Turbocharger petrol engine

The quest for higher efficiency of the internal combustion engine will always be pursued. Increasingly stringent emission regulations are forcing the manufacturers to downsize on engine displacement and increase the specific power. By adding the turbocharger, the air flows through the engine and hence specific power can be increased.

The advantage of a small turbocharged engine over a naturally aspirated (NA) engine of a similar power is that it is lighter, having better part load efficiency when operating at the same load, while producing less emission.

The objective in this study is to investigate a turbocharger in a naturally aspirated engine and testing the engine before the installation of the turbocharger.

Boost refers to the increase in the manifold pressure that is generated by the turbocharger in the intake path or specifically that exceeds the normal atmospheric pressure. This study also aims to develop a strategy for the control of boost for the engine.

1. 0 Introduction

1. 1 Background

Turbocharged spark ignition engines have been around since the 1970s, but their popularity outside the motorsport sector has been small until the recent advances in engine control. The lack of popularity could partly be due to the drivability issues associated with early turbocharged engines. The engine’s response to a sudden increase in driver’s demand was delayed due to a turbocharger lag.

The lag was then usually followed by a rapid increase of power which resulted in loss of traction and possible loss of control over the car. The developments made in the electronic control and management of internal combustion engine made it possible to overcome most of these drivability limitations. Passenger vehicles with turbocharged SI engines are now becoming more common. A number of companies such as Audi and Volvo now offer different passenger vehicle models with turbocharged SI engine whereas Mercedes offers supercharged and turbocharged engines.

The operating principle of the turbocharger is to use the energy recovered from the exhaust gases to force more air into the combustion chamber. This increases the amount of oxygen in the combustion chamber and hence more fuel can be burned and more power can be produced. Therefore a turbocharged engine can produce more power than a similar sized naturally aspirated engine. It is claimed that the displacement of the turbocharged engine can be reduced by up to 40% relative to NA engine, without compromising power output. Thus the turbocharged engine could be smaller, lighter and more fuel efficient as well as produce less emissions.

1. 2 Aim

To design and specify turbocharger in a Vauxhall 2. 2 litre engine

1. 3 Objectives:

1. Critical literature review of the project.
2. To investigate turbo system, develop a system for the Vauxhall 2. 2, produce drawings and design.
3. Testing the engine before installation of turbocharger

4. To investigate and develop strategy for control of boost for the engine over a wide range of condition.

2. 0 Initial Critical Review of Literature

This project is related to the turbocharging of a four stroke petrol engine. In this discussion a turbocharged four stroke diesel engine will also be discussed briefly and the differences will be highlighted. However, it omits to discuss two stroke engines due to their different gas exchange processes.

Supercharging

The term supercharging refers to increasing the air density by increasing its pressure prior to entering the cylinder. This allows a proportional increase in the fuel that can be burned and hence raises the potential power output. Three basics categories are used to accomplish this.

The first is mechanical supercharging where a separate pump or compressor, usually driven by power taken from the engine, provides the compressed air. The second method is turbocharging, where a turbocharger, a compressor and turbine on a single shaft is used to boost the inlet air density. The third method is pressure wave supercharging which uses wave action in the intake and exhaust systems to compress the intake mixture.

The main advantage of turbocharging as opposed to supercharging is that turbocharging uses the energy in the exhaust gas that would have been lost. Supercharging uses power from the engine’s crank shaft and thus less power is available for propulsion

Turbocharging

The author acknowledges that the theory represented in this section is extracted from Watson and Janota (1982).

The exhaust driven turbocharger was invented by a Swiss engineer named Alfred Buchi, his patent applied to a diesel engine in 1905. It took a very long time to establishe a turbocharger, but it is now proved that their characteristics are suited to the diesel engine, the reason being that only air is compressed, and no throttling is used.

A turbocharger consists basically of a compressor and turbine coupled on a common shaft. The exhaust gases from the engine are directed by the turbine inlet casing on the blades of the turbine and subsequently discharged to atmosphere through a turbine outlet casing. The exhaust gases are utilized in the turbine to drive the compressor, which compresses the air and directs it to the engine induction manifold, to supply the engine cylinder with air of higher density than is available to a naturally aspirated engine.

Figure1: Automotive Turbocharger

Since diesel engines having no knock limitations, the maximum allowable boost on CI engines depends only on the mechanical strength of the engine. On an SI engine, the boost pressure is limited by knock. Thus if boost pressure is high on SI engines, the compression pressure must be low, high octane number fuel must be used or ignition timing must be retarded.

Turbocharger Theory

The operating characteristics of turbo machines such as turbines and compressors are totally different from the reciprocating internal combustion engine. The most common turbocharging assembly used in the automotive industry is made up of radial compressor coupled to radial turbine. Between the two is a wide supporting plain journal bearing, because an ordinary roller bearing would not survive the high rotational speed of up to 25000 rev/min of which a small turbine is capable. For racing application, ceramic ball bearings are being used more frequently.

Axial turbine coupled with a radial compressor is a common configuration. Axial turbines are preferred for their superior efficiency to those of a radial turbine, but according to manufacturer's radial flow turbines are simpler and cheaper to manufacture and also the operating range of radial flow compressors are limited to certain pressure ratios, because a high pressure ratio will cause the supersonic flow and cause shockwaves at the inlet, this will impair the efficiency of compressor.

Turbocharging Diesel (CI) or Petrol (SI) engines

Today turbocharged diesel engines are common but turbocharged petrol engines are rare. There are sound reasons, both technical and economic for this situation. The principal reasons stem from the difference between the combustion systems of petrol and diesel engines. The petrol engine uses a carburettor or fuel injection system to mix air and fuel in the inlet manifold so that a homogeneous mixture is compressed in the cylinder.

A spark is used to control the initiation of combustion which then spreads throughout the mixture. This is because the mixture temperature during the compression must be kept below the self-ignition temperature of the fuel. Once the combustion has started it takes time for the flame front to move across the combustion chamber burning the fuel. During this time unburnt ‘ end gas’ is heated by further compression and the radiation from the flame front.

If it reaches the self-ignition temperature before the flame front arrives, a large quantity of mixture may burn extremely rapidly producing severe pressure waves in the combustion chamber. This situation is commonly referred to as ‘ knock’ and may result in severe cylinder head and piston damage. This is due to the fact that the compression ratio of the engine must be low enough to prevent knock occurring.

In the CI engine cylinder, air alone is compressed. Fuel is sprayed directly into the combustion chamber from an injector only when combustion is required. This fuel self-ignites as in a diesel engine the compression ratio must be high enough for the air temperature on compression to exceed the self-ignition temperature of the fuel. As injection takes time, only some of the fuel is in the combustion chamber when ignition starts, and since much of this fuel is not as damaging as the knocking situation in a petrol engine.

The maximum CR of the petrol engine, but not the diesel engine, is therefore limited by the ignition properties of the fuel. The minimum CR is limited by resulting low efficiency. Turbocharging results not only give a higher boost pressure, but a higher temperature. Unless the compression ratio of a petrol engine is reduced the temperature at the end of compression stroke will be too high and the engine will knock.

The engine may remain knock free under mild boost – but only because there should be a sufficient safe knock free margin, or a fuel of higher self-ignition temperature/octane number has been used. Thus the potential power output of a turbocharged petrol engine is limited. The diesel engine has no such a limitation and can therefore use a much higher boost pressure.

Petrol engines cost substantially less to produce than diesel engines of equivalent power output. The cost of the turbocharger on a diesel engine is more than offset by reduced engine size required for a specified power output (with the exception of very small engines). This situation will rarely occur in the case of petrol engine.

Energy Available In the Exhaust Gas:

Figure 2 shows the ideal limited pressure engine cycle in terms of pressure/volume diagram for the naturally aspirated engine. Superimposed is a line representing isentropic expansion from point 5, at which the exhaust valve opens, down to the ambient pressure (P_a) which could be obtained by further expansion if the piston were allowed to move to point 6. The maximum theoretical energy that could be extracted from the exhaust system is represented by the shaded area 1-5-6. This energy is called as ‘ blow-down’ energy.

Figure2: Naturally Aspirated Ideal Pressure Limited Cycle (Watson and Janota, 1982)

Considering the supercharged engine, the ideal four stroke pressure/volume diagram would appear as shown in figure, where P_1 is the supercharging pressure and P_7 is the engine back pressure in the exhaust manifold. Process 12-1 is the induction stroke, during which fresh charge at the compressor delivery pressure enters the cylinder. Process 5-1-13-11 represents the exhaust process.

When the exhaust valve first opens (point 5) some of the gas in the cylinder escapes to the exhaust manifold expanding along line 5-7 if the expansion is isentropic. Thus the remaining gas in the cylinder is at P_7, when the piston moves towards the TDC, displacing the cylinder contents through the exhaust valve into the exhaust pipe against the back pressure.

At the end of the exhaust stroke the cylinder retains a volume (V_{cl}) of residual combustion products, which for simplicity can be assumed to remain there. The maximum possible energy that could be extracted during the expulsion stroke will be represented by area 7-8-10-11, where 7-8 represents isentropic expansion down to the ambient pressure.

Figure3: Turbocharged Ideal Pressure Limited cycle (Watson and Janota, 1982)

There are two distinct areas in figure 3 representing energy available from the exhaust gas, the blow down energy (area 5-8-9) and the work done by the piston (area 13-9-10-11). The maximum possible energy available to drive a turbocharger turbine will clearly be the sum of these two areas. Although the energy associated with one area is easier to harness than the other, it is difficult to devise a system that will harness all of the energy.

To achieve that, the turbine inlet pressure must rise instantaneously to P_5 when the exhaust valve opens, followed by isentropic expansion of the exhaust gas through P_7 to the ambient pressure ($P_8 = P_a$). During the displacement part of the exhaust process, the turbine inlet pressure must be held at P_7. Such a series of process is impracticable.

Considering the simpler process in which a large chamber is fitted between the engine and the turbine inlet in order to damp down the pulsating exhaust gas flow. By forming a restriction to the flow, the turbine may maintain its inlet pressure at P_7 for the whole cycle. The available work at the turbine will then be given by area 7-8-10-11. This is the ideal constant pressure system. Next consider an alternative system, in which a turbine wheel is placed directly downstream of the engine close to the exhaust valve.

If there were no losses in the port, the gas would expand directly out through the turbine along line 5-6-7-8, assuming isentropic expansion. If the turbine area were sufficiently large, both cylinders and the turbine inlet pressure would drop to P_9 before the piston had moved significantly up the bore.

Hence the available energy at the turbine would be given by area 5-8-9. This can be considered the ideal pulse system. The system commonly used and referred to as ‘ constant pressure’ and ‘ pulse’ are based on the above principles but in practice they differ from these ideals.

Constant Pressure Turbocharging

In constant pressure turbocharging exhaust ports from all the cylinders are connected to a single exhaust manifold, whose volume is sufficiently large to dampen down the unsteady flow entering from each cylinder. When the exhaust valve of a cylinder opens, the gas expands down to the (constant) pressure in the exhaust manifold without doing useful work.

However, not all of the pulse energy is lost. From the law of conservation of energy, the only energy actually lost between the cylinder and turbine will be due to heat transfer. With a well-insulated manifold, this loss will be very small and can be neglected.

Consider what happens to the gas leaving the cylinder, expanding down into the exhaust manifold, and then flowing through the turbine. At the moment of the exhaust valve opening, the cylinder pressure will be much higher than the exhaust manifold pressure. During the early stages of valve opening (when the effective throat area of the valve is very small) the pressure ratio across the valve will be above the choked value.

Hence gas flow will accelerate to sonic velocity in the throat followed by the shock wave at the valve throat and sudden expansion to the exhaust manifold pressure. Due to the turbulent mixing and throttling, no pressure recovery occurs. The stagnation enthalpy remains unchanged and hence flow from the valve to turbine is accompanied by an increase in entropy.

As the valve continues to open the cylinder pressure will fall and flow through the valve which becomes subsonic. The flow will continue to accelerate to the valve throat and then expand to a pressure in the exhaust manifold. The energy available to useful work in the turbine is given by isentropic enthalpy change across the turbine, whereas the actual energy recovered is given by the enthalpy change across the turbine.

Clearly it is a lack of recovery of the kinetic energy leaving the valve throat and throttling gases that lead to poor exhaust gas energy utilization with the constant pressure system.

If the exhaust manifold is not sufficiently large, the blow down or the first part of the exhaust pulse from the cylinder will raise the general pressure in the manifold. If the engine has more than three cylinders, it is inevitable that at the moment when the blow down pulse from the cylinder arrives in the manifold, another cylinder is nearing the end of its exhaust process.

The pressure in the latter cylinder will be low; hence any increase in exhaust manifold pressure will impede or even reverse its exhaust processes. This will be particularly important where the cylinder has both intake and exhaust valves partially open and is relying on a through-flow of air for scavenging of the burnt combustion products.

There are some advantages and disadvantages of using a constant pressure system:

- Conditions at the turbine entry are steady with time. Therefore losses in the turbine that result from unsteady flow are absent.
- A single entry turbine may be used, eliminating ‘ end of sector losses’.
- Single turbocharger can be used on all multi-cylinder engines, it will be a large turbocharger unit and since it is a large unit it will have low leakage losses and hence have higher efficiency. Turbines designed for constant pressure turbocharging have a high degree of reaction (50%)

which, coupled with exhaust diffuser, brings additional gains in efficiency.

- From a practical point of view, exhaust manifold is simple to construct although it may be rather bulky, particularly relative to small engines with few cylinders.
- Transient response of the system is poor. Due to the large volume of gas in the exhaust manifold, the pressure is slow to rise, resulting in poor engine response and making it unsuitable for applications with frequent load or speed changes.

Pulse turbocharging

Although the constant pressure system is commonly used on certain types of engines, the vast majority of turbocharged engines in Europe use a pulse turbocharging system. In the practical pulse system an attempt is made to utilize the energy represented by both pulse and constant pressure areas of figure 2.

The objective is to make the maximum use of high pressure and temperature exist in the cylinder at the moment of exhaust valve opening, even at the expense of the creating highly unsteady flow through the turbine. In most cases the benefit from increasing the available energy will more than offset the loss in the turbine efficiency due to the unsteady flow.

Now consider small exhaust manifold as shown in figure. Due to the small volume of exhaust manifold, a pressure build up will occur during the exhaust blow-down period. This results from a flow rate of gases entering the manifold through the valve exceeding that of gas through the turbine.

At the moment the exhaust valve starts to open, the pressure in the cylinder will be 6 to 10 times more than the atmospheric pressure, whereas the pressure in the exhaust manifold will be close to atmospheric. Therefore the initial pressure drop across the valve is above the critical value at which choking occurs and the flow will be sonic.

Further expansion of the gas to the exhaust manifold pressure occurs by sudden expansion at the exhaust manifold recovery occurs due to turbulent mixing. The stagnation enthalpy remains constant hence the flow from the valve throat is accompanied by an increase in entropy.

Finally the gas expands through the turbine to atmospheric pressure, doing useful work. The out-flowing gas from the cylinder loses a very large part of its available energy in throttling and turbulence after passing the minimum section of the exhaust valve. If the ratio of valve throat area to manifold cross section area is very small then throttling losses are very large and pressure drop across the valve is very large, during the initial stages of valve opening.

Following further opening of the exhaust valve the cylinder pressure increases, reducing the throttling losses across the valve. The pressure drop across the turbine is now much larger, transferring the available energy to the turbine, which represents a much larger proportion of the available energy in the cylinder.

At the end position of the valve opening the flow is sub-sonic and the throttling loss is reduced and is equivalent to the kinetic energy at the entry to the exhaust manifold. During the exhaust stroke, the flow process follows approximately the constant pressure pattern as described in the previous section. At the exhaust valve, the pressure in the exhaust manifold approaches atmospheric value.

With pulse operation, a much larger portion of the exhaust energy can be made available to the turbine by considerably reducing throttling losses across the exhaust valve. The speed at which the exhaust valve opens to its full area and the size of the exhaust manifold become important factors as far as energy concerned. If the exhaust valve can be made to open faster, the throttling losses become smaller during the initial exhaust period.

Furthermore, if the area of exhaust manifold is smaller than the rise in pressure of exhaust manifold will be faster, contributing to a further reduction in throttling losses in the early stages of the blow-down period. A small exhaust manifold also causes a much more rapid fall in pressure towards the end of the exhaust process improving scavenging and reducing pumping work. This discussion has therefore focused on the single cylinder engine connected to the exhaust manifold.

However, in the case of a multi-cylinder engine this problem becomes more complicated. Because the turbocharger may be located at the one end of the engine, narrow pipes are used to connect the cylinders to the turbine to keep the exhaust manifold size as small as possible. By using the narrow pipes the area increase following the valve throat is greatly reduced, keeping throttling losses to a minimum.

Scan dig7. 2

Consider again a single cylinder engine, connected to a turbine by a long narrow pipe as shown in figure. Since the large quantity of exhaust energy becomes available in the form of a pressure wave, which travels along the pipe to the turbine at sonic velocity, the conditions at the exhaust valve and the turbine are not the same at a given time.

Therefore the flow process at the exhaust pipe and at the turbine end, have to be presented separately as shown in figure. For simplicity, pressure wave reflections in the pipe are ignored. During the first part of the exhaust process, in the choked region of flow through the valve, the gas is accelerated to sonic velocity at the throat. Since the contents of the pipe are initially at rest at atmospheric pressure, sudden expansion takes place across the valve throat. However some of the kinetic energy is retained as dependent on the valve throat area to pipe cross-section area.

As the valve opens further the pressure at the exhaust pipe entry rises rapidly. This is firstly because a certain amount of time is required for the acceleration of the outgoing gases, and secondly because the gases enter the exhaust pipe from the cylinder at a higher rate than they are leaving the exhaust pipe at the turbine end.

The sudden pressure rise at the pipe entry is transmitted along the pipe in the form of a pressure wave and will arrive at the turbine displaced in time. This displacement is a function of length of pipe and properties of gas. The pressure drop across the valve is noticeably reduced due to the rapid drop in cylinder pressure and the rise in the pipe pressure and also because the valve throat area to pipe area ratio has increased. Both effects considerably reduce throttling losses. The velocity at the turbine end of the pipe is greater than velocity after the valve, due to the arrival of high pressure wave at the turbine end.

In the subcritical flow region of blow down period, the pressure in the exhaust falls at the same time as that in the cylinder. The velocity at the valve throat is equal to the velocity in the pipe, since the valve is fully open. At the turbine exhaust gas expands to atmospheric pressure, doing useful work in the turbine.

It has been established that the pulse turbocharging system results in greater energy availability at the turbine. As the pressure wave travels through the pipe, it carries a large portion of pressure energy and small portion of kinetic energy, which is affected by friction. The gain obtained through the use of a narrow exhaust pipe is achieved partly by reducing throttling losses at the early stages of the blow down period and partly by preserving kinetic energy.

Thus the small diameter exhaust pipe is essential because this will preserve high gas velocity from the valve to the turbine. However if pipes are made too narrow, viscous friction at the pipe wall will become excessive. The optimum exhaust manifold pipe diameter will be a compromise, but the cross sectional area should not be significantly greater than the geometric valve area at full lift.

The actual flow through a pulse exhaust system is highly unsteady and is affected by pulse reflections from the turbine and closed exhaust valves. It will be evident that effective time of arrival of a reflected pulse changes as per the engine speed. Hence the exhaust pipe length is critical and must be optimized to suit the speed range of the engine.

The interference of reflected pressure waves with the scavenging process is the most critical aspect of a pulse turbocharging system, particularly on the engine with a very long valve overlap. Because of this phenomenon it is impossible to connect an engine with more than three cylinders to the same turbine without using a twin-entry turbine or introducing losses on the intake or exhaust processes.

The advantage of pulse over the constant pressure turbocharging is that the energy available for conversion to useful work in the turbine is greater. The ideal pulse turbocharging must have following characteristics:

- The peak of blow-down pulse must occur just before the bottom dead centre of the cylinder, followed by a rapid pressure drop to below boost pressure.
- The boost pressure must be above the exhaust manifold pressure to aid the scavenging process during the valve overlap.
- The effectiveness of pulse system is governed by the gas exchange process and overall efficiency of the turbocharger under unsteady flow conditions.

Pulse converters in turbocharging

The pulse turbocharging system has been found to be superior as compared to the constant pressure system on the majority of today’s diesel engines. In the previous section it is made clear that the pulse turbocharging is most effective when groups of three cylinders are connected to the turbine entry.

When one or two cylinders are connected to a turbine the average turbine efficiency and expansion ratio tend to fall due to the wide spacing of exhaust gases pulses. To overcome some of these advantages ‘ pulse converter’ has been developed.

Birmann was the first to use the term ‘ pulse converter’. His main objective was to design a device that preserved the unsteady flow of gases from the cylinder during the exhaust and valve overlap periods, yet to maintain a steady flow at the turbine, so that it might be possible to achieve good scavenging and high turbine efficiency. For good scavenging he proposed a ‘ jet pump system’, by using high velocity of gas issuing from a central nozzle to decrease the pressure in short pipes at the exhaust valves.

The system shown in figure 8. 1 has some disadvantages as following:

- Each nozzle must be larger than last which results in high manufacturing cost.
- The whole installation is bulky and complex.
- Because much of exhaust gases will pass through several ejectors and diffusers, the frictional and diffusion losses will be high.
- There is insufficient length between exhaust ports to permits efficient pressure recovery in the diffusers.

The majority of pulse converters in use today are based on the concept of minimum energy loss, even if this means not only a loss of all suction effect, but also some pressure wave difference during scavenging. To avoid high mixing losses at the junction, the area reduction in the inlet nozzles is usually small (junction area > 50% of pipe area), while the mixing length and plenum and often even the diffuser are omitted completely, as suggested by Petak (as cited in Watson and Janota, 1982).

These simple pulse converters have the added advantage of adding little over-all length to the exhaust system. A typical example from a four stroke engine is shown in figure 8. 4. The pulse converter is specified by the nozzle and throat area ratios. Clearly such a pulse converter will generate no suction, but the flow losses through it will be very much less than in more complex designs.

Tests on a model pulse converters by Watson and Janota (1971) have shown that the area reduction at nozzles has to be severe to reduce pulse propagation substantially. The penalty accompanying large area reductions in the inlet nozzles is higher internal losses and hence reduces the amount of energy available for useful expansion through the turbine. In practice this means that the minimum possible area reduction is used, consistent with reasonable scavenging.

It follows that the design of the pulse converter is a compromise between minimum losses and reduction of the pulse interaction between the inlet branches. The compromise adopted may vary from one engine design to another, depending on the amount of pulse interference, etc.

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