the detonation-safe margin. In addition, as the overall operating temperatures of the engine are reduced, thermal loading on valves and pistons and heat-rejection requirement of the engine are also reduced.

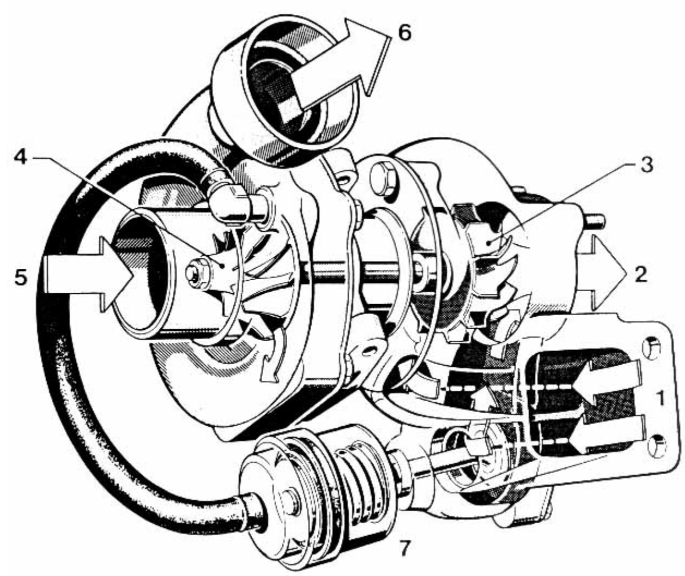
2.4.4 Boost Controls

The need for effective boost controls in a turbocharger system is because the turbocharger's characteristic of increasing its rate of airflow faster than the ability of the engine to accept that flow. If unchecked, the turbocharger can quickly produce damagingly high boost pressures that lead to engine detonation. There are various boost-control devices such as intake restrictor, exhaust restrictor, vent valve and waste gate. Of all boost-control devices commercially available, the waste gate is the best in terms of effectiveness and control.

A waste gate is used to control the exhaust gas flow rate to the turbine. On the turbine side of the turbocharger, exhaust gases can pass through two different openings. The normal opening routes the exhaust through the turbine wheel and the other opening bypasses the turbine wheel and sends the exhaust gasses directly to the exhaust system or directly to the atmosphere (Figure 2.9). By wasting or bypassing a portion of the exhaust gas energy around the turbine, the actual speed of the turbine (therefore the boost produces by the compressor) can be controlled. The two types of waste gates are integral and external. Integral waste gate is built into the turbocharger itself as shown in

Figure 2.9. The external waste gate is placed at any appropriate location at the exhaust manifold where the pulses from all cylinders have been collected (Figure 2.10).

The waste gate opening may be operated by boost pressure, manually or by a servomotor. A common method practiced in the real world is by using the boost pressure. Boost pressure applied to the waste gate is referred to as the actuator signal. Whenever the boost reaches the waste gate's diaphragm setting, it actuates the actuator that opens the bypass opening, which directs the exhaust gases directly to the exhaust system or atmosphere without passing through the turbine wheel.



Key 1 Exhaust gases to turbine. 2 Exhaust gas outlet. 3 Turbine wheel. 4 Compressor wheel. 5 Intake air to compressor. 6 Compressed air to engine. 7 Waste gate.

Figure 2.9: Turbocharger with integral waste gate, shown with the exhaust gas and intake air flow routes.

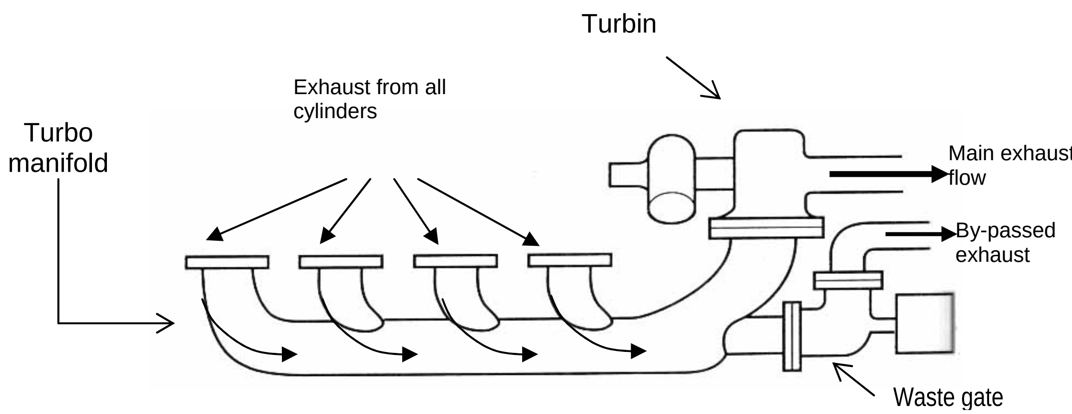


Figure 2.10: External waste gate placed prior to exhaust gas entering the turbocharger, where all pulses have been collected.

2.4.5 Blow-off Valve

When the throttle valve suddenly closed, such as during gear shifting, the compressed air from the turbocharger, which was rushing into the engine has no place to go. At the same time, the compressor continues to spin trying to compress air and make boost. . Because the throttle body is closed, the charged air is pushed back on to itself (backpressure), which slows down the turbo. This condition is known as 'surge'. When the engine start to accelerate again, the turbo has to re-spool up to develop boost again.

Blow-off valve is a pressure-relief valve and it is installed between the compressor outlet and the throttle body. It detects the vacuum on the intake manifold, which opens the valve to dump excess pressure between the throttle body and the turbocharger. This reduces the restriction and allows the turbo to almost spin freely, which reduces the lag. This backpressure is can also be very damaging to the bushings or bearings and seals in the turbocharger's center section.

2.4.6 Electronic Fuel Injection System

The atomization of fuel into the intake charge is extremely significant to the functioning of the internal combustion engine. The purpose of a carburettor or electronic fuel injection (EFI) system is to add fuel to the air entering in engine at the correct ratio so it will burn efficiently in the combustion chamber and at the same time, not to create hot fire that would cause early destruction of the engine. The duty is the same for a naturally aspirated and turbocharged engine except the intake manifold pressure is higher than ambient in a turbocharged condition. As the majority of automotive engines today utilize EFI system instead of the carburettor, only EFI system will be the subject of this discussion. Further discussions on carburetion of the turbocharged engine can be found in Allard (1986), Setright (1976), MacInnes (1976) and Bell (1997).

Electronic fuel injection is the most accurate method of fuel metering, mixing and distribution system, resulting in the best economy and lowest emissions. The fuel delivery system must be able to compensate for the additional airflow generated and add a corresponding amount of fuel under boost. Correct size of the injectors and fuel pumps must be selected properly for the vehicles' application. Incorrect fuel delivery may lead to harmful knocking problem. Bell (1997) discussed detail on EFI requirements and modifications of the turbocharged engine.

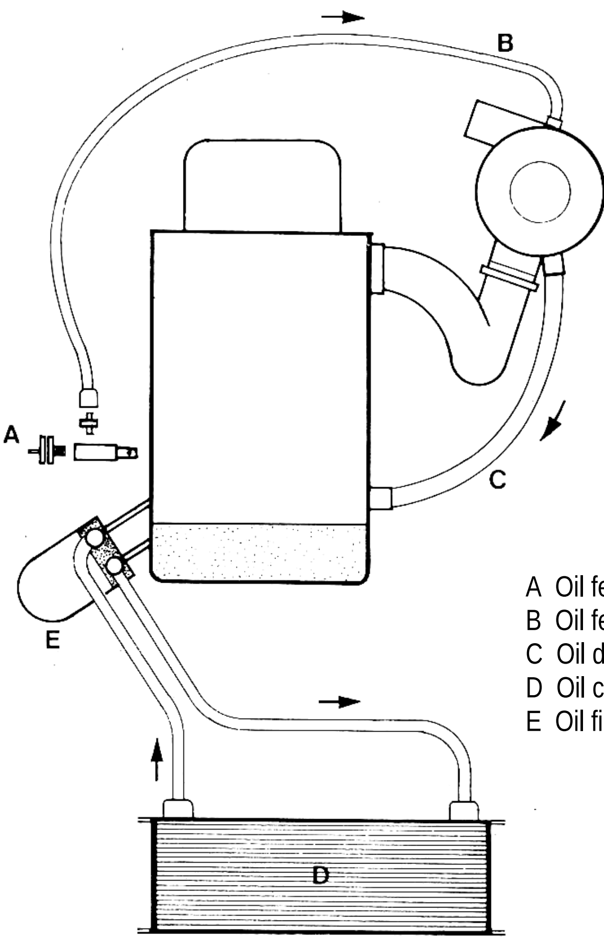
A fuel injection system is designed for a given engine. So, to turbo charge a naturally aspirated engine, alterations must be done to the stock fuel injection system to permit increasing fuel flow as the boost pressure rises.

2.4.7 Turbocharger Lubrication

A turbocharger requires a continuous and adequate supply of clean oil to lubricate and cool the bearings that support the turbine and compressor shaft and wheel assembly. According to Allard (1986), even the most lightly loaded turbocharger will be spinning at a speed of not less than 25,000rpm for most of its working life, with a turbine temperature excess of 500°C. It can be readily realized therefore, if the oil supply is inadequate, rapid wear or the destruction of the whole turbocharger unit can be the result.

There are two types of lubricants; synthetic-based and mineral-based. Synthetic lubes are manufactured fluids (not necessary from oil) in which the basic molecular structure is uniform, very consistent and with high-temperature stability. Mineral-based lubes are less expensive and have an inadequate high-temperature stability, which makes they lose their lubrication properties. The turbocharger survives with low oil pressure and flow. It is virtually certain that all engines in production today have enough excess oil-pumping capacity to adequately take on the additional requirement of lubricating the turbocharger. Too much oil pressure may give rise to the problems with the turbocharger oil seals. The turbocharger needs no special filtering requirements, as far as good, stock filtration equipments are concern.

The oil lines feeding the turbocharger must meet the requirements of pressure and temperature (usually twice the maximum oil temperature allowable) and be hydrocarbon-proof. It is usually safe to draw oil from the opening where the low-oil-pressure light is normally connected. Oil that has passed through the turbocharger bearings must be free to drain out quickly by gravity and without any serious restriction. Ideally, the drain line should swoop smoothly downward and arc gently above the oil level in the oil pan without sharp bends (Figure 2.11). The oil cooler is not necessary for street engine if synthetic oil is used.



Key
A Oil feed tee piece
B Oil feed to turbocharger
C Oil drain from turbocharger
D Oil cooler
E Oil filter

Figure 2.11: Typical turbocharger lubrication system.

2.5 Engine Performance Parameters

According to Heywood (1988) in his book, *Internal Combustion Engines Fundamentals*, engine performance is more precisely defined by; the maximum power (or maximum torque) available at each speed within the useful engine operating range and the range of speed and power over which engine operation is satisfactory. In order to study and obtain the performance of an engine, such as power and fuel consumption by experiment, there are associated formulas to be utilized. Heywood (1988) and Pulkrabek (1997) discussed detail on the engine geometrical and operating characteristics. Some basic geometrical and the parameters commonly used to characterize engine operation are discussed in the following sections.

2.5.1 Engine Geometrical Properties

Displacement volume is the volume that displaced by the reciprocating movement of the piston. Displacement volume:

$$V_d = (\pi / 4) B^2 S \qquad \text{for one cylinder} \tag{1}$$

$$V_d = n_c (\pi / 4) B^2 S \qquad \text{for entire engine} \tag{2}$$

Compression ratio is the ratio of the cylinder's clearance volume (or the volume of the combustion chamber) to the displacement volume. Compression ratio:

$$r_c = V_{BDC} / V_{TDC} = (V_{cl} + V_d) / V_{cl} \tag{3}$$

where: B = cylinder bore [cm]

S = stroke [cm]

n_c = number of engine cylinders

V_{cl} = clearance volume [cm^3]

2.5.2 Mean Effective Pressure

Pressure inside the cylinder of an engine is continuously changing during the cycle. An average or mean effective pressure (mep) is defined by;

$$mep = W / V_d \tag{4}$$

where: mep = mean effective pressure [N/m^2]
W = work of one cycle [Nm]

Work generated inside the combustion chamber is called indicated work; W_i. Thus, indicated mean effective pressure (imep):

$$imep = W_i / V_d \tag{5}$$

Actual work delivered at the crankshaft is called brake work, W_b. It is less than indicated work due to mechanical friction and parasitic loads of the engine. Thus, brake means effective pressure (bmep);

$$bmep = W_b / V_d \tag{6}$$

2.5.3 Mechanical Efficiency

$$\eta_m = bmep / imep \tag{7}$$

Mechanical efficiency will be on the order of 75% to 95% at wide-open throttle. It then decreases with engine speed to zero at idle, when no work is taken off from the crankshaft.

2.5.4 Volumetric Efficiency

$$\eta_v = n\dot{m}_a / \rho_a V_d N \tag{8}$$

where: $\dot{m}_a$ = steady state air flow into the engine [kg/s]

ρ_a = air density at atmospheric conditions outside the engine [kg/m^3]

N = engine speed [rpm]

n = number of revolutions per cycle

Typically, volumetric efficiency for an engine at wide-open throttle is in the range of 75% to 90%, going down much lower as the throttle is closed. Closing the throttle is the primary means of power control for a gasoline engine.

2.5.5 Thermal Efficiency

$$\eta_t = p / \dot{m}_f Q_{HV} \eta_c \tag{9}$$

where: Q_{HV} = heating value of fuel [kJ/kg]

η_c = combustion efficiency (usually 95% ~ 98%)

$$\eta_m = (\eta_t)_b / (\eta_t)_i \tag{10}$$

where: $(\eta_t)_b$ = brake thermal efficiency

$(\eta_t)_I$ = indicated thermal efficiency

2.5.6 Engine Torque

Torque is a good indicator of an engine's ability to do work. It is a turning-effort about the crankshaft's axis of rotation and is equal to the product of the force acting along the connecting rod and the perpendicular distance between this force and the center of the crankshaft rotation. It is expressed in N-m or lbf-ft.

$$T = Fr \tag{11}$$

where: T = engine torque [Nm]

F = force applied to crank [N]

r = effective crank-arm radius [m]

Torque is related to work by:

$$2\pi T = W_b = (bmep)V_d / n \tag{12}$$

where: W_b = brake work of one revolution [Nm]

V_d = displacement volume [m^3]

n = number of revolutions per cycle

For a four-stroke cycle engine which takes two revolutions per cycle:

$$T = (bmep)V_d / 4\pi \qquad \text{four-stroke cycle} \tag{13}$$

In these equations, bmep and brake work W_b are used because torque is measured off the output crankshaft.

The point of maximum torque is called maximum brake torque speed (MBT). A major goal in the design of a modern automobile engine is to *flatten* the torque-versus-speed curve and to have high torque at both high and low speed.

2.5.7 Engine Power

Power is defined as the rate of work of the engine. The actual power developed inside the cylinder, is called indicated power (i.p):

$$i.p = [(imep)\ S\ A\ N\ n_c] / 60 \qquad (14)$$

where: $i.p$ = indicated power [W]
A = cross-sectional area of piston [m^2]
N = engine speed [rpm]

The output power measured at the crankshaft is called brake power (b.p):

$$b.p = 2\pi\ T\ N / 60 \qquad (15)$$

Both torque and power are functions of engine speed. At low speed, torque increases as engine speed increases. As engine speed increases further, torque reaches a maximum and then decreases because the engine is unable to ingest a full charge of air at higher speeds. Indicated power increases with speed, while brake power increases to a maximum and then decreases at higher speeds. This is because friction losses increase with speed and become the dominant factor at very high speeds.

2.5.8 Specific Fuel Consumption

Specific fuel consumption is defined by:

$$sfc = \dot{m}_f \,/\, p \tag{16}$$

where: $\dot{m}_f$ = rate of fuel flow into engine [gm/hr]

p = engine power [kW]

Brake power gives brake specific fuel consumption:

$$bsfc = \dot{m}_f \,/\, b.p \tag{17}$$

Brakes specific fuel consumption decreases as engines speed increases, reaches a minimum, and then increases at high speeds. Fuel consumption increases at high speed because of greater friction losses. At low engine speed, the longer time per cycle allows more heat loss and fuel consumption goes up. It decreases with higher compression ratio due to higher thermal efficiency. It is lowest when combustion occurs in a mixture with a fuel equivalence ratio near one, ($\phi = 1$). The further from stoichiometric combustion, either rich or lean, the higher will be the fuel consumption. Sfc is generally given in units of gm/kW-hr or lbm/hp-hr.

2.5.9 Air/Fuel Ratio

For combustion reaction to occur, the proper relative amounts of air (oxygen) and fuel must be present. Air/fuel ratio (AF) and fuel/air ratio (FA) are parameters used to describe the mixture ratio:

$$AF = m_a / m_f = \dot{m}_a / \dot{m}_f \tag{18}$$

$$FA = 1/AF \tag{19}$$

where: m_a = mass of air

m_f = mass of fuel

$\dot{m}_a$ = mass flow rate of air

$\dot{m}_f$ = mass flow rate of fuel

The ideal or stoichiometric AF for gasoline is about 15:1 with combustion possible in the range of 6:1 to 19:1.

Equivalence ratio ϕ is defined as the ratio of actual AF to stoichiometric AF:

$$\phi = (AF)_{stoich} / (AF)_{actual} = (FA)_{actual} / (FA)_{stoich} \tag{20}$$

when: $\phi < 1$ engine running lean, O_2 in the exhaust

$\phi > 1$ engine running rich, CO and fuel in exhaust

$\phi = 1$ stoichiometric, maximum energy released from fuel

2.6 Emissions

The three main gasoline engine exhaust emissions that must be controlled are oxides of nitrogen (NO_x), carbon monoxide (CO) and hydrocarbons (HC).

2.6.1 Oxides of Nitrogen (NO_x)

Nitrogen oxides (NO_x) are formed throughout the combustion chamber during the combustion process due to the reaction of atomic oxygen and nitrogen. The reactions forming NO_x are very temperature dependent, so the NO_x emissions from an engine scale proportionally to the engine load, and are relatively low during engine start and warm-up. In spark ignition (SI) engines, the dominant component of NO_x is nitric oxide (NO). Some experimental results are presented to illustrate how nitric oxides in the exhaust depend on various engine parameters. Although not universal, for homogenous-charge, SI engines lead to the following observations:

- For lean mixtures, NO_x emissions are strongly depends on spark timing and inlet pressure.
- NO are maximum for slightly lean mixtures.
- Increased coolant temperature or the presence of deposits reduce heat transfer efficiency thus increases the NO.
- Dilution of the intake charge by exhaust gas recirculation (EGR) or by moisture in the inlet air, reduces the NO. Released NO_X reacts in the atmosphere to form ozone and is one of the major causes of photochemical smog.

2.6.2 Carbon Monoxide (CO)

Carbon monoxide, a colourless, odourless, poisonous gas, is generated in an engine when it is operated with a fuel-rich equivalence ratio. When there is not enough oxygen to convert all carbon to CO_2, some fuel does not get burned and some carbon ends up as CO. Typically the exhaust of an SI engine will be about 0.2% to 5% carbon monoxide. Not only CO is considered an undesirable emission, but it also represents lost chemical energy that was not fully utilized is the engine. CO is a fuel that can be combusted to supply additional thermal energy . Maximum CO is generated when an engine runs rich, such as when starting or when accelerating under load. Even when the intake air-fuel mixture is stoichiometric or lean, some CO will be generated in the engine. Poor mixing, local rich regions, and incomplete combustion will create some CO. A well-designed SI engine operating under ideal conditions can have an exhaust mole fraction of CO as low as 10^{-3}.

2.6.3 Hydrocarbons (HC)

Exhaust gas leaving the combustion chamber of an SI engine contain up to 6000 ppm of hydrocarbon components, the equivalent of 1-1.5% of the fuel. About 40% of this is unburned gasoline fuel components. The other 60% consists of partially reacted components that were not present in the original fuel. These consist of small non-equilibrium molecules, which are formed when large fuel molecules break up (thermal cracking) during the combustion reaction. HC emissions are greatest during engine start and warm-up, due to decreased fuel vaporization and oxidation. The makeup of HC emissions will be different for each gasoline blend, depending on the original fuel components. Combustion chamber geometry such as the crevices in the combustion chamber, and engine operating parameters also influence the HC component spectrum. When hydrocarbon emissions get into the atmosphere, they act irritants and odorants; some are carcinogenic. All components except CH_4 react with atmospheric gases to form photochemical smog.

2.7 Engine Performance Test

To study the performance of an engine, one must control the speed of the engine and the load applied to it. One also needs to instrument the engine to determine the value of parameters such as the engine torque, engine speed, fuel flow rate, airflow rate, emissions, cylinder pressure, coolant temperature, oil temperature and spark or fuel injection timing. Some measurements are rather straightforward and some of the measurements require analysis to obtain the desired result. The general instrumentation required to perform engine performance or combustion studies are discussed below:

2.7.1 Dynamometers

The dynamometer is a device used to resist the rotation of the engine shaft, thus provides an external load to the engine. It also absorbs the power produced by the engine. Figure 2.12 illustrates the engine-dynamometer installation.

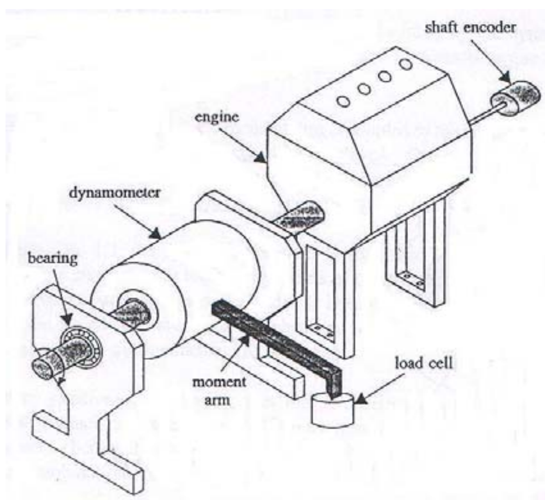


Figure 2.12. Engine-dynamometer arrangements.

The classification of dynamometers and their brief explanations are as below:

Friction brake dynamometer **–** the earliest type of dynamometer that used mechanical friction to absorb the engine power, hence the power absorbed was called the “brake horsepower”.

Electrical dynamometer **–** the rotation of the dynamometer shaft drives the electrical generator (electromagnetic field principle). The strength of the electromagnetic field can be adjusted almost instantly in order to increase or decrease the resistance offered to the engine rotation. With electrical dynamometers, the engine power is converted to electrical energy that can be transferred away by cables to power another system. Alternatively, the electrical energy can be dissipated within the dynamometer as heat and carried away by the cooling water. Electrical dynamometer includes direct current, regenerative alternating current and eddy-current types.

***Hydraulic dynamometer* –** the commonly used dynamometer, also known as “water brake”. It is constructed of a vaned rotor mounted in a casing. The rotor shaft then coupled to the rotating engine shaft. A continuous flow of water is maintained through the casing. The power absorbed by the rotor is dissipated in fluid friction as the rotor shears through the water. Adjusting the water level in the casing varies the torque absorbed by the dynamometer.

The method most commonly used to measure torque is shown in Figure 2.13. The dynamometer is supported by trunnion bearings and restrained from rotation only by a strut connected to a load cell. When the dynamometer is absorbing power, a reaction torque is applied to it. Hence, if the force applied by the strut is F and R_O as defined in Figure 2.13, then the torque applied to the engine is; $T = FR_O$

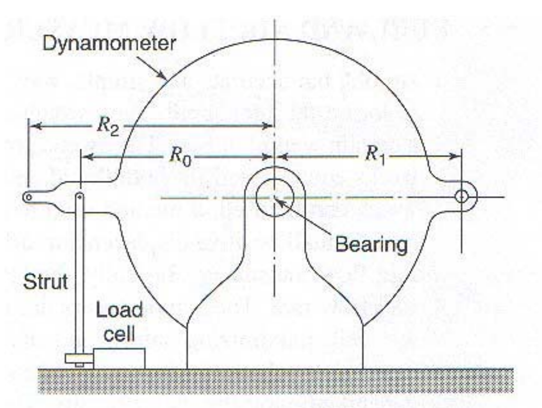


Figure 2.13. Torque measurement using cradle mounted dynamometer.

Most research engines are tested under one or a combination of the three dynamometer operating modes;

***Constant engine speed mode* –** In this mode, the objective is to always keep the engine speed constant while the engine is being tested at different torque output levels.

***Constant torque mode* –** In this mode, the objective is to keep the torque developed by the engine always constant, while the engine is being tested at different throttle openings.

***Constant throttle opening mode* –** In this mode, the objective is to keep the engine throttle opening constant while the engine is tested at different combinations of torque and speed. This test mode is useful for obtaining full-throttle torque/speed curves for an engine (also called maximum power curves).

2.7.2 Engine Speed

The engine speed is measured with optical or electrical techniques. One optical technique uses a disk with holes mounted on the revolving engine shaft. A light emitting diode is mounted on one side of the disk and a phototransistor mounted on the other side. Each time a hole on the disk passes by the optical sensor, a pulse of light impinges on the phototransistor, which generates a periodic signal, the frequency of which is proportional to engine speed. Another method used to determine the engine speed is by using a notch in the flywheel and an electromagnetic sensor which induced voltage where it varies with the change in the magnetic flux as the notch passes the sensor.

2.7.3 Air Flow Measurement

Often, researchers are interested in the average air flow rate (quasi-steady) over the entire engine cycle rather than instantaneous air flow rate. The simplest method in quasi-steady air flow rate measurement is the air box, as shown in Figure 2.14 . The air drawn from the box by the engine is replenished by atmospheric air which enters the box through a calibrated sharp-edged orifice plate or a venturi. The volume of the air box must be sufficiently large to damp out the air pulsation generated by the engine, so that the air flow through the orifice plate is as steady as possible. Designs of suitable orifice plates and the required volume of air box can be obtained in BS 1042 (1964) and in Hua Zhao (2001) respectively.

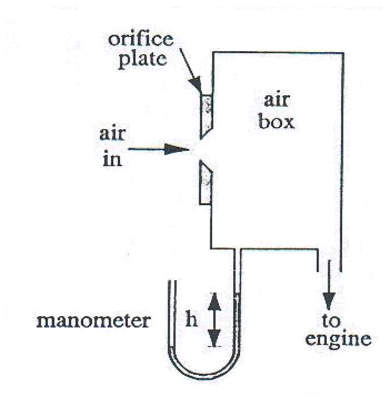


Figure 2.14. Air flow measurement device.

2.7.4 Fuel Flow Measurement

There are a number of methods to measure fuel consumption rate. Different methods may be used to measure the average fuel consumption and the instantaneous flow rate. The common and simple measuring system of average fuel consumption is to record the time taken (usually 2 to 3 minutes) for the engine to consume a certain volume of fuel. To convert this volumetric measurement to a gravimetric rate, the fuel density must be known. There are also methods that measure the fuel mass consumption (gravimetric fuel consumption) directly rather than the volume consumption as shown in Figure 2.14. These methods can be automated using electronic controlled force transducer or optoelectronic liquid-level sensor, valves and timer. The suitable methods for instantaneous flow rate measurement are the “Flowtron” meter and Coriolis flow meter. The “Flowtron” meter is the hydraulic equivalent of the Wheatstone bridge circuit that consists of four orifices and a recirculation pump. The Coriolis meter measures the magnitude of the twist of a vibrating U-tube (as a result of fuel flow through it), which is proportional to the flow rate.

2.7.5 Intake Air Pressure

The intake air static pressure can be measured by U-tube or inclined manometer. The measurement shall be carried out downstream the air filter at a distance as specified by whatever test standard used in the testing. The intake manifold pressure may be obtained by tapping a pressure gauge at the engine’s vacuum line.

2.7.6 Temperatures

The simplest method to acquire the temperature value is by the use of the thermocouples. When two different types of conductors wound together, they form a closed-loop electrical circuit that permits current flow if there is any temperature difference between the junctions. The voltage generated then was converted to the temperature values and displayed by the thermocouple scanner.

2.7.7 Exhaust Gas Analyzer

Measurements of exhaust gas emissions are needed not only because of legislation, but also because of the insight that they provide into the combustion processes. The exhaust gases that are currently legislated for gasoline engines include CO, HCs and NO_x. In addition, O_2 and CO_2 are also measured to determine the air/fuel ratio. In most engine research laboratories, exhaust emissions measurements are often carried out on an engine test bench. Exhaust gas is sampled directly from the exhaust systems and analyzed without dilution.

***CO and CO_2 analyzer* –** Infrared analyzers are used for measuring concentrations of CO and CO_2. Their principle of operation is based on the absorption of infra red radiation by these gases.

***NO_x analyzer* –** NO_x is measured by the chemiluminescence's technique. Chemiluminescence's is the light emission from an atom or molecule that is electronically excited by a chemical reaction.

***Unburned HCs* –** Infra red analyzers can also be used to measure the unburned HCs in the exhaust but with certain limitation. The preferred technique for the total unburned HC measurement is the flame ionization detector (FID).

***O_2 sensor* –** Measurement of O_2 concentration in the exhaust is used to evaluate the air/fuel ratio of the in-cylinder charge. Devices commonly used are the Galvanic Oxygen sensor, lambda sensor, Lambda scan Air/Fuel Ratio Analyzer, Universal Exhaust Gas Oxygen sensor and Paramagnetic Oxygen Analyzer.

2.7.8 Engine Performance Test Standards

There are a number of engine power test codes or standards that may be followed in performing engine performance tests;

BS 5514 Reciprocating internal combustion engines

ISO 3046 Reciprocating internal combustion engines

ISO 1585 Road vehicles – Engine test code – Net power

ISO 2534 Road vehicles – Engine test code – Gross power

SAE J1995 Engine power test code – Spark ignition and compression ignition – Gross power rating

SAE J1349 Engine power test code – Spark ignition and compression ignition – Net power rating

JIS D1001 (1982) Engine power test code for Road Vehicles

CHAPTER III

METHODOLOGY

3.1 Apparatus and Instrumentation

Complete apparatus with instrumentations installed to the gasoline engine that was coupled to the hydraulic dynamometer were utilized in this experiment. The specifications of the engine, the dynamometer and all the instrumentations are as below:

a) *Engine*

Model:	Proton 4G92
Valve train:	16V SOHC
No. of cylinders :	4
Bore x Stroke (mm):	81 x 77.5
Total displacement (cc):	1597
Compression ratio:	10:1
Fuel system:	Multi-point injection (MPI)
Cycle type:	4-stroke

b) *Dynamometer*

Type:	Hydraulic Brake
Model:	SAJ AWM50 (80kW)
Coupling:	Direct coupling through universal joint
Controller:	SAJ Electrical
Arm length:	29cm

c) *Gas Analyzer*

Make:	Techno test
Model:	481 Multigas
Type:	Nondispersive infra red

d) *Fuel Flow rate Meter*

Make:	Ono Sokki
Model:	DF-2420
Sensors:	Pressure, Temperature, Flow

e) *Thermocouple Scanner*

Make:	Cole-Parmer
Channels:	12-channel
Thermocouple:	Type K (-190 ~ 1260°C)

f) *Pressure Gauge*

Make:	Auto Gauge
Type:	Bourdon Tube
Pressure range:	-30 ~ 20psi

g) *Air Tank*

Type:	Air box with sharp-edged orifice plate and manometer

h) *Fuel*

Type:	Gasoline
Density:	718.96 kg/m^3
Low Heating Value:	43 000 kJ/kg

3.2 Test Bed Setup

The engine was coupled to the dynamometer to absorb the engine power and at the same time, to measure the torque produced by the engine or to apply loads to the engine. Temperature measurements at several points such as engine coolant outlet and inlet, intake manifold, exhaust and lubrication oil were done by plugging in the K-type thermocouple to the thermocouple scanner. In order to determine the fuel consumptions of the engine tested, the digital fuel flow meter was utilized. Pressure difference between the air tank and ambient air pressure was measured using U-tube manometer while the gas analyzer was used to measure the exhaust gas emissions. Figure 3.1 shows the engine test bed photograph and Figure 3.2 illustrates the schematic diagram of the test bed setup.

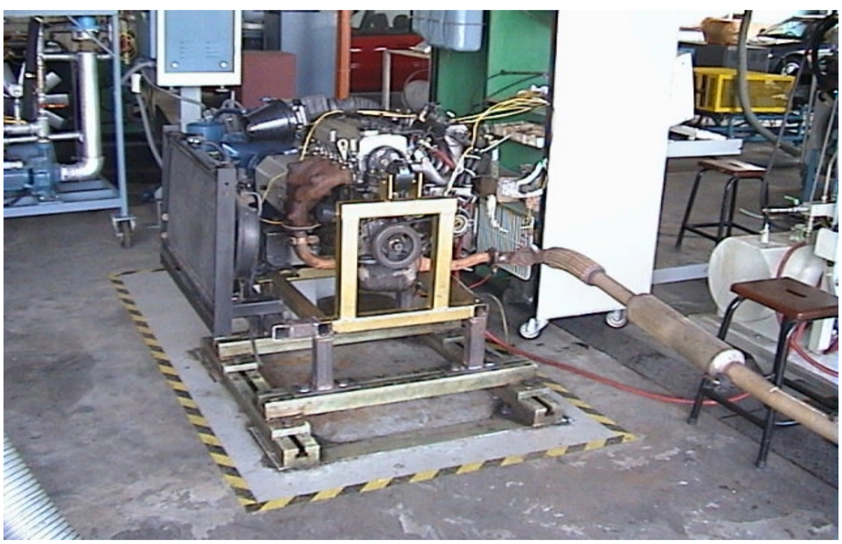
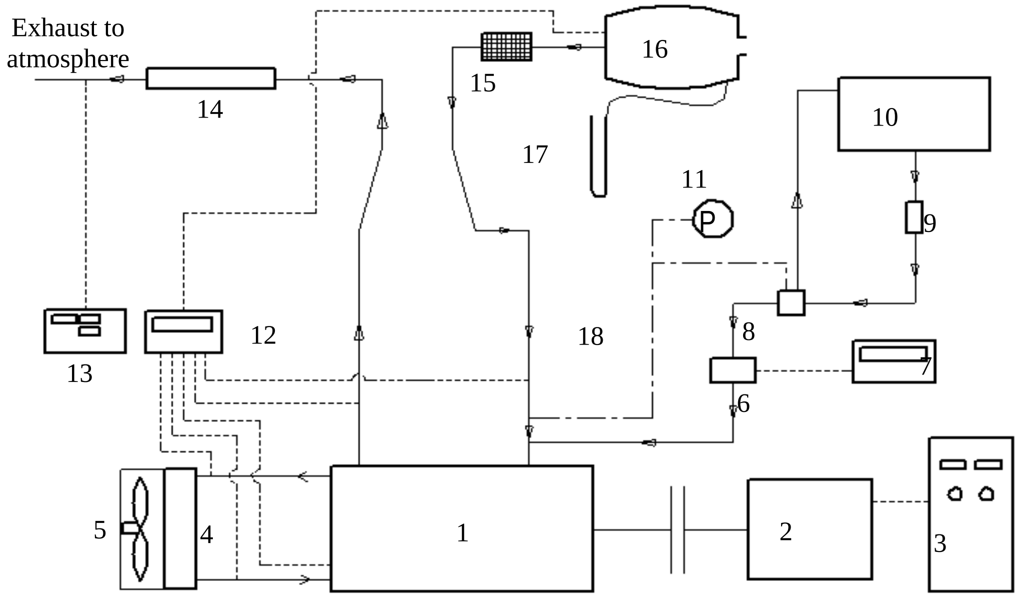
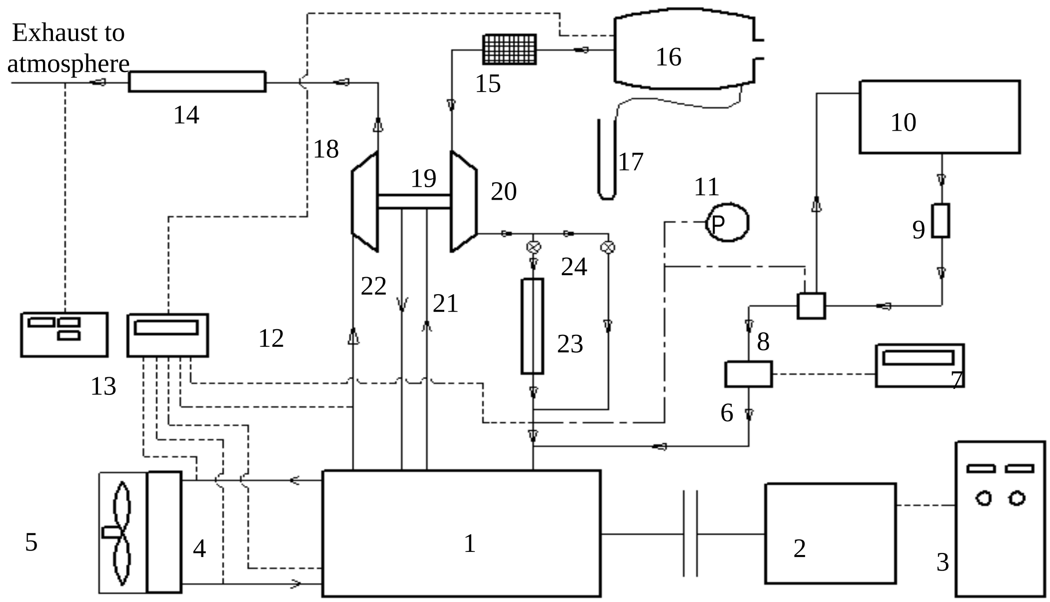


Figure 3.1. Engine test bed



1. Test engine
2. Hydraulic dynamometer
3. Dynamometer controller
4. Radiator
5. Fan
6. Fuel flow meter
7. Fuel flow meter controller
8. Fuel pressure regulator
9. Fuel filter
10. Fuel tank
11. Pressure gauge
12. Thermocouple scanner
13. Gas analyzer
14. Exhaust tailpipe
15. Air filter
16. Air tank
17. U-tube manometer
18. Vacuum line

Figure 3.2: Schematic arrangement of the naturally aspirated engine test bed setup.



1. Test engine
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8. Fuel pressure regulator
9. Fuel filter
10. Fuel tank
11. Pressure gauge
12. Thermocouple scanner
13. Gas analyzer
14. Exhaust tailpipe
15. Air filter
16. Air tank
17. U-tube manometer
18. Turbine
19. Bearing housing
20. Compressor
21. Oil feed line
22. Oil return line
23. Intercooler
24. Valve

Figure 3.3: Schematic arrangement of the turbocharged engine test bed setup.

3.2.1 General inspections before start-up.

1.0 Engine/dynamometer alignment is within the set limits and shaft bolts are tightened to correct torque.

1.0 Shaft guard in place and centred so that no contact with shaft is possible.

2.0 Rock the engine mounts to see the rigging system, including exhaust tubing, is secure and flexes correctly.

3.0 All loose bolts, tools, etc removed from the test bed.

4.0 Engine support system tightened down.

5.0 Fuel systems are connected and leak proof.

6.0 Engine oil at the correct level and engine coolant fluid are sufficient.

7.0 Dynamometer water and exhaust fan are on.

3.2.2 Instruments preparation.

1. The exhaust fan and cooling tower controller switches were turned on.
2. The isolator, cooling tower pump and the AWM50 dynamometer switch at the control panel were turned on.
3. The Dynamometer Controller, thermocouple scanner and fuel flow rate meter were turned on.
4. The gas analyzer was turned on to let it warm-up for 10 minutes.

3.2.3. Engine warm-up

1. The engine was started and allowed to warm-up in idle condition for about 15 minutes. The dynamometer was checked to assure that no load is applied to the engine.

2. Engine fuel, oil leaks, cable or pipes chafing or being blown against the exhaust system were looked for and abnormal noises were listened to.

3.2.4 Engine Testing – Constant speed, WOT

1.0 The throttle opening increased until the engine speed reaches 1500 rpm.

2.0 The engine is allowed to stabilize before loading it by pushing the ‘load’ button at the dynamometer controller.

3.0 To maintain the engine speed at 1500 rpm after the brake load applied, the throttle opening was increased further.

4.0 Procedure (1) – (3) are repeated until the throttle is fully open.

5.0 The brake load applied to the engine was stopped when the throttle is fully open.

6.0 Measurement of torque, temperature, exhaust emissions and fuel consumption were recorded in Table 4.1 after allowing stable running of the engine at this condition for 1 minute (JIS Standard – Engine Power Test Code).

7.0 Procedures (1) – (6) were repeated for the engine speeds of up to 5000 rpm with 500-rpm intervals.

3.2.5. Checks immediately after shut-down

1.0 The exhaust fans were left on to allow for engine cooling period.

2.0 The load applied by the dynamometer was checked at zero (fully unload).

3.0 The dynamometer water was left running for cooling period.

4.0 The external fuel system was shut off.

5.0 Data saving and checking was carried out.

3.3 Conversion of Proton 4G92 1.6L Natural Aspirated To Turbo Charged Operation

In order to prepare this engine to be turbocharged, there were some modifications done. The turbo manifold (Fig. 3.4) directs the exhaust gases from all four cylinders to the turbine inlet port and supports the turbo itself. A cast iron turbo manifold was installed rather than a fabricated one because it is comparatively stronger and retains heat, which is good for the turbine performance.

The turbocharger fitted to the engine is the Mitsubishi Heavy Industries (MHI) TD05H-16G with internal waste gate (Fig. 3.4). The waste gate is an exhaust gas bypass valve built into the turbocharger assembly that allows only enough exhaust gas flow to the turbine to produce the desired boost. It is because the turbo's characteristic is to increase its airflow rate faster than the engine's ability to accept that flow, and if unchecked, the turbo can quickly produce damaging high boost pressures that lead to engine knock. A manual waste gate controller was added as a means to gain more control of the boost pressure during the engine-testing period (Fig. 3.6). The expanded exhaust gas then was routed to the exhaust tailpipe via the cast iron turbine discharge pipe (Fig. 3.4).

An air-to-air intercooler was installed between the turbo and the intake manifold (Fig. 3.7). The intercooler is a heat exchanger used to reduce the increased temperature of the charge air due to compressive heating. The use of an intercooler is important for three reasons. First, the reduction of temperature makes the intake charge denser. Higher charge density will allow more mass flow of air into the engine at any given intake-manifold pressure, hence greater quantity of air/fuel mixture can be combusted to produce more power. Second, it reduces the thermal loading on the engine for given boost and power output, as a result of the proportional drop of the exhaust temperature to the intake charge temperature drop. This contributes to a reduction in thermal loading on valves and pistons and reduces the amount of heat that has to be dissipated by the engine. Lastly, the benefit of inter cooling is to the combustion process where detonation is reduced by reduction in the intake temperature.

An air filter was connected to the compressor inlet by a flexible hose (Fig. 3.8). The stock intake manifold (Fig. 3.9) was retained, as this engine is equipped with the multipoint fuel injection system, where the manifold inlet can be connected to the intercooler outlet by a pipe and a flexible hose without any modifications.

The most important aspect to take into account in turbo charging a gasoline engine is its compression ratio. A reduction in compression ratio from that of a similar naturally aspirated engine is an essential feature of a turbocharged engine. A turbocharger compresses air and thus, raises the temperature of the air induced into the engine. The effect is to increase the peak cylinder temperature and approaches the temperature at which detonation commences. Detonation is the uncontrolled burning of the air/fuel mixture, leading to a further rapid increase in cylinder temperature, that will results in a blown cylinder head gasket or piston failure if the process continues for more than a few seconds. A set of low-compression pistons with a dish in the center (Fig. 3.10) was installed in order to reduce the original compression ratio of 10:1 to 7:1. The new pistons were installed complete with new rings and connecting rod bearings. A new metal head gasket (Fig. 3.5) replaced the stock asbestos-based cylinder head gasket. The metal gasket was coated with copper layers, sprayed at both sides.

The turbocharger, which operate at high speeds and temperatures require a continuous and adequate supply of clean lubricating oil to give reliable long-term service. The oil is required to lubricate and cool the bearings, which support the turbine and compressor shaft and wheel assembly. To provide lubrication for the turbocharger on this engine, the oil feed line was screwed into the oil pressure switch tapping in the cylinder head (Fig. 3.11). A metal-braid-protected line was chosen due to its high-pressure stand ability, chafing, abrasion and vibration resistance characteristics. Oil that has passed through the turbo bearings must be free to drain out quickly without any serious restriction. The drain hose installed was larger than the feed line because the high-speed rotation of the turbine shaft causes the oil to pick up air that causes the hot oil to foam. As the oil return is by gravity, the flexible drain hose was routed smoothly downward and arc gently into the oil sump with no kinks, sharp bends or rises. The oil returned to the sump above the oil level in the sump. If not, the foamy oil will build up in the hose and back up into the bearing housing, causing leakage through the oil seals. The method used to attach the oil drain fitting to the oil sump is by brazing a segment of tube into a drilled hole at the sump (Fig. 3.11).

To monitor the intake manifold pressure, a boost gauge was tapped using a tee piece at the vacuum line on the intake manifold. All other engine components were retained including the fuel injectors and the fuel pump. The engine control unit (ECU) was also not reprogrammed and no other additional electronic control system was utilized in this engine. Other components that did not mentioned above were completely retained in stock configurations. Fig. 3.12 shows the engine with the completed installed turbo kit and Fig. 3.13 illustrates the engine coupled to the dynamometer with complete instrumentations on, ready to be tested.

Fig. 3.4: TD05H-16G turbocharger unit with internal waste gate, turbo manifold and turbine discharge pipe.

Fig. 3.5: Turbocharger kit complete with metal cylinder head and exhaust gasket.

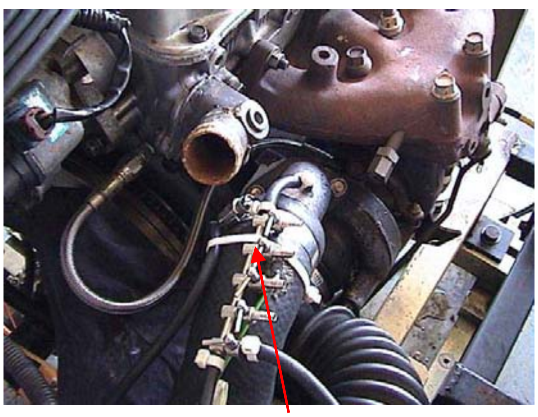

Fig. 3.6: Manual waste gate controller and oil feed line to turbo bearing housing.

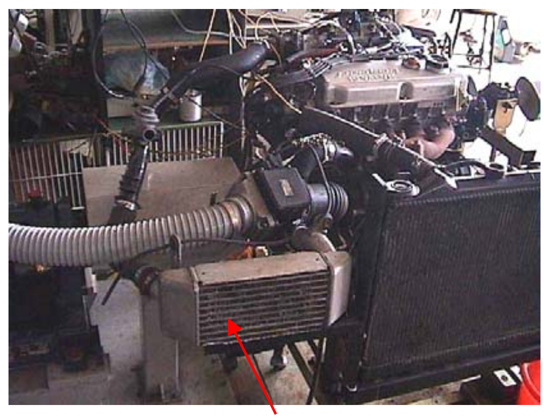

Fig. 3.7: Air-to-air intercooler positioned between compressor outlet and intake manifold.

Fig. 3.8: Air filter and flexible hose with mass flow sensor.

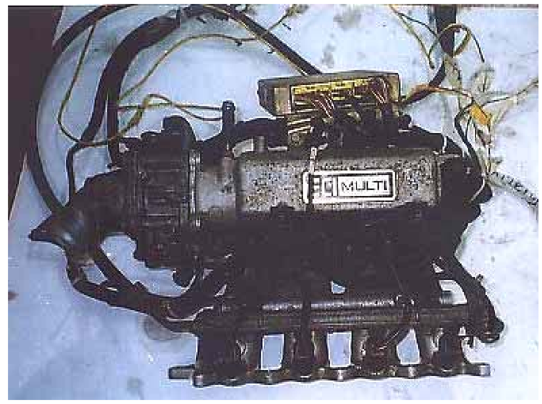

Fig. 3.9: Stock intake manifold shown with stock 4G92 ECU.

Fig. 3.10: Stock high-compression piston (left) and new low-compression turbo piston (right).

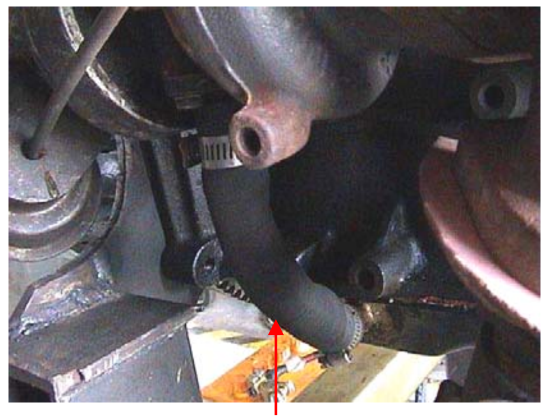

Fig. 3.11: Lube oil drain hose from turbo bearing housing to oil sump.

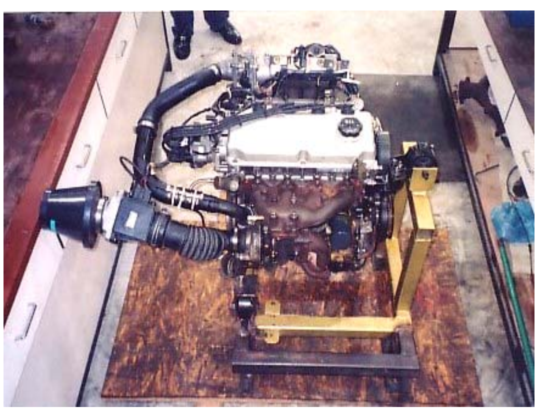

Fig. 3.12: 4G92 engine with complete turbo kit installed.

Fig. 3.13: Engine coupled to dynamometer on test bed complete with instrumentations.

CHAPTER IV

RESULTS

4.1 Test data

The testing done on the engine was to determine the maximum torque produced by the engine, fuel consumption rate, temperatures and emissions of the exhaust gas at the specified speed. This was achieved by setting the throttle valve at fully opened position while applying the dynamometer load. Maximum torque generated by the engine at specified rpm is achieved when the engine was unable to maintain the speed as the load applied further. This type of test is known as the steady-state, full load testing because the data will be acquired after allowing a stable running of the engine for 1 minute.

After the engine operation parameter data were recorded in Table 4.1, the next step is to analyze the performance of the engine. The analysis done based on the performance parameters formulae described previously in section 2.5. The results of the analysis are shown in Table 4.2 and the calculation sample in the analysis also shown below as a reference.

Sample calculation

Performance analysis at 5000rpm:

Data; Speed = 5000rpm

Torque = 111.37 Nm

Fuel consumption = 150.30mL / 25sec

Analysis;

1) Brake power (b.p)

$$b.p = \frac{2\pi NT}{60} = \frac{2\pi \times 5000 \times 111.37}{60} = 58.31kW$$

2) Brake mean effective pressure (bmep)

$$bmep = \frac{4\pi T}{V_d} = \frac{4\pi \times 111.37}{1597 \times 10^{-6}} = 876.34kPa$$

3) Indicated mean effective pressure (imep)

$$imep = \frac{bmep}{\eta_m} = \frac{876.34}{0.85} = 1030.99kPa$$

4) Mass flow rate of fuel ($\dot{m}_f$)

$$\dot{m}_f = \frac{150.30 \times 10^{-3} L}{25s} \times \frac{3600s}{1hr} \times \frac{718.96kg}{1m^3} \times \frac{1m^3}{1000L} = 15.56kg / hr$$

$= 15560.60 gm/hr$
$= 0.00432 kg/s$

5) Brake specific fuel consumption (bsfc)

$$bsfc = \frac{\dot{m}_f}{b.p} = \frac{15560.60}{58.31} = 266.85 gm/kW - hr$$

6) Indicated specific fuel consumption (isfc)

$$isfc = \frac{bsfc}{\eta_m} = \frac{266.85}{0.85} = 313.94 gm/kW - hr$$

7) Brake thermal efficiency $(\eta_t)_b$

$$(\eta_t)_b = \frac{b.p}{\dot{m}_f Q_{HV} \eta_c} = \frac{58.31}{0.00432 \times 43000 \times 0.97} = 0.32$$

8) Indicated thermal efficiency $(\eta_t)_b$

$$(\eta_t)_i = \frac{(\eta_t)_b}{\eta_m} = \frac{0.32}{0.85} = 0.38$$

4.2 Data analysis

The data were recorded in Table 4.1 for the naturally aspirated and Table 4.3 for the turbocharged engine. The analysis results are best presented in a graph form for the ease of comparisons, explanations, evaluations and understanding of the engine performance as a function of the engine speed. The data for the turbocharged engine were plotted at maximum boost condition for each speed. The graphs plotted are shown in Fig. 4.1 to Fig. 4.8.

4.2.1 Comparison tables

After the engine operation parameter data were recorded in Table 4.1, the next step is to analyze the performance of the engine. The results of the analysis are shown in Table 4.2 for the naturally aspirated and Table 4.4 for the turbocharged engine. Table 4.7, Table 4.8 and Table 4.9 shows the comparisons made between the naturally aspirated and the turbocharged engine on engine performance, engine efficiencies and exhaust emissions respectively.

Table 4.1 : Engine Testing Data Table for Natural Aspirated Engine

Title: 4G92 N/A Performance Test - Data (Full load)

Date: 25-Jun-03

Time: 2.30pm - 4.30pm

Test person: Ismail, Regis

Speed	**Torque**	**Fuel cons**		**Manifold**	**Temperature [deg C]**					**Emissions**			
[rpm]	**[Nm]**	**[ml]**	**[sec]**	**press[psi]**	Exhaust	Lube oil	Engine in	Engine out	Inlet mnfld	O2 (% vol)	**CO[%vol]**	**CO2[%vol]**	**HC[ppm]**
1500	87.07	43.45	25	0	578.03	94.70	57.50	87.90	47.10	2.50	1.80	14.10	120
2000	100.07	58.04	25	0	700.8	102.90	77.10	89.20	52.70	2.10	2.08	14.80	90
2500	104.00	73.11	25	0	713.17	109.90	84.30	90.60	50.40	5.30	1.49	12.00	70
3000	105.40	86.21	25	0	755.27	117.20	91.80	96.90	50.40	8.70	1.19	13.90	60
3500	106.70	98.81	25	0	789.85	120.00	94.15	99.10	57.70	5.10	1.50	14.60	50
4000	110.80	113.23	25	0	782.95	113.70	83.00	89.80	50.30	4.20	1.54	15.50	40
4500	111.07	136.27	25	0	833.6	120.20	86.20	93.00	50.00	6.00	1.30	16.40	30
5000	111.37	150.30	25	0	901.45	125.00	87.70	94.50	55.10	4.20	1.73	18.00	20

Table 4.2 : Analysis Results Table for Natural Aspirated Engine

Title: **4G92 N/A Performance Test - Analysis (Full load)**

Additional data used in analysis;

V_d, Displacement volume = 1597 cc

$_{fuel}$, Fuel density = 718.96 kg/m^3

$_c$, Combustion efficiency = 97%

$_m$, Mechanical efficiency = 85%

Q_{HV}, Fuel heating value = 43 000 kJ/kg

Speed, N	**Torque, T**	**Fuel cons.**		**b.p**	**i.p**	**bmep**	**imep**	$\dot{m}_f$	**bsfc**	**isfc**	$\dot{m}_f$	$(\eta_t)_b$	$(\eta_t)_i$
[rpm]	**[Nm]**	**[mL]**	**[sec]**	**[kW]**	**[kW]**	**[kPa]**	**[kPa]**	**[gm/hr]**	**[gm/kW-hr]**	**[gm/kW-hr]**	**[kg/s]**		
1500	87.07	43.45	25	13.68	32.18	685.13	806.04	4498.39	328.90	139.78	0.00125	0.26	0.62
2000	100.07	58.04	25	20.96	49.31	787.42	926.38	6008.90	286.70	121.85	0.00167	0.30	0.71
2500	104.00	73.11	25	27.23	64.06	818.35	962.76	7569.10	278.00	118.15	0.00210	0.31	0.73
3000	105.40	86.21	25	33.11	77.91	829.36	975.72	8925.34	269.55	114.56	0.00248	0.32	0.75
3500	106.70	98.81	25	39.11	92.02	839.59	987.76	10229.82	261.58	111.17	0.00284	0.33	0.78
4000	110.80	113.23	25	46.41	109.20	871.86	1025.71	11722.73	252.58	107.35	0.00326	0.34	0.80
4500	111.07	136.27	25	52.34	123.15	873.98	1028.21	14108.07	269.54	114.56	0.00392	0.32	0.75
5000	111.37	150.30	25	58.31	137.21	876.34	1030.99	15560.60	266.85	113.41	0.00432	0.32	0.76

Table 4.3 : Engine Testing Data Table for Turbo charged Engine

Title: 4G92 Turbocharged Performance Test – Non-intercooled Data (Full load)

Date: 30-Jul-03

Time: 10.00am - 4.30pm

Test person: Ismail, Regis, Subki

Speed	**Boost**	**Torque**	**Fuel cons**		**Temperature [deg C]**					**Emissions**			
[rpm]	**[psi]**	**[Nm]**	**[ml]**	**[sec]**	Water jacket	Lube oil	Intake air	Engine out	Engine in	O2[%vol]	CO[%vol]	CO2[%vol]	HC[ppm]
1500	**1**	87.70	45.00	25	87.0	92.0	40.9	90.7	82.3	4.7	0.76	12.9	100
2000	**1**	98.80	62.50	25	90.5	103.8	44.1	91.3	89.9	2.4	0.67	15.3	
	1.5	100.20	62.50	25	85.4	95.2	46.2	87.2	85.8	2.4	0.93	15.1	60
2500	**1**	92.50	73.60	25	95.7	108.5	48.7	97.4	95.4	1.7	0.27	15.5	
	2	110.00	81.90	25	87.5	101.2	50.5	89.5	87.9	2.1	0.66	15.3	40
3000	**1**	105.80	94.90	25	88.7	122.1	50.9	88.8	86.3	1.3	0.38	15.6	
	2	111.30	97.26	25	88.5	111.3	52.2	86.2	84.0	1.5	0.46	15.5	
	3	118.00	104.17	25	84.1	110.1	56.8	85.1	83.1	1.6	0.65	15.7	30
3500	**1**	110.50	115.70	25	93.0	126.2	51.3	95.3	93.1	1.0	0.15	15.3	
	2	116.00	116.13	25	92.7	124.7	55.9	93.0	91.1	1.2	0.34	15.6	
	3	120.00	116.68	25	92.0	120.9	60.4	92.8	90.7	1.7	0.72	15.9	
	4	123.50	116.99	25	92.0	116.1	64.1	91.3	89.3	1.9	1.14	16.3	20

Table 4.4 : Analysis Results Table for Turbocharged Engine

Title: **4G92 Turbocharged Test – Non-intercooled Analysis (Full load)**

Additional data used in analysis;

V_d, Displacement volume = 1597 cc

$_{fuel}$, Fuel density = 718.96 kg/m^3

$_{c}$, Combustion efficiency = 97%

$_{m}$, Mechanical efficiency = 85%

Q_{HV}, Fuel heating value = 43 000 kJ/kg

Speed,N [rpm]	**Boost [psi]**	**Torque,T [Nm]**	**Fuel cons. [mL]**	**[sec]**	**b.p [kW]**	**i.p [kW]**	**bmep [kPa]**	**imep [kPa]**	$\dot{m}_f$ **[gm/hr]**	**bsfc [gm/kW-hr]**	**isfc [gm/kW-hr]**	$\dot{m}_f$ **[kg/s]**	$(\eta_t)_b$	$(\eta_t)_i$
1500	**1**	87.70	45.00	25	13.78	32.41	690.09	811.87	4658.86	338.19	143.73	0.00129	0.26	0.60
2000	**1**	98.80	62.50	25	20.69	48.69	777.43	914.62	6470.64	312.70	132.90	0.00180	0.28	0.65
	1.5	100.20	62.50	25	20.99	49.38	788.45	927.59	6470.64	308.33	131.04	0.00180	0.28	0.66
2500	**1**	92.50	73.60	25	24.22	56.98	727.86	856.30	7619.83	314.66	133.73	0.00212	0.27	0.65
	2	110.00	81.90	25	28.80	67.76	865.56	1018.31	8479.13	294.44	125.13	0.00236	0.29	0.69
3000	**1**	105.80	94.90	25	33.24	78.21	832.51	979.43	9825.02	295.60	125.63	0.00273	0.29	0.69
	2	111.30	97.26	25	34.97	82.27	875.79	1030.34	10069.35	287.98	122.39	0.00280	0.30	0.71
	3	118.00	104.17	25	37.07	87.23	928.51	1092.37	10784.75	290.92	123.64	0.00300	0.30	0.70
3500	**1**	110.50	115.70	25	40.50	95.29	869.50	1022.94	11978.45	295.76	125.70	0.00333	0.29	0.69
	2	116.00	116.13	25	42.52	100.04	912.77	1073.85	12022.97	282.79	120.18	0.00334	0.31	0.72
	3	120.00	116.68	25	43.98	103.49	944.25	1110.88	12079.91	274.65	116.73	0.00336	0.31	0.74
	4	123.50	116.99	25	45.27	106.51	971.79	1143.28	12112.00	267.58	113.72	0.00336	0.32	0.76

Table 4.5 : Engine Testing Data Table for Turbocharged Engine

Title: 4G92 Turbocharged Test; Intercooled Data (Full load)

Date: 14.8.03

Time: 11.00am - 4.30pm

Test person: Ismail, Regis

Speed	Boost	Torque	Fuel cons		Temperature [deg C]						h	Emissions			
[rpm]	[psi]	[Nm]	[ml]	[sec]	Water Jacket	Lube oil	Engine out	Engine in	Intake air	Exhaust	[mm H2O]	O2[%vol]	CO[%vol]	CO2[%vol]	HC[ppm]
3000	**1**	109.2	93.52	25	86.9	116.4	89.1	81.3	39.6	-	24	2.7	0.78	14.5	90
	2	113.6	103.98	25	87.2	111.1	89.8	82.8	40.1	-	29	2.6	1.69	13.6	90
	2.5	116	105	25	85.2	107.9	90.3	83.6	42.7	-	34	2.6	1.26	14.1	80
3500	**1**	112.8	106.9	25	88	119.1	91.7	85.4	42.2	754.3	32	2.5	1.05	14.5	60
	2	117.4	113.64	25	88.3	116.2	91.7	86.3	42.9	775.1	37	2.4	1.65	14	70
	3	122	115.63	25	89.2	114.2	92.3	86.4	43	788.7	45	2.5	1.63	13.5	70

Table 4.6 : Analysis Results Table for Turbocharged Engine

Title: **4G92 Turbocharged Test; Intercooled Analysis - Full load**

Additional data used in analysis;

V_d, Displacement volume = 1597 cc

$_{fuel}$, Fuel density = 718.96 kg/m^3

$_{c}$, Combustion efficiency = 97%

$_{m}$, Mechanical efficiency = 85%

Q_{HV}, Fuel heating value = 43 000 kJ/kg

Speed,N	**Boost**	**Torque,T**	**Fuel cons.**		**b.p**	**i.p**	**bmep**	**imep**	$\dot{m}_f$	**bsfc**	**isfc**	$\dot{m}_f$	$(\eta_t)_b$	$(\eta_t)_i$
[rpm]	**[psi]**	**[Nm]**	**[mL]**	**[sec]**	**[kW]**	**[kW]**	**[kPa]**	**[kPa]**	**[gm/hr]**	**[gm/kW-hr]**	**[gm/kW-hr]**	**[kg/s]**		
3000	**1**	109.20	93.52	25	34.31	80.72	859.27	1010.90	9682.15	282.23	119.95	0.00269	0.31	0.72
	2	113.60	96.31	25	35.69	83.97	893.89	1051.63	9971.00	279.39	118.74	0.00277	0.31	0.73
	2.5	116.00	103.98	25	36.44	85.75	912.77	1073.85	10765.07	295.40	125.54	0.00299	0.29	0.69
3500	**1**	112.80	106.90	25	41.34	97.28	887.59	1044.23	11067.38	267.69	113.77	0.00307	0.32	0.76
	2	117.40	113.64	25	43.03	101.25	923.79	1086.81	11765.18	273.42	116.20	0.00327	0.32	0.74
	3	122.00	115.63	25	44.72	105.21	959.99	1129.39	11971.20	267.72	113.78	0.00333	0.32	0.76

Table 4.7 : Performance Comparison of Naturally Aspirated and Turbocharged Proton 4G92 1.6L Engine

Speed (rpm)	Boost (psi)	Brake Power, b.p (kW)	% b.p to NA (baseline)	Bsfc (gm/kW-hr)	% bsfc to NA (baseline)
1500 (NA)	0	13.68	-	328.90	-
1500 (T)	1	13.78	0.7 % ↑	338.19	2.8 % ↑
2000 (NA)	0	20.96	-	286.70	-
2000 (T)	1	20.69	1.3 % ↓	312.70	9.0 % ↑
	1.5	20.99	0.1 % ↑	308.33	7.5 % ↑
2500 (NA)	0	27.23	-	278.00	-
2500 (T)	1	24.22	11.1 % ↓	314.66	13.2 % ↑
	2	28.80	5.8 % ↑	294.44	5.9 % ↑
3000 (NA)	0	33.11	-	261.58	-
3000 (T)	1	33.24	0.4 % ↑	295.60	9.7 % ↑
	2	34.97	5.6 % ↑	287.98	6.8 % ↑
	3	37.07	12.0 % ↑	290.92	7.9 % ↑
3500 [NA]	0	39.11	-	261.58	-
3500 [T]	1	40.50	3.6 % ↑	295.76	13.0 % ↑
	2	42.52	8.7 % ↑	282.79	8.1 % ↑
	3	43.98	12.5 % ↑	274.65	5.0 % ↑
	4	45.27	15.8 % ↑	267.58	2.3 % ↑

Table 4.8 : Engine Efficiencies Comparison of Naturally Aspirated and Turbocharged Proton 4G92 1.6L Engine @ maximum boost for each speed

Speed (rpm)	**Boost (psi)**	**Volumetric efficiency, η_v**	**% Increase in η_v**	**Brake thermal eff., $(\eta_t)_b$**	**% Decrease in $(\eta_t)_b$**
1500 (NA)	0	0.768	-	0.262	-
1500 (T)	1	0.841	9.5 % ↑	0.255	2.7 % ↓
2000 (NA)	0	0.773	-	0.301	-
2000 (T)	1.5	0.892	15.4 % ↑	0.280	7.0 % ↓
2500 (NA)	0	0.798	-	0.310	-
2500 (T)	2	0.944	18.3 % ↑	0.293	5.5 % ↓
3000 (NA)	0	0.787	-	0.320	-
3000 (T)	3	1.016	29.1 % ↑	0.297	7.2 % ↓
3500 (NA)	0	0.793	-	0.330	-
3500 (T)	4	1.009	27.2 % ↑	0.323	2.1 % ↓

Table 4.9 : Exhaust Emissions Comparison of Naturally Aspirated and Turbocharged Proton 4G92 1.6L Engine @ maximum boost for each speed

Speed (rpm)	Boost (psi)	O_2 (% vol)	CO_2 (% vol)	CO (%vol)	HC (ppm)
1500 (NA)	0	2.5	14.1	1.80	120
1500 (T)	1	4.7	12.9	0.76	100
2000 (NA)	0	2.1	14.8	2.08	90
2000 (T)	1.5	2.4	15.1	0.93	60
2500 (NA)	0	5.3	12.0	1.49	70
2500 (T)	2	2.1	15.3	0.66	40
3000 (NA)	0	8.7	13.9	1.19	60
3000 (T)	3	1.6	15.7	0.65	30
3500 (NA)	0	5.1	14.6	1.50	50
3500 (T)	4	1.9	16.3	1.14	20

4.3 Graph plotting

The analysis results are best presented in a graph form for the ease of explanations, evaluations and understanding of the engine performance as a function of the engine speed. The graphs plotted are as below:

4.3.1 Effect of the engine speed to the engine torque

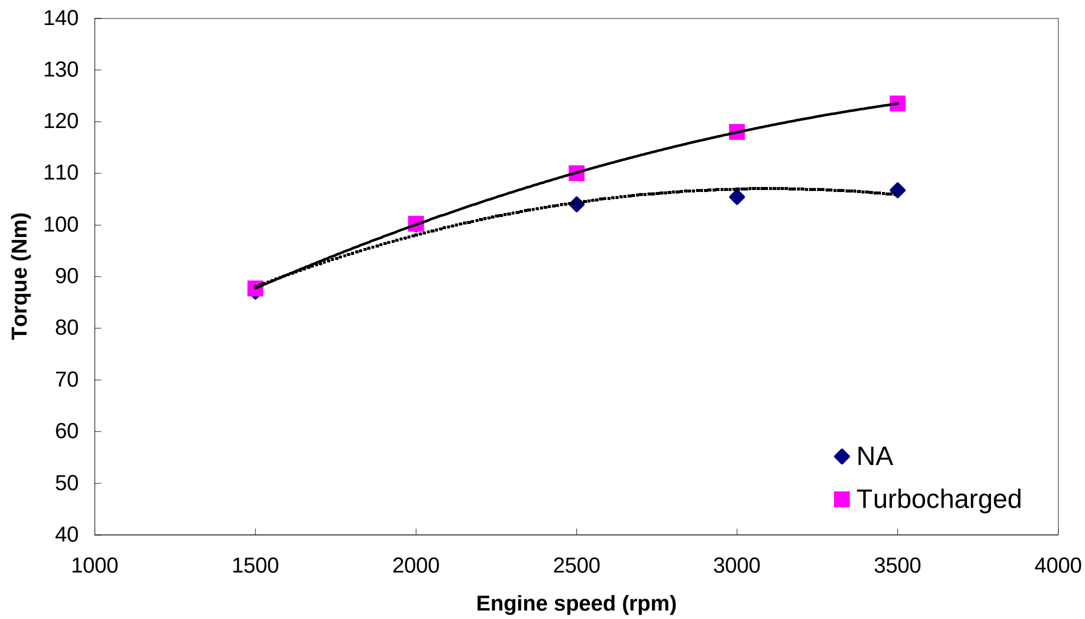

Figure 4.1. Engine torque as a function of engine speed.

Fig. 4.1 clearly illustrates that by the addition of the turbocharger, the same engine was capable of increasing its torque compared to the lower torque curve generated when it is naturally aspirated. Table 4.7 shows that at 3500 rpm, the torque increment is up to approximately 16% at 4 psi of boost pressure. It is 4 psi above the atmospheric pressure. The increased intake pressure, therefore the density of the induced air into the engine results in the increment of mean effective pressure on the pistons and the volumetric efficiency. This increases the engine's ability for doing work; that is the turning effort about the crankshaft's rotation axis compared to the naturally aspirated condition where the maximum pressure at WOT is atmospheric pressure. Fig. 4.1 also illustrates that turbocharger offsets the speed at which maximum torque generated to a higher speed. In naturally aspirated condition, the engine produces peak torque of approximately 105Nm at 3000rpm, whereas by adding the turbocharger the same torque value can be achieved at 2400rpm.

4.3.2 Effect of the engine speed to the brake power

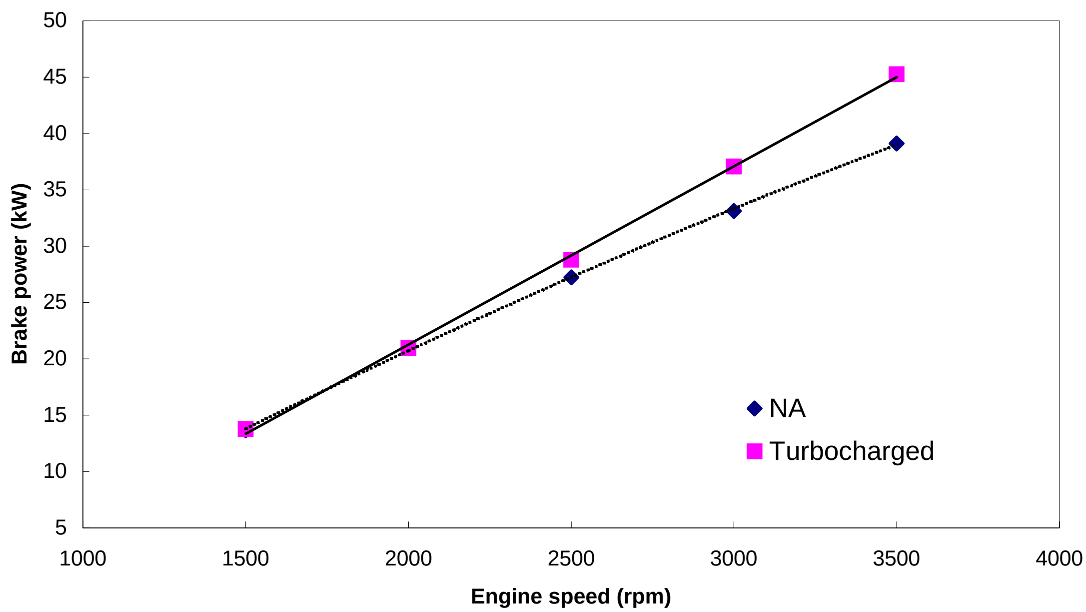

Figure 4.2. Engine brake power as a function of engine speed.

Fig. 4.2 illustrates the difference of brake power between the naturally aspirated and turbocharged condition. The brake power for all the speeds tested shows a direct proportional increment to engine speeds. Power is the product of torque and engine speed. Power keeps increasing although the torque reached maximum due to the higher engine speed. However, the brake power will reach its maximum value and starts to decrease, as the engine torque will be much lower at high speeds. The turbocharger starts producing noticeable power increase at 2000rpm upwards, with higher increments as the engine speed increase. By referring to Table 4.7, the power increase at 2000rpm is 0.1% at maximum boost of 1.5psi and achieved approximately 16% increments at 3500rpm at maximum boost of 4psi. Below 2000rpm, the engine behaves as the naturally aspirated condition as the turbocharger was in an idling condition due to insufficient exhaust gas energy to spin the turbine, hence the compressor fast enough to starts producing a noticeable boost. The turbocharger increases the power without the need to raise the engine speed. For example, from Fig. 4.2, the turbocharged version of the engine produces 37kW at 3000rpm whereas in naturally aspirated condition, this power output can only be generated when the engine is revved to 3400rpm. This results from the fact that the pressurized induction done by the turbocharger did not increase the in-cylinder peak pressure substantially, but *fattens* the overall cylinder pressure indicated curve. The *fatter* curve increased the mean effective pressure and therefore the power output for a particular speed.

4.3.3 Effect of the engine speed to brake mean effective pressure (bmep)

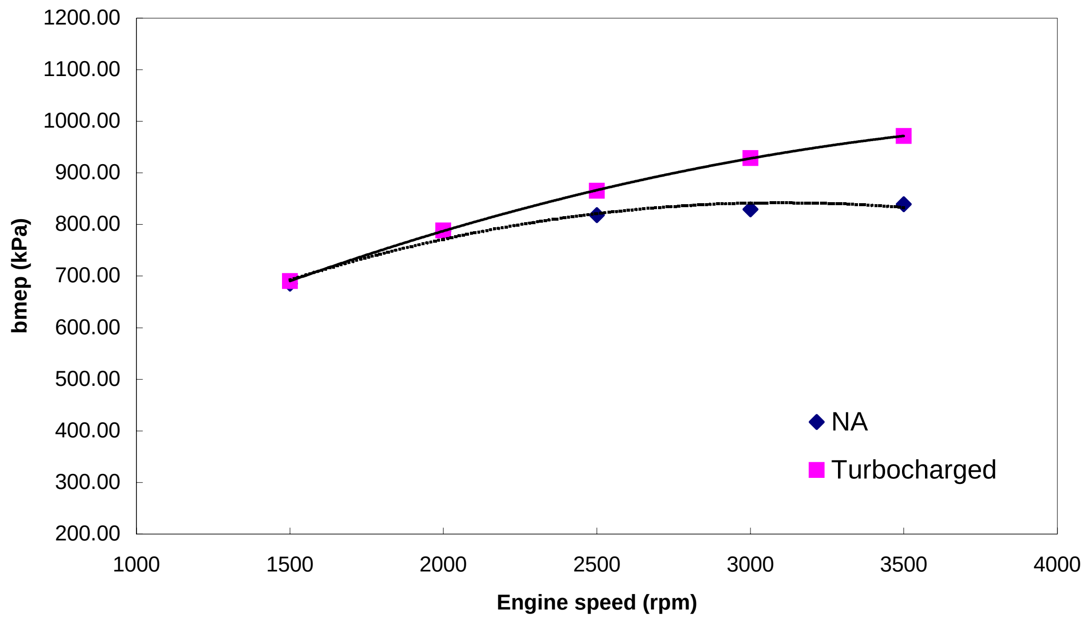

Figure 4.3. Brake mean effective pressure as a function of engine speed.

Fig. 4.3 shows the comparison of bmep between the naturally aspirated and turbocharged condition. The bmep is a measure of the work output for a given swept volume or cylinder capacity. Brake power and torque is the product of the bmep. The trend of the curve in Fig. 4.3 is of similar to the torque curve in Fig. 4.1 because torque is directly proportional to bmep. For the speeds below 2000rpm, the bmep of the turbocharged engine is about the same with the naturally aspirated engine. This is due to the fact that the turbocharger is spinning with inadequate speed of producing a noticeable boost. At 2000rpm, an increase of 0.1% in bmep and it raised to approximately 16% as the boost of 4psi produced at 3500rpm as compared to the naturally aspirated bmep. The intake pressure higher than atmospheric pressure increases all pressures through the cycle and increased air-fuel mixture give greater energy output during the combustion process. In naturally aspirated condition, the engine produces peak pressure of approximately 830kPa at 3000rpm, whereas by adding the turbocharger the same pressure can be achieved at 2400rpm. By using turbocharger, the peak pressure of non-turbocharged condition is achieved at 0.8 times the speed. At 3000rpm, the turbocharged engine produces about 930kPa, which is 1.12 times greater and yet the pressure still increases and does not reached its maximum value even at 3500rpm. Turbo charging increases mep rather than the need to rev the engine higher or increase the engine displacement to produce higher power output.

4.3.4 Effect of the engine speed to the brake specific fuel consumption (bsfc)

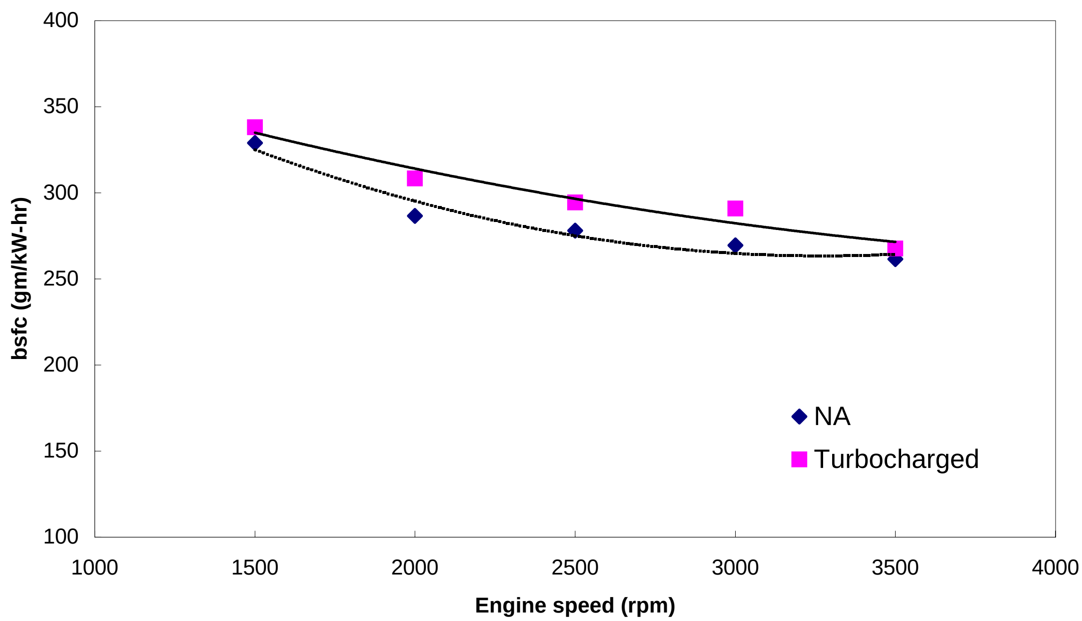

Figure 4.4. Brake specific fuel consumption as a function of engine speed.

Fig. 4.4 compares the bsfc of the naturally aspirated and turbocharged engine as a function of speed. Both conditions illustrate the same trend; the bsfc decreases as the engine speed increases. This is due to the longer time per cycle at low speeds allows more heat loss to occur. As can be seen in Fig. 4.4, the bsfc of turbocharged engine is higher compared to the naturally aspirated engine. This results from the substantial reduction in the compression ratio from 10.0 to 7.0:1 of this particular engine. Allard (1986) states that, tests indicate that for a reduction of one ratio in compression, there is four to five percent increase in fuel consumption at low and medium speeds. Table 4.7

shows the percentage increase of fuel consumption at 1500rpm to 3500rpm, with different boost pressures for each speed. For this engine, the reduction of three ratios in the compression ratio may leads to about 12 to 15% increase in fuel consumption. As can be seen in Table 4.7, the percentage increment in bsfc is within the range. He also states that too large a reduction in compression ratio will make the engine thermally very inefficient at lower rpm with poor fuel consumption. By referring to Table 4.7, for a given speed, say 3500rpm, as boost raised, the percentage increment of fuel consumption is reduced and correspondingly, the percentage gain in brake power is increased higher. This can be explained by the increased mass of air ingested into the cylinders caused the effective compression ratio (that depends on intake charge pressure, therefore the real-time amount of mass air-fuel mixture) at higher boost to increase, thus increasing the thermal efficiency that leads to the lower fuel consumptions and higher energy released during combustion.

4.3.5 Effect of engine speed to the mass flow rate of air and air-fuel (AF) ratio

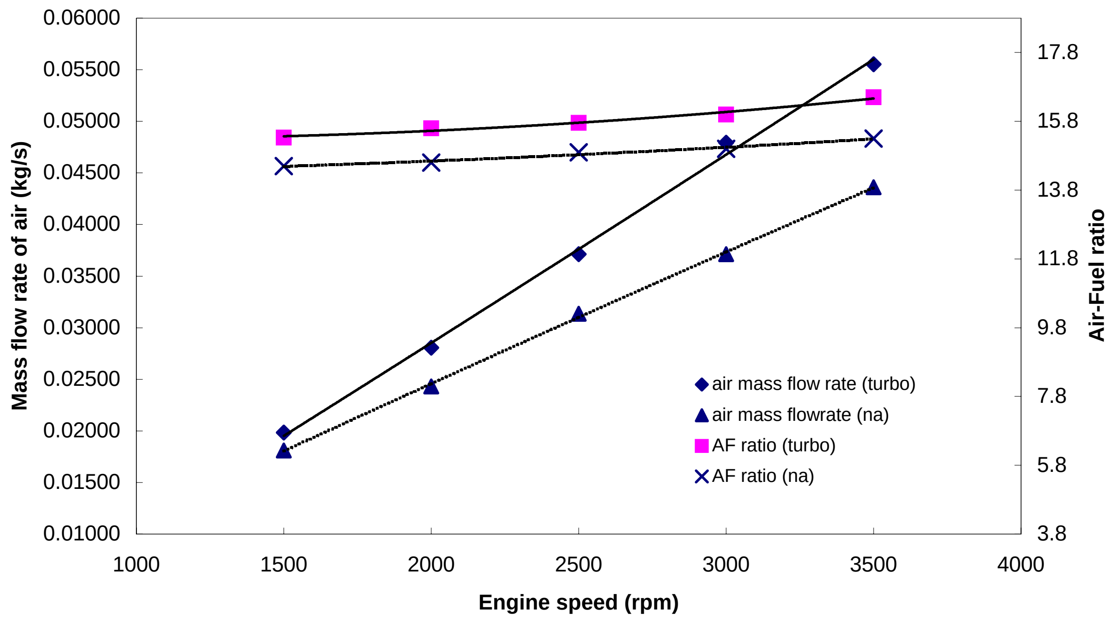

Figure 4.5. Mass flow rate of air and AF ratio as a function of engine speed.

Fig. 4.5 shows the comparisons of mass air flow rate and AF ratio between the naturally aspirated and turbocharged engine. The turbocharger increases the intake pressure, thus increases the air density. Higher air density contains more air molecules per volume. That explained the higher mass flow rate of air of the turbocharged engine as compared to the naturally aspirated one. Higher boost pressures induced higher air mass flow rate into the cylinders in the available time as the intake valves opened per cycle.

Correct AF ratio means the engine is getting all the fuel it can efficiently burn in every cycle. The fuel injection system must be able to regulate the proper amount of fuel for any given air flow. Pulkrabek (1997) states that gasoline-fueled engines usually have AF ratio in the range of 12 to 18 depending on the operating conditions at the time (e.g. accelerating, cruising, starting). Fig. 4.5 also shows the AF ratio of the naturally aspirated and turbocharged engine at full load, wide-open throttle (WOT) condition. For the naturally aspirated engine, the AF ratio at 1500rpm to 3500rpm is in the range of

14.5 to 15.3. The AF ratio of the turbocharged engine is in 15.3 to 16.5 ranges. These results confirms with Pulkrabek. Higher AF ratio of the turbocharged engine is due to more quantity of air forced into the cylinders compared to the naturally aspirated. The electronic fuel injection (EFI) system components such as fuel injectors, fuel pump, engine control unit, fuel pressure regulator and the associated sensors of this engine were retained as the stock configuration even after the turbo conversion. According to Corky Bell (1997), the full-throttle AF ratio of a turbocharged engine (when under boost) should be close to 12.5 or 13.0 to 1 and according to Setright (1976), the enrichment of a petrol/air mixture beyond 1:12 leads to substantial loss in power due to the optimum combustion temperatures. Charge air under pressure is hot. The introduction of fuel can cool it. The cooling effect of the fuel lowers the charge temperature and increases its density before it reaches the cylinders. This leads to lower temperatures throughout the cycle and more efficient combustion due to a reduction in temperature of piston crown, combustion chamber walls, exhaust valves and spark plugs. The slightly rich mixture needed when the engine is under boost for the purpose to reduce the compressed charge air temperature. As can be seen in the case of this engine, the AF ratio in the turbocharged condition is leaner. This is due to, as mentioned earlier, there was no modifications done to the EFI equipment to deliver extra fuel flow in proportions to the boost pressure to maintain proper fuel delivery relative to air mass flow rate.

4.3.6 Effect of the engine speed to the engine volumetric efficiency

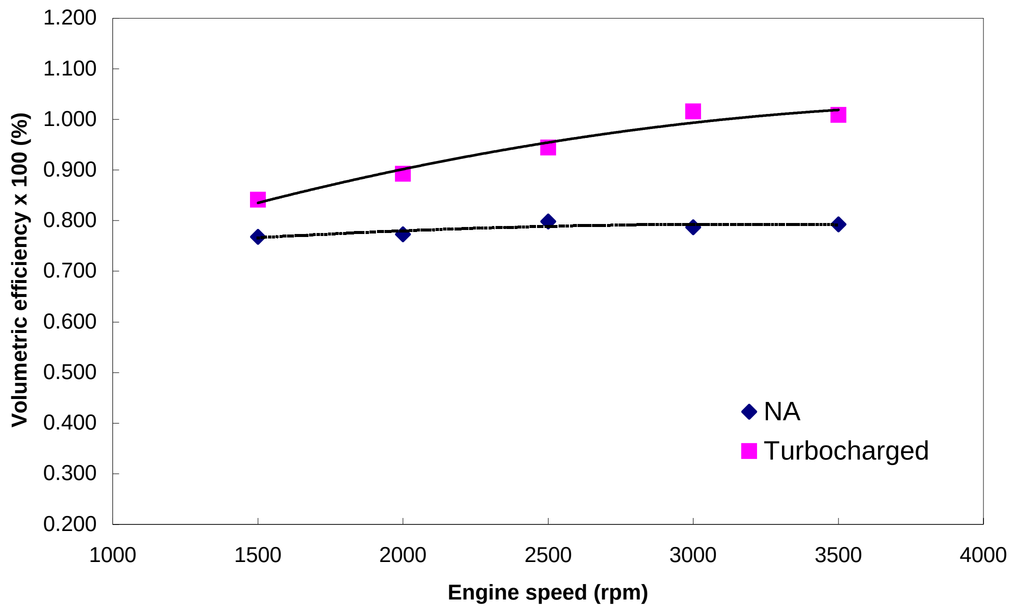

Figure 4.6. Volumetric efficiency as a function of engine speed.

Fig. 4.6 illustrates the volumetric efficiency as a function of the engine speed for the naturally aspirated and turbocharged engine at wide-open throttle. Table 4.7 shows the percentage increase of the turbocharged engine's volumetric efficiency at maximum boost for each speed from the naturally aspirated. As can be seen in Fig. 4.6, the volumetric efficiency of naturally aspirated engine falls around 75% to 80%. However, with the use of turbocharger, the increment of up to 29% can be achieved. At any given speed, the ratio of the mass of new charge in the cylinder to the mass of charge that would fill the displacement volume at atmospheric pressure is a measure of the volumetric efficiency of the engine. The turbocharged engine, by pumping in extra air into the combustion chamber, is clearly superior to the normally aspirated one. The volumetric efficiency may exceed 100% as shown in the table 4.8, at 3000rpm and 3500rpm with boost pressures of 3psi and 4psi respectively. This is due to the pressurized intake manifold had raised the effective compression ratio higher than the geometric compression ratio (determined by the displacement and clearance volume), making the mass of new charge greater than the mass of charge that would fill the displacement volume at atmospheric pressure.

4.3.7 Effect of the engine speed to the engine thermal efficiency

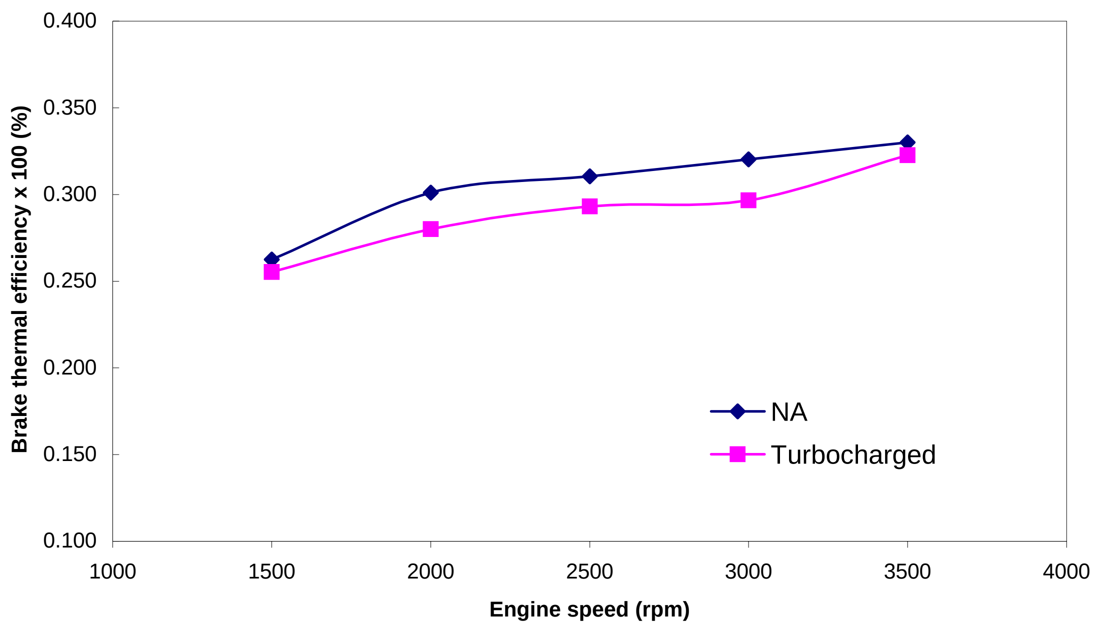

Figure 4.7. Engine brake thermal efficiency as a function of engine speed

If the strength of the turbocharged engine is in its volumetric efficiency, its weakness is in thermal efficiency. As shown in Fig. 4.7, the thermal efficiency of the naturally aspirated engine is higher compared to the turbocharged one. Setright (1976) states that the thermal efficiency of the naturally aspirated petrol engine seldom exceeds 33%. By referring to Table 4.8, the thermal efficiency of this engine in naturally aspirated condition is within the range. The table also shows the percentage decrease of the turbocharged condition compared to the naturally aspirated. About 35% of the total chemical energy that enters an engine in the fuel is converted to useful crankshaft work, about 30% is carried away in the exhaust flow and the remainder is dissipated to the engine surroundings by heat transfer processes. As the piston descends during the working stroke (expansion stroke), it causes the volume of the cylinder to increase, and by expanding the heated gas, they are cooled, the heat loss being converted into useful work. As previously discussed, the volumetric efficiency of the turbocharged engine is higher than 100% due to the effective compression ratio. However, during the expansion stroke, the expansion ratio remains unchanged for the same engine either it is naturally aspirated or turbocharged. So, there is no possibility of recovering during the expansion phase the potential energy was created during the increased compression phase. It is because this imbalance between the effective compression ratio and the expansion ratio that the thermal efficiency of the turbocharged engine is lower than that of the naturally aspirated one.

4.3.8 Effect of the engine speed to the exhaust emission products

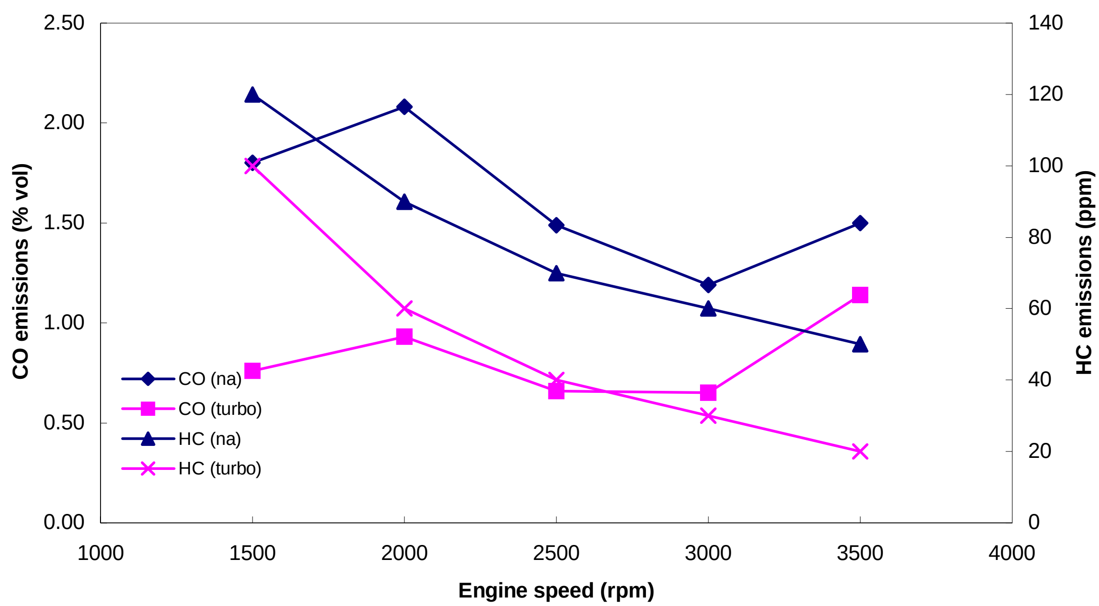

Figure 4.8. Exhaust emissions as a function of engine speed.

Mac Innes (1976) states that tests have shown that a turbocharger applied to a gasoline engine in good running condition will not significantly increase any of the measured exhaust emissions and in almost every case will decrease them. An emission test for this particular engine was run before and after the turbocharger was installed. The results of these tests are shown in Table 4.9 and Fig. 4.8. It can clearly be seen that by adding a turbocharger, the emissions at WOT are low as compared to the naturally aspirated. Because of the added turbulence due to the compressor impeller and because the combustion temperature will not be any higher, exhaust emissions will be about the same or a little less with turbocharged engine. AT WOT, intake manifold pressure will be higher than the exhaust manifold pressure. This will remove the clearance gases and reduce the combustion temperature of the turbocharged engine even further.

CHAPTER V

CONCLUSIONS

A 1.6-litre gasoline engine was converted from the naturally aspirated condition to a turbocharged engine. Testing was done on both the naturally aspirated and turbocharged condition. Data gathered from the dynamometer and instrumentations coupled to the engine. The purpose of the testing is to study the effects of turbocharger on engine performance. The following conclusions have been reached from the comparative analysis:

1. The addition of the turbocharger increased the maximum output of the engine at the speeds above 2000rpm.

2. Below 2000rpm, the turbocharger is not supplied with enough exhaust energy to spin fast enough to produce above-atmospheric pressures in the intake manifold.

3. If full-throttle (WOT) is not used, the turbo makes no contribution to the torque curve and the engine behaves like a naturally aspirated engine.

4. Turbocharger offsets the speed at which maximum torque generated to a higher speed and produces a given torque at a speed lower than naturally aspirated engine.

5. The intake pressure higher than atmospheric pressure increases all pressures through the cycle, include the mean effective pressure inside the cylinders.

6. The turbocharger increases the power without the need to raise the engine speed. Turbo charging increases mean effective pressures rather than the need to rev the engine higher or increase the engine displacement to produce higher power output.

7. The reduction in the compression ratio from 10.0 to 7.0:1 of this particular engine leads to about 2 to 8% increase in fuel consumption.

8. A slightly rich mixture needed when the engine is under boost for the purpose to reduce the compressed charge air temperature. This leads to lower temperatures throughout the cycle and more efficient combustion due to a reduction in temperature of piston crown, combustion chamber walls, exhaust valves and spark plugs.

9. The turbocharged engine, by pumping in extra air into the combustion chamber, may achieve volumetric efficiency of over 100%. Instead of being wasted, the exhaust gases were expanded through the turbine that converts it into useful work to drive a compressor that supply the engine extra air. As the load and the volumetric efficiency and the effective compression ratio are increased, so is the expansion ratio. Therefore the turbocharged engine can achieve thermal efficiency almost too naturally aspirated engine.

10. HC and CO emissions are reduced considerably when the engine is turbocharged compared to its stock naturally aspirated condition.

RECOMMENDATION

In order to improve the engine performance and the experimental setup for better results to be achieved, there are several suggestions for future works:

1. A computerized data acquisition system that comprises of a computer that gathers, manipulate and displays data through a signal conditioning device, pressure transducer, crank angle encoder and thermocouples may be utilized to obtain better accuracy of the results and control. By utilizing this data acquisition system, the indicated pressure inside the cylinder as a function of the crank angle can be obtained.

2. Full-scale testing in the test cell is needed to run the engine at higher speed and the maximum torque and power can be obtained. It is dangerous to run higher speed than 3500rpm in the present open test bed area.

3. In order to supply the turbocharged engine with adequate air-fuel ratio in all condition, a modification to the stock EFI system to increase the fuel flow is needed. A boost-pressure-powered fuel pressure regulator can be made to drive the fuel pressure up rapidly to keep pace with rising boost pressure. The stock injectors may be retained but the fuel pump needs to be substituted with high-pressure pump.

4. The turbocharged engine ignition curve needs a small retard as boost rises and the mixture becomes denser and more turbulent. The correct ignition timing under all circumstances is achieve only if the timing curve can be designed right along with the fuel curve. This can be accomplished with aftermarket engine management systems such as Motec, Microtek and Haltech fully programmable engine management systems that can control both ignition and fuel curves.

5. Smaller size turbocharger unit such as TD04H may substitute the TD05H unit presently used to enable faster response to provide boost at lower speeds.

6. The use of fully synthetic engine oil is reasonable because of its high-temperature stability and toughness of the molecular structure. The semi-synthetic oil currently used is less stable at very high temperature in turbocharger bearing section caused by the heat in the turbine side.

7. The engine coolant feed and return lines, connecting the water jacket around the turbocharger's bearing housing with the engine cooling system capable of carrying away most of the heat that migrates from the turbine side, is necessary.

8. The installation of spark plugs that is one or two range colder from the stock plugs, which conduct lots of heat away from the electrode should also be considered.

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