Palestine Polytechnic University

College of Engineering and Technology
Mechanical Engineering Department

Bachelor's Thesis

Graduation Project

Enhancement of the Engine Acceleration by Using Compressed Air

Project Team

Hussein M. Nassar **Motaz bilah I. Ihrabat**

Project Supervisor

Dr. Momen Suhgayyer

Hebron – Palestine

June - 2014

Abstract

Experimental work has shown that injecting air into the manifold greatly improves the transient response of the turbocharger. This work has shown that air injection has a large effect upon improving the transient response when performing rapid load changes, which result in increases engine torque output while minimizing the potential for exceed various operating limits to the maximum practicable extent, moreover the transient performance has improved which lead to a significant reduction in the harmful emissions.

The vehicle's air injection system controller implements strategies for shaping the rate of the air injection during a boost event, by controlling the timing, duration, quantity and/or injection pattern during a boost event to achieve a refined distribution of compressed air injection over the course of the boost event to provide desired engine torque output and fuel efficiency while minimizing the potential for exceeding a wide variety of operation limits, regulatory, engineering and passenger comfort limits.

ملخص

أظهرت النتائج العملية أن حقن الهواء في مجاري السحب يحسن كثيرا من الاستجابة للشاحن التوربيني. وقد أظهر هذا العمل أن حقن الهواء له تأثير كبير على تحسين المرحلة الانتقالية لعمل المحرك عند تغير الحمل بشكل مفاجئ، مما يؤدي إلى زيادة عزم دوران المحرك مع التقليل من احتمالات تتجاوز حدود التشغيل المختلفة إلى أقصى حد ممكن، وعلاوة على ذلك يؤدي إلى انخفاض كبير في الانبعاثات الضارة بالبيئة في هذه الفترة.

نظام حقن الهواء المضغوط يتحكم بمعدل حقن الهواء خلال فترة الحقن، عن طريق التحكم في توقيت ومدة وكمية ونمط الحقن لتحقيق التدفق المناسب للهواء المضغوط خلال فترة الحقن لتوفير عزم الدوران المطلوب للمحرك والكفاءة في استهلاك الوقود والتقليل من احتمالية تتجاوز حدود التشغيل والحدود التنظيمية والهندسية وراحة الركاب.

Table of Contents

List of Figure

CHAPTER ONE: INTRODUCTION

CHAPTER TWO: SYSTEM OPERATION PRINCIPLES

CHAPTER THREE: SYSTEM CONTROLLER

CHAPTER FOUR: ENGINE MEAN VALUE MODEL

CHAPTER FIVE:SIMULATION

CHAPTER SIX:SYSTEM OPERATION AND RESULTE

List of Tables:

CHAPTER ONE

INTRODUCTION

Content:

1.1 General Outlook

Turbocharging is an increasingly popular method of reducing the harmful emissions of an Internal Combustion (IC) engines. A turbocharger utilizes energy from the exhaust gases and uses it to pressurize the air at the inlet to the cylinders. This allows for the density of air in the cylinder to be greater than the density of ambient air, which allows more power to be produced for a given engine size. It has also been shown that use of turbochargers results in reduction of the harmful emissions.

Turbocharged engines have the disadvantage of poor drivability under transient running conditions. The phenomenon known as "Turbo-lag" is particularly apparent in conditions where a rapid load change is applied at lower engine speeds with rapid acceleration. Quick changes in accelerator pedal do not result in instantaneous response of the turbocharger and consequently vehicle acceleration. This delay has the side effect of increased harmful emissions and engine efficiency deterioration because of unfavorable air-fuel ratio. Reduced air flow in the combustion chamber while fuel is injected causes inadequate air fuel (rich mixture) resulting in increased formation of emissions such as particulates, CO, CO_2 and NO_x which contribute to environmental pollution and increase fuel consumption.

1.2 Objectives of this Project

This Project investigates the turbocharging technology with a particular focus on control for transient engine operation. The investigation is based on a Diesel engine due to the availability of suitable equipment, the interest of industrial collaborators and the consistency with current trends in industrial research and development.

The project aims to improve vehicle performance fig.1.1, in a number of areas; including acceleration, fuel economy and emissions reduction. In particular, the method of injecting fresh air in the intake manifold for vehicle engines, including diesel engines having at least one turbocharger supplying air to the engine's intake manifold, in a manner which increases engine torque output in order to improve drivability performance .

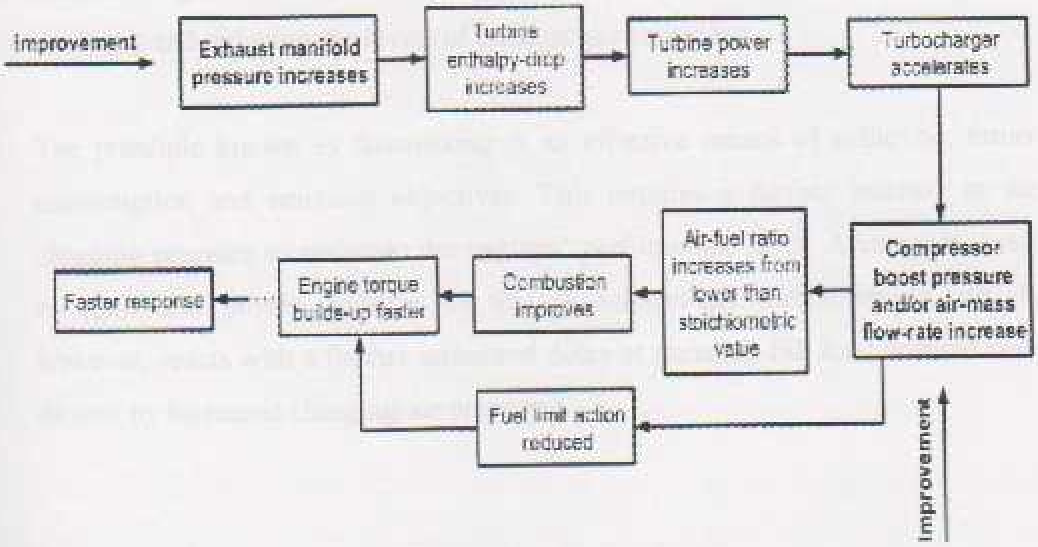

Figure 1.1 Main mechanism of transient response improvement

The importance of this project can be summarized in the following points:

1. Increasing the delivery pressure will increase the density of air in the cylinder to be greater than the density of ambient air, which increases engine volumetric efficiency and results in higher exhaust pressure, which results also in faster turbocharger response; therefore, it leads to increase engine power output in earlier stage.

2. Modern engine developers face the modern challenge of improving vehicle economy and reducing the levels of exhaust gas emissions.

3. The principle known as downsizing is an effective means of achieving future consumption and emission objectives. This requires a further increase to the charging pressure to maintain the engines' performance level. A compact, cost-efficient and proven solution to this is exhaust-gas turbocharging, which, however, reacts with a further undesired delay at transient full load demand conditions by increased charging-air pressures.

1.3 Literature Review

The present work is focused on improving turbocharged diesel engines responses. This response is a major issue limiting their application for a broader use. A study has been carried out looking at different methods of alleviating these problems. The various techniques that have been studied or developed so far for improving transient response can be classified as follows:

1) Methods that utilize an external energy source such as electrical or hydraulic assistance, e.g. electrically assisted turbocharging, or air-injection onto the compressor impeller.

2) Methods that focus on the turbocharger, e.g. Variable geometry turbine (VGT).

1.3.1 Electrically Assisted Turbocharging

The ability of a fixed geometry turbocharger assembly to extract power from the exhaust gases depends on the mass-flow, pressure and temperature. Consequently, at low engine speeds and loads the energy potential of the turbine is limited, producing low boost pressure and delaying engine torque response. A possible solution would be an electrically assisted turbo charging configuration: this makes use of an asynchronous (induction) electric motor or a synchronous type (e.g. reluctance motor or permanent magnet) mounted on the turbocharger shaft.

The electric motor is used to offer additional power to the compressor when a transient or starting event is detected, with its primary application being in automotive engines. The power to drive the electric motor is typically drawn from the battery of the vehicle, a fact that is expected to affect efficiency, particularly at low engine speeds.

Zellbeck et al. [1] studied vehicle performance with an electrically assisted turbocharger in comparison to its 'nominal' counterpart fitted with a waste-gate valve. The former was able to improve vehicle full-load acceleration from 40–80 km/h in sixth gear up to 20% compared with the nominal case. The results are depicted in Fig.1.2, showing mean effective pressure, boost pressure and vehicle speed for a four-stroke, four-cylinder diesel engine.

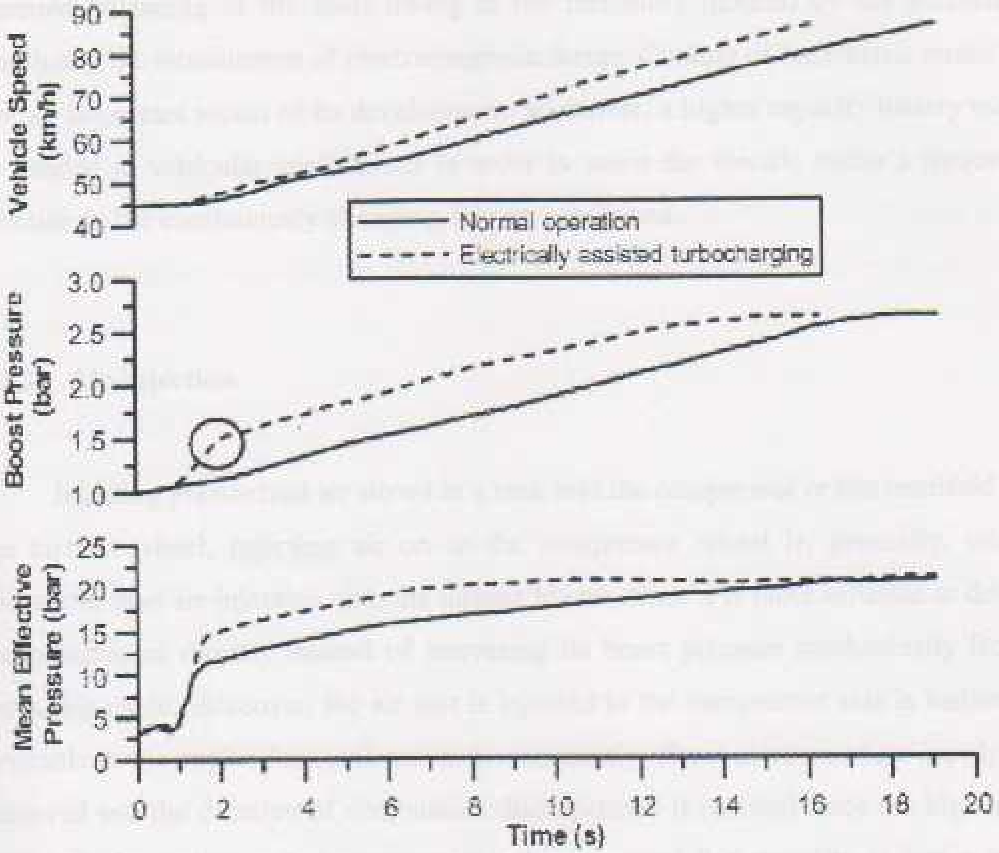

Figure 1.2 Acceleration performance of a diesel-engine vehicle with electrically assisted Turbocharging. [1]

In fact, the brake mean effective pressure of the electrically assisted turbocharged engine managed to achieve steady-state values a few seconds after initiation of acceleration, much sooner compared with the 'normal' engine operation. After the initial sharp increase in the boost pressure, however, the air-supply build-up of the electrically turbocharged engine proceeded at a similar rate compared with its conventional counterpart, since the electrical assistance was discontinued.

A particular aspect in electrically assisted turbocharging units that needs attention is the fact that incorporation of an electric motor on the turbocharger shaft might require a renewed matching for optimum performance and surge avoidance, and a re-assessed balancing of the shaft owing to the instability induced by the increased length and the introduction of electromagnetic forces. Cooling of the electric motor is also an important aspect of its development. Moreover, a higher capacity battery may be needed in vehicular applications in order to assist the electric motor's frequent function in the continuously changing driving conditions.

1.3.2 Air-injection

Injecting pressurized air stored in a tank into the compressor or inlet manifold or the turbine wheel. Injecting air on to the compressor wheel is, generally, more successful than air-injection onto the turbine blades, since it is more efficient to drive the compressor directly instead of increasing its boost pressure mechanically from the turbine side; moreover, the air that is injected in the compressor side is instantly available to the engine for combustion. Consequently, direct increase of air-supply is achieved and the duration of combustion discrepancies is reduced since the high air-flow into the engine cylinders can match the increased fuel quantity in just a few cycles; with the resulting high exhaust gas energy aiding faster turbocharger and engine response.

Ledger [3] and Winterbone [5], proposed a method of air-injection into the centrifugal compressor with a control system interfaced to the four-stroke engine in order to support the modifications required. This consisted of a closed-loop controlling the air-injection via the air–fuel ratio. The latter was measured as the quotient of air manifold pressure and fuel pump rack position. When the value was lower than a specified threshold (being a function of engine speed), air-injection was initiated. Typical results are reproduced in Fig.1.3 for a 0 – 60% load acceptance aided by a 2s air-injection duration right after the onset of the transient event.

Following the assisted compressor operation, the air–fuel equivalence ratio was kept at adequately high values during the early cycles of the transient event, preventing combustion deterioration and establishing faster turbocharger response. As a result, the engine response time was approximately halved and the transient speed droop was significantly reduced. Further tests were performed with 100% step load applications; the standard engine was not capable of accepting this load, whereas when air-injection was applied (again for 2 s) satisfactory load acceptance was achieved [6].

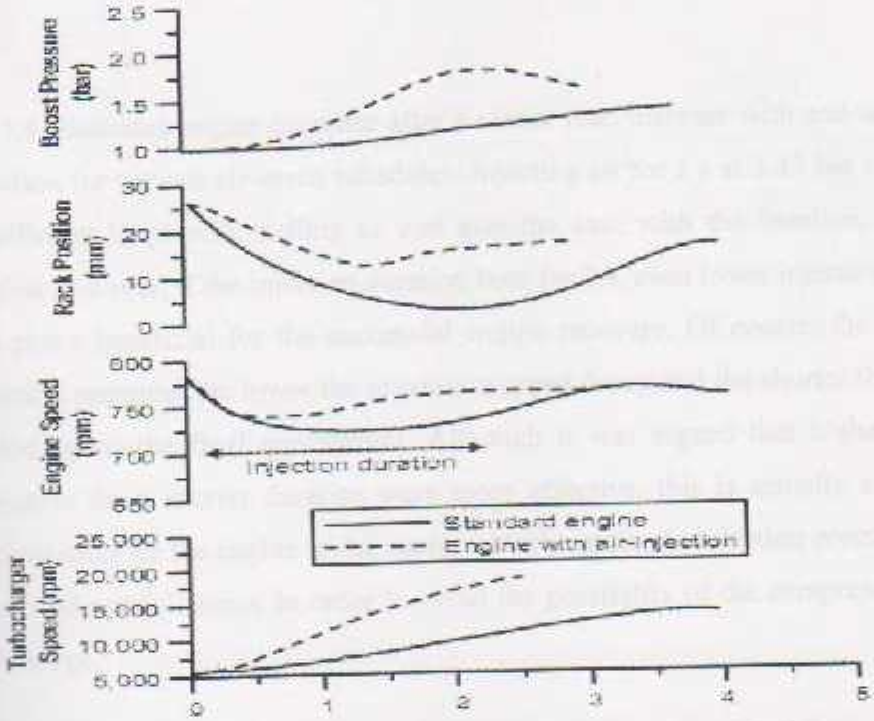

Figure 1.3 Effect of air-injection on transient response after a load step of 60% full-load (medium-speed, turbocharged diesel engine at 750 rpm). [6]

Further application of the latter air-injection assist on a six-cylinder, light-duty turbocharged diesel engine of 11.32 L displacement volume revealed a reduction in the time period with peak smoke from 3 to 0.5 s as well as improvement in the speed response from 4 to 1.5 s for a severe load acceptance transient at 1000 rpm. This was accomplished by applying 1s air-injection duration, which provided an air consumption of about 0.1 kg. Similarly encouraging results were reported for acceleration transients. Harndorf and Kuhnt [2] quantified the transient soot emissions improvement for a similar engine configuration at approximately 50% the amount of emissions under operation without air-injection; when combined compressor and turbine air-injection was applied, the improvement was even higher.

The particular importance in air-injection configurations is the exact injection schedule, i.e., onset and duration of injection as well as the pressure of the injected air.

Fig.1.4 illustrates engine response after a severe load increase with and without air-injection for various air-assist schedules. Injecting air for 1 s at 3.45 bar was proven insufficient to prevent stalling as was also the case with the baseline, unassisted engine; however, if the injection duration lasts for 2 s, even lower injection pressures can prove beneficial for the successful engine recovery. Of course, the higher the injection pressure, the lower the maximum speed droop and the shorter the recovery period up to the final equilibrium. Although it was argued that higher injection pressures for a shorter duration were more effective, this is actually a subject of optimization for the engine under study, with the optimum injection pressure being a matter of careful choice in order to avoid the possibility of the compressor moving onto surge.

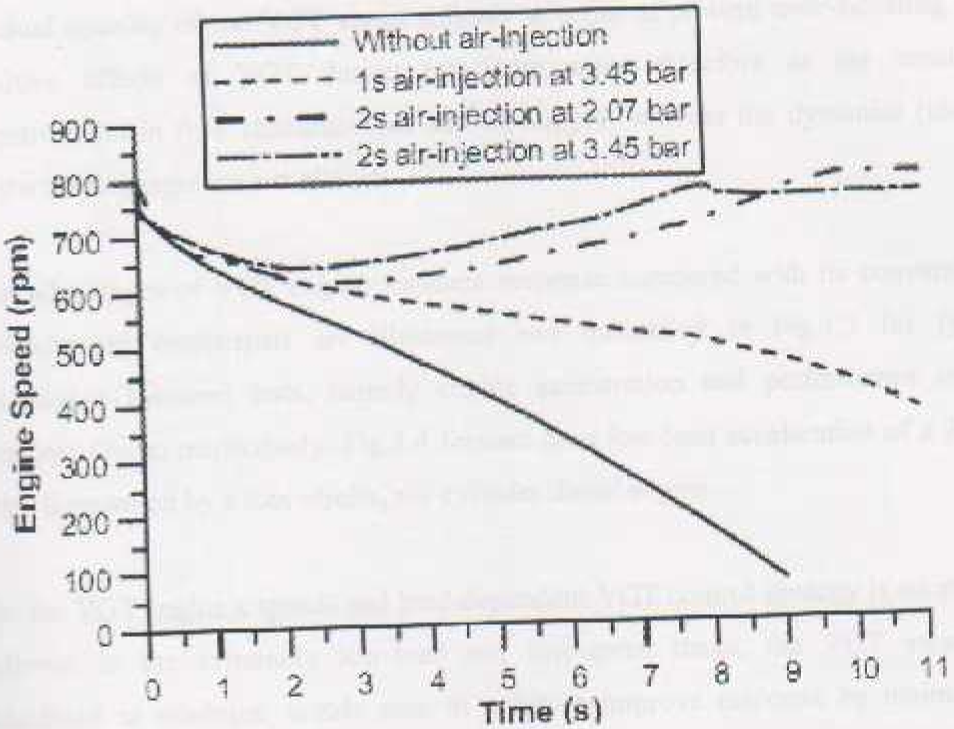

Figure 1.4 Comparison of transient response after a 0–16.88 bar brake mean effective pressure load increase with various air-injection schedules (four-stroke, six-cylinder, turbocharged diesel engine of 68 L displacement volume, rated at 746 kW) [4]

1.3.3 Variable Geometry Turbine

During transient operation the main strategy of the variable geometry turbine(VGT) is the reduction of the nozzle area by closing down the vanes so as to increase the back-pressure and enthalpy drop across the turbine, thereby boosting the compressor operating point. By so doing, faster build-up of the engine air-supply is established, minimizing fuel limiting function and improving derivability and emissions. At the same time, the EGR valve is closed to support filling of the cylinders with fresh air. As soon as the air-supply to the engine has been built-up, gradual opening of the VGT vanes follows in order to prevent over-boosting. The positive effects of VGT during transients come therefore as the result of improvement in flow characteristics and air-supply, whereas the dynamics (inertia) of the turbocharger are not altered.

The advantages of VGT engine transient response compared with its conventional turbocharged counterpart are illustrated and quantified in Fig.1.5 for typical automotive transient tests, namely engine acceleration and performance over a transient Cycle, respectively. Fig.1.4 focuses on a low-load acceleration of a 20 ton vehicle powered by a four-stroke, six-cylinder diesel engine.

For the VGT engine a speed- and load-dependent VGT control strategy is adopted as follows: in the extremely low-load and low-speed range, the VGT vanes are scheduled to minimize nozzle area in order to improve response by maintaining adequately high boost pressure. In the middle-load range, the vanes maximize nozzle area in order to improve fuel consumption by reducing the pumping loss caused by the positive difference between exhaust- and inlet manifold pressure. For the high-load operating range, nozzle vanes occupy three positions (minimum, middle and maximum) according to the increase in engine speed. Closing down of the VGT vanes right after the onset of the transient event results in much faster turbocharger acceleration and boost pressure build-up compared with the conventionally turbocharged engine, hence speed recovery is faster. For example, time up to the intermediate speed of 2100 rpm is reduced by 0.6 s (or 15%) for the VGT compared with the conventional vehicle. As soon as the boost pressure has assumed an adequate value, VGT vanes gradually open to avoid over-boosting, that is why a reduction in the delivery pressure at approximately $t = 3$ s is observed.[7]

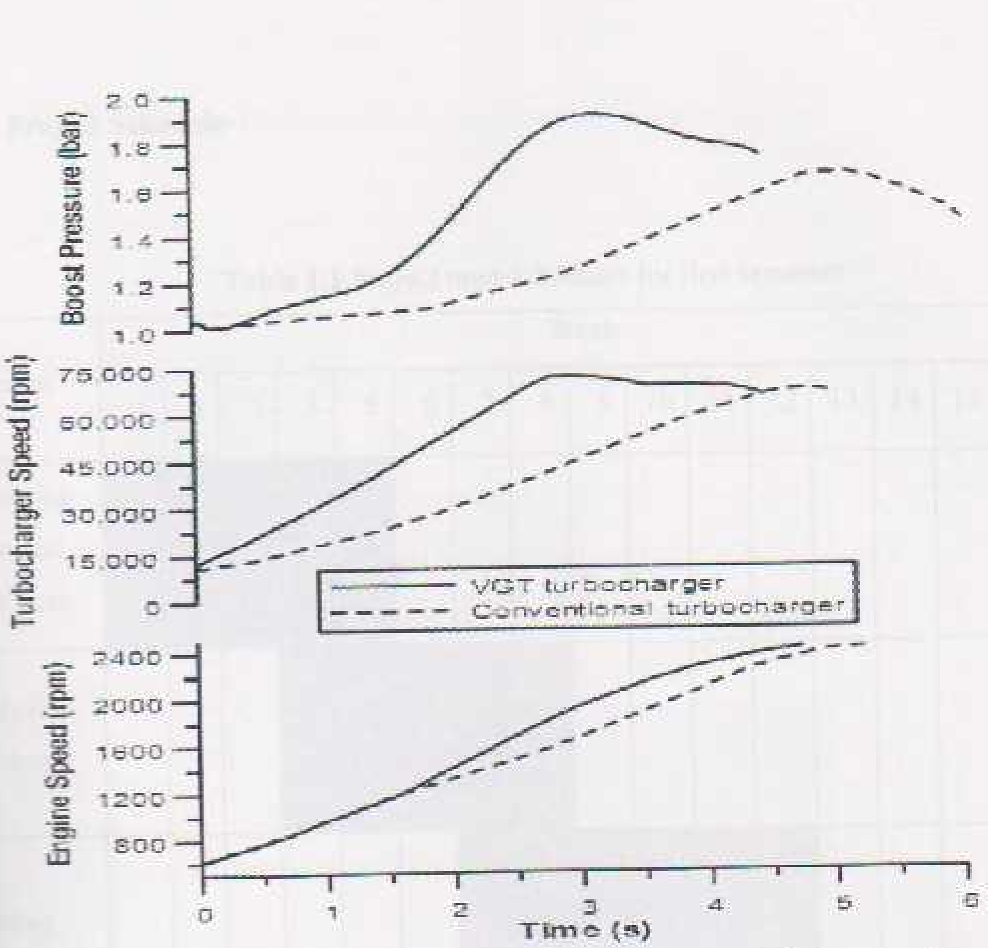

Figure 1.5 Effect of VGT on engine speed development during second gear acceleration from 600 rpm (six-cylinder, turbocharged diesel engine of 11 L displacement volume, installed on a 20 ton truck).[7]

1.5 Project Schedule

Table 1.1 Project time-schedules for first semester

Process	Week															
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Collecting Data and Literature																
Analyzing of data																
Modeling																
Simulation																
Writing The documentation																
First Presentation																

Table 1.2 Project time-schedules for second semester

Process	Week															
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Preparing the combustion engine	■	■	■	■												
Buying mechanical and pneumatic part				■	■	■	■	■								
Building experimental setup							■	■	■	■	■	■				
Chick the project parts and perform the initial experiment												■	■	■		
Perform final experiment													■	■	■	
Writing The documentation													■	■	■	
Final Presentation																■

1.6 Project Budget

The apparatus requirements are Turbocharger, Air injector, pressure sensor…). The budget of the project also includes printing costs and local study and survey. The following table shows the estimated cost of each one.

Table 1.3 Project budget

Element	Description	Availability	Students	DSR Grant
	Turbocharger and exhaust manifold			1350
	Manifold Pressure sensors			150
	Pressure regulator			140
	Pressure sensors, chick valves (on air tank)			100
	Diesel oil			100
	Air injector (Solenoid valve)			150
Experimentation	Workshop tools	Workshop in PPU		
	Work agreements and its factories (Blacksmith)			400
Transportation				200
Total cost (NIS)				2590

CHAPTER TWO

SYSTEM OPERATION PRINCIPLE

Content:

2.1 Introduction

Through the combination of parallel developments of recent years in the automotive industry, electronic injection, e.g. Common Rail, in the commercial vehicle diesel engine sector, and highly dynamic, electronically controlled compressed air, it can easily be integrated into the charge air pipe, and realizes an immense improvement to the response and acceleration behavior of the engine and the vehicle.

Automotive development is caught between the conflicting priorities of increasing demands placed by customers and the legislature, which can only be met with technological progress and in-depth knowledge. This challenge is compounded by diverging markets with regional preferences and increasing energy costs caused by growing demand, increasing fuel prices and more stringent emission regulations are the most powerful driving force behind the developments. This particularly applies in the commercial vehicle sector, where optimal efficiency is the permanent objective. The commercial vehicle of the future is characterized by high cost-effectiveness and meets all applicable emission regulations. Mobility will remain at least at the current level, but will more likely increase.

The principle known as downsizing is an effective means of achieving future consumption and emission objectives. This requires a further increase to the charging pressure to maintain the engines' performance level. A compact, cost-efficient and proven solution to this is exhaust-gas turbocharging, which, however, reacts with a further undesired delay at transient full load demand conditions by increased charging-air pressures. Even multi-stage and variable charging systems are only able to lessen this effect. The delay to the charging-air pressure buildup impairs the vehicle's drivability, particularly when setting off, changing gears, driving up hills or overtaking, and can only be compensated for by changing gear more frequently or increased clutch slipping.

In general, the technologies mentioned above, with further increasing of the charging-air pressure, cause a deterioration of the engine's response behavior, which is why new, alternative approaches to improving the dynamic drivability are required.

The development of such an approach, the air injection system (AIS), will be presented in this project. It effectively improves the dynamic properties of turbocharged diesel engines, particularly for commercial vehicles and buses. The AIS utilizes the compressed-air conditioning available for the brake system in these vehicles and assists the turbocharger with additional, very quick air injection in transient situations.

2.2 Description

The present project relates to an apparatus and method for improving vehicle performance in a number of areas; including acceleration, fuel economy and emissions reduction. In particular it relates to an apparatus and method for application of air boost to vehicle engines, including vehicle diesel engines having at least one turbocharger supplying air to the engine's intake manifoldFig.2.1, in a manner which increases engine torque output in a manner which meets design, regulatory and other requirements.

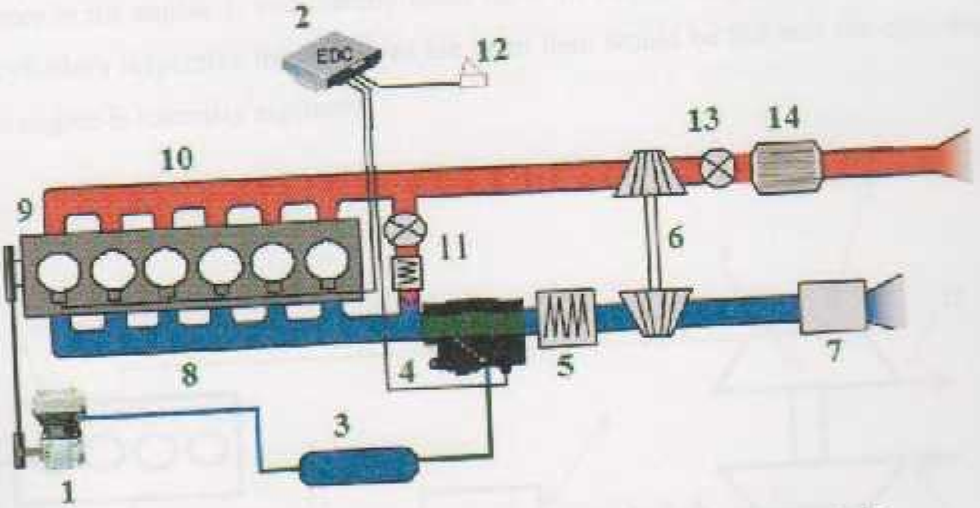

Figure 2.1A schematic illustration of an engine and related components in accordance with an embodiment of the project.

1. Compressor
2. Engine control
3. Vehicle air reservoir
4. AIS module
5. Intercooler
6. Turbocharger
7. Air filter
8. Intake manifold
9. Diesel engine
10. Exhaust manifold
11. EGR valve
12. Accelerator pedal
13. Exhaust flap break
14. Diesel particle filter

2.3 Background of the project

Internal combustion engines, such as diesel engines, are often fitted with exhaust-gas turbochargers. For example, fig.2.2 shows a schematic illustration of an internal combustion engine 1 having an exhaust line 10 which is coupled to an exhaust-gas turbocharger 2. The exhaust-gas turbocharger has a turbine 4 which is driven by exhaust gas from exhaust line 10. The turbine 4 is coupled to a compressor 3 (together these components form turbocharger impeller unit) which compresses intake air from an intake air inlet 12. The compressed air discharged from the compressor 3 is fed to an intake line 11 for the engine 1 in order to increase the air pressure in the engine 1, and thereby feeds more air into the engine's cylinders when the cylinder's respective intake valves are open than would be fed into the cylinders if the engine is naturally aspirated.

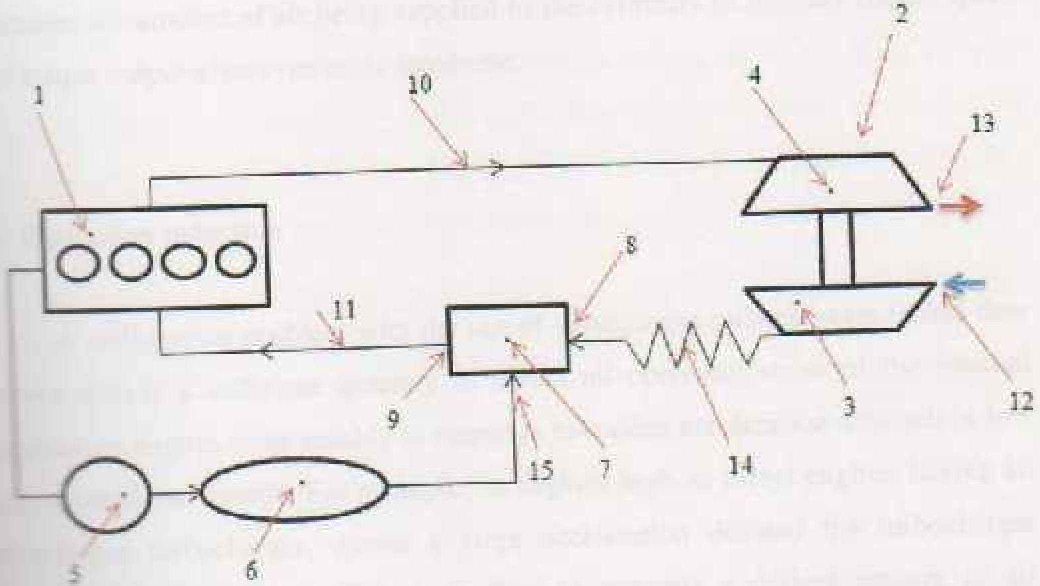

Figure 2.2 Schematic illustrations of diesel engine have turbocharger and air booster system.

As a result of the turbocharger's supply of additional air into the engine cylinders, along with associated additional fuel from the engine's fuel injection system, the torque output of the engine is increased and the engine operates at a higher efficiency. Specifically, the additional pressure delivered by the turbocharger to the intake manifold results in greater pressure in the engine cylinder when the cylinder's intake valve closes. The greater mass of air present in the cylinder, when combined with additional fuel and ignited, results in higher combustion pressure, and thus higher piston force to be converted by the engine's crankshaft into higher engine torque output.

In addition, the increased combustion mass and pressure generates a higher pressure and volume of exhaust gases, which in turn provides additional energy in the exhaust for driving the turbine of the turbocharger. The increased exhaust energy further increases the rotational speed of the turbocharger compressor and thereby further increases the amount of air being supplied to the cylinders to increase engine speed and torque output at an even more rapid rate.

2.4 Operation principle

A well-known problem with the use of exhaust-gas turbochargers is that they cannot deliver a sufficient quantity of air in all operating states of the internal combustion engine, most notably in response to sudden acceleration demands at low engine rotational speeds. For example, in engines such as diesel engines having an exhaust-gas turbocharger, during a large acceleration demand the turbocharger typically cannot supply sufficient air flow to generate a desired amount of air pressure in the intake manifold due to the low engine speed and correspondingly low mass flow rate of air intake and exhaust output to drive the turbocharger. As a result, the internal combustion engine reacts slowly, with significant torque output and rotational speed increases occurring only after a notable delay after the accelerator pedal is pressed (an effect known as "turbo lag").

Various solutions have been proposed to improve the effects of "turbo lag," including arrangements in which compressed air is supplied to the intake manifold of the engine. An example of such an "air injection" system is illustrated in fig.2.2. In this example, reservoir 6 stores compressed air generated by an air compressor 5. The compressed air is introduced into the intake line 11 of the engine 1 in response to a demand for increase engine torque output during the transient period between the start of the acceleration demand and the time at which the turbocharger has built up enough pressure to equalize with the intake manifold pressure and begin to meet the torque output demand on its own.

The additional air supplied into the intake line 11 from reservoir 6 has at least two primary effects. The additional combustion air fed to the cylinders of the engine 1 provides an immediate increase in engine torque output. The additional air also results in a more rapid increase in exhaust gas flow from the engine, which in turn helps the turbocharger turbine 4 to more rapidly increase its rotational speed, thus enabling the turbocharger compressor 3 to build pressure in the intake line 11 faster.

Further, the sooner the turbocharger compressor can supply enough pressure to support the torque output demand, the sooner the flow of additional air being supplied from reservoir 6 may be to stop, preserving compressed air for other uses and reducing the duty cycle of the vehicle's air compressor.

The injection of compressed air from reservoir 6 takes place via an intake air control device 7. The intake air control device 7 is arranged between the intake line 11 and either the compressor 3 of the turbocharger, as shown in Fig.2.2, the charge-air cooler 14 downstream from the compressor 3. The intake air control device 7, illustrated schematically in fig.2.3, is connected with an inlet 8 to the charge-air cooler 14 and with an outlet 9 to the intake line 11.

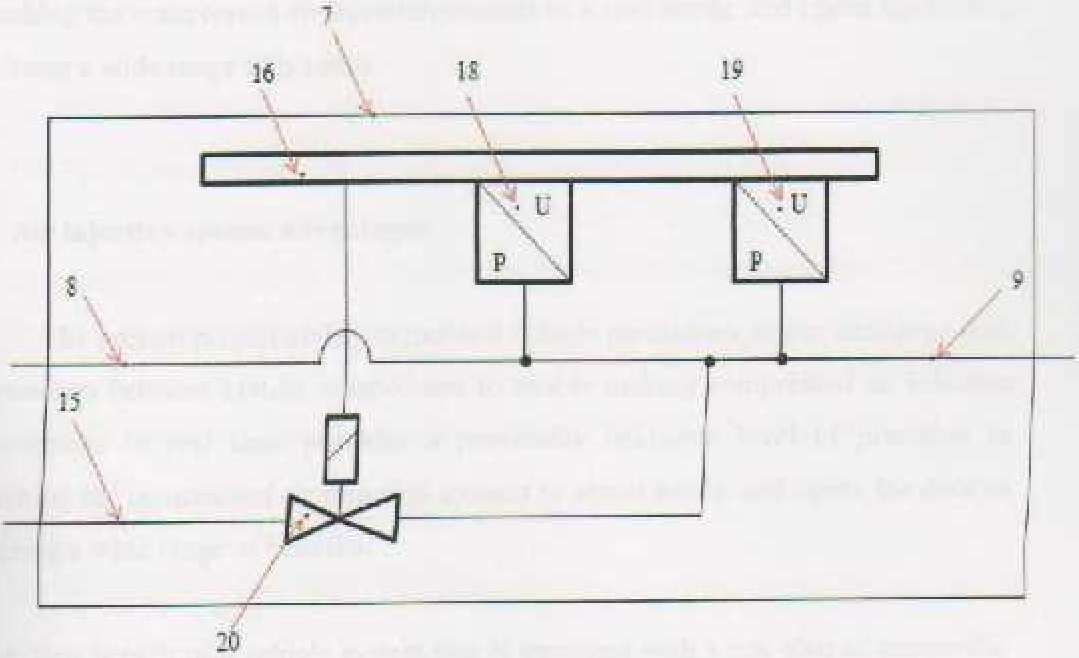

Figure 2.3 A schematic illustration of the intake air control device

A compressed air inlet 15 is connected to the outlet 9 to the reservoir 6 via a flow-regulating device 20. A controller 16 serves to control the flow-regulating device 20. The control device 16 receives inputs from pressure sensors 18 and 19, which measure, respectively, an outlet pressure at the outlet 9 and an inlet pressure at the charge-air inlet 8.

In operation, the flow-regulating device 20 supplies compressed air to the engine intake manifold by opening the connection from the compressed-air inlet 8 to the outlet 9.

The present project ability to monitor vehicle parameters and/or exchange such parameters between system components to enable making compressed air injection adjustments in real time provides a previously unknown level of precision in matching the compressed air injection amount to actual needs, and opens the door to realizing a wide range of benefits.

2.5 Air injection system advantages.

The present project ability to monitor vehicle parameters and/or exchange such parameters between system components to enable making compressed air injection adjustments in real time provides a previously unknown level of precision in matching the compressed air injection amount to actual needs, and opens the door to realizing a wide range of benefits:

1. A first benefit of a vehicle system that is equipped with a rate-shaped pneumatic booster system is that significant fuel efficiency increases may be obtained. The fuel savings result from the use of rate-shaped compressed air injection to improve combustion and exhaust generation to more rapidly get the engine into the engine speed range at which the engine is operating at its most efficient fuel efficiency (often referred to as the engine's "sweet spot") and thereby get the vehicle to the desired cruising speed in the least amount of time, and with the least amount of fuel consumption possible while still avoiding operational, emissions and/or equipment engineering limits.

2. An additional benefit with the system rate shaping is that the vehicle designer can avoid unnecessary compressed air use and thus decrease the size and cost of the vehicle's installed compressed air generation and storage equipment. Specifically, by injecting only the actual amount of compressed air required to obtain a desired

vehicle acceleration while still maintaining compliance with operating limits, and doing so only at the actual times the compressed air is needed during the pneumatic boost event, this system can obtain a desired level of engine torque output with less compressed air than typically consumed by another pneumatic booster systems.

3. The increased precision in compressed air injection decreases the volume of compressed air required during vehicle operations, allowing the vehicle designer to reduce the size of the compressed air generation and storage components to match the lower compressed air demands. These reductions in component size and capacity provide further fuel economy benefits, both due to reduced vehicle weight and due to reduced energy loss from the vehicle's air compressor.

2.6 Air injection systems disadvantages:

While it has previously been known to inject compressed air into the intake manifold of an engine to reduce "turbo-lag," work in this field has primarily concentrated on maximizing the amount of compressed air available to flow into the engine intake manifold, and on minimizing the response time from the initiation of the pneumatic boost event to the actual injection of compressed air so as to immediately begin to increase engine torque output and avoid undesired operator-perceived delays in delivery of torque from the engine.

1. One problem with this system is the sometimes very abrupt increase in engine torque output at the beginning of a pneumatic boost event injection resulting from very rapid compressed air injection. Such sharp engine torque output transients may also be experienced at the subsequent termination of compressed air injection this transient can create significant discomfort to the vehicle operator and passengers.

2. Another problem with air injection systems is that, in the rush to quickly boost engine torque output until the turbocharger has built up sufficient pressure, regulatory limits such as pollution emissions limits may be exceeded. The sudden application of excessive pneumatic boost also has the potential to impose sudden loads on the engine components. For example, sudden application of excessive pneumatic boost can apply a large amount of torque to the vehicle drivetrain which may approach engine, transmission and/or drive axle stress limits.

3. Excessive pneumatic boost may also generate a sudden high volume, high pressure flow of exhaust gases from the engine which can cause the speed of the turbocharger turbine-compressor assembly to rise to high levels. Similarly, sudden compressed air injection and accompanying increased exhaust gas flow can create the potential for over-pressuring the engine's intake air intercooler and its associated piping.

CHAPTER THREE

SYSTEM CONTROLLER

Content:

3.1 Introduction

The air injection system controller is configured to receive vehicle operating information from the vehicle prior to initiating the event of boosting air and to control the operation of the compressed air flow control valve to adjust of air injection rate, duration and timing for a first of the compressed air injection pulses based on the received vehicle operating information.

The air injection controller is programmed to adjust of the compressed air injection rate, injection duration and injection timing to maintain the at least one operating parameter within the predetermined range by pulse-width modulation of compressed air injection control device.

3.2 Operating parameters

The controller is programmed to control the pneumatic boost event based on at least one monitored operating parameter of the vehicle by coordinating the operation of the at least one compressed air flow control valve to supply compressed air to the engine via the intake, and the air booster controller is programmed to set the compressed air injection rate in accordance with the at least one monitored operating parameter.

1. A pressure of compressed air stored in a compressed air storage vessel, and the predetermined range is a pressure higher than a minimum compressed air pressure required to ensure vehicle safety systems have sufficient compressed air available to perform the safety system function.

2. A pressure at least one of an inlet and an outlet of an air injection system compressed air injection module, and the predetermined range is a predetermined allowable difference between the pressure at least one of an inlet and an outlet of an air injection system compressed air injection module, and a pressure at least one of the turbocharger compressor and a location in the intake downstream of the pneumatic booster system compressed air injection module.

3. An intake pressure and the predetermined range is a pressure below an intercooler over-pressure limit.

4. A pressure downstream of a turbocharger compressor and the predetermined range is a pressure variation rate which is indicative of a turbocharger impeller speed variation being below a predetermined impeller speed variation corresponding to turbocharger surging.

5. A rotational speed of the engine and the predetermined range is an engine speed which is lower than an engine over speed limit.

6. A rotational speed of a compressor of a turbocharger which supplies the engine with combustion air, and the predetermined range is a turbocharger impeller speed which is lower than a turbocharger compressor over speed limit.

7. Vehicle acceleration and the predetermined range is an acceleration which is lower than a maximum acceleration limit for maintaining passenger comfort in the vehicle.

8. Vehicle acceleration and the predetermined range is an acceleration rate which is lower than an operator-selectable acceleration profile.

9. An engine operating temperature and the predetermined range is a temperature above a predetermined minimum engine operating limit temperature.

10. An operating parameter of the vehicle indicative of an actual air flow rate in the intake, and the *predetermined range is a predetermined allowable difference* between the actual air flow rate in the intake and an air flow rate demanded by at least one controller of the vehicle.

11. A driver acceleration request and the predetermined range is at least one of an accelerator pedal position a frequency of accelerator position pedal exceeding a predetermined position.

3.3 Flowchart of control

The air booster controller is programmed to set the compressed air injection rate by reference to the at least one monitored operating parameter and a look-up table correlating the at least one operating parameter to predetermined compressed air injection rate profiles.

The following describes criteria and logic flow for the initiation of air boost event in an embodiment of the project, with reference to Figures 3.1 to 3.4.

As will show in Fig.3.1, a number of inputs are received by the air injection system controller, either from a CAN bus connection or separate communication links. These inputs include, for example:

1. Engine status and parameter information received from, e.g. the engine controller and/or directly from engine-related sensors.
2. Information in the boost state of the engine from, e.g. the engine controller and/or pressure sensors in the intake tract.

3. Vehicle emissions performance information obtained, e.g. directly from exhaust sensors and/or other control modules.

4. Vehicle emissions performance information obtained, e.g. directly from exhaust sensors and/or other control modules.

5. Air brake system status information from, e.g. sensors (such as a compressed air storage reservoir pressure sensors, a brake pedal position sensor and/or a wheel speed sensor), a vehicle brake controller and/or a vehicle stability control system controller, and other vehicle equipment status information (such as air compressor engaged/disengaged status and/or other power take-off equipment operating state).

Preferably, input parameters to be considered in the evaluation of air boost event activation and deactivation include engine speed, intercooler pressure (a measure of air boost status), pressure present in the air booster system compressed air supply, accelerator pedal position and position rate of change (and/or alternatively, frequency of acceleration position exceeding a predetermined position), and the transmission gear, clutch state and current shifting status (e.g. upshifting or downshifting).

At a minimum, knowledge of intake manifold pressure and accelerator pedal position is needed, however alternative and/or supplemental inputs include: for engine-related information, turbocharger rpm engine torque output, engine load, coolant temperature and exhaust gas mass flow rate; for engine air boosting-related information, intake manifold pressure, intake pressure measured in the intake tract

upstream of the intake manifold and the intake air mass flow rate; for emissions-related information, EGR mass flow rate, DPF (diesel particulate filter) regeneration state and NOx after-treatment system availability (e.g., status of exhaust line SCR and/or NOx absorber components); for air brake system information, status of anti-lock brake system activation (in the case of tractor-trailer vehicles, preferably the ABS status of both the tractor and the trailer brakes), brake pedal position, parking brake status and trailer stability status; for other vehicle systems, the vehicle ignition status and cruise control status.

It will be readily apparent to those of skill in the art that the foregoing is an illustrative, but not exhaustive, listing of parameters and system status indications which may be considered as inputs to the air booster system controller for determining whether to activate or deactivate air booster system, and that in the course of implementation of an embodiment of the present project the system design will determined from each vehicle's various available parameter and system status sources which inputs will be provided to the air booster system controller.

Applying the control logic in the embodiment illustrated in figures 3.2 to 3.4, the air booster system controller outputs control signals to activate or deactivate an air boost event via control of the solenoid valves in the air control device.

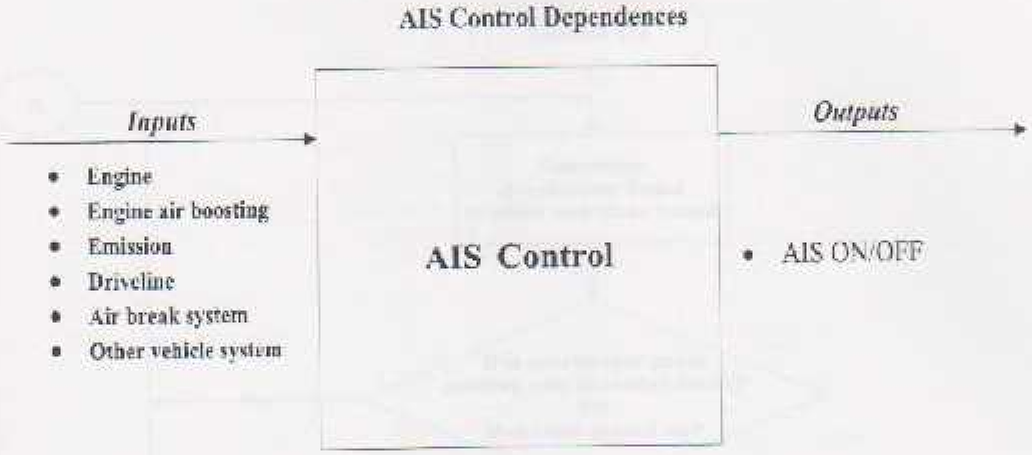

Figure 3.1 Identifying control dependencies for initiation and deactivation of air boost event in accordance with an embodiment of the project.

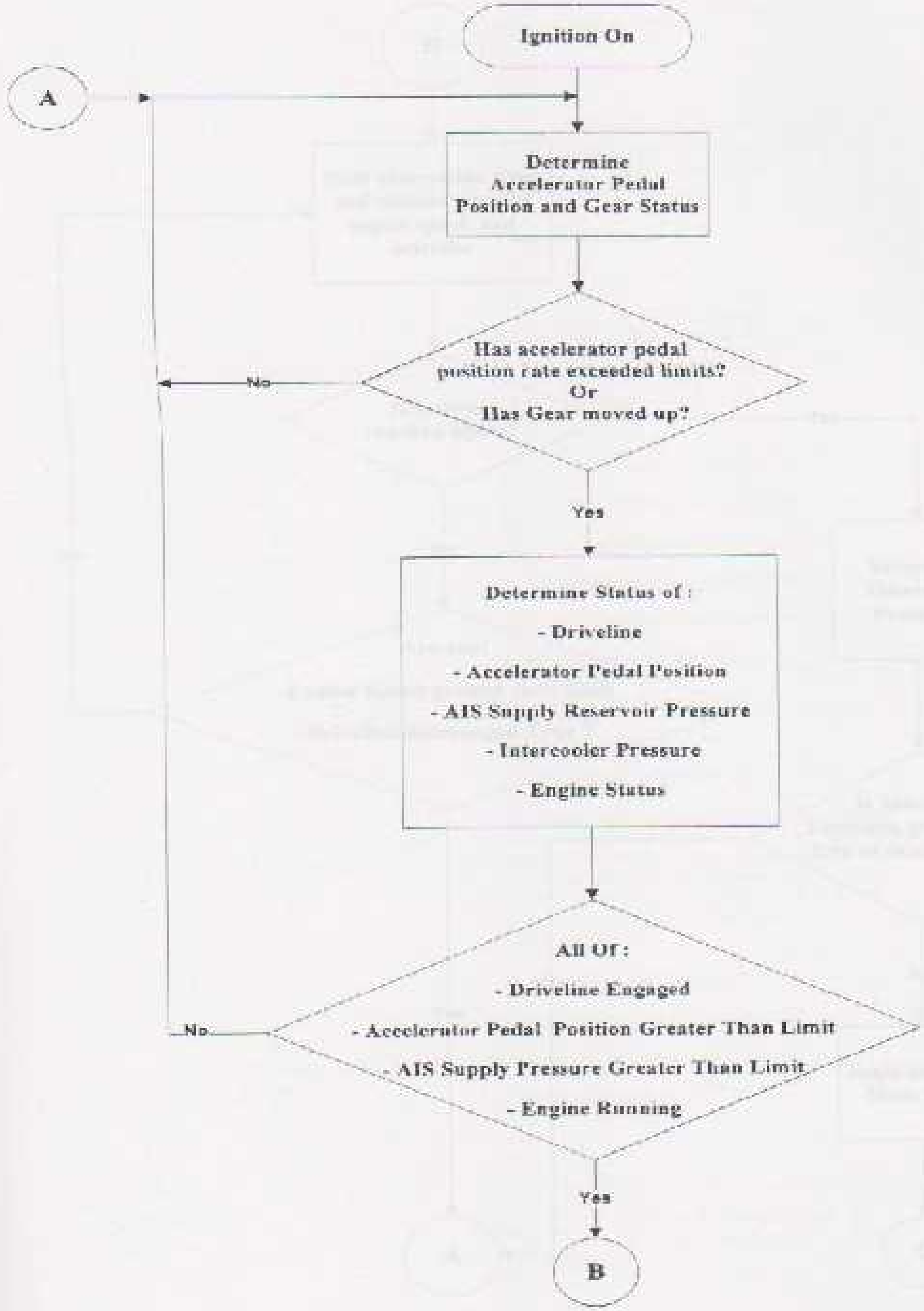

Figure 3.2 Is a flow chart illustrating a first portion of control decisions and dependencies for initiation and deactivation of air boost event in accordance with an embodiment of the project.

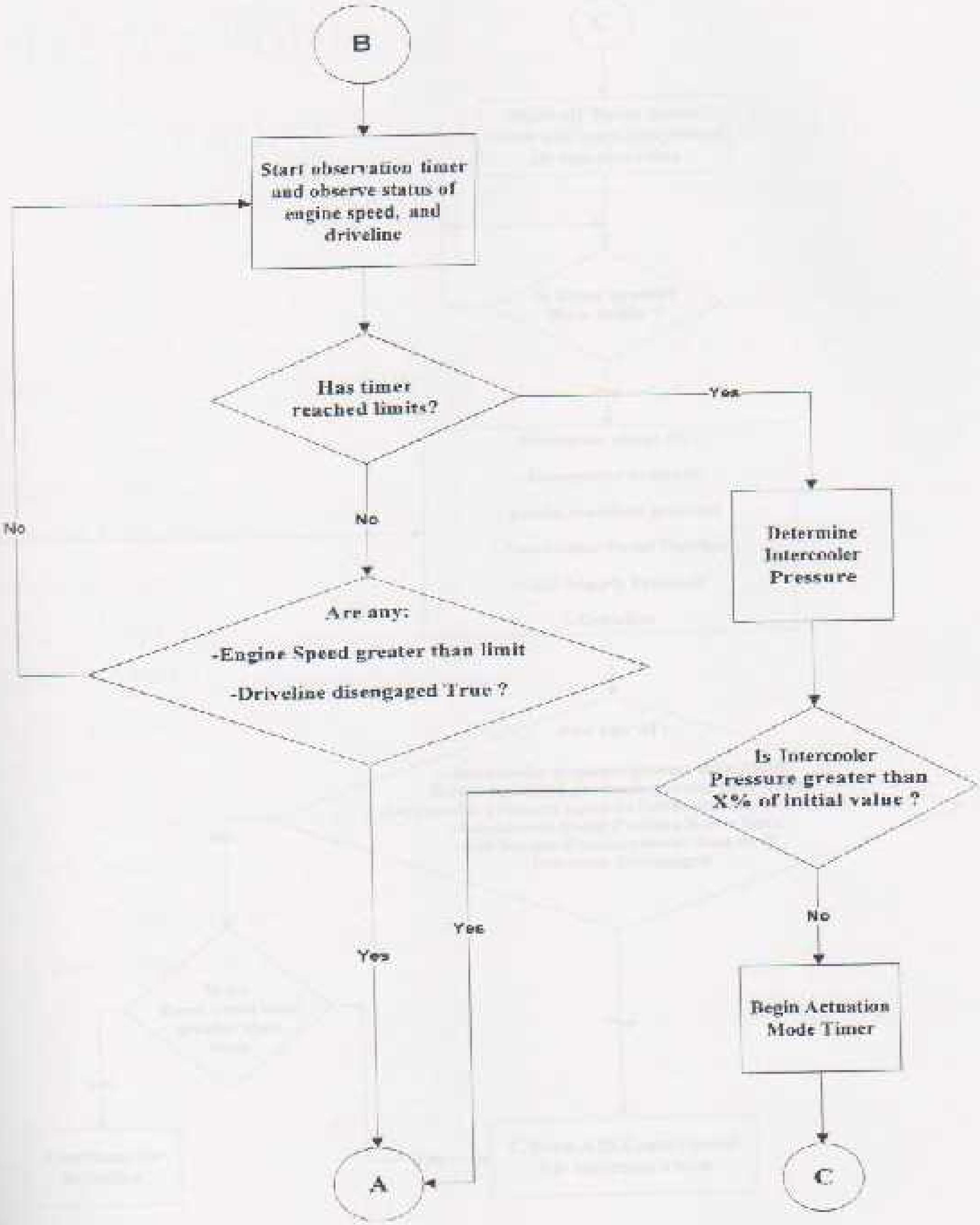

Figure 3.3 Is a flow chart illustrating a second portion of control decisions and dependencies for initiation and deactivation of air boost event in accordance with an embodiment of the project.

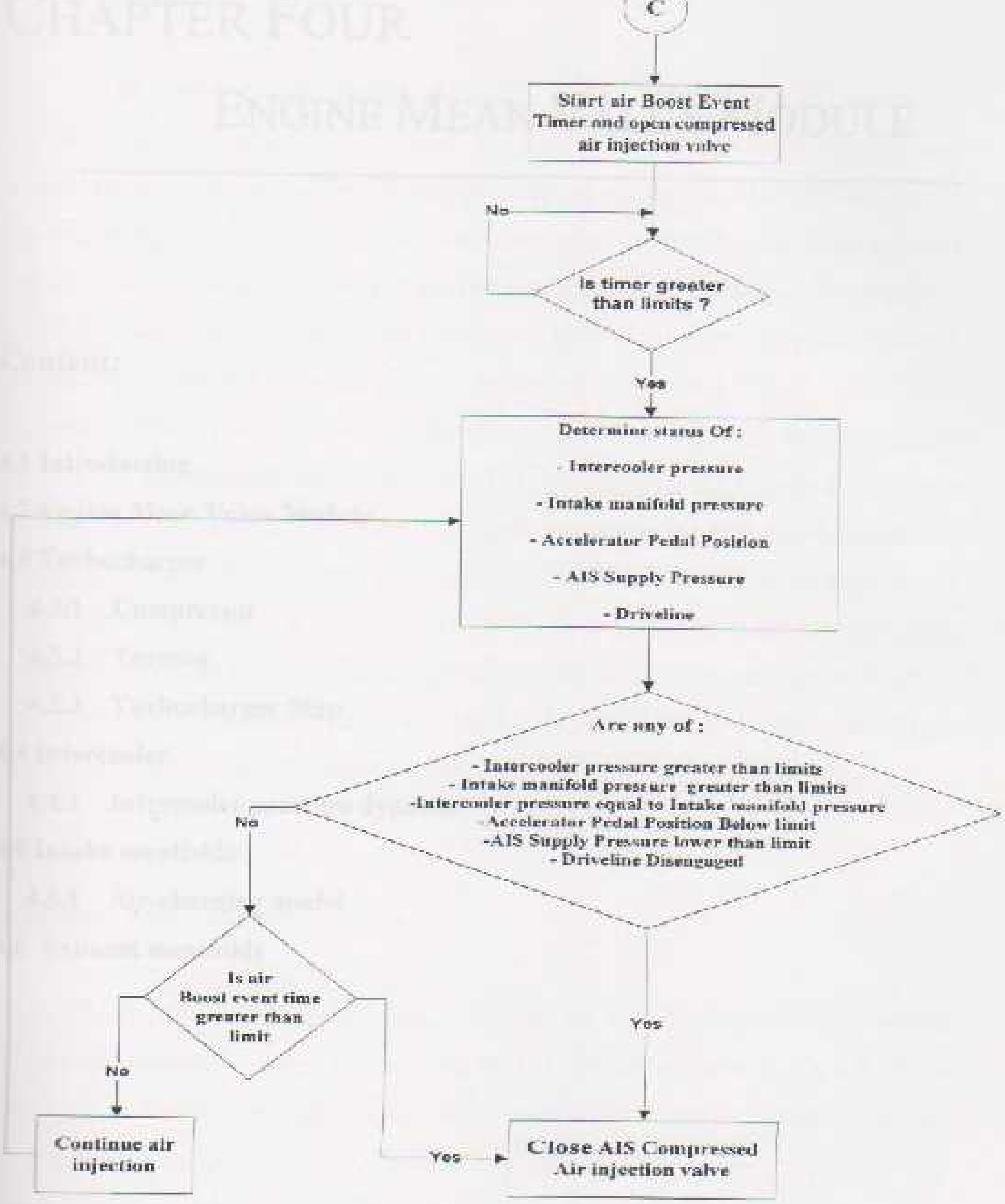

Figure 3.4 Is a flow chart illustrating a third portion of control decisions and dependencies for initiation and deactivation of air boost event in accordance with an embodiment of the project.

CHAPTER FOUR

ENGINE MEAN VALUE MODULE

Content:

4.1 Introduction

Mean Value Engine Model (MVEM) is the basis of control design for advanced internal combustion engines. The engine performance transient process usually takes a few cycles. The MVEM provides an adequate accurate description of the engine dynamics with reasonable approximation by ignoring the heat loss and sub-cycle events. MVEM is very important for engine system control development, especially when the modern engine becomes more and more complicated when equipped with throttle, turbocharger and after-treatment systems. Usually the MVEM is developed based on data from engine tests, which is a costly and time consuming process. In this chapter, the air path MVEM modeling method is discussed for a turbocharged diesel engine. Simulation is applied to demonstrate the effectiveness of this new method. This approach could be used to get the MVEM for control design even before the prototype engine is available. It reduces the cost, risk and labor compared with the test data based approach. This MVEM model can be built in modules and the parameters can be validated for a specific engine. These advantages make it applicable to a wide range of engines.

4.2 Mean Value Engine Module

The control oriented sub-system models of a turbocharged diesel engine system are presented in this section. The system layout is shown in fig.4.1. These sub-system models include compressor, intercooler, intake manifold, engine combustion, exhaust manifold, turbine, waste gate, and turbocharger dynamic. The detail process of modeling and verification of each module are introduced.

For the mean values model, it was assumed that the air obeys the ideal gas law, the pressure is uniform in the intake and exhaust manifold, and there is not any heat losses to the walls. In combustion, heat is released in the whole combustion chamber at homogeneous conditions, and the gases can be regarded as ideal gases.

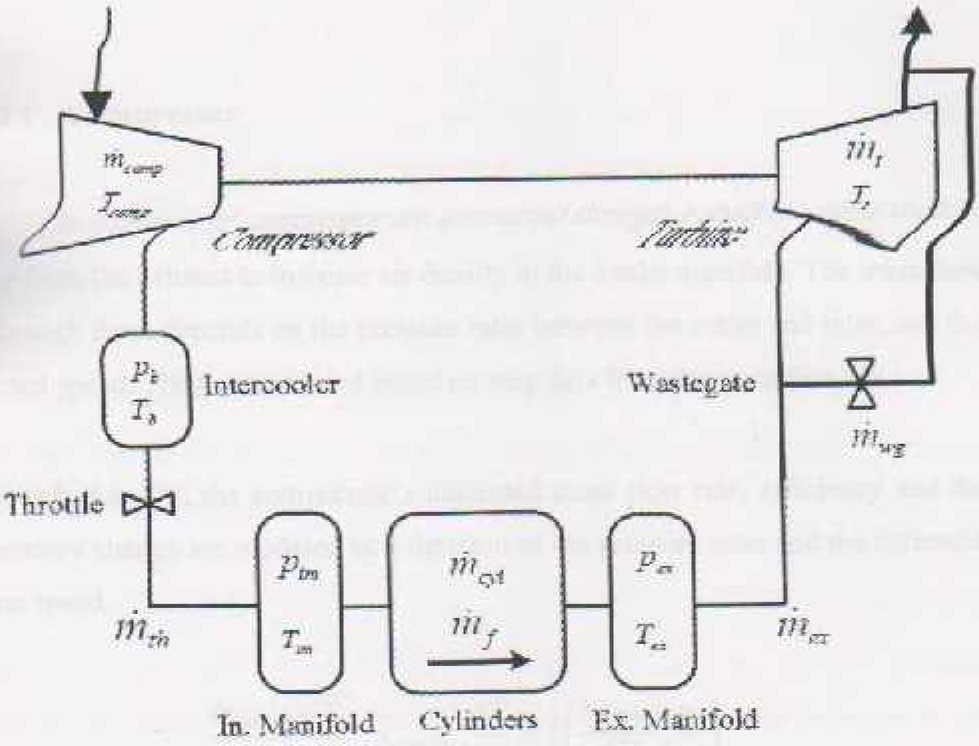

Figure 4.1 Engine structure and main variables of MVEM

4.3 Turbocharger

The turbocharger consists of a turbine driven by the exhaust gas and connected via a common shaft to the compressor, which compresses the air in the intake. The rotational speed of the turbocharger shaft $N_{turb.}$ can be derived as a power balance between the power of turbine $P_{turb.}$ and the power of compressor side $P_{comp.}$ which is it the rate of change in the turbocharger speed $\dot{N}_{turb.}$

$$\dot{N}_{turb.} = \left(\frac{60}{2\pi}\right)^2 \frac{P_{turb.} - P_{comp.}}{I_{turb.} - N_{turb.}} \tag{4.1}$$

Where the turbocharger speed is measured in revolutions per minute (rpm) and $I_{turb.}$ is the inertia of the turbocharger. Subsequently, the expressions for the compressor and turbine power are derived separately.

4.3.1 Compressor

The turbine and compressor are connected through a shaft to utilize the kinetic energy from the exhaust to increase air density in the intake manifold. The mass flow rate through them depends on the pressure ratio between the outlet and inlet, and the corrected speed. They are modeled based on map data from the manufacturer.

As described in [8], the compressor's corrected mass flow rate, efficiency and the temperature change are modeled as a function of the pressure ratio and the corrected turbine speed.

$$\frac{\dot{m}_{comp.}\sqrt{T_a}}{p_a}, \eta_{comp.}, \frac{\Delta T}{T_a} = f\left(\frac{N_{turb.}}{\sqrt{T_a}}, \frac{p_c}{p_a}\right)$$

The inlet of the compressor is assumed at ambient conditions.

$$p_a = p_{atm.}$$
$$T_a = T_{atm.}$$

The air flow into the intake manifold from the compressor($\dot{m}_{comp.}$) is determined from the compressor power equation

$$\dot{m}_{comp.} = \frac{\eta_{comp.}}{c_p T_a} * \frac{P_c}{\left(\frac{p_c}{p_a}\right)^{\frac{\gamma-1}{\gamma}} - 1} \qquad 4.2$$

The compressor efficiency ($\eta_{comp.}$) cannot be measured directly and has to be calculated based on the pressure and temperature ratios across the compressor.

$$\eta_{comp.} = \frac{\left(\frac{p_c}{p_a}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{T_c}{T_a}\right)^{\frac{\gamma-1}{\gamma}} - 1} \qquad 4.3$$

Where the P_c is the compressor power and p_c and T_c is the air pressure and temperature developed by the compressor respectively, p_a and T_a is pressure and temperature ofambient air respectively, c_p is the specific heat of ambient air, and γ is the air specific heat ration.

To avoid using complex function and regression, we have to use two pressure sensors in the intake and exhaust manifold, these two sensor used to measure the compressor pressure and the exhaust gas pressure.

4.3.2 Turbine

The expressions for the turbine power ($P_{turb.}$) and efficiency($\eta_{turb.}$)

$$P_{turb.} = \dot{m}_{turb.} c_{p\ ex.}\ \eta_{turb.} T_{ex.} \left(1 - \left(\frac{p_{turb.}}{p_{ex.}}\right)^{\frac{\gamma-1}{\gamma}}\right) \qquad 4.4$$

$$\eta_{turb.} = \frac{1 - \left(\frac{T_{turb.}}{T_{ex.}}\right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{p_{turb.}}{p_{ex.}}\right)^{\frac{\gamma-1}{\gamma}}} \qquad 4.5$$

Where the $p_{turb.}$ and $T_{turb.}$ is the pressure and temperature after the turbine respectively and $p_{ex.}$ and $T_{ex.}$ is pressure and temperature of the exhaust gas respectively, and $\dot{m}_{turb.}$ is the mass flow rate of the exhaust gas and $c_{p\,ex.}$ is the specific heat of the exhaust gas.

The power transfer between the turbine and the compressor is modeled by

$$\tau\dot{P}_{comp.} + P_{comp.} = \eta_{turb.}P_{turb.} \qquad 4.6$$

Where $\eta_{turb.}$ and τ are the turbo efficiency and the turbo-lag time constants respectively and $\dot{P}_{comp.}$ is the rate of change of the compressor power.

4.3.3 Turbocharger map

The main characteristic of a turbocharged engine is the lack of mechanical connection between turbocharger and engine crankshaft. Consequently, turbocharger and engine speeds are not mechanically but only indirectly interrelated. Turbocharger compressor boost pressure and air-mass flow rate are typically interconnected via maps such as the one illustrated in fig.4.2 showing also contours of constant isentropic efficiency and rotational speed.

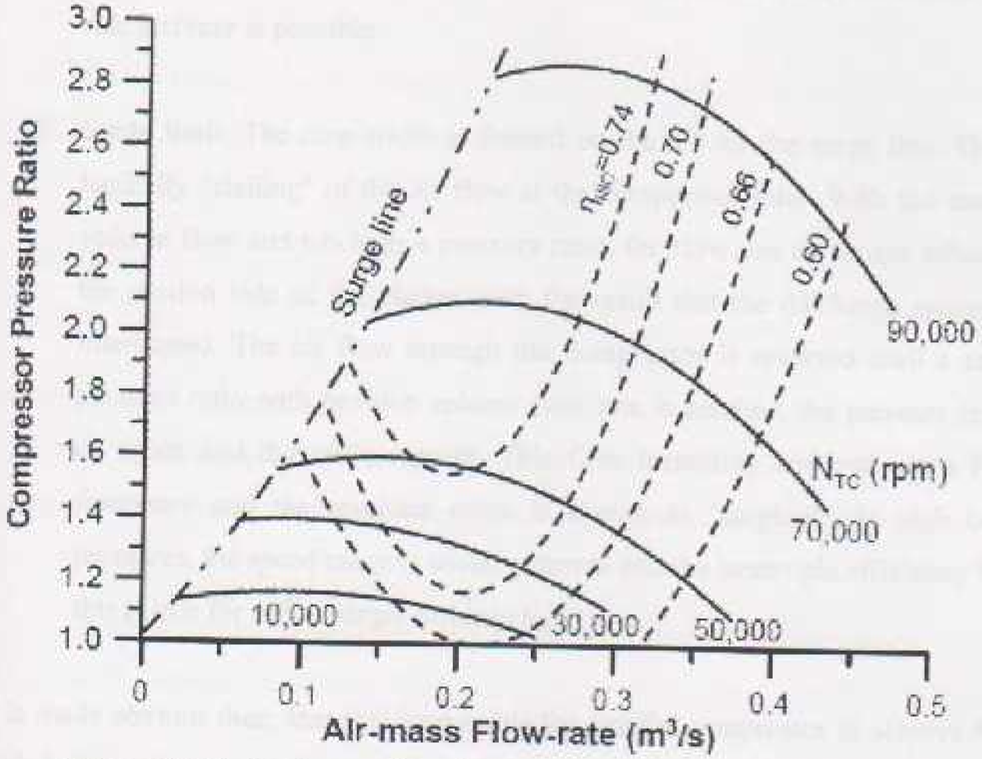

Figure 4.2 Typical map of aerodynamic type turbocharger compressor

The non-linearity of the compressor characteristics is apparent. Considering the constant speed parameter lines it is observed that their slopes vary from almost perfectly horizontal near the surge limit to almost vertical at high flows. The contours of constant efficiency are roughly elliptical.

There are several limits of compressor operation:

1. Maximum speed limit: Turbocharger speed should be kept below this speed to avoid excessive centrifugal forces.

2. Choke limit at the right of the map: The maximum centrifugal compressor mass flow rate is normally limited by the cross-section at the compressor

inlet. When the flow at the wheel inlet reaches sonic velocity, no further flow rate increase is possible.

3. Surge limit: The map width is limited on the left by the surge line. This is basically "stalling" of the air flow at the compressor inlet. With too small a volume flow and too high a pressure ratio, the flow can no longer adhere to the suction side of the blades, with the result that the discharge process is interrupted. The air flow through the compressor is reversed until a stable pressure ratio with positive volume flow rate is reached, the pressure builds up again and the cycle repeats. This flow instability continues at a fixed frequency and the resultant noise is known as "surging". At high boost pressures, the speed range is usually narrow and the isentropic efficiency low; this is true for turbocharger turbines too.

It is made obvious then, that it is impossible for a radial compressor to achieve both high boost pressure and air-mass flow-rate when operating at low rotational speed. Instead, acceleration of the turbocharger to a higher speed is required, which, unfortunately, is delayed considerably by the turbocharger's mass moment of inertia. At steady-state conditions, the turbine power is equal to the sum of the compressor power and the losses in the turbocharger shaft bearings, so both compressor and turbine are in an equilibrium state. If the engine operating condition is changed after a ramp increase in load or fueling, the turbocharger needs to accelerate to a new operating point in order to provide the required higher boost. To achieve that, the turbine power must exceed the compressor power.

4.4 Intercooler

The air temperature T_c increases after the compression, so the intercooler was used to cool it down. The air temperature at the exit of the intercooler T_b is given by

$$T_b = T_c - \varepsilon(T_c - T_{coolant}) \qquad 4.7$$

Where $T_{coolant}$ is assumed to be same with ambient temperature $T_{amb,}$ for simplicity. ε is the effectiveness of the intercooler. The pressure drop, filling and emptying effect of the intercooler volume is considered together with the intake manifold.

4.4.1 Intercooler pressure dynamic

The inter-cooler pressure dynamic is model by a control volume filled with ideal gas, with assumption of the constant temperature. Based on ideal gas question

$$p_b V_b = mRT_b \qquad 4.8$$

$$\dot{p}_b V_b = \dot{m}RT_b = \dot{m}RT_b + mR\dot{T}_b \qquad 4.9$$

The temperature was assumed constant; the second term of the model was ignored for simplicity. So the intercooler pressure dynamic model becomes

$$\frac{dp_b}{dt} = \frac{R}{V_b} T_b \dot{m} = \frac{R}{V_b} T_b(\dot{m}_{com} - \dot{m}_d) \qquad 4.10$$

$$\dot{m}_d = \frac{C_d A_d\ p_h}{\sqrt{RT_h}} \left(\frac{p_{im}}{p_p}\right)^{\frac{1}{\gamma}} \left(\frac{2\gamma}{\gamma - 1}\left[1 - \left(\frac{p_{im}}{p_b}\right)^{\frac{\gamma-1}{\gamma}}\right]\right)^{\frac{1}{2}} \qquad 4.11$$

Where $\dot{m}_d$ is the out flow to the intake manifold, p_{im} is the intake manifold pressure, p_b is the boost pressure after intercooler. A_d is the area of the duct, while C_d is the discharge coefficient through the duct and R is the gas constant of the intake manifold mixture

4.5 Intake manifolds

The intake and exhaust manifolds are modeled as open thermodynamic systems, where the mass of gas can increase or decrease with time (so-called filling and emptying model). The two governing equations for such systems are the Conservation of Mass and the Conservation of Energy. Considering the adiabatic conditions and assuming an ideal gas with constant specific heats, the rate mass flow change inside this control volume can be given by the continuity equation given by

$$\frac{dm_{im}}{dt} = \dot{m}_i - \dot{m}_o \qquad 4.12$$

Where m_{im} is the intake manifold mass content, $\dot{m}_i$ is the input mass flow to the intake manifold, and $\dot{m}_o$ is the out flow to the engine cylinders. For an ideal gas, the mass in this case can be expressed as follows.

$$m_{im} = \frac{p_{im} V_{im}}{R\, T_{im}} \qquad 4.13$$

Where p_{im} and T_{im} is the pressure and temperature of the intake manifold respectively, and R is the gas constant of the intake manifold mixture, while V_{im} is the intake manifold volume. Accordingly, the pressure rate of change inside the intake manifold is modeled as:

$$\frac{dp_{im}}{dt} = \frac{RT_{im}}{V_{im}}(\dot{m}_i - \dot{m}_o) \qquad 4.14$$

The total mass flow into the intake manifold depends on the pressure difference between the source and the intake manifold, and the accumulated masses in the manifold are:

$$\dot{m}_i = \dot{m}_{com} + \dot{m}_{at} + \dot{m}_{egr} \qquad 4.15$$

Where $\dot{m}_{com}$ is the mass flow rate from the compressor, $\dot{m}_{at}$ is the mass flow rate from the air tank, and $\dot{m}_{egr}$ is the re-circulated exhaust gas mass flow, which is equal to zero in this period of modeling, because during acceleration the re-circulated exhaust gas valve is completely closed.

For the purpose of this modeling, it is sufficiently accurate to adopt subsonic flow through restriction modeling to account for the flow for the intake manifold.
For subsonic flow the mass flow rate from the air tank to the intake displacement can be expressed as:

$$\dot{m}_{at} = \frac{C_v A_v\, p_t}{\sqrt{RT_r}}\left(\frac{p_{im}}{p_t}\right)^{\frac{1}{\gamma}}\left(\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p_{im}}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right)^{\frac{1}{2}} \qquad 4.16$$

And for sonic flow, the mass flow rate from the air tank to the intake displacement can be expressed as:

$$\dot{m}_{at} = \sqrt{\gamma \left[\frac{2}{\gamma + 1}\right]^{\gamma+1/_{\gamma-1}}} \qquad 4.17$$

Where C_v the coefficient of discharge of the compressed air is, A_v is the flow area of the injector valve, p_t is the pressure of the air in the tank, and γ is the air specific heat ratio.

4.5.1 Air-charging model

Air charging from the intake manifold to the cylinder is a highly nonlinear process depending on the volumetric efficiency η_{vol}, engine speed N_e and intake manifold states of T_{im} and p_{im} which describes the engine pumping process as

$$\dot{m}_o = \eta_{vol} \frac{p_{im}}{T_{im} R_{im}} \frac{N_e V_d}{120} \qquad 4.18$$

Where V_d is the displacement volume, the volumetric efficiency is mainly a function of engine speed, intake manifold pressure.

4.6 Exhaust Manifold

To model the conditions in the exhaust manifold, the temperature of the mass flow from the cylinder into the exhaust manifold is necessary. It is a function of fuel flow, air flow into the cylinders, and engine speed.

$$T_{ex} = f(m_f, m_{air}, N_e)$$

The exhaust pressure dynamics are modeled as in the intake manifold.

$$\dot{p}_{ex} = \frac{R_{ex}T_{ex}}{V_{ex}}\left(\dot{m}_{ex} - \dot{m}_{turb} - \dot{m}_{wg}\right) \qquad 4.19$$

Where p_{ex} is pressure, T_{ex} is the temperature, and R_{ex} is the gas constant of the intake manifold, while V_{ex} is the manifold volume and $\dot{m}_{ex}$ exhaust mass flow rate, which consist of fuel and air flow rate into the cylinder; $\dot{m}_{turb}$ and $\dot{m}_{wg}$ are turbine flow rates and waste gate flow rates respectively. The waste gate openings used to adjust the bypass flow as $\dot{m}_{wg}$, in order to control the exhaust manifold pressure. The engine air density is indirectly adjusted through the turbocharger power by the exhaust manifold pressure.

CHAPTER FIVE

COMPUTER SIMULATION

Content:

5.1 Simulation strategy

The Simulation is based on a typical automotive diesel engine [9]. The engine has a radial turbine and centrifugal compressor. With this engine specification:

Table 5.1 Engine specification for simulation

Engine Type	**Stroke Diesel**
Stroke	82.0 mm
Bore	78.1mm
Number of Cylinders	4
Combustion System	Direct Injection
Compression Ratio	22.0

To generate the results for this project it was important to understand what effects the air injection would have. The aim of the air injection system was to reduce the time that the turbocharger takes to reach steady state and also the overall improvement to performance needed to be considered to make sure that the increased turbocharger performance had the desired effects.

The transient period for the simulation was carried out for 4 seconds. During this time the engine speed was increased from 1000rpm to 4000rpm. This transient simulation was set up to represent a simple engine acceleration event.

A Quasi-Steady model was used to simulate the transient behavior of the air injection system. Primarily a range of values for different engine speeds ranging from 1000rpm to 4000rpm were selected.

These results were then used to judge the critical area. Once identified more results in this important region were generated to improve the resolution. This reduced the computing time by decreasing the resolution in areas of less importance in the transient period.

The simulations were run with and without air injection, the air injection included pressures of 2.5 bar and 3 bar. The air was injected for the entire time of the transient period.

To assess the effect the air injection had on the turbocharger pressure readings from the compressor outlet were taken. This showed how the turbocharger was responding to the air injection system. To review the effect on overall engine performance, the power output from the engine was simulated. Using power output meant that it was possible to see if the system would have an effect on the drivability of the car.

5.2 Results

Fig.5.1 shows the pressure response of the compressor outlet. The results show an initial delay in response before rapidly increasing to an almost steady state. The results show that when the air injection system is used the time taken for the system to reach a steady state is greatly reduced. This means that the air pressure going into the cylinders is higher at an earlier stage of the process which would ensure that the air/fuel ratio is ideal.

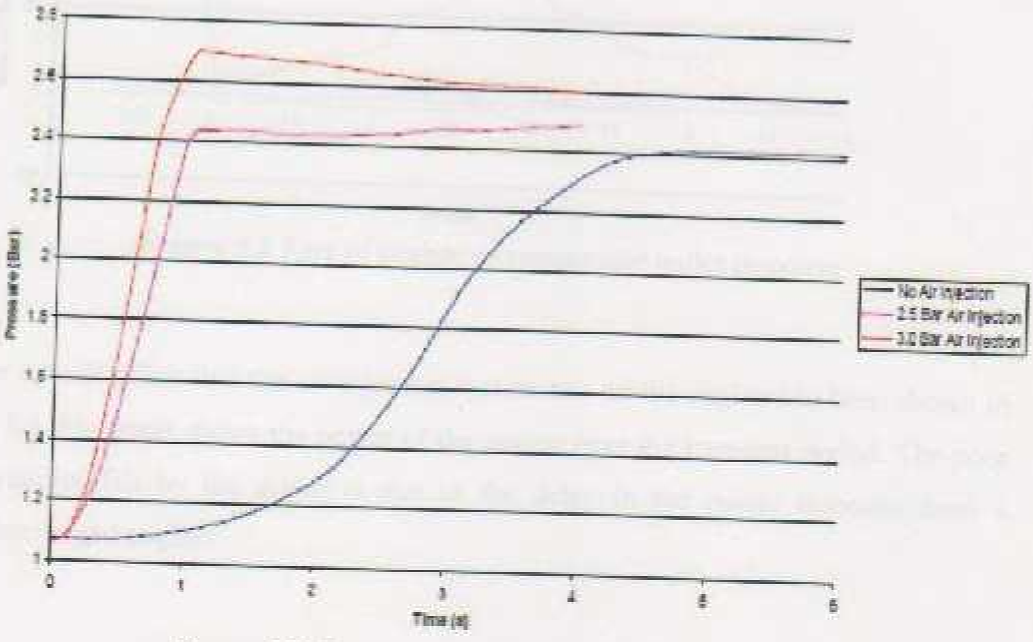

Figure 5.1 Compressor outlet pressure responses

Fig.5.2 shows the rate of change of the pressure response for the compressor outlet. The rate of change is shown to be much greater and to have been completed much faster for the air injection systems. The 3 bar air injection reduces the response time by three seconds. This means that the compressor reaches its steady state operating conditions significantly earlier than a turbocharger without the air injection system.

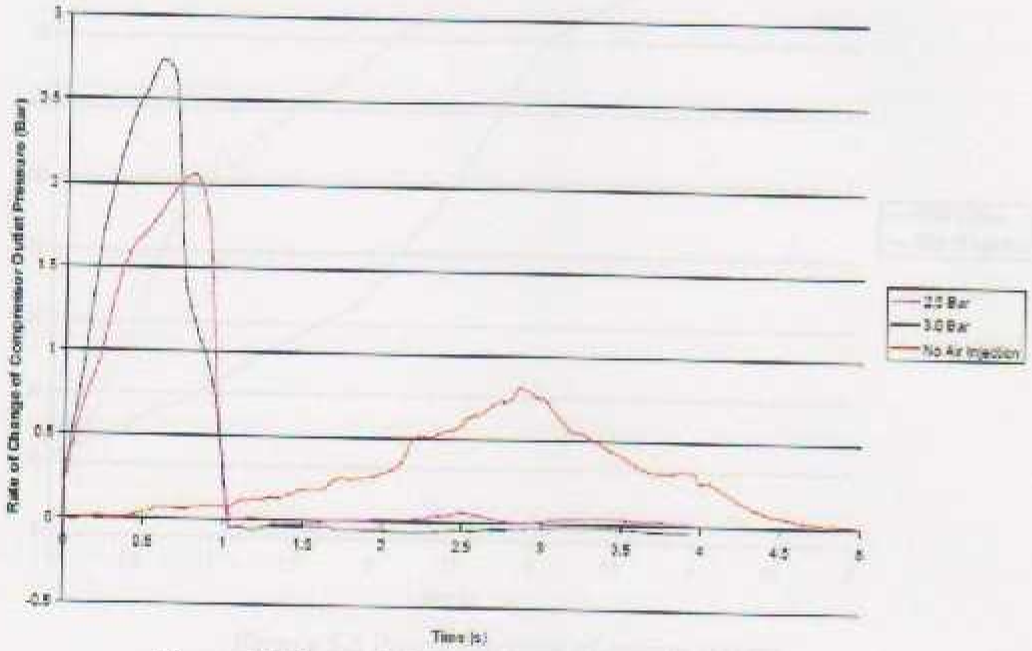

Figure 5.2 Rate of change of compressor outlet response

The overall affect that the air injection system has on the engine has been shown in fig 5.3 this graph shows the power of the engine over the transient period. The poor drivability felt by the driver is due to the delay in the power response from a turbocharged engine.

The Graph shows that the Air injected system shows a considerable reduction in the time that the engine reaches equilibrium. This subsequently leads to an improvement in the efficiency of the engine, with of improved drivability.

Figure 5.3 Power response of engine system

CHAPTER SIX

SYSTEM OPERATION AND RESULTS

Content:

6.1 Introduction

Turbocharged vehicles exhibit a weak point of poor drivability under transient running conditions. Reduced air flow in the combustion chamber while fuel is injected causes inadequate air fuel ratio (i.e. rich mixture). The result is an increased formation of emissions such as CO, NO_x which contribute to environmental pollution.

Air injection increases the air flow during the transient phase reducing the harmful emissions. Experimental work has shown that injecting air into the manifold greatly improves the transient response of the turbocharger and engine power.

6.2 System Components

Figure 6.1 System components

6.2.1 The air injection unit

The air injection unit consists of:

1. Air compressor.
2. High-pressure air tank with mechanical pressure sensor.
3. Air injection unit (Pressure regulator, solenoid valve).

6.2.1.1 Air compressor

The compressor that is used is the compressor for the A/C system that used in the vehicle, which allow charging the air inside the tank under high pressure.

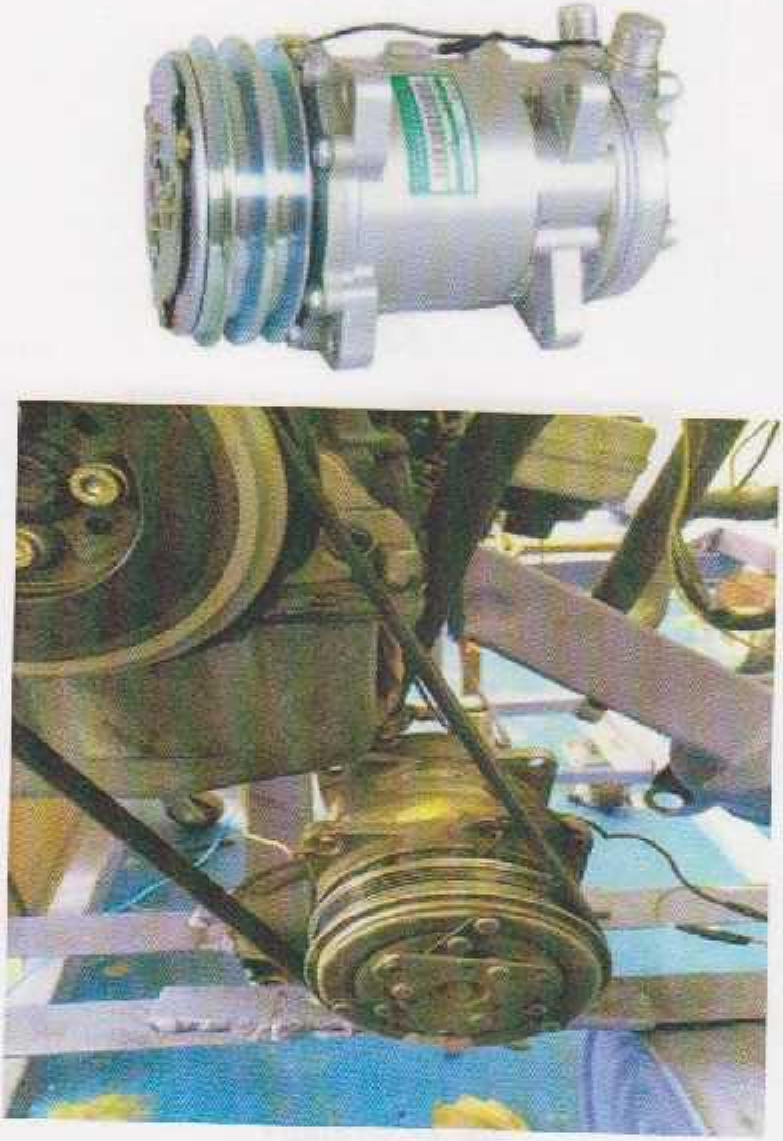

Figure 6.2 Air compressor

6.2.1.2 High-pressure air tank with mechanical pressure sensor

It can accumulate the air under pressure of 8 bar and its volume is 40 liter, including mechanical pressure sensor and pressure gage where the mechanical pressure sensor is used to regulate the pressure in the tank preventing it from increasing above 8 bar and decreasing it less than 4 bar by connecting and disconnecting the voltage to the compressor.

Figure 6.3 Air tank

6.2.1.3 Air injection unit (pressure regulator, solenoid valve)

a. Air pressure regulator

Its function to reduce the pressure coming from the high-pressure air storage tank (8 bar) to the desired pressure in the intake manifold.

Figure 6.4 Pressure regulator

b. Air injection solenoid valve

Its function is to pass the pressurized air from the tank to the intake manifold when it needed in the transient period.

Figure 6.5 Solenoid valve

Figure 6.6 Pressure regulator and Solenoid valve

6.2.2 The dynamometer (Water brake absorber)

Hydraulic dynamometers are machines that measure the power of an engine by using a cell filled with liquid to increase its load. The turbulent action of the water absorbs the power of the engine. The load is controlled by the water inlet. The power is converted into heat, which is carried away by the continually flowing of water.

Dynamometer with vertical instrument panel measures engine torque and power continuously from an absorption brake.

The schematic shows the most common type of water brake, known as the "variable level" type. Water is added until the engine is held at a steady RPM against the load, with the water then kept at that level and replaced by constant draining and refilling (which is needed to carry away the heat created by absorbing the horsepower). The housing attempts to rotate in response to the torque produced, but are restrained by the scale or torque metering cell that measures the torque.

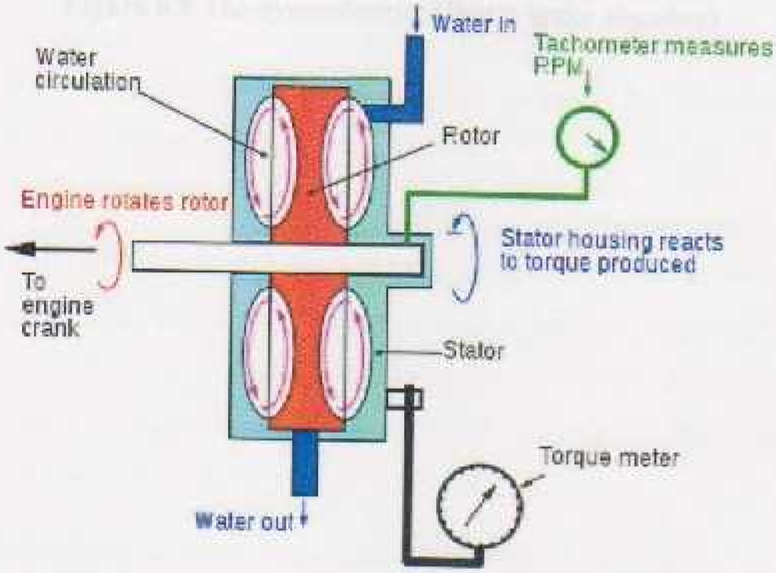

Figure 6.7 A water brake

Figure 6.8 The dynamometer (Water brake absorber)

6.3 Results

The project was applied on a diesel engine. The engine has a turbocharger. With this engine specification.

Table 6.1 Engine specification

Engine Type	**4 Stroke Diesel**
Stroke	**79.5**
Bore	**95.5mm**
Number of Cylinders	**4**
Combustion System	**Direct Injection**
Compression Ratio	**18**

Figure 6.9 shows the pressure response of the turbocharger compressor outlet. The results show an initial delay in response before rapidly increasing to an almost steady state. The results show that when the air injection system is used the time taken for the system to reach a steady state is greatly reduced.

This means that the air pressure going into the cylinders is higher at an earlier stage of the process, which would ensure that the air/fuel ratio is correct. The 2 bar air injection reduces the response time by two seconds. This means that the compressor reaches its steady state operating conditions significantly earlier than a turbocharger without the air injection system.

By comparing several readings were taken by using a scan tool system, with and without using system and put it in the curve.

Clearly shows the difference in response time to reach's the maximum value of pressure which take about 3 s without system and 1.5 s by using system.

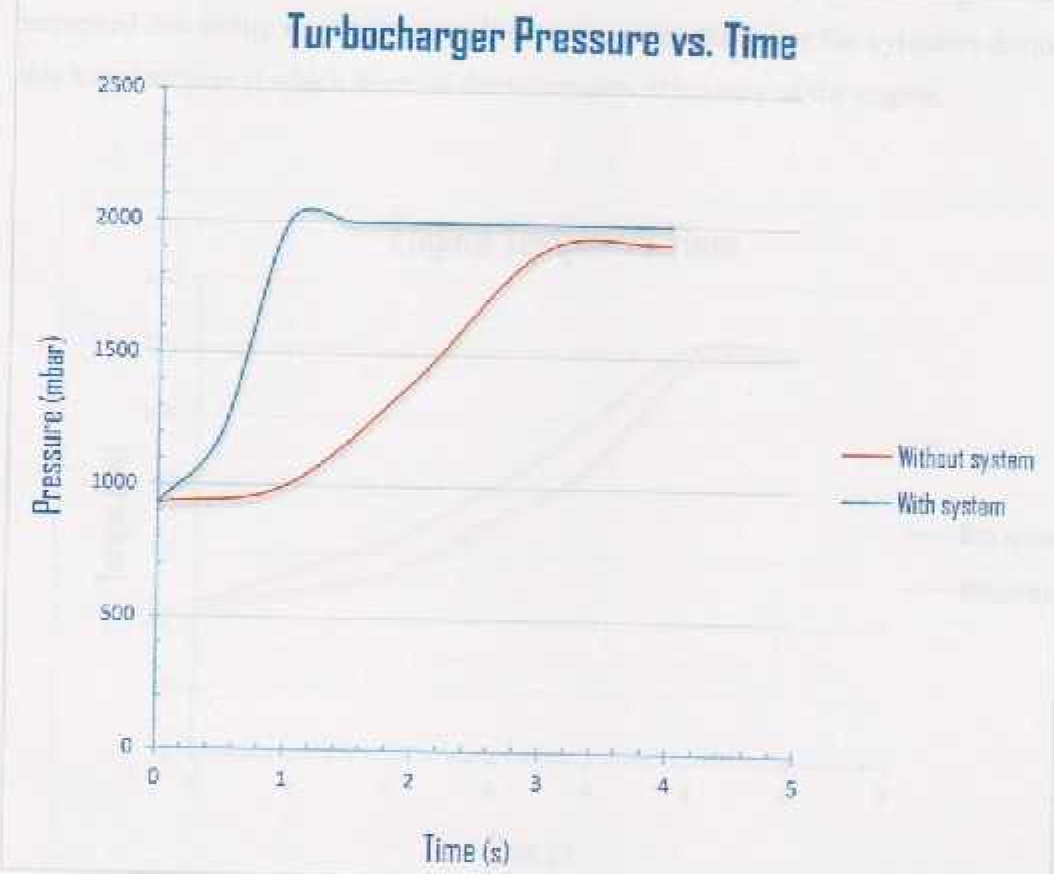

Figure 6.9 Compressor outlet pressure responses

The overall effect that the air injection system on the engine torque and engine power has been shown in fig. 6.10 and fig. 6.11 respectively; these graphs shows the torque and the power of the engine over the transient period.

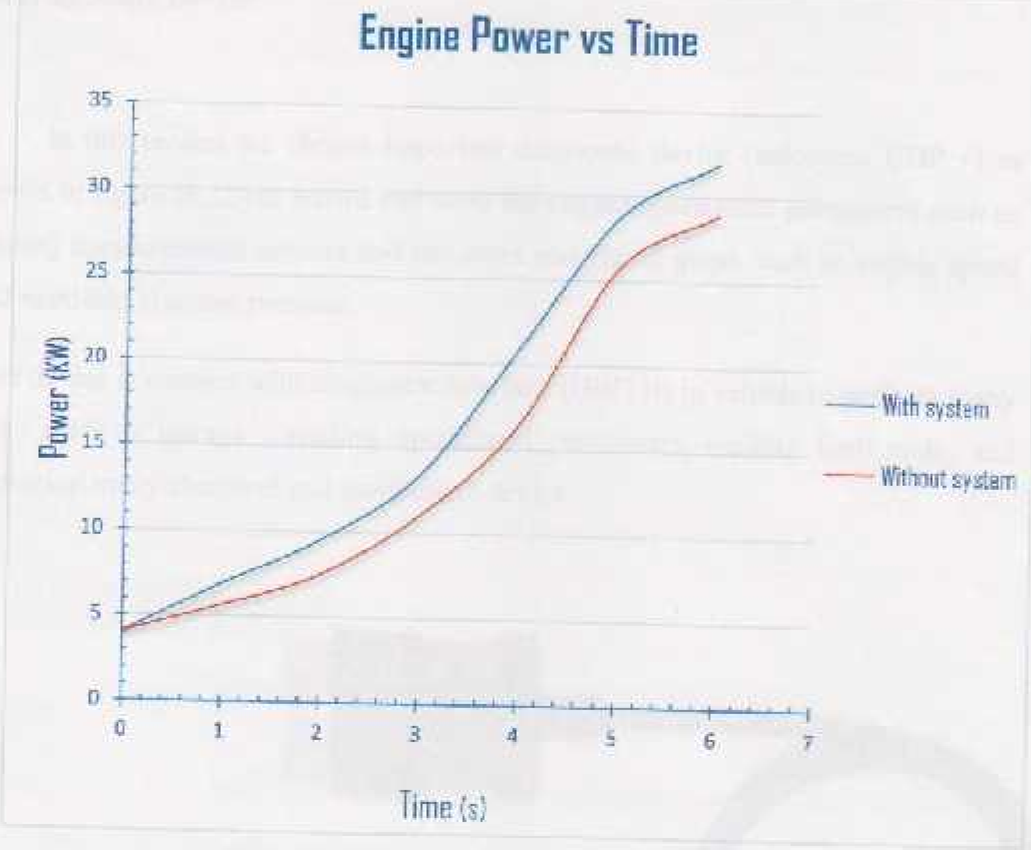

Figure 6.11 Power response of engine system

6.4 Diagnostic Device

In this project we choose important diagnostic device (autocome CDP +) as shown in figure (6.12) to record and store the engine operational parameters such as reading measurements sensors and actuators and signal graph such as engine speed and manifold absolute pressure.

This device is connect with diagnostic interface (OBD II) in vehicle to perform many tasks such as storage , reading operational parameters, reading fault code, and activation many electrical and mechanical device .

Figure 6.12 Diagnostic device autocome CDP+.

The following graphs was obtained from autocom diagnostic tool. Fig. 6.13 shows that at 3108 RPM the intake manifold pressure take 3 seconds to reaches 1920 mbar.

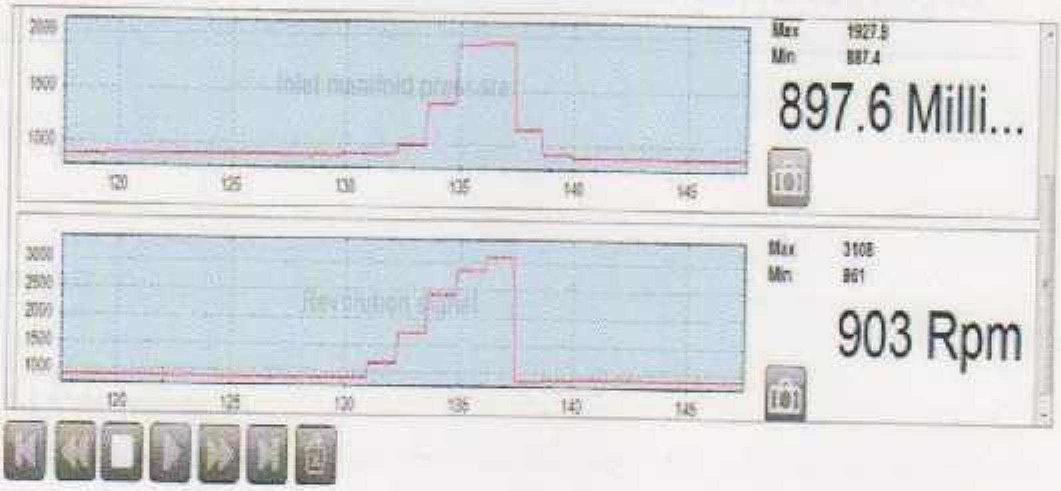

Figure 6.13 Autocom graph without using system

Fig. 6.14 shows that at 2370 RPM the intake manifold pressure take 1.5 seconds to reaches 1990 mbar.

Which means that the air injection reduces the response time by 1.5 seconds

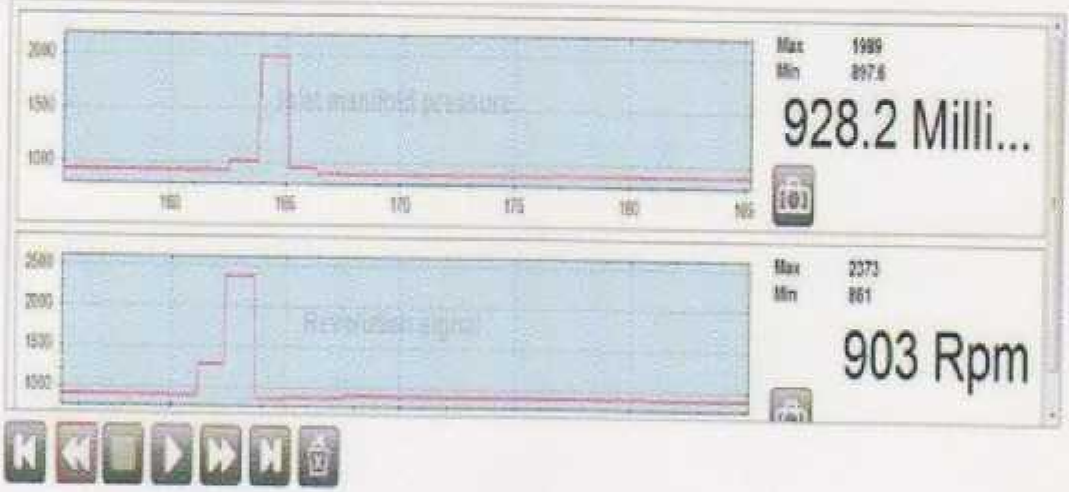

Figure 6.14 Autocom graph with using system

Chapter Seven

Conclusions And Recommendations

Content:

7.1 Conclusions

This project presented modeling technique and experimental results of a transient turbocharged engine system. The initial results show that this is a promising area to reduce emissions and improving performance of internal combustion engines during transient operation.

The obtained results show that:

1. Significant improvement to the transient response of the compressor outlet which should reduce the formation of CO_2 and No_x.

2. Larger Injection pressures have an increased effect on the turbochargers transient response.

3. Overall engine performance has been improved.

4. High injection pressures improve performance.

7.2 Recommendations

The obtained results are promotion and need further study for potential improvement of engine transient response and emission model.

Engine development is a multi-disciplinary area and has the interaction and cooperation at multiple levels and aspects. This research discusses the co-design and integration of detail modeling and control design. Some proposed methods are not finished and some ideas need further development for verification and application.

Following are potential topics:

1. Mean Value Model transient verification.
2. Control design for system.
3. Emission model integration.
4. Generic Mean Value Model with extension of emission model and after-treatment.
5. Fault diagnosis design and verification with the Integrated Engine Simulation Model.

REFERENCES

References:

[1] Zellbeck H, Friedrich J, Berger C. Electrically assisted turbocharging as a new boostingconcept. MTZ 1999;60: 386–391.

[2] Harndorf H, Kuhnt H-W. Improvement of transient behavior of turbocharged dieselengines through additional air injection in the turbocharger. MTZ 1995;56:20–28

[3] Ledger J. D. Benson RS, Furukawa H. Performance characteristics of a centrifugal compressor with air injection. ProcInstMechEng 1973;187:425–434.

[4] Ledger JD, Benson RS, Furukawa H. Improvement in transient performance of a turbocharged diesel engine by air injection into the compressor. SAE Paper No.730665, 1973.

[5] Winterbone DE, Benson RS, Mortimer AG, Kenyon P, Stotter A. Transient response of turbocharged diesel engines. SAE Paper No. 770122, 1977.

[6] Rakopoulos, C.D., and Giakoumis, E.G. (2009). Diesel Engine Transient Operation Principles of Operation and Simulation Analysis. National Technical University of Athens, School of Mechanical Engineering. Springer-Verlag London Limited, 390 pp.

[7] Filipi Z, Wang Y, Assanis D. Effect of variable geometry turbine (VGT) on dieselengine and vehicle system transient response. SAE Paper No. 2001-01-1247, 2001.

[8] J. B. Heywood. Internal Combustion Engines Fundamentals, McGraw-Hill, New York, 1988.

[9] Gilkes, Oliver S. and Mishra, Rakesh (2006) Comparison of passive and active methods for improving transient performance of turbocharged engine systems. In: Proceedings of Computing and Engineering Annual Researchers' Conference 2006: CEARC'06. University of Huddersfield, Huddersfield, pp. 1-6. Simulation REF 10