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Experimental and Numerical Investigations on the Transient Performance of a Turbocharged Diesel Engine

EXPERIMENTAL AND NUMERICAL INVESTIGATIONS ON THE TRANSIENT PERFORMANCE OF A TURBOCHARGED DIESEL ENGINE

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Abstract

In the highly competitive automotive market the share of diesel engine is continuously increasing. They are extensively used in passenger cars as well as in long distance haulage sector vehicles to impart motive force. Regulations for diesel engine emissions as well as public concern for fuel economy have forced the research community to address the combustion and emission issues associated with the use of the diesel engines in general and during the transient operation mode in particular. Turbocharging of diesel engine is the most extensively used technology for the improvement of power density, emissions and enabling downsizing of engines without compromising power. However, turbocharged diesel engine suffer from turbo-lag which is a common phenomenon especially during the rapid transient conditions. Turbo lag causes engine performance deterioration and increased emissions during transient events. The studies on reducing the turbo lag and maintaining the desirable air fuel ratio are considered to be very important for making turbocharged engine compliant with the current environmental regulations.

The thesis covers the important aspects of a turbocharged diesel engine with a specific focus on the transient response of the diesel engine. In the first facet of this research, computer-based investigations using commercial engine simulation packages are performed to simulate the transient response of the system using different methods. Torque assistance of 0.16 Nm reduces the turbo lag by 3.6 sec for both compressor exit pressure and compressor speed. Optimum value of inertia reduction is found to be -10% which reduces the turbo lag by 2.9 sec for compressor exit pressure and by 0.6 sec for compressor speed. The effect of 2.5 bar air injection is found to reduce the turbo lag by 3.9 sec for both compressor exit pressure and compressor speed. A comparison is made for the assessment of relative improvement in transient response brought by the three methods and based on this comparison, air injection system is chosen that yields maximum benefit in the performance of the engine. The experimental results from the real CI engine is used to validate the simulation model. A good agreement is achieved between the simulation and experimental results.

The effect of air injection on a heavy-duty turbocharged diesel engine under various operating conditions forms the second facet of this study. The turbocharger response parameters are analyzed under the effect of air injection for different transient operating conditions of speed and load transients. For

speed transients, considering the energy imparted for air injection, 1 bar is the optimum injection pressure with turbo lag reduction per unit energy as 0.290, 0.392 and 0.555 per joule for compressor exit pressure, turbine inlet pressure and turbine inlet temperature (TIT) respectively. Faster recovery time is noted for 1 sec rapid acceleration than that of 2 sec with the application of air injection. For load transients, optimization of air injection is also performed for injection pressure and orifice diameter. Under constant load step, maximum improvement in turbo lag reduction is observed at 1000 – 1400 rpm whereas its effect on maximum attainable value is at 1600 – 1800 rpm. For load magnitude variations, the optimum injection pressure of 3 bar at 15 mm orifice diameter brings more improvement for lighter load than for stronger load. Maximum improvement is noted for 50-70% load step. 3 bar air injection at 15mm orifice diameter is the optimum injection pressure which brings more improvement in terms of turbo lag reduction for faster load application. The effect of this optimized air injection is more beneficial for 1 sec load schedule than for 2 sec.

During the transient operation of diesel engines, exhaust emissions are the imperative issue that needs to be addressed. In the third facet of the research exhaust emissions are analyzed to make sure that the application of air injection technique does not need compromise on emissions. The air injection technique is evaluated for exhaust emissions through simulation. 3 bar air injection at 10 mm orifice diameter is the optimum air injection for emissions under speed transient which satisfies Euro 6 standard for CO and HC emissions. This air injection reduces the concentration of CO and HC emissions by 5% and 0.4% respectively. Under load transient the optimum value of air injection is 1.2 bar at 10 mm orifice diameter which reduces the concentration of CO, HC and NO_x emissions by 0.8%, 0.01% and 0.4% respectively.

The study reveals that air injection technique while improving the transient response of the turbocharged diesel engine doesn't increase the emissions, highlighting the magnitude of contribution of the technique to the overall performance of the system. The novelty in the research is the compilation of these methods in a cohesive approach of modeling the transient response of turbocharged engine system.

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Dedicated to my late parents - Mr Sayeed Ul Hasan and Mrs Noorus Sabah

Declaration

I declare that I have presented my own work in this thesis and relevant citations have been included to acknowledge the work of others. I have two papers published in the journals and I have referred them wherever the material from these papers are reproduced in the thesis.

Publications and contributions

1. Saad, S. M. and Mishra, R. (2019). Performance of a Heavy-Duty Turbocharged Diesel Engine Under the Effect of Air Injection at Intake Manifold During Transient Operations. Arabian Journal for Science and Engineering.

   I developed the engine model, performed data analysis, interpreted the results and presented in manuscript form.

2. Mishra, R., and Saad, S. M. (2017). Simulation based study on improving the transient response quality of turbocharged diesel engines. Journal of Quality in Maintenance Engineering. ISSN 1355-2511.

   I developed the engine model, validated the steady state results against the experimental data which I collected from the test bed at University of Huddersfiled, developed the test strategies and wrote the manuscript.

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1

INTRODUCTION

Invention of internal combustion engine is considered as one of the most significant inventions of the last century which had a significant impact on the society, particularly on human mobility. It became the foundation of the successful development of several commercial technologies. It has transformed the entire transportation industry with its relatively low cost, high efficiency and favorable power to weight ratio [1].

Diesel engines are also known as compression ignition (CI) engines. The air that is sucked from the intake manifold into the cylinder is compressed to above the auto ignition temperature of the fuel. At the end of the compression stroke the fuel is sprayed into the combustion chamber under high pressure and temperature conditions and combustion takes place. The energy hence released from the burning of fuel forces the piston which moves from top dead center to bottom dead center and this motion is converted to forward motion of the vehicle through integrated and well design power train. Compression ignition engines plays a vital role in transportation industry where a very high amount of torque is required such as trucks and buses etc. [2]. There are four processes involved in diesel cycle as shown in Figure 1.1:

1-2: Isentropic compression

2-3: Constant pressure heat addition

3-4: Isentropic expansion

4-1: Constant pressure heat rejection

Intake stroke: The intake valve opens as the piston moves from top dead center to bottom dead center and the air is introduced inside the cylinder.

Compression stroke: The movement of the piston back to top dead center from the bottom dead center compresses the air inside the cylinder and the temperature inside the cylinder is raised over the auto-ignition temperature of the fuel.

Power stroke: The fuel is injected into the combustion chamber by the injectors at the end of the compression stroke. The fuel auto ignites after the ignition delay period. The mixture of fuel and air combust to produce pressure on the cylinder head which pushes the cylinder downward.

Exhaust stroke: After completion of the power stroke, the piston moves from bottom dead center to top dead center and the exhaust valve opens. Due to the sweep effect of the piston, the exhaust gas leaves the cylinder.

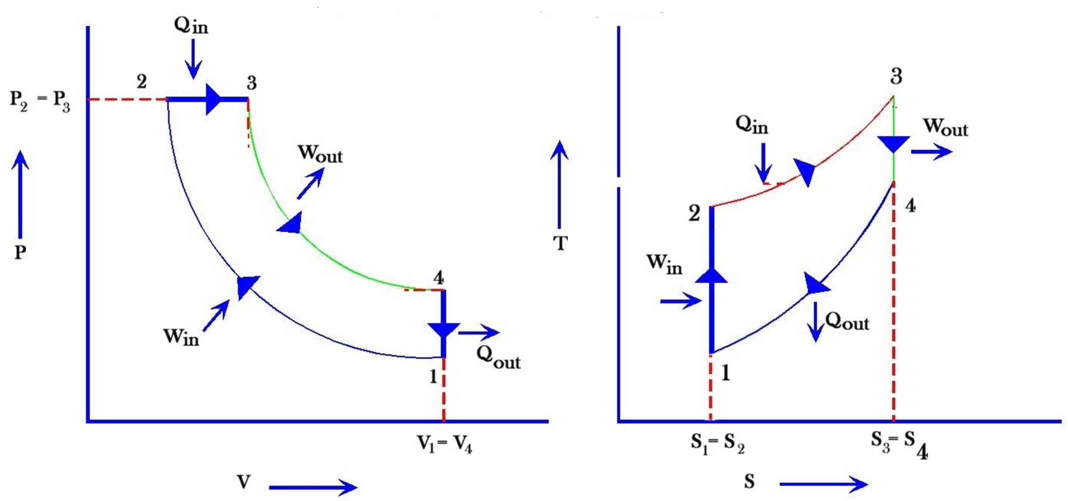

Figure 1.1: P-V and T-S diagram of four stroke CI engine

The speed and power of the diesel engines are regulated by the amount of fuel injected [3]. Due to their operational simplicity, diesel engines are more preferable in the transport sector, agriculture and forestry machinery [4].

The excellent fuel efficiency is the most attractive feature of the compression ignition engine which can surpass 40% in vehicle application and 50% in large two stroke units for marine propulsion or electricity generation. Therefore, for the whole lifetime of the vehicle and over the entire operating range, lower specific fuel consumption and reduced CO_2 emissions are achieved for the vehicles equipped with the diesel engine in comparison to its similarly rated spark ignition counterparts [5]. The issue of potential global warming demands reduction of green house gases. Diesel engine with its significant efficiency over gasoline engine can meet the low carbon emission requirement [6].

The difference between the engine built 100 years ago and the modern-day engine can be seen by comparing their thermal efficiency, reliability and emission level. Internal combustion engines are the dominant prime mover in several areas. Efficiency optimization and emission reduction are the main focus of development in internal combustion engines [7]. Increased use of technology such as engine downsizing, turbocharging, supercharging and variable valve timing is being extensively investigated to meet mileage requirement [1].

1.1 Turbocharged diesel engine

Continuous improvement in internal combustion engine technology is necessary to meet stringent performance and emission requirements. The developments in CI engine technologies have led to the increase in the demand of fossil fuels causing the rapid depletion of fossil fuel reserves [8]. Recent estimates for fossil fuel predict that the fossil fuels may be depleted in the coming 45 years [9]. Fuel saving in the transportation sector is a critical issue across the globe [10]. In addition, the transport sector is undergoing rapid transformation due to stringent emission regulations in most of the developed countries with a target of improving fuel efficiency of the vehicles [11, 12]. In order to reduce engine emissions, improvement in engine efficiency and combustion process technology is important to meet the increasing land transport requirements [13].

Trend of down-sizing is increasing in internal combustion engine development since last two decades [14]. Engine down-sizing is a key measure to reduce fuel consumption and lower exhaust emissions [15]. Engine manufacturers are forced to produce significantly down-sized engines due to the increase in fuel prices and the stringent emission regulations. However, the engine down-sizing results in lower power output. The output power can be improved by boosting or introducing extra air into the engine, resulting in burning more fuel to maintain specific air fuel ratio. Several methods of boosting are available such as superchargers, turbochargers or the combination of both can be used. Operating supercharged is a very favourable feature of diesel engine among its many merits. By supercharging, inlet pressure and density increases by means of a compressor located upstream of the engine. This enhances the air supply entering into the cylinder enabling efficient burning of higher amount of fuel and in turn increasing the engine power. A special case of supercharging is turbocharging which is the most preferred supercharging versions for many decades [16]. It is a promising way for energy saving and CO_2 reduction [17, 18, 19]. Turbochargers have the capability to enhance the engine power and torque without increasing the swept volume of each cylinder [20]. In a study [21], a comparison was made between turbocharged engine and naturally aspirated engine in terms of engine performance and emissions. While turbocharging decreases brake specific emissions, brake torque and brake power were boosted as compared to naturally aspirated engine as shown in Figure 1.2 and Figure 1.3 respectively.

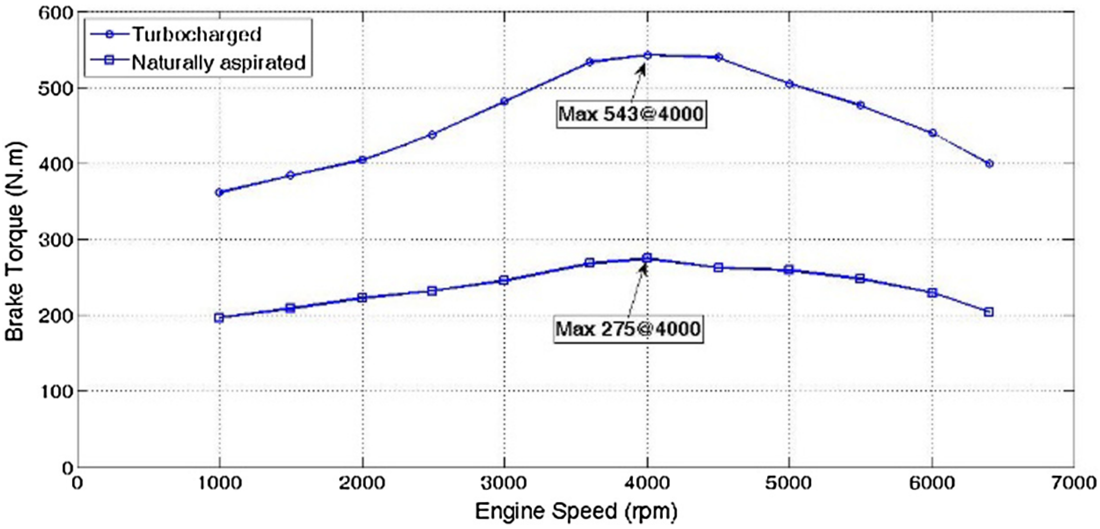

Figure 1.2: Comparison of brake torque in turbocharged and naturally aspirated engine [21]

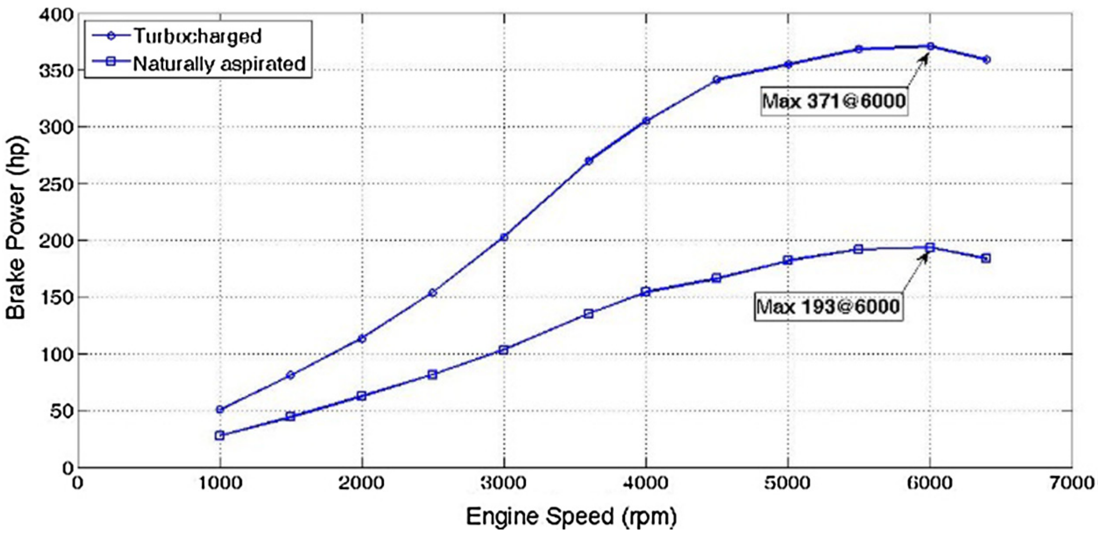

Figure 1.3: Comparison of brake power in turbocharged and naturally aspirated engine [21]

The reliability, widespread availability and excellent thermal efficiency of turbocharged diesel engine make them the prime mover for small and medium sized vehicles [22, 23].

A typical turbocharged diesel engine is shown in Figure 1.4.

Figure 1.4: Turbocharged diesel engine [24]

1.1.1 Geometry and performance of turbocharged diesel engines

The layout of turbocharging system is shown in Figure 1.5. Turbocharger is a device consisting of a turbine and a compressor attached to a single rotating shaft. In turbocharged diesel engines the heat that is generally lost in exhaust gases (around 30-40% in a naturally aspirated diesel engine [25, 26]) is recovered for compressing the fresh air into the engine cylinders. The exhaust gases are expanded in the turbine blades and part of the wasted energy is converted into useful work by the compressor. The compressor outlet is connected to the inlet manifold through an inter-cooler to increase the air density flowing into the intake manifold. In this way the volumetric efficiency of the diesel engine is enhanced by the use of turbocharger.

Turbocharger boosts the power output of the engine with the same displacement volume of the engine by providing intake air with increased density and hence increased mass. So, it is a boosting system enabling the engine to breathe more air resulting in more power output for a given size of the engine. A distinctive advantage of a turbocharged engine is that it operates more efficiently than its naturally aspirated counterpart, hence CO_2 production is proportionately less. This contributes to the increase in the production of turbocharged vehicles as concerns over greenhouse gas

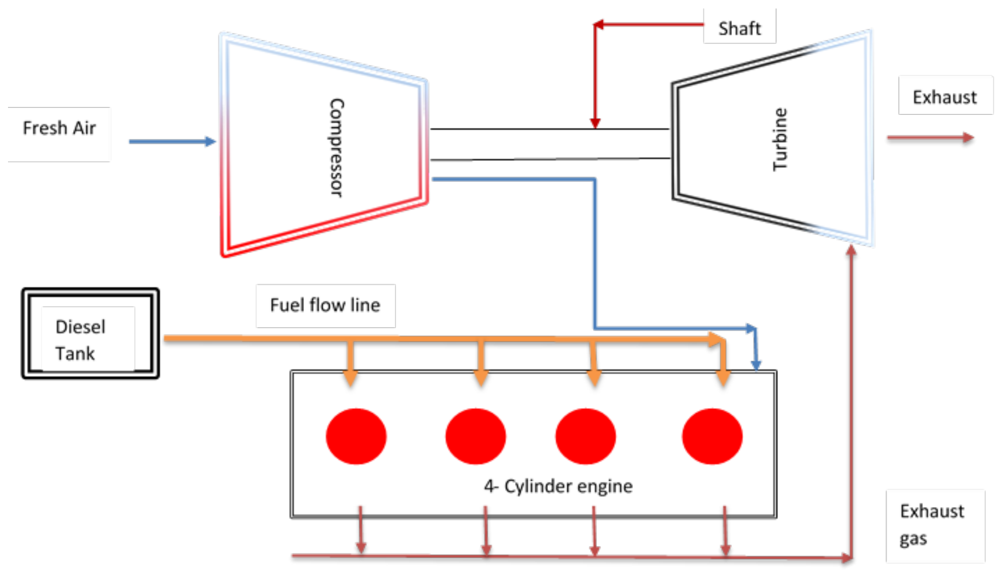

Figure 1.5: Layout of turbocharged diesel engine

emissions are now reflected in the automotive legislation also [16]. Turbocharging is considered as the most common boosting method used in the automotive industry.

1.2 Transient response of turbocharged diesel engine

Despite its several benefits, a serious operating disadvantage of turbocharger is its problematic transient behaviour which is due to the fact that at low loads and speeds the available exhaust energy is not sufficient enough to produce substantial boost [16].

There are essential operating discrepancies between the steady state and transient conditions of diesel engine. Crankshaft rotational speed, fueling and all other engine as well as turbocharger properties remain practically constant during steady state conditions. On the other hand, both engine speed and amount of injected fuel change continuously during the transient operation. The result is the variation of exhaust gas energy that affects the enthalpy of the turbine. Air supply and boost pressure are henceforth influenced through the turbocharger shaft torque balance [5]. Due to the often non-optimum performance of the diesel engine, the transient response is of crucial importance. Figure 1.6 shows the difference between steady state and transient operation of diesel engine in terms of engine's torque response. The torque response of naturally aspirated engine is linear. Contrary to this, the transient torque build-up of turbocharged diesel engine is highly non linear. The engine is unable to attain its quasi steady state performance leading to torque deficit [16].

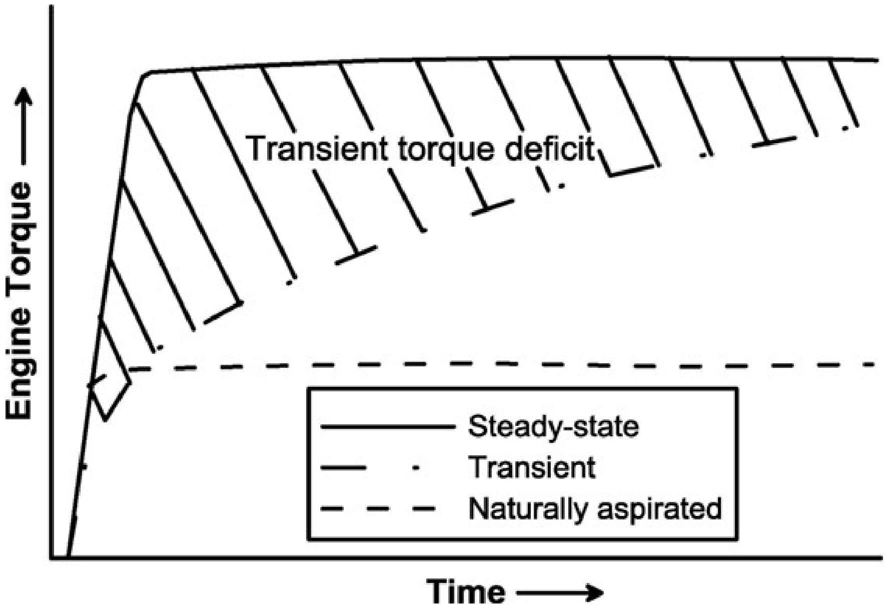

Figure 1.6: Comparison between steady and transient operation of diesel engine [16]

At high load transient conditions, faster response of the turbocharger is needed for the improvement in the drivability of diesel engines. However, turbochargers have their own inefficiencies and there are associated problems such as the turbo lag. The turbo lag refers to the time required for the exhaust system and turbocharger to generate the required boost. When a sudden load change is applied at lower engine speeds with rapid acceleration, the turbocharger must also accelerate to its new steady state. But the turbocharger doesn't respond effectively as compared to its naturally aspirated counterparts. Turbo-lag is the biggest disadvantage of turbocharged diesel engines due to which sufficient boost pressure can not be achieved at low engine speeds [27]. The reasons for turbo-lag are the inertia of the turbocharger rotor and the compressibility of exhaust gas within the engine as will be discussed in the next paragraphs.

A typical transient event of acceleration from low load and speed is described by Giakoumis [16] to highlight the concept of turbo-lag. With the fueling increase command, the fuel pump rack position changes instantly leading to higher fueling. At this moment higher amount of air is required to accommodate this elevated fueling. However, breathing is strongly influenced by the operation of the turbocharger in turbocharged diesel engine. Hence, the amount of air inducted into the cylinder is not only determined by the engine speed as in naturally aspirated counterpart but also by the compressor operating point. To illustrate the problematic behaviour of turbo systems, following equation can be referred which describes Newton's second law of motion for rotational system:

$$\eta_{\mathrm{mtc}} T_{\mathrm{qt}} - |T_{\mathrm{qc}}| = J_{\mathrm{tc}} \frac{dw_{\mathrm{tc}}}{dt} \tag{1.1}$$

In the above equation, T_{qt} and T_{qc} are the turbine torque and compressor torque respectively, J_{tc} is the turbocharger inertia, η_{mtc} is the shaft mechanical efficiency and w_{tc} is the turbocharger speed. From equation 1.1 it is clear that turbocharger's acceleration depends on turbine torque and inertia of the turbocharger. The turbine torque T_{qt} does not raise instantly due to two reasons: increased heat transfer lost to the cylinder and exhaust manifold walls and the subsequent filling of exhaust manifold with higher enthalpy gas. In addition, higher turbocharger inertia J_{tc} is the most influential factor that prohibits fast turbocharger shaft acceleration. Hence, during the early cycles the compressor operating point moves slowly towards increasing the boost pressure and air mass flow rate. The delay between fueling and air-supply is known as turbo lag and during this slow response period the engine runs with very limited boost. Consequently, the air fuel ratio becomes very low and the combustion deteriorates in diesel engines. This leads to slow engine response and increased exhaust emissions. One of the major challenges in turbocharged engines is achieving good transient performance and this is still being actively pursued. Combustion and emission performance of turbocharged diesel engines under transient operating conditions has been the focus of research in order to meet the increasingly stringent emission regulations [28, 29, 30].

1.3 Diesel engine exhaust emissions during transient operation

In addition to air-supply deficiency, the problematic transient response pattern gets amplified with in-cylinder combustion issues. As compared to steady state, boundary condition response delay under transient conditions is the characteristics of turbocharged diesel engine [31, 32]. The response delay causes poor in-cylinder physical condition leading to fuel economy deterioration and pollutant emissions [33, 34]. At the beginning of the transient event of speed or load increase, the higher pressure jets are injected into an environment that remains almost the same to the previous steady state conditions. The temperature adapts slowly to new loading conditions. Hence, liquid fuel impingement on the cool combustion walls increases that lowers the rate of mixture preparation. Subsequently, combustion gets deteriorated leading to increased exhaust emissions [16]. As compared to steady state operation, the combustion products are produced in an unpredictable manner under transient operating conditions [35].

Major emissions from internal combustion engine include nitrogen oxides (NO_x), carbon monoxide (CO), hydrocarbons (HC), particulates (PM) and aldehydes. These products are the major source of air pollution. There may be many unregulated emissions as well. With the recognition of environmental issues such as impact of air quality on health, a great emphasis is made on the reduction of various emissions from the engine. Meeting the emission requirement is currently one of the major factors in designing and operation of the internal combustion engines [1].

1.4 Research problem and aims

Due to the fact that turbo-lag is the major reason for poor transient response of the diesel engine resulting in increased exhaust emissions [8], several technologies are developed over the years to cope with this problem. These technologies can be categorised as active and passive methods based on how the inappropriate air-fuel ratio is addressed. Quantification of turbo-lag for these technologies and its comparison for relative improvement in transient response to find the optimum technology is an important task which has hardly been investigated. Based on this, optimization of transient combustion and emission performance can be done which is essential to address the increasingly severe ecological issues in addition to the strict emission protocols [36]. With this motivation detailed investigations are performed in the present study to find the optimum technology and examine its effect on the overall performance of turbocharged diesel engine. The important aspects that needs to be addressed in the context of transient response of turbocharged diesel engine are described in the following.

In the event of rapid speed and or load changes, the operational capability of the diesel engine can be enhanced by improving the transient response of the diesel engine [37]. Since majority of daily driving schedules are transient in nature, hence the experimental and modeling analyses of transient diesel engine operation is an important aspect to be focused by the engine manufacturers. Numerical simulation are being used for several years to simulate the actual real working conditions in order to get more effective results. In actual physical tests, for example, intake manifold design and optimization, the trial and error method could be effective but very expensive and time consuming. Simulation technologies play a vital role in reducing time and cost as well as getting optimized design with little iterations [38]. Diesel engine modeling using advanced engine simulation software to simulate the effect of different technologies on the transient response of diesel engine can be a useful and reliable tool to compare them for optimum results which otherwise would be very difficult and costly using the real setups. This forms the first aspect of this research.

Rapid increase in fueling rate for turbocharged diesel engine represents a transient event [39] as due to this elevated fueling and insufficient air flow in the combustion chamber, air-fuel ratio becomes very low. Mostly transient events reported in literature are engine speed or load changes such as acceleration, load increase or even starting. Fueling rate change representing a transient event is never investigated before and so it can be a novel operating condition for which technologies can be tested for transient response improvement. Nevertheless, speed and load transients can't be neglected as these are the most popular transient events. Acceleration tests are the best representative of the transient operation of turbocharged diesel engine and turbo lag is more prominent during the acceleration from low engine speeds. Similarly, load increase transients are more difficult for the engine to cope with, as unlike acceleration, it initially causes drop in engine speed which poses another burden for the engine. Engine might even stall in circumstances where very high loading is applied instantly [16]. For these reasons it is worth evaluating the effect of optimum technology under these operating conditions and this forms the second research area.

In order to produce engines that meet stringent regulations concerning exhaust emissions, the study of transient emission is very important to the manufacturers [40]. For the overall performance of the engine, the effect of optimum technology should be evaluated for exhaust emissions to ensure that the chosen technology does not require compromise on emissions. This forms the third aspect of the current research.

A detailed literature review is performed in the next chapter on different methods of reducing the turbo-lag and improving the transient response. Specific research objectives will be discussed after identifying the knowledge gap through literature review. Based on the research problem, aims of the study are formulated below.

1. Development of engine model for the techniques of improving the transient response of turbocharged diesel engine and investigation on selecting the optimum technology based on turbo lag quantification.

2. To investigate the effect of selected technology on the transient response of diesel engine under various operating conditions.

3. To investigate the effect of selected technology on the environmental performance of turbocharged diesel engine.

1.5 Thesis structure

The entire thesis is organized into 7 chapters. Chapter 1 presents the overview of the turbocharged diesel engine and the basic information related to its transient response and subsequent exhaust emissions. Based on this overview, motivation for carrying out this research is described and aims of the study are formulated. Chapter 2 covers the literature review followed by specific research objectives. Chapter 3 examines the numerical/experimental methods employed to achieve the objectives. This section is broadly divided into two sub sections. Each sub section contains the details of the methodology employed. Simulation based investigations are carried out in Chapter 4 for the effect of passive methods of improving the transient response. Transient performance of the engine has been simulated through the rapid change in fueling rate. Parametric study has been undertaken to analyze the quality indicators. Turbo lag is quantified and taken as a measure of the transient response of the turbocharged engine. Chapter 5 provides experimental and simulation based studies on active method of improving the transient response and a comparison is made between active and passive methods. The potential improvement in the response of turbocharger is assessed through turbocharger response parameters and the technologies are evaluated for their relative contribution for transient response improvement. Based on maximum improvement in transient response, optimum technique is chosen. Detailed investigations are carried out to observe the effect of the selected technique on the transient response of the turbocharger under acceleration and load tests. In addition, the effect of selected technology on the environmental performance of the engine is reported in Chapter 6. Chapter 7 discusses the findings of the study and the contribution of the work is highlighted. Recommendation for the future work is also included.

2

BACKGROUND AND LITERATURE REVIEW

This chapter provides the background information on the transient response of turbocharged diesel engine, looking into the various methods of controlling turbo lag and hence improving the transient response of the turbocharger. Major exhaust emissions from the engine during transient conditions are also reviewed. The relevant literature is reported on the efforts made by the researchers to improve the engine response under different transient conditions affecting the overall performance of the engine. Finally the objectives of the study are presented based on the literature review.

2.1 Transient operating conditions

Turbocharged diesel engine still dominates the market of medium and large vehicles as well as ships with the advantages of high power and low fuel consumption. The speed of turbocharger can be determined by engine load and speed under steady state operation. However, in rapid changing conditions such as acceleration or load change, the turbocharger will show delay in response due to the inertia of turbocharger rotor and the volume of intake and exhaust pipes connected [41]. Great emphasis is made on transient operating conditions due to increasingly stringent fuel consumption and emission regulations [42].

Proper inter connection between engine, governor, fuel pump, turbocharger and load is important to address the non-optimum performance of turbocharged CI engine during transient operations. Diesel engine simulations have contributed significantly towards this aim [43, 44]. In terms of engine response and in the reliability of governers and fuel pumps, rapid load changes are very demanding.

Transient load of a diesel engine can be described as either speed variation under constant torque or variation in torque under a constant speed or simultaneous changes in both. The effect on system response, emissions and fuel efficiency are of main interest to researchers and developers [45]. The transient operating conditions include [22]:

1. Changing torque and changing speed - decreasing speed and decreasing torque (DSDT), decreasing speed and increasing torque (DSIT), increasing speed and decreasing torque (ISDT), increasing speed and increasing torque (ISIT)

2. Constant speed and changing torque - constant speed and increasing torque (CSIT), constant speed and decreasing torque (CSDT)

3. Constant torque and changing speed - constant torque and increasing speed (CTIS), constant torque and decreasing speed (CTDS)

4. Starting

5. Parking

As per the studies conducted by Qiang Liu et al.[22], transient operation under constant speed and changing torque is most popular due to its higher weight (61.13%) as compared to other conditions, weights assigned based on the transient characteristics and some assumptions by He [46]. Moreover, smoke emission degradation is observed at CSIT rather than CSDT, so CSIT is the main focus for the investigations under load transient in the present study.

Rakopoulos and Giakoumis [47] presented detailed analysis of the operation of an indirect naturally aspirated diesel engine under various load conditions. These include - load intensity, load schedules and load types. The results show that the speed droop and recovery time were seriously affected by the load intensity as well as by the load types. However, they were not significantly effected by the load schedule. The term speed droop meant maximum deviation of speed from its initial value divided by the initial speed. Recovery period was the time needed for the engine to reach 98% of its final speed. These results are shown in Figure 2.1 and Figure 2.2. Figure 2.1 shows that greater load leads to lower final speed. However the recovery time was not much affected for final load greater than 70%. The effect of load schedule is shown in Figure 2.2. Although the final speed and recovery period were not significantly affected yet faster load application shows lower final speed and recovery period.

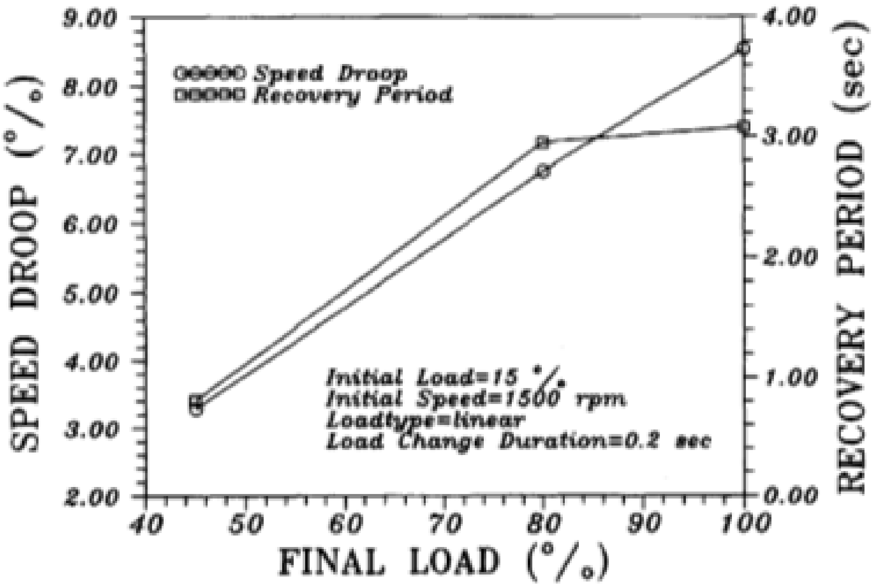

Figure 2.1: Effect of load change on final speed and recovery period [47]

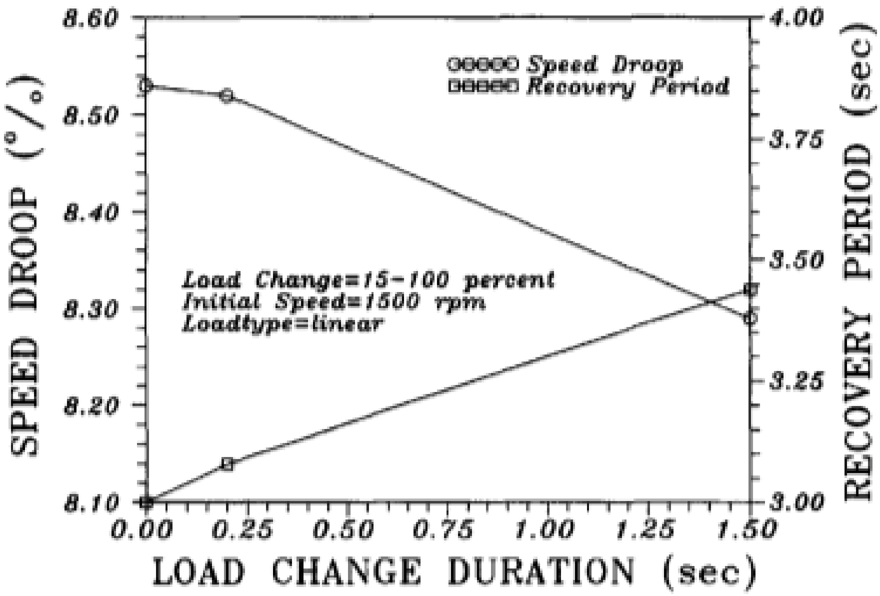

Figure 2.2: Effect of load schedule on final speed and recovery period [47]

2.2 Existing methods for controlling turbo lag

In turbocharged diesel engines the problem of turbo lag arises during the transient operating conditions. This problem arises from the lack of air induction due to the slower response of the turbocharger during a transient. To meet the requirement of complete combustion, sufficient air should be supplied to the engine. Hence a good air injection system is important to achieve higher diesel engine performance [48]. For sudden load application in turbocharged diesel engine, the transient performance of the engine can be improved by controlling the fuel flow and by accelerating the turbocharger by some external means [49]. Control of fuel flow can be achieved through the use of boost pressure sensitive fuel limiters that limits the fuel flow until the boost pressure achieves a value so that very low air fuel ratio is prohibited. Soot emissions are reduced by this method, however, the drawback is slower engine response and poorer vehicle drivability. So, controlling only the fuel flow can not be the sole measure during transients rather turbocharger specific measure should be taken to address the transient delay due to slower turbocharger response. Six such measures are reported by Giakoumis [16], which are listed below:

1. Lower-inertia turbocharger
2. Variable-geometry turbine (VGT)
3. Combined supercharging
4. Two-stage turbocharging
5. Sequential turbocharging
6. Electrically turbocharger assist (ETA)

The above listed methods are the indirect method of improving the transient response achieved through faster turbocharger response to provide sufficient boost in order to meet the elevated fueling during transients. Such methods are referred as passive methods. Another method is active method of meeting the air-supply demand by means of air-injection as reported in [49, 50, 51].

Figure 2.3 shows *Combined supercharging* in which two compressors are arranged in series realized through the use of a positive displacement compressor which is located upstream of the turbocharger compressor. At low engine load and speeds when the enthalpy capacity of exhaust gases leaving the cylinder is low and the turbocharger is incapable of producing the required boost, the positive displacement compressor becomes active at this moment and provides sufficient boost thereby improving the transient response. Nevertheless, the drawback of this configuration is reduced engine efficiency owing to the extra power needed to drive the compressor [16].

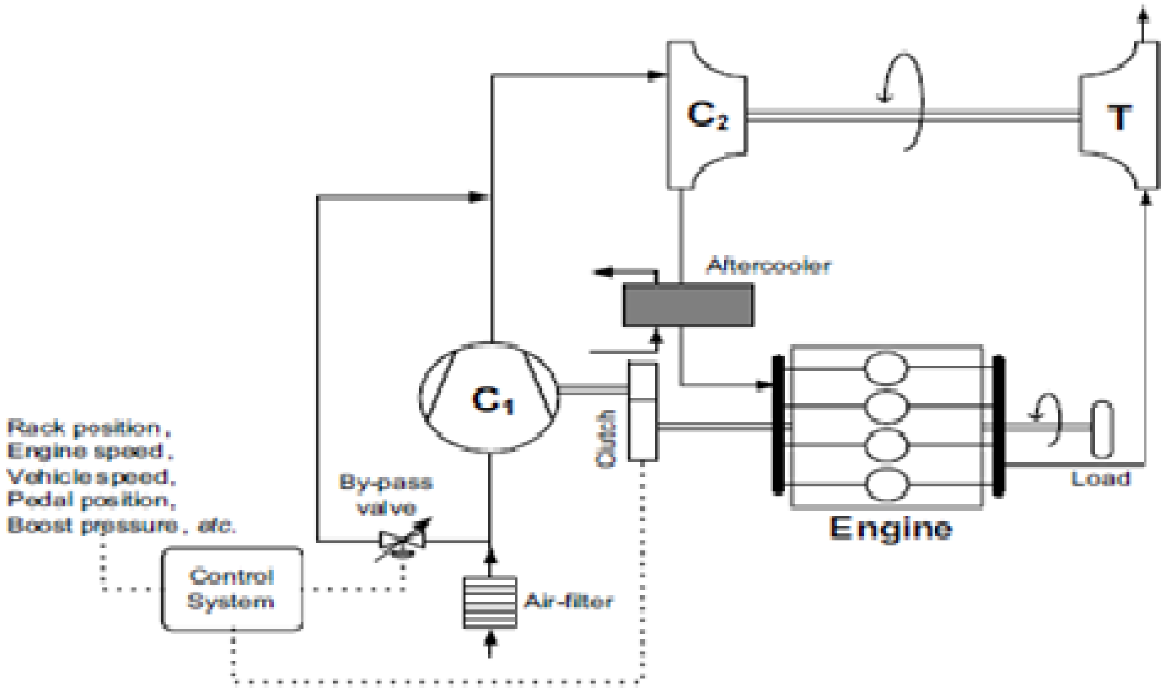

Figure 2.3: Combined supercharging [16]

In *Two-stage turbocharging* as shown in Figure 2.4, two units (turbochargers) as low pressure (LP) and high pressure (HP) are connected such as LP unit is fitted upstream of the HP one with an inter cooler between the two compressors. High compression ratios are employed by this method and efficiency is increased as greater fuel saving is achieved due to elevated boost pressure. But two issues need to be addressed for this configuration - Firstly, due to highly changing operating conditions, a control system is required. Secondly, engine and turbocharger matching is more difficult due to the added complexity by the second turbocharger [16].

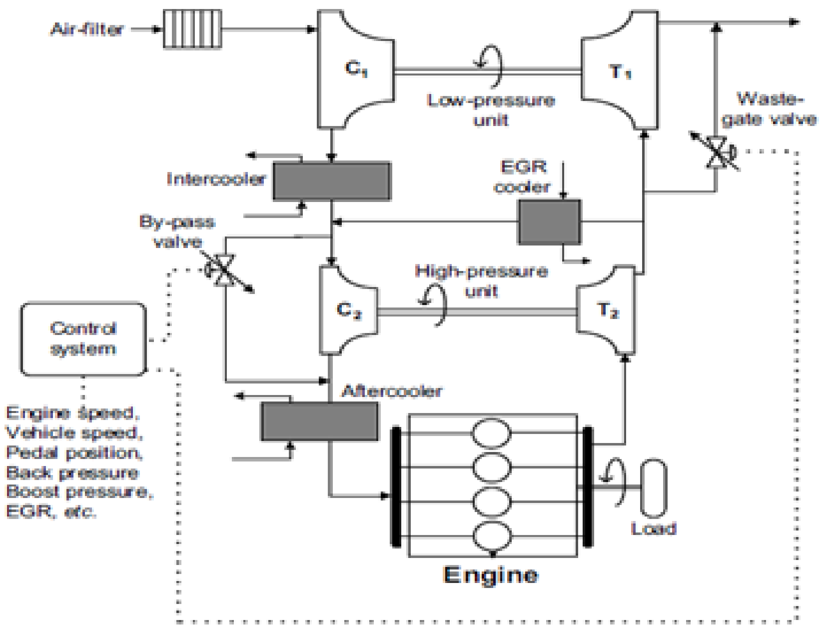

Figure 2.4: Two-stage turbocharging [16]

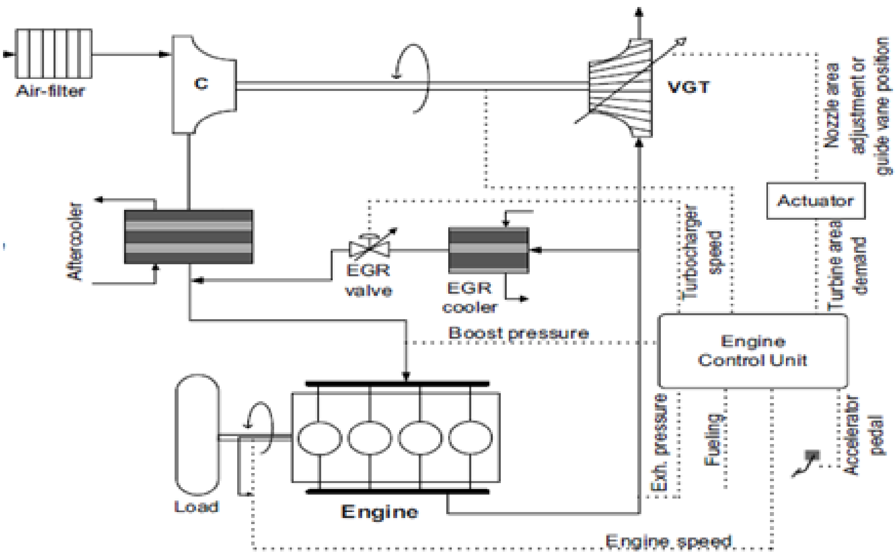

Figure 2.5: Variable geometry turbine (VGT) [16]

VGT is shown in Figure 2.5. The turbine area gets adapted according to the continuously changing engine's operating conditions. During the transient period when the boost pressure is typically low, the turbocharger control system reduces the turbine area by closing down the vanes. This in-turn increases the pressure and temperature of the turbine resulting in increased turbine torque. Thus according to equation 1.1, higher acceleration of turbocharger shaft is achieved that promotes faster transient response. However, extensive matching procedure is needed in addition to sophisticated control system to determine the exact opening and closing pattern according to engine and turbocharger parameters.

In a recent study by Giakoumis and Tziolas [52], a GT-Power model is constructed for a variable geometry turbocharged (VGT) truck diesel engine. Turbocharger as well as engine performance parameters are investigated. The study highlights the benefit of VGT as compared to fixed-geometry operation in terms of higher boost pressure and air supply to the engine cylinders. An instrumental test bed was set up by Rakopoulos et al. [53] to investigate the key engine and turbocharger parameters. A variety of starting conditions are taken into consideration. The results indicate that turbo lag is the major contributor for increased pollutant emissions in all the cases.

Two-stage turbocharger and VGT are compared by Choi et al. [54] and they repoerted faster response at 1000 and 2000 rpm by using two-stage turbocharger as compared to VGT. Further, two-stage turbocharger achieved higher torque at low engine speed as compared to VGT.

In *Sequential turbocharging* configuration (shown in Figure 2.6), two or more turbocharger are arranged in parallel which is in contrast to two-stage turbocharging. In this configuration also smaller units for faster response are connected as in two-stage turbocharging. Sequential turbocharging configuration is complex and accurate matching process is required to avoid the risk of surge when a unit is engaged and disengaged [16]. As per the studies conducted by [55] and [56], the things become more complicated if aftercooler(s) and VGT are incorporated in sequential turbocharging or when more than two units are connected in sequential turbocharging.

ETA is shown in Figure 2.7, an electric motor/generator is used to assist the turbocharger operation. ETA device are also referred as hybrid turbocharger or turbocharger motor/generator [57]. For the reduction of turbo lag during transient operations, electrical hybridization of turbocharger has gained widespread attention as it has potential to be an effective method of improving air delivery into the engine. The motor/generator is added to the turbocharger which can be integrated in various different arrangements. The scope of electrical turbocharger assistance is reduction in time to boost the pressure by the application of the motor/generator which increases the power available to accelerate the rotor assembly [58].

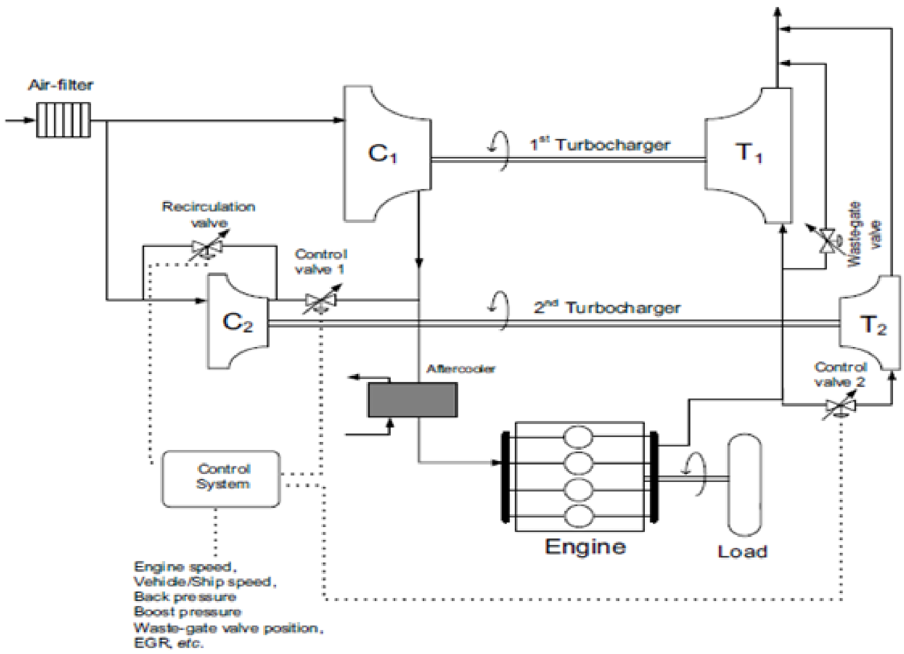

Figure 2.6: Sequential turbocharging [16]

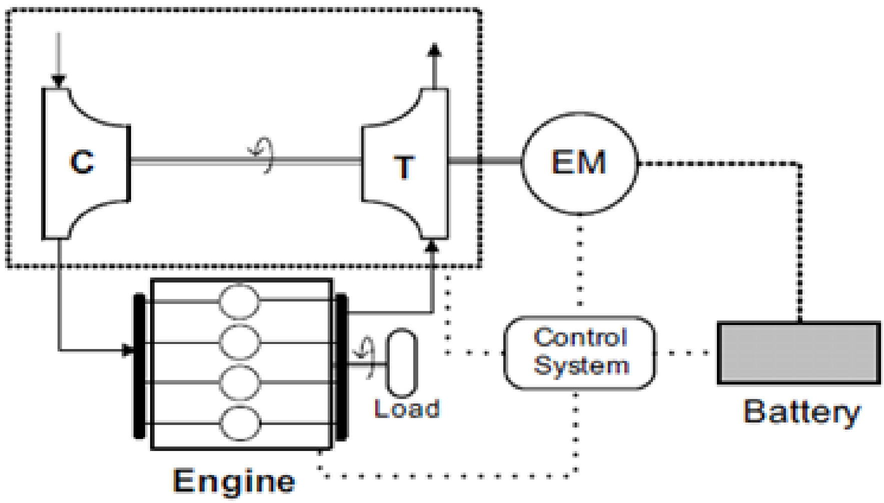

Figure 2.7: Electric torque assist (ETA) [16]

Among the passive methods inertia reduction using lighter materials and ETA are still promising options to improve the transient response. Furthermore, it is worth considering active method of air injection also for review in view of improving the transient response. Hence in the next sub section, review will be carried out on the three main technologies for the reduction of turbo lag: Electric torque assistance, air injection system and by the method of reducing the inertia of the compressor

2.2.1 Electric torque assistance

Tavcar et al. [59] compared the performance of a baseline turbocharged high speed direct injection engine with different types of electrically assisted turbochargers (EATs) as given below:

- Electrically assisted turbocharger (EAT)
- Additional electrically driven compressor (TEDC)
- Electrically split turbocharger (EST)

Analyses were carried out for different driving conditions, including tip in the gear and the new European drive cycle. They found that all EAT topologies improved the transient response of the engine and the driveability of the vehicle. These topologies could also improve the steady state torque output of the engine with retained fueling.

Results reveal that an engine utilizing TEDC delivers the highest torque output and fastest transient response at low engine speeds. While at high engine speeds, EAT provides high torque output in addition to the transient response improvement. EST provides the same transient response as that of EAT, since both are constrained by compressor surge limit. For urban driven cycles, the utilization of EST enables the reduction in fuel consumption. The study concluded that the ideal arrangement of electric turbocharger depends mainly upon the application [60]. The major disadvantage of such systems is increased inertia and their inability to meet low packaging space requirements in modern engines.

The simulation results obtained by Ibaraki et al. [61] reveal 50% improvement in engine torque by a 1 kW motor assist while 100% improvement was achieved with a 2 kW motor assist. Furthermore, the fuel efficiency improved by 8% with a 1 kW motor assist while 12% improvement in fuel efficiency is observed by 2 kW motor assist [61]. Similar investigation performed by Cieslar [50] on time to achieve torque rise shows better performance by the 2 kW motor in comparison to 1 kW motor. However, the 3 kW motor gives very little additional benefit. These studies suggest that the assistance provided by the 2 kW motor is the optimum and indicate that it is equivalent to 0.16 Nm of additional torque [50, 61]

2.2.2 Inertia reduction of the turbocharger

Decreasing the turbocharger's mass moment of inertia plays important role for faster turbocharger and in-turn engine response as clear from equation 1.1. There are following three basic approaches reported in [16] for reducing turbocharger's mass moment of inertia:

1. The most straightforward is through employing lighter materials in order to decrease the turbine rotor mass. For this purpose silicon nitride and ceramic materials are the two successful choices.

2. Rather than using a single turbocharger, two units can be installed with smaller diameters. By the dependency of inertia on the fifth power of rotor diameter [62], this reduces the inertia of the system.

3. Smaller frame turbocharger can be utilized for lower inertia values.

In case of smaller frame turbocharger, efficiency decreases at higher engine speeds. Overboosting at higher engine speed and load is another adverse effect associated with smaller turbocharger frame which could be dangerous for safe engine and turbocharger operation [16].

In a study by Gilkes [63], different inertia values were taken as a percentage from the original compressor. Ricardo Wave simulation software was utilised in the study to investigate the effect of reducing inertia. The pressure response of turbocharger with different inertia values was plotted and the result shows that delay in the turbocharger response was reduced by reducing the inertia of compressor. Moreover, they compared the active method of injecting air into the inlet manifold and passive method of reducing the inertia of the compressor wheel and the results show that both methods reduce turbo lag, however the active system produces superior results. Turbo lag reduction achieved by inertia reduction method was found to be very small.

2.2.3 Air injection strategies

During the operation of CI engines, the air fuel ratio should be maintained within tight limits to obtain maximum engine output. During the turbo lag this ratio reduces and hence the quantity of air available may not be sufficient to meet the torque demand. During the period of low speed and fast acceleration, the air can be injected using different strategies to improve the air supply during transient operation of the turbocharger. Lee and Choi [51] performed detailed experimental study to improve the low speed torque and acceleration performance of the turbocharged engine system by injecting air into intake manifold during transient conditions. They found that the air injection into the intake manifold during the rapid acceleration greatly improves the combustion performance of turbocharged diesel engine.

Air injection is an effective technique to improve the transient response of marine diesel engine [48]. When a ship encounters a sudden danger in the sea, effective stopping of the ship is required followed by acceleration in the reverse direction rapidly. Turbocharged diesel engine is considered as a prime mover of a ship and hence the success of the ship's emergency stop relies on the improvement of the rapid acceleration after the engine is put in reverse. The response of the turbocharger under the transient event of rapid acceleration plays an essential role in this situation. Wei [48] employed air injection technique to improve the turbocharger response. Air injection was applied at the compressor exit. Injection timings and durations were varied and compared under different working conditions. The results of the study demonstrate that additional air injection helps to stop a marine engine faster in case of sudden danger.

Gilkes et al. [64] analyzed the effect of injecting air at either an inlet manifold and compressor or both the manifolds and compressor which is also known as two-point injection system. They performed a parametric study analyzing four parameters that are vehicle speed, outlet pressure of compressor, turbine inlet temperature and driver demand. It is found that the transient response of the turbocharger as well as the overall performance of the engine was improved using air injection systems. With air injection system the turbo lag is reduced by 71% in achieving the vehicle response of 30 km/hour. However, two-point air injection has shown small improvement.

Ceislar [50] examined the effect of air injection at various locations within the engine system and compared the results to establish the optimal injection location. The results revealed that air injection at exhaust manifold overcomes the problem of compressor surge while improving the transient response. In this investigation, the air injected at exhaust manifold is adjusted by a control valve and is stored in a compressed gas tank. The gas tank is recharged during the engine overrun when the pressure in the exhaust manifold is high. So, the compressed gas is collected in the tank during braking to be used later for assistance during a tip-in (acceleration phase). The system design for this functionality is very simple as it requires only a compressed gas tank, an additional exhaust manifold connection and a control valve. Standard diesel EGR/VGT is included in engine air path for the Braking Exhaust Energy Storage (BREES). The transient response of BREES is analyzed when the turbo lag is more pronounced i.e. in high gears. It is worth noting that the time to torque during 3rd gear tip-in acceleration is reduced by 60%. However, the recharging of Braking Exhaust Energy Storage (BREES) is slow, demanding the clutch to be disengaged for about 3 seconds. Also the volume of gas tank is relatively larger for passenger cars.

The pros and cons of various methods of improving the transient response as discussed above are summarized in Table 2.1.

Table 2.1: Various methods of improving the transient response with their pros and cons

Technique	Pros	Cons
Combined supercharging [16]	Improved transient response by means of positive displacement compressor	Reduced engine efficiency
Two stage turbocharging [16]	Increased efficiency with less fuel consumption	Matching difficulty between the two turbochargers, control system requirement
Variable geometry turbine [52]	Increased turbine torque, faster transient response	Requirement of highly sophisticated control system
Sequential turbocharging [16]	Faster response	Complex configuration, accurate matching process was required to avoid surging
Electric torque assist [57]	Reduction in time to boost the pressure, power available to accelerate the rotor assembly was increased	Increased inertia and more space required to install the motor
Inertia reduction of the compressor wheel [63]	The delay in turbocharger response was reduced	Limited literature is available on inertia reduction
Air injection system [51]	Improves the transient response of turbocharger as well as the over all performance of the engine.	The effect of air injection on emissions was not considered.

A common rail direct injection (CRDI) engine was taken into consideration for air flow rationalization as well as turbo lag reduction [65]. The authors proposed a strategy for air injection directly into the combustion chamber. Accumulator and pressure regulator valve are the two additional components in the proposed system. At high engine speed and low torque demand conditions, the pressure regulator allows the high pressure air in the inlet manifold to flow into the accumulator, to be stored there for later use during the turbo lag. So, during low inlet manifold pressure when the torque demand is high, the pressure regulator valve allows the air flow from the accumulator to the inlet manifold. Hence the accumulator supplies the air according to the torque demand and balances acceleration and deceleration.

2.3 Diesel engine exhaust emissions under transient events

To meet the stringent emission regulations, it is important to improve the emission performance of the diesel engine. Like most of the fossil fuels, diesel fuel is mainly composed of carbon and hydrogen. Only CO_2 and H_2O would be generated by the combustion process in an ideal thermodynamic equilibrium. However, many harmful products such as CO, HC, NO_x and PM are generated during combustion due to the following reasons [66]:

- Air fuel ratio
- Ignition timing
- Turbulence in the combustion chamber
- Combustion form
- Combustion temperature etc.

The incomplete combustion leads to CO emission. The concentration of CO largely depends on the air/fuel mixture. For a rich mixture the excess air factor (λ - which is the ratio of stoichiometric air fuel ratio to actual air fuel ratio) is less than 1. Due to air deficiency and reactant concentration in the rich mixture, all the carbon cannot convert to CO_2 and CO is formed. Formation of CO in diesel engine is minimum because they are lean combustion engines having high air fuel ratio (λ >1). However if more droplets are there in the engine and if insufficient turbulence or swift is generated CO is produced. CO_2 emission is considered as the main contributor of global warming. The combustion of carbon in the fuel results in CO_2 emission [67]. In the presence of adequate oxygen and high temperature of the gases, CO may get oxidized to CO_2.

Highly compressed hot air is used in diesel engine to ignite the fuel. Due to very high temperature (above 1600 °C) in the cylinders, nitrogen reacts with oxygen and generates NO_x (nitrogen oxides) emission. Therefore temperature and concentration of oxygen in the combustion are the main factors for the formation of NO_x. This is illustrated in Figure 2.8 [40]. 10-80% load increase event was taken into consideration. The Figure shows the development of NO concentration in the exhaust along with the maximum gas temperature and oxygen concentration in the fuel spray. In the beginning when the load is low, NO emissions were very low. With the increase in load, NO emissions were also increased due to its dependence on temperature. Diesel engines need higher temperature than gasoline engines because they are compression ignition engines. Therefore diesel engines are the main contributors to NO_x emissions. Nitrogen oxide (NO) and nitrogen dioxide (NO_2) are together called as nitrogen oxides. NO gradually converts to NO_2 at atmospheric pressures.

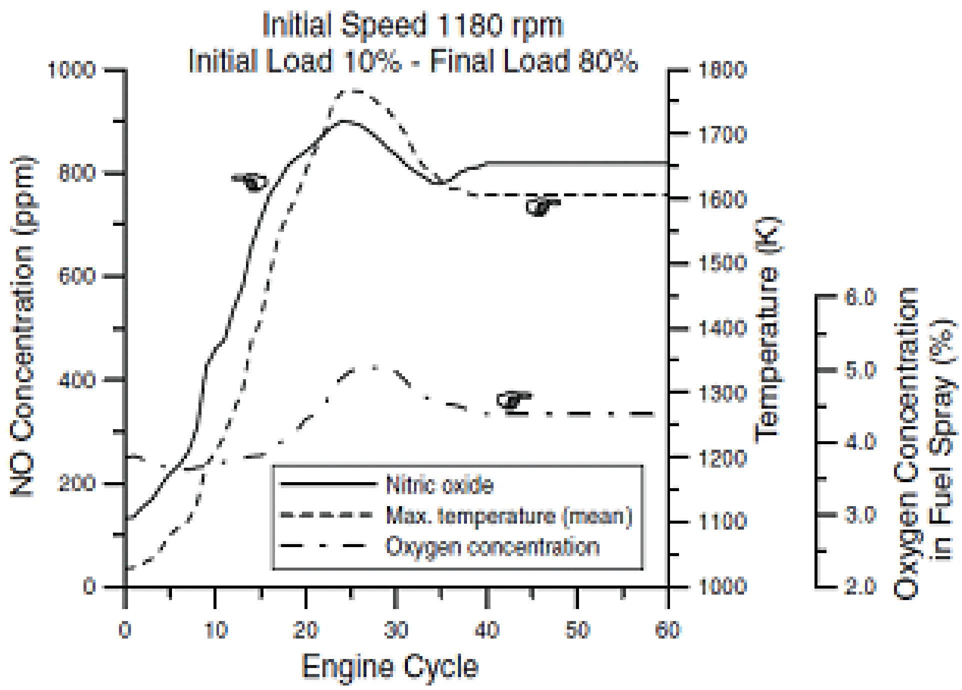

Figure 2.8: NO concentration in the exhaust, maximum gas temperature and oxygen concentration in the fuel spray [40]

Fuel evaporation from the open areas and the incomplete combustion leads to the formation of hydrocarbons [67]. Although diesel engine for automobile has the advantage of superior dynamic and economy [68] but NO_x emissions are high which is a threat on human health and environment [69, 70]. Where as CO, HC and greenhouse emissions are low. Relatively high NO_x emissions in diesel engine is because of high combustion temperatures and lean air-fuel mixtures [71]. The engine exhaust typically contains 70-90% NO and 10 - 30% NO_2 [72].

Emission directives are published by European Commission for engine manufacturers regarding emission requirements. 'Euro 6' standard came into picture in January 2015. Since more emissions are produced by diesel engines, therefore it becomes difficult for engine manufacturers to adhere to these emission regulations. Maximum limit by Euro 6 for NO_x is 80 mg/km and 170 mg/km for hydrocarbons and nitrous oxide emissions. In addition to being very expensive, it is technically challenging to produce a vehicle that adheres to the Euro standards [73].

In a study [74], the effect of air injection by an air cell on the combustion and emission characteristics of a direct injection diesel engine was experimentally demonstrated. The study concluded that the use of an air cell in engine improves the pollutant emissions while BSFC remains constant. In another study [75], researchers experimentally studied the effect of establishing an air-cell inside of the piston body at different injection timings and observed the soot, NO_x emissions, combustion and performance parameters. They reported simultaneous reduction in soot and NO_x emissions in addition to engine performance improvement. Zhou et al. [76] studied heavy duty truck engine with an intake valve closing mechanism and a two-stage turbocharger. They highlighted transient emissions spikes reduction. In an experimental study [77], the effect of intake throttle valve opening on NO_x emissions under world Harmonized transit cycle (WHTC) was examined and 41.5% reduction was observed in NO_x emissions while the fuel economy increased slightly. Guan et al. conducted an experiment on a HD diesel engine using exhaust gas recirculation (EGR) and Miller cycle and they reported minimum impact on fuel economy and smoke emission while lower engine-out NO_x emissions and higher exhaust gas temperatures (EGT) were achieved [78]. Strict regulations on emissions demand the implantation of after treatment devices along the exhaust line of diesel engines [79]. By means of boost and EGR control the lack of temperature at the inlet of the turbine can be overcomed leading to power management in the exhaust gas [80].

Fuel efficiency and NO_x emissions were analyzed under cyclic transient loads for a turbocharged diesel engine by Yum et al. [45]. Load profile consisting of a series of sinusoidal wave with a fixed amplitude at different frequencies and mean values were applied. They reported the average effect of cyclic load on the fuel consumption and NO_x emissions. The effect depends upon the average load level and as the load level was lowered, it became more visible.

2.4 Summary and knowledge gap on methods of controlling turbo lag

In most of the passive methods reviewed above for improving the transient response, more than one unit is combined to achieve lower turbocharger inertia. These include Combined supercharging, Two-stage turbocharging and Sequential turbocharging. As discussed in the previous paragraphs, these configurations require compromise in terms of efficiency, matching and sometimes fuel consumption as in case of combined supercharging. In case of VGT, the main drawback is reduction in the efficiency of engine and/or turbocharger as the turbine can not operate efficiently throughout the whole operating range [16].

Electrically assisted turbochargers are widely used for the improvement of transient response of the engine and hence drivability. Findings of simulation based investigations using software described in the literature are very important to model the optimal real systems before its implementation. As discussed before, rapid increase in fueling rate is rarely investigated before. Electric torque assistance will be simulated using Lotus Engine Simulation under the event of fueling rate change and its effect on transient response characteristics will be reported.

Furthermore, although it is well documented that inertia reduction of the compressor is one of the promising method of improving transient response of turbocharged diesel engine, nevertheless very limited literature is available on the investigations performed using inertia reduction. In the study [63], effect of inertia reduction is observed only on compressor outlet pressure. Compressor speed is another important parameter which represents the transient response of the turbocharger, as during the transient the compressor doesn't accelerate appropriately to meet the air supply demand. Moreover, since reductions in inertia are difficult to achieve, hence it is important to perform further studies which could be using different simulation program. In addition to benchmark the results of [63], the effect of inertia reduction on compressor speed will provide useful insight on the transient response improvement. Use of Lotus engine simulation for this purpose will be a new study to model this effect.

Investigations on air injection technique reported in the aforementioned review are under taken for light duty and medium duty vehicles. However, the effect of air injection on the response of heavy duty vehicles is scarcely reported. There is essential difference between the passenger cars and heavy duty vehicles. The effect of air injection on heavy duty vehicle is considered for marine applications only [48]. But the working conditions of marine applications and road vehicles are quite different. Marine engines normally run at steady state conditions for most of its service. The load remains constant and speed is also relatively fixed while the speed and load of automobile engines keep changing depending upon the conditions on the road. This indicates the need for considering the effect of air injection on heavy duty turbocharged road vehicles [50]. Furthermore, as reported in the review on air injection, for the realization of air injection pressure a compressed gas tank is required which is difficult to incorporate in passenger cars. While in heavy duty vehicles it is not challenging. This is another reason for choosing heavy duty engine used in road vehicles for investigations on air injection.

Use of fossil fuels in automotive sector is one of the primary causes of greenhouse emissions. The automotive engines need to perform at their best efficiency point to limit these emissions. Most of the quality indicators in this regard are based on near steady state global operational characteristics for engines without considering local performance [81]. Considerable amount of research has been carried out as reported in [40] on modelling the thermodynamic and gas dynamic processes as well as combustion in diesel engines. However, exhaust emissions under the transient operation is investigated mainly on experimental rather than simulation basis due to high computational times required for the analysis of each transient cycle. Only few studies [73, 82, 83] are conducted predicting the exhaust emissions through modeling. Furthermore, the techniques employed for transient response improvement are rarely investigated to reduce the exhaust emissions during transient operation. It is important to evaluate the technique used for improving the transient response for exhaust emissions.

2.5 Aims and objectives of the study

Based on the three research problems identified through literature review, the overall aim of the project is to investigate the techniques for improving the transient response of turbocharged diesel engine through modelling and making sure that the technology doesn't need compromise on emissions. The research aim can be achieved by addressing the following objectives:

- **Objective 1:** Engine model development to quantify turbo-lag through torque assistance

- **Objective 2:** Engine model development to quantify turbo-lag by reducing the inertia of rotating component.

- **Objective 3:** Engine model development to quantify turbo-lag through air injection at intake manifold.

- **Objective 4:** Optimization of transient response improvement technique by comparing the three different methods of turbo lag reduction under the event of fueling rate change.
- **Objective 5:** Evaluation of the effect of selected technology for acceleration tests.
- **Objective 6:**Evaluation of the effect of selected technology for load magnitude variation.
- **Objective 7:** Evaluation of the effect of selected technology for load schedule variation.
- **Objective 8:** Evaluation of the effect of selected technology on emission characteristics of the engine under speed transient.
- **Objective 9:** Evaluation of the effect of selected technology on emission characteristics of the engine under load transient.

3

ENGINE SIMULATION

Calculation models are the research tools not only to guide the expensive experimental studies towards optimal solution for engines but they can also generate more information than experiment on the physical phenomenon taking place in the engines [84]. Computer simulations play important role in predicting the performance of engine systems. For studying the engine performance, diesel engine simulation modeling is an effective tool that contributes to the evaluation and new development in this field [85]. Furthermore, since turbocharger is a vital part of most modern diesel engines, the importance of turbocharger simulation can't be neglected. The simulation of turbocharger can provide accurate predictions of performance and emissions of turbocharged diesel engines [86].

Currently, within the engineering sector in particular, computer simulation is gaining widespread attention to model complex real world problems which otherwise require expensive experimental setups. Several commercial engine simulation packages are available in the engineering industry related to the design and optimization of internal combustion engines. The four engine simulation commercial packages most commonly used in the automotive industry are Ricardo Wave (RW), Lotus engine simulation software (LESoft), AVL Fire and GT-Power. These simulation software are similar in purpose and functionality. In a study performed by Chan et al. [87] a comparison was made for Ricardo Wave and Lotus engine simulation. These two software are tested on engine control capabilities. The result of the study reveals that the two codes predict similar pressure and temperature profiles in the engine. Figure 3.1 shows the flow diagram of the simulation model used in the modeling studies. The simulation modeling consists of preprocessing, run and postprocessing. In preprocessing, simulation model is constructed using different components such as engine, turbocharger, intake valves, exhaust valves and fuel injectors. Strategies are developed for different tests and a control system is formulated to model a specific problem. The model is then run and the results obtained in postprcessing are analyzed.

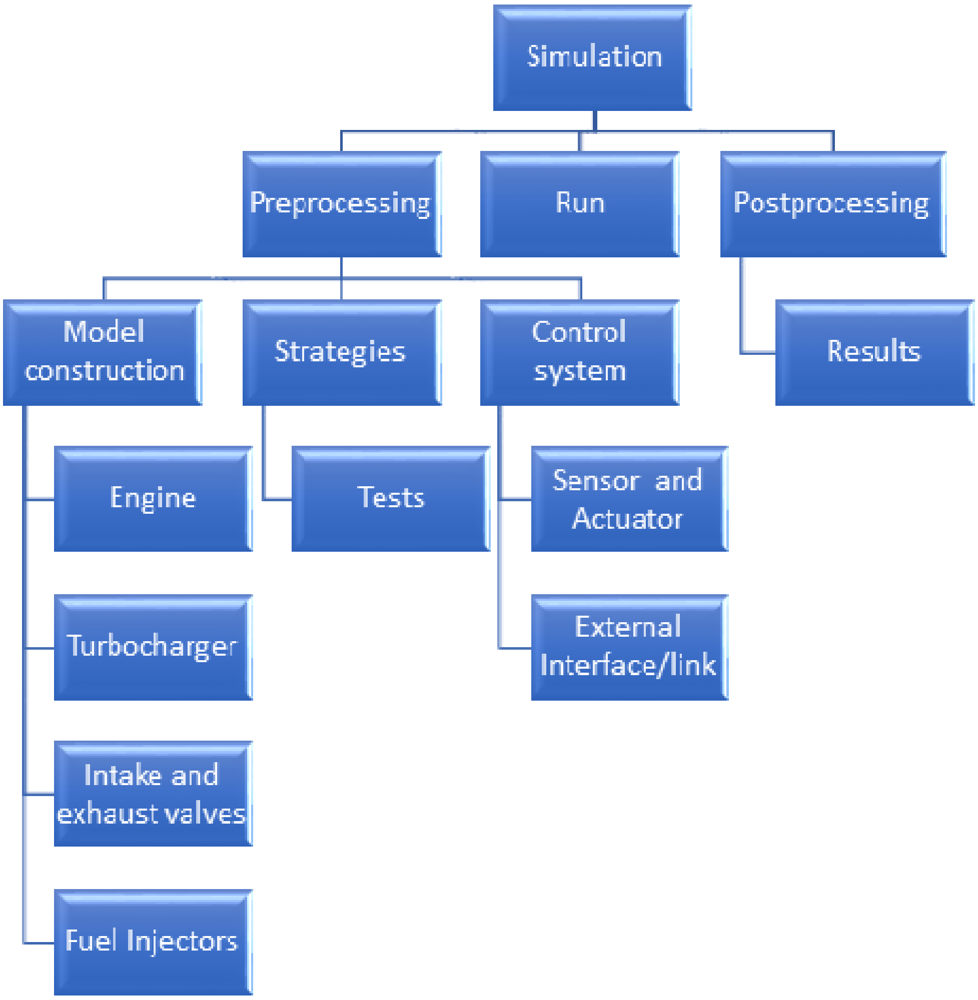

Figure 3.1: Steps in simulation modelling

The present study is performed in two stages. Initially the author could only avail the Lotus Engine Simulation software. So, in the first stage based on the literature review on the capabilities of the software, LES is used to simulate electric torque assistance and further to investigate the effect of reduction in inertia of compressor wheel. These investigations cover the objectives 1 and 2 respectively.

In the next stage Ricardo wave is used to investigate the effect of air injection systems. Switching to Ricardo is done because of its flexibility for sub models, efficient calculation and well tested transient simulation capabilities. The Ricardo wave model is validated against the data collected from the real CI engine and after validation detailed investigations on air injection system are carried out to satisfy objective 3 to objective 9.

In LES software the engine simulation is processed in two modules named as the data module and the solver module. The user inputs the engine data through the data module. The solver module involves combustion and heat transfer zero-dimensional equations and fuel/gas composition solver based on the data provided in data module. The code can predict the gas flow, combustion and overall performance of internal combustion engines. Ricardo Wave analyzes the dynamics of pressure waves, mass flows, energy losses in ducts, plenums and manifolds. It provides time dependent fluid dynamics simulation and thermodynamics using two-zone model [59].

3.1 Lotus engine simulation

The engine model is developed using Lotus Engine Simulation software which can predict the complete performance of an engine system. The software provides a Concept Building Tool to build a complete engine model based on these three key parameters: number of cylinders, swept volume and engine speed at maximum power. As presented by Pearson et al. [88], the engine power is given by

$$W = \frac{P}{1200} V_{\text{swept}} N \tag{3.1}$$

where P is the brake mean effective pressure (BMEP) in bar, V_{swept} is the engine swept volume in litres and N is the engine speed in rev/min. The primary factor for determining the power output from an engine of a given swept volume is its maximum rated speed. Hence the maximum rated speed together with the number of cylinders and swept volume is responsible for defining the engine performance. The BMEP at a fixed air-fuel ratio (AFR) and given fuel specific heat value (Q_v) is related to the volumetric efficiency (η_v), thermal efficiency (η_{therm}) and the combustion efficiency (η_{comb}) by the equation:

$$P \alpha \frac{Q_v}{AFR} \eta_v \eta_{therm} \eta_{comb} \tag{3.2}$$

Lotus Engine Simulation uses a 'heat release' model to simulate the energy release mechanism from the mixture. In this model the mass fraction of burn fuel is calculated using a Wiebe function [89]. To model the combustion inside the cylinder, two-part

Wiebe function is utilized. It has two consecutive combustion periods: premixed and diffusion. For predicting the burned mass fraction in each combustion period, following equations are used [90]:

$$m_{\text{premix}} = 1 - \left[1 - \left[\theta/\theta_{\text{b}}\right]^{\text{C}_1}\right]^{\text{C}_2} \tag{3.3}$$

$$m_{\text{diff}} = 1 - e^{\text{-A}(\theta/\theta_{\text{b}})^{\text{M}+1}} \tag{3.4}$$

In the above equation, C_1, C_2, A and M are Wiebe coefficients. θ is the burn angle and θ_{b} is the total burn angle. Heat transfer to the three major surfaces cylinder head, piston and liner is evaluated from the difference between the instantaneous gas temperatures and the metal surface temperature. Annand heat transfer model is used to calculate cylinder heat losses [39]. The convective heat transfer coefficients are derived from the well- known semi-empirical relationships of Annand model. Simulation of the energy release rate and the heat transfer processes enables to predict the cylinder pressure during combustion and hence a complete cycle simulation is performed [88]. Annand model is defined as:

$$ARe^{\text{B}} = hB_{\text{cyl}}/k \tag{3.5}$$

where A and B are the Annand coefficient for open and closed cycles, B_{cyl} is the cylinder bore, h is heat transfer coefficient and k is the thermal conductivity of the gas in the cylinder. Further, following equation determines the heat transfer per unit area [91]:

$$dQ/A = h(T_{\text{g}} - T_{\text{w}}) + C({T_{\text{g}}}^4 - {T_{\text{w}}}^4) \tag{3.6}$$

In the above equation, T_{g} is the temperature of gas, T_{w} is the temperature of cylinder wall and C is the Annand coefficient only for closed cycle.

As the turbocharger is used to deliver more power out of a smaller engine, a relatively smaller engine is chosen to be modeled. A four cylinder, four stroke, CI engine is modeled in this study. The engine specifications are shown in Table 3.1. It shows the specification of turbocharged diesel engine based on engine type, stroke, bore, number of cylinders, combustion system, compression ratio, connecting rod and displacement. The specifications match with the specifications of the engine used in a mid range car as mentioned in [81]. A fully connected turbocharged model is shown in Figure 3.2. Engine cylinders, components of intake and exhaust manifolds, sensors, actuators and turbocharger are explicitly demonstrated on the figure. The manifolds have been modeled using plenum, pipes, valves and ports (blue at intake and orange at exhaust). Cylinders specifications are provided as shown in Table 3.1. Bore is taken as 88.0 mm, stroke as 85.0 mm, connecting rod length as 144 mm and compression ratio as 19.5:1. Firing order is set as 1-3-4-2 where cylinder 1 is the top cylinder. Valves are selected from the intake component tab and attached to the cylinder elements via connectors. Inlet and exhaust boundaries are added to complete the basic model. Fuel element is added to the model and fuel type is selected as diesel and the fuel system is chosen as direct injection. Compressor is connected to the inlet plenum and the turbine is connected with the exhaust manifold.

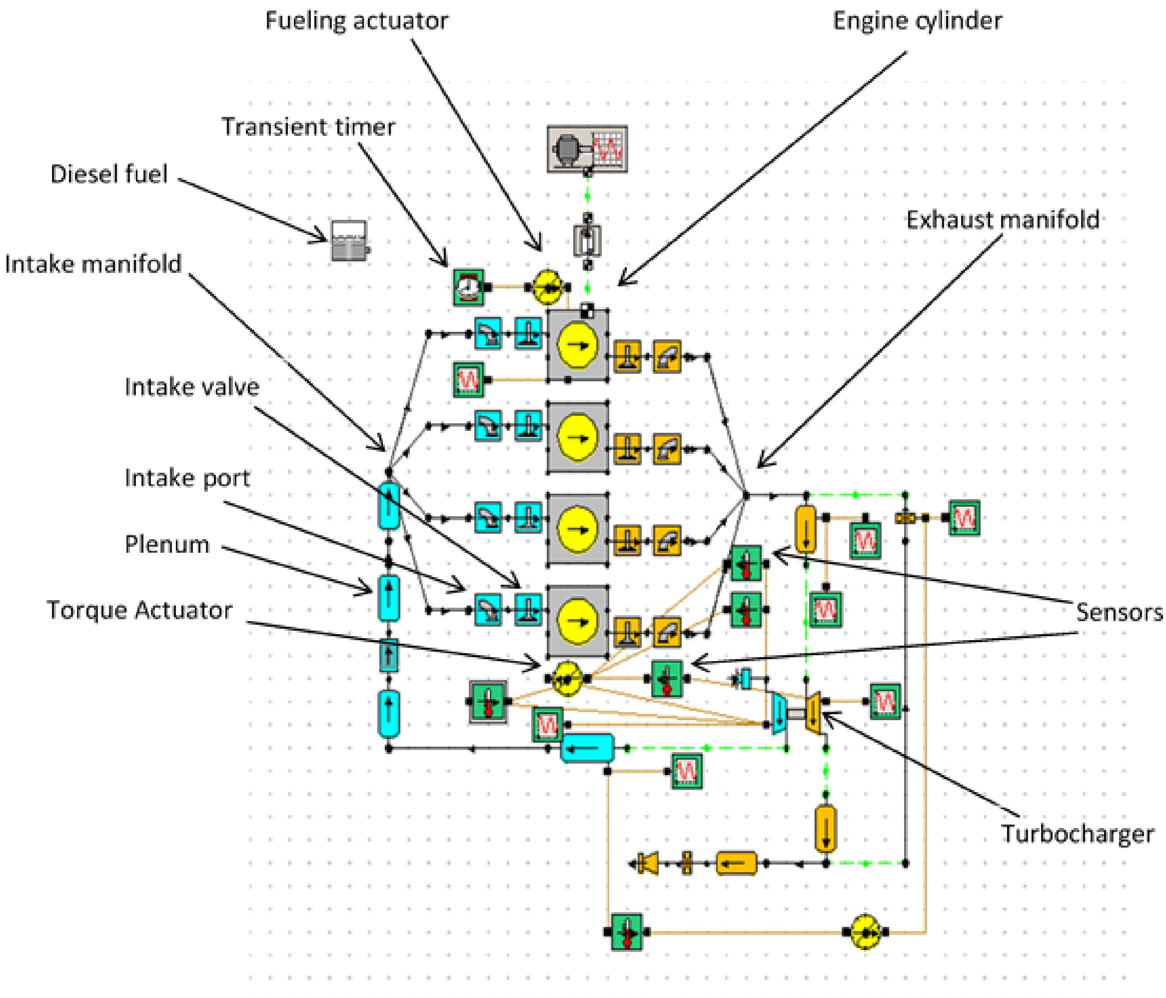

Figure 3.2: Turbocharged engine model in Lotus Engine

Table 3.1: Engine specification

Engine Type	4 Stroke Diesel
Stroke	85 mm
Bore	88 mm
Number of Cylinders	4
Combustion System	Direct Injection
Compression Ratio	19.5:1
Connecting rod	144 mm
Firing order	1-3-4-2

3.2 Ricardo wave simulation

Currently, within the engineering sector in particular, computer simulation is gaining widespread attention to model complex real world problems which otherwise require expensive experimental setups. Ricardo Wave is a 1 D engine and gas dynamic simulation software package commonly used worldwide. In this model, the combustion process is modeled by Diesel Weibe combustion model, which is an appropriate heat release model. The model uses functions to determine the correlations for premixed and diffusion burn with addition of tail burning which represent the slow late burning. The mass fraction burned, W is given by the equation [92]:

$$W = P_f[1 - [1 - (0.75\tau)^2]^{5000}] + d_f[1 - [1 - (cd_3\tau)^{1.75}]^{5000}] + t_f[1 - [1 - (ct_3\tau)^{2.5}]^{5000}] \tag{3.7}$$

where

P_f - mass fraction of the premix

d_f - mass fraction of the diffusion

t_f - mass fraction of the tail burn curves

cd_3 - coefficient of burn duration for the diffusion

ct_3 - coefficient of burn duration for the tail burn curves

The following equations are used to find d_f, t_f, cd_3 and ct_3:

$$d_f = 1 - P_f(1 - \alpha) \tag{3.8}$$

$$t_f = 1 - P_f(\alpha) \tag{3.9}$$

where

$$\alpha = 0.6 * [min(\phi,0.85]^2 \tag{3.10}$$

$$cd_3 = \frac{0.055}{1 + 0.5min(\phi,0.85)} \tag{3.11}$$

$$ct_3 = \frac{3.7cd_3}{1.12min(\phi,0.85) + 1} \tag{3.12}$$

In the above equations, ϕ is the equivalence ratio.

Finally, the equation used to determine the burn duration term:

$$\tau = \frac{\theta - \theta_b}{125(\frac{RPM}{BRPM})^{0.3}} \tag{3.13}$$

where

θ = Crank angle

θ_b = crank angle at the start of combustion

RPM = Engine speed

BRPM = User-entered reference speed

For heat transfer process, the Woschni correlation model is used. The Woschni model expresses the convective heat transfer coefficient as given by the equation [92]:

$$h_g = 0.0128D^{-O.20}P^{*0.80}T^{0.53}V_c^{0.80}C_{enht} \tag{3.14}$$

where

D: Cylinder bore

P: Cylinder pressure

T: Cylinder temperature

V_c: Characteristic velocity

C_{enht}: User entered multiplier (For diesel jet combustion sub-model $C_{enht} = 1$)

In the present study a four cylinder, four stroke, turbocharged diesel engine available at the Advanced Automotive Laboratory of University of Huddersfield, UK is used to validate the engine model constructed using Ricardo Wave simulation software. The description of the diesel engine is provided in Table 3.2. The engine specifications provided are applicable to the heavy duty diesel engine [67].

Different parameters in Ricardo wave package are modelled using a number of mathematical as well as empirical formulae. For heat release, Diesel Wiebe function [89] is used as due to the unavailability of actual heat release profiles, this is the most appropriate heat release model. Model parameters are used to adjust the burn duration over the engine speed range. An ignition delay model is included in the Wiebe function to define the crank angle at which combustion is initiated. Woschni model for convective heat transfer is used to calculate the heat transfer [89]. In this model, simple heat transfer is assumed from a confined volume surrounded by all sides by walls representing the cylinder head, cylinder liner and piston face areas exposed to the combustion chamber [64].

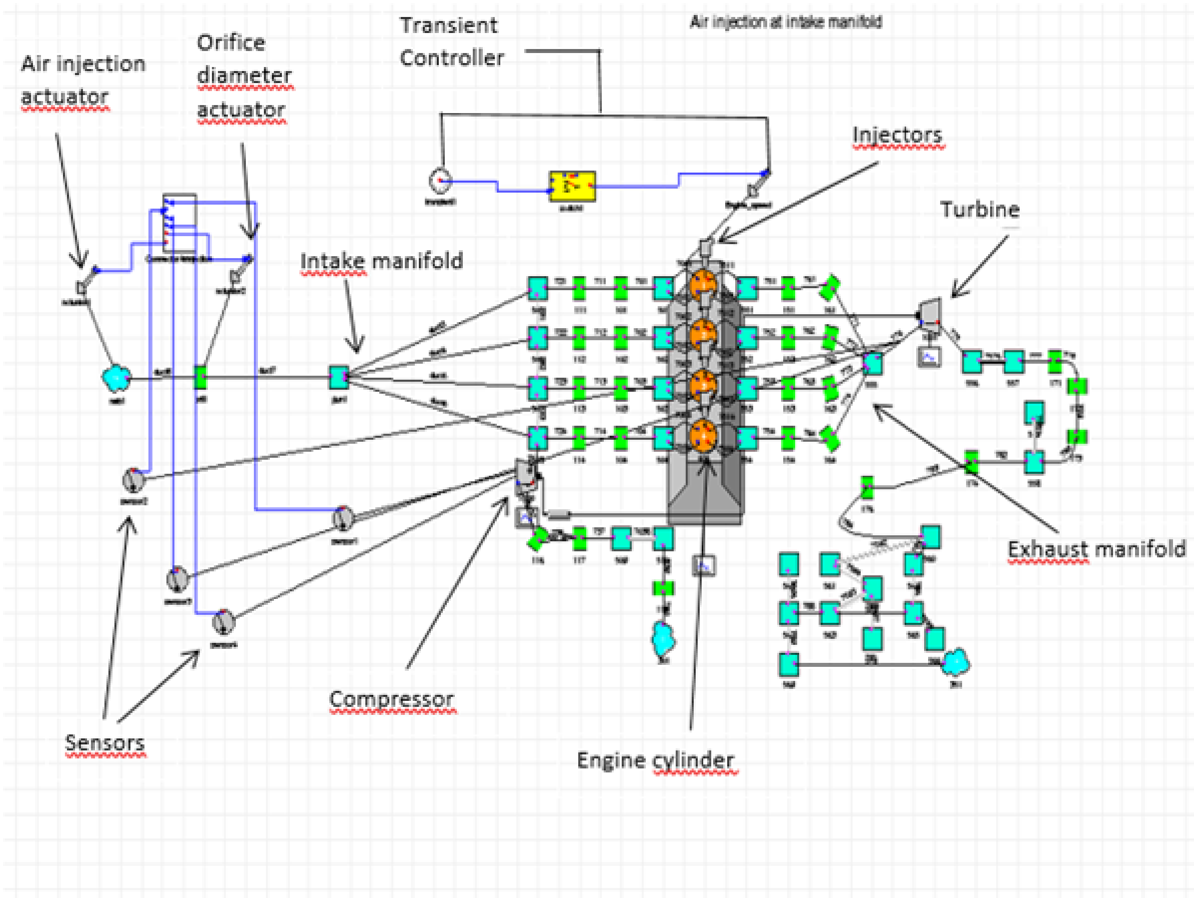

Figure 3.3: Turbocharged engine model in Ricardo Wave

Wave Build is a GUI interface to build a wave model. A fully connected turbocharged engine model is shown in Figure 3.3. Elements representing different engine components are connected by ducts. Cylinders are selected from the flow element and as per Table 3.2 bore is taken as 103 mm, stroke as 132 mm and compression ratio as 18.3:1. For intake and exhaust valve, the basic valve parameters are entered as shown in Table 3.3 [93]. The intake and exhaust manifolds are modeled using ambients, ducts, orifices and junctions.

From the engine general panel, engine type is chosen as diesel and the specifications are entered as described in Table 3.2. As discussed earlier, Diesel Weibe is chosen as the combustion model and Woschni model is chosen for heat transfer. Fuel injectors are added to the model. From the turbo junction flow element, compressor and turbine are added on the canvas and a turbo shaft is selected from the mechanical element to connect the compressor and turbine. Compressor is connected to the intake manifold and the turbine is connected with the exhaust manifold. Turbocharger is modeled with its performance maps. Ricardo WAVE provides pre-processed standard compressor and turbine map files. Due to the unavailability of turbocharger map data for the actual system, these default map files provided by Ricardo wave are used in the present investigations. This may affect the accuracy of the simulation, however, good agreement between the results of simulation model and the experimental data for steady state shows that this affect is negligible on the accuracy of the simulation. The maps consist of 4 dimensional dataset: rotational speed, mass flow rate of the gas, pressure ratio and efficiency. The performance maps need pre-processing before they can be used by WAVE. There exists a routine in WAVE Build that has an algorithm to convert a performance map into a TCMAP object that WAVE uses to model both the compressor and the turbine [92].

In order to model the transient event, a controller is constructed. Three control elements are added to the model: a transient element, a switch and an actuator. Based on the transient event, the profile of the transient element can be set.

3.3 Modelling of techniques

This section provides details of the modeling strategies for torque assist, inertia reduction and air injection method. As stated earlier, the first two techniques are modelled using Lotus engine simulation and the third technique is modelled using Ricardo wave simulation software.

3.3.1 Modelling of torque assist and inertia reduction

A control system for providing the additional power to the turbocharger is constructed using sensors and actuators. The sensors in LES give the compressor power and turbine power. To simulate electric turbocharger assist system, an actuator is used which adds an additional torque to the turbocharger shaft. The angular velocity of the shaft is calculated using Newton's second law for rotational systems as given in the below equation [94]:

$$\dot{w} = \frac{1}{J_{tc}}(T_{qt} + T_{qja} - T_{qc}) \tag{3.15}$$

where
T_{qt} = Torque generated by turbine (Nm)
T_{qja} = Torque from Jet-Assist (Nm)
T_{qc} = Torque consumed by compressor (Nm)
J_{tc} = Inertia of turbo system (kgm^2)

The effect of change in inertia of compressor wheel on engine performance is investigated using different inertia values found as a percentage from the original compressor modelled with inertia as 6e-5 kgm^2. The reduction in inertia is then observed for 10%, 25% and 40%. The first two reductions are taken as motivation from Gilkes et al. [63]. As reported by [16], 50% inertia reduction brings faster engine response. Close to this value, the third reduction is chosen to observe the effect on the transient response of the engine.

3.3.2 Modelling of air injection

To regulate the air injection pressure, a control system is constructed using Simulink [95]. Ricardo wave software has limited control system capabilities when compared to other dedicated control packages such as Simulink. However, the wave package includes an interface to Simulink [96]. The control system in Simulink coupled with wave model is shown in Figure 3.4. Simulink accesses the wave signal through sensors and actuators. Signals from the sensors are send "Out of WAVE" whereas the signals to the actuators are sent "Into WAVE". In Simulink Mux block is used to create a vector of input signals while the Demux block is used to split the vector of output signals [92].

The air injection system is modeled by the addition of third ambient. Air is supplied through an extra ambient into the intake manifold. The mass flow of air is regulated by an orifice. Orifice models a restriction in the system such as to restrict the mass flow of the air. Its a joint style junction used to connect the two ducts. The orifice's diameter can be set equal to or smaller than the smaller of the two ducts [92]. The air with injection pressure P_1 is passed through an orifice for which the orifice flow (equation 3.16) gives the mass flow rate of the air [50].

$$\dot{m} = \frac{AP_1}{\sqrt{RT'}}\psi\left(\frac{P_1}{P_2}\right) \tag{3.16}$$

where ψ is defined as per the following equations:

For $P_2/P_1 > 0.528$,

$$\psi\left(\frac{P_1}{P_2}\right) = \left(\sqrt[\gamma]{\frac{P_2}{P_1}}\right)\sqrt{\frac{2\gamma}{\gamma-1}\left(1-\sqrt[\frac{\gamma}{\gamma-1}]{\frac{P_2}{P_1}}\right)} \tag{3.17}$$

And for $P_2/P_1 \leq 0.528$,

$$\psi\left(\frac{P_1}{P_2}\right) = \left(\frac{2\gamma}{\gamma-1}\right)\left(\frac{\gamma+1}{2(\gamma-1)}\right)\sqrt{\gamma} \tag{3.18}$$

where

P_1: Pressure before the orifice

P_2: Pressure after the orifice

A: Orifice area

T': Gas temperature at the orifice inlet

R: Gas constant

γ : Ratio of specific heats

Orifice diameter can be changed in modeling through an actuator. In actual running case, an orifice flow meter can be used to monitor the flow rate of air and the flow of air can be controlled with a precision flow control valve [97]. A compressed gas tank can be utilized for providing the compressed air.

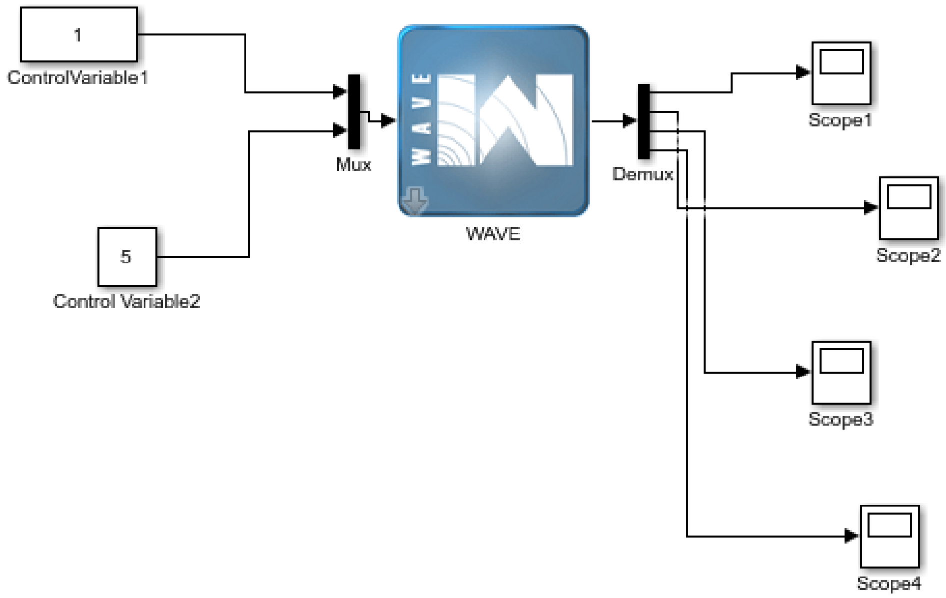

Figure 3.4: Wave coupled Simulink Model

Further, to observe the effect of air injection on emission characteristics of the engine, the parameters of the sensors to the Simulink model are altered. Following emissions are chosen at the sensors to investigate the effect of air injection:

1. CO emission
2. NO_x emission
3. HC emission

Ricardo wave simulation software enables to analyze the exhaust emissions. From the engine panel CO, HC and NO_x emissions are enabled under the combustion tab for the diesel engine.

3.4 CI engine test bed

The CI engine used to validate the simulation studies is available at the Advanced Automotive Laboratory of University of Huddersfield, UK. The schematic diagram of the experimental facility is shown in Figure 3.5 and the engine's characteristics are presented in Table 3.2. The test facility consists of a turbocharger, 220 kW AC dynamo-meter, pressure transducers, speed sensors, fuel flow meter and in-line torque meter [98]. It is equipped with CADET software for programming the steady state and transient cycle. The test rig was utilized to conduct the steady state experiments. Before performing the experiment, the engine was run for 10 minutes to reach the steady state. Engine parameters (Brake power, Brake specific fuel consumption and Brake torque) were observed for the engine speed varying from 1000 - 2200 rpm. These steady state results provide useful data to validate the simulation based studies.

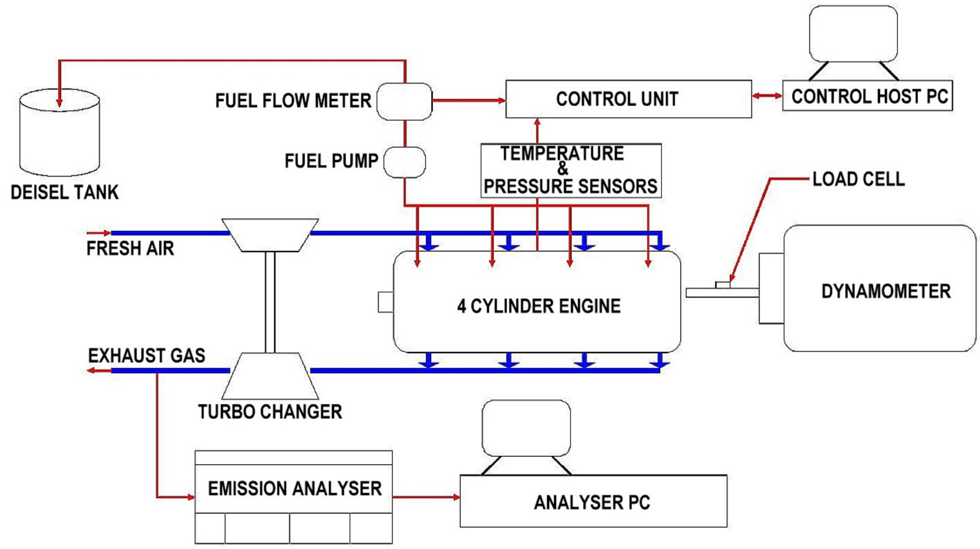

Figure 3.5: Schematic of experimental Facility [98]

Table 3.2: Engine block specifications

Engine Type	4 Stroke Diesel
Stroke	132 mm
Bore	103 mm
Number of Cylinders	4
Combustion system	Direct injection
Compression Ratio	18.3:1
Connecting rod	144 mm
Displacement	4.399 liter

Table 3.3 shows the basic valve parameters of the engine. There are two valves per

cylinder for both intake and exhaust manifold. The diameter of intake valve is 36.2 mm and the exhaust valve is 33.5 mm.

Table 3.3: Basic valve parameters

Intake valves per cylinder	2
Intake valve diameter	36.2 mm
Exhaust valves per cylinder	2
Exhaust valve diameter	33.5 mm

3.5 Test procedures

In this section, two broad topics are covered: strategies for the transient events that the study covers and modelling of techniques investigated. The strategies that are applied in the present work are discussed which will provide a clear basis for carrying out the investigations under various operating conditions. Experimenting with different transient operating conditions is difficult as it requires expensive setups. Strategies are developed to model the techniques identified in literature review using the simulation codes mentioned in this chapter.

3.5.1 Strategies for transient events

In urban roads, the working condition of vehicle engine are mostly transient and constant speed and changing torque conditions are more popular [36] Hence, in the present study, more rigorous analysis is performed for load transients although acceleration transients are also investigated but to a lesser degree.

Furthermore, rapid increase in fueling rate also corresponds to a transient event as due to this elevated fueling and insufficient air flow in the combustion chamber, air fuel ratio becomes very low. Hence three types of transient events are considered for investigation: fueling rate change, speed transients and load transients. While comparing the technologies for relative improvement in transient response, fueling rate change is chosen to represent a transient event. After finding the optimum technology, detailed investigations are then performed for speed and load transient operating conditions.

3.5.1.1 Fueling rate change

The performance characteristics of the engine is affected by several parameters such as gas temperature and pressure in the cylinder, temperature and pressure characteristics of the turbocharger and the pressure variations during inlet and exhaust processes. The transient performance of the IC engine can be improved by supplying additional air into the cylinder and by controlling the fueling rate [97].

The transient performance of the engine is simulated through the rapid change in fueling rate which represents the transient event. Initially the model runs at 1000 rpm. During the transient, the quantity of fuel injected is increased from 40 mm^3/injection to 52 mm^3/injection and hence the boost pressure rises. The values of fueling rate have been referred from LES for modelling the transient response of turbocharged diesel engine [39].

3.5.1.2 Speed transients

Acceleration time and air injection pressure are the two important transient performance factors for which the turbocharger response is measured [97].

For the sake of studying the effect of acceleration time and injection pressure three tests are carried out as shown in Table 3.4. Test 1 presents the case of no air injection and only the acceleration time is varied by 1 sec and 2 sec. Test 2 is taken as the case of more rapid acceleration with the speed change in 1 sec and the injection pressure is varied. Finally, test 3 highlights the comparative improvement in turbo lag reduction for 1 sec and 2 sec injection duration respectively for the injection pressure obtained from test 2.

Table 3.4: Test cases

Tests	Turbocharger response	
	Acceleration time	Air injection pressure
Test1	Varied	Not applied
Test2	Constant	Varied
Test3	Varied	Constant

3.5.1.3 Load transients

As discussed in section 2.1, constant speed and increasing torque (CSIT) is more popular than other load conditions. So, CSIT is the main focus for the investigations under load transient and these tests will be referred as "Load magnitude".

Nevertheless, narrow speed range transients are also demanding as discussed in the previous section. To this aim, investigations are also performed for constant torque and changing speed. These tests will be referred as "Constant load" and the results are presented under section 5.4.1. Finally, scheduling of transient event is covered under section 5.4.3 to observe the effect of faster or slower load application on key performance indicators (KPI), the test will be referred as "Load schedule". For speed transients at a constant load step (section 5.4.1) and for load magnitude variations (section 5.4.2), transient time is taken as 1 sec which represents the case of rapid transient as discussed in section 3.5.1.2. For load scheduling, transient time is taken as 2 sec to observe the effect of slower load application (section 5.4.3). The tests described above for the three different load strategies are presented in Table 3.5.

Rather than randomly choosing the load steps, identification of load step is done based on engine performance degradation. It would be a novel approach of selecting the loads for the present investigation. Figure 3.6 shows the methodology for choosing the load steps. Different load step values are investigated for engine torque which is an important performance measure to observe the effect of various load steps. By the term torque deficit, it is meant the maximum percentage deviation of torque from its steady state value. Air injection is then simulated for the identified load steps and optimization of injection pressure is done based on the mass flow of the air passing through the orifice. The effect of optimized air injection is evaluated on turbocharger response parameters and performance improvement of the diesel engine under the transient events of load change is reported.

Table 3.5: Tests cases for load strategies

Tests	Load steps	Speed	Acceleration time
Constant load	Constant	Varied	Constant
Load magnitude	Identification based on torque deficit	Constant	Constant
Load schedule	Identification based on torque deficit	Constant	Varied

3.5.1.3.1 Constant load

Optimum injection pressure is considered based on the acceleration tests (to be carried out in Chapter 5) and the effect of this air injection on the transient response of the engine is evaluated for a load change from 80% to 100% in 1 sec at different speeds. This represents the case of higher engine load at a particular speed. Within the working range of the engine, five different engine speeds 1000 rpm, 1200 rpm, 1400 rpm, 1600 rpm and 1800 rpm are considered and the turbocharger response is observed for load change process at each speed.

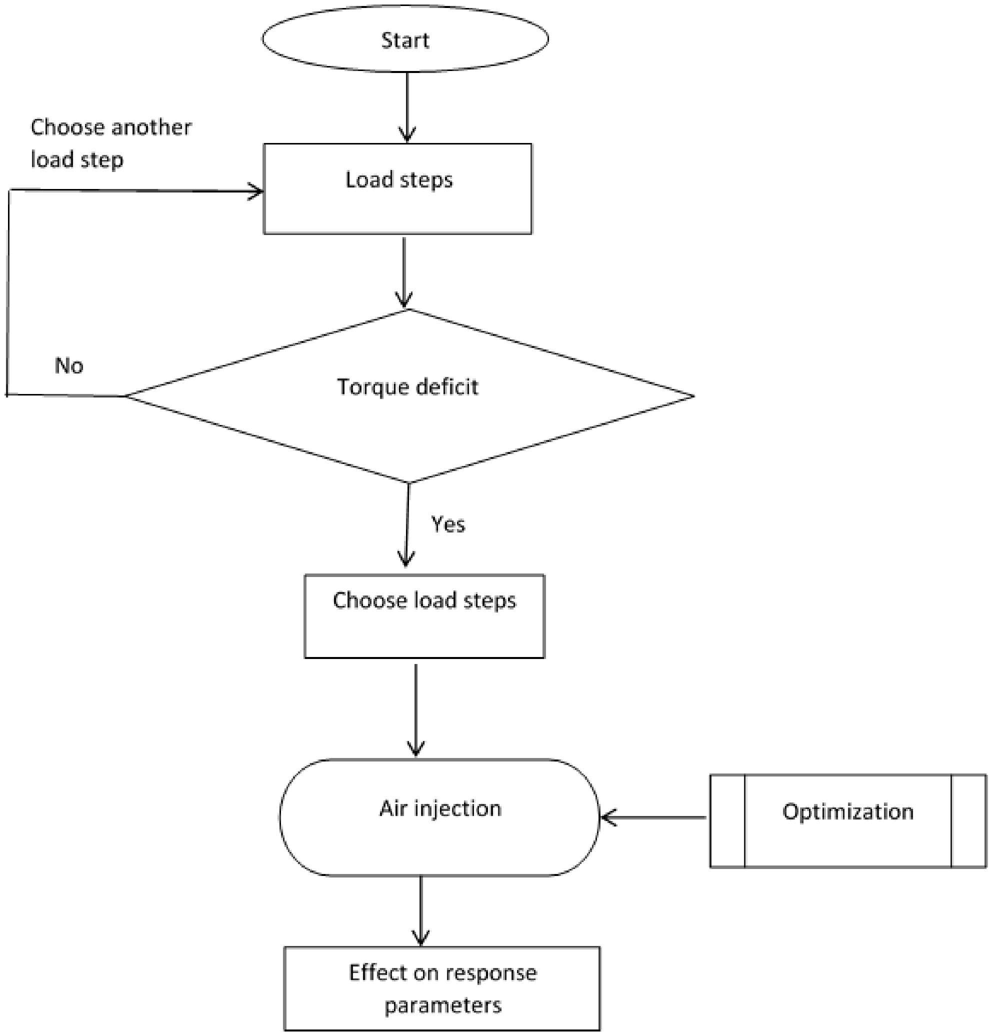

Figure 3.6: Identification of load steps in load tests

3.5.1.3.2 Load magnitude

For final speed drop, the magnitude of applied load plays the prime role. In the initial steady state condition, the engine and the load torques are equal. With the application of new load, torque deficit occurs, when the engine torque can not instantly match with the increasing load torque [99]. The result is the engine speed drop which will be shown in the investigations.

To identify the load steps as discussed in section 3.5.1.3, initial load is taken as 10% and the final load is varied from 20% to 80%. As shown in figure 3.7, the effect of load change is more prominent for 70% to 76% final load. During the early load steps

(10% to 66%) engine torque continues to increase with time which indicates that during these load changes turbocharger is able to produce the required boost and the engine is capable to accept these loads. However, engine torque deteriorates during 70% to 76% final load and the engine speed becomes zero if the final load is further increased.

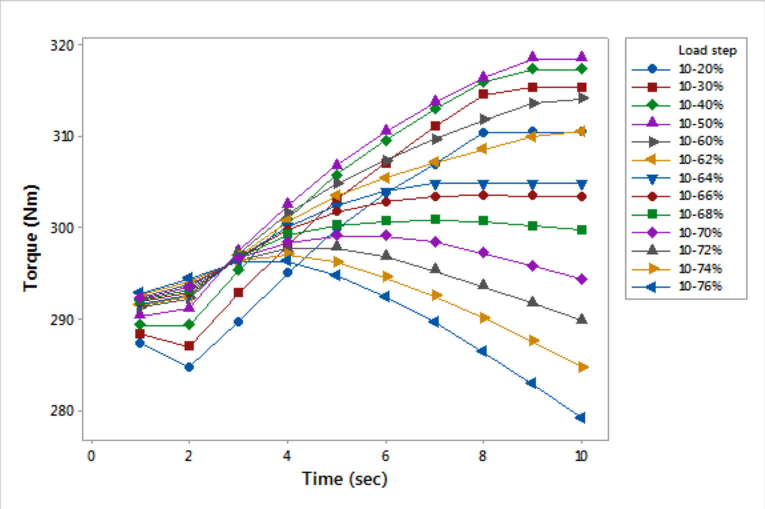

Figure 3.7: Variation of engine torque with different load steps in 1 sec, initial load 10%

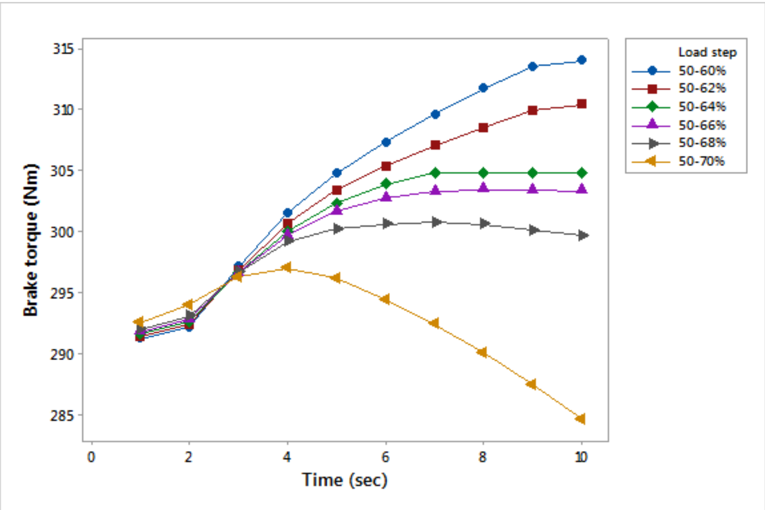

Figure 3.8: Variation of brake torque with different load steps in 1 sec, initial load 50%

Similarly, decrease in torque is also observed for initial load 50% as shown in Figure 3.8. Torque deteriorates for 68% to 70% final load and engine speed becomes zero if the final load is increased beyond 70%. The load steps with torque deficit are identified as load tests and are presented in Table 3.6. From the table it is clear that there are three load steps with 10% initial load and two with 50% initial load that need investigation for load acceptance.

Table 3.6: Load tests resulting in torque deficit

Load Test	Initial load	Final load	Torque deficit
L1	10%	70%	1.5%
L2	10%	72%	2.7%
L3	10%	74%	4.0%
L4	10%	76%	6.0%
L5	50%	68%	0.4%
L6	50%	70%	4.5%

3.5.1.3.3 Load schedule

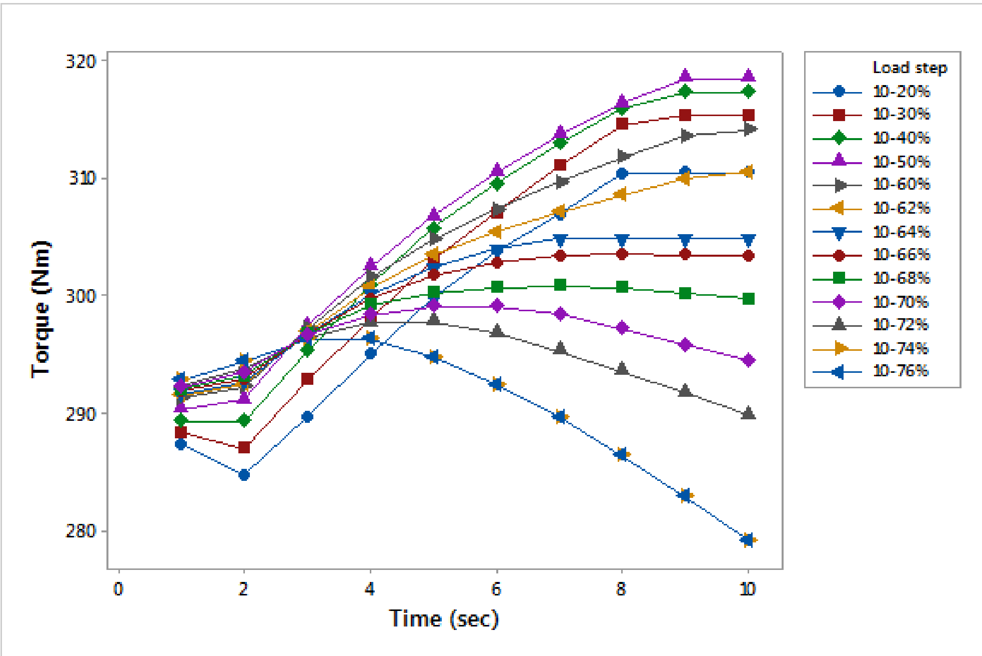

Figure 3.9: Variation of brake torque with different load steps in 2 sec, initial load 10%

In load magnitude variation, load step is changed in 1 sec. In the present test, load step is varied in 2 sec to observe the effect of slower load application. As shown in Figure 3.9

and Figure 3.10, various initial and final load are investigated to find the decrease in torque under different load steps, in the same way as done in section 3.5.1.3.2 . It is found that 10% to 68-76% and 50% to 68-74% load step corresponds to the decrease in torque which can be investigated under air injection for improvement.

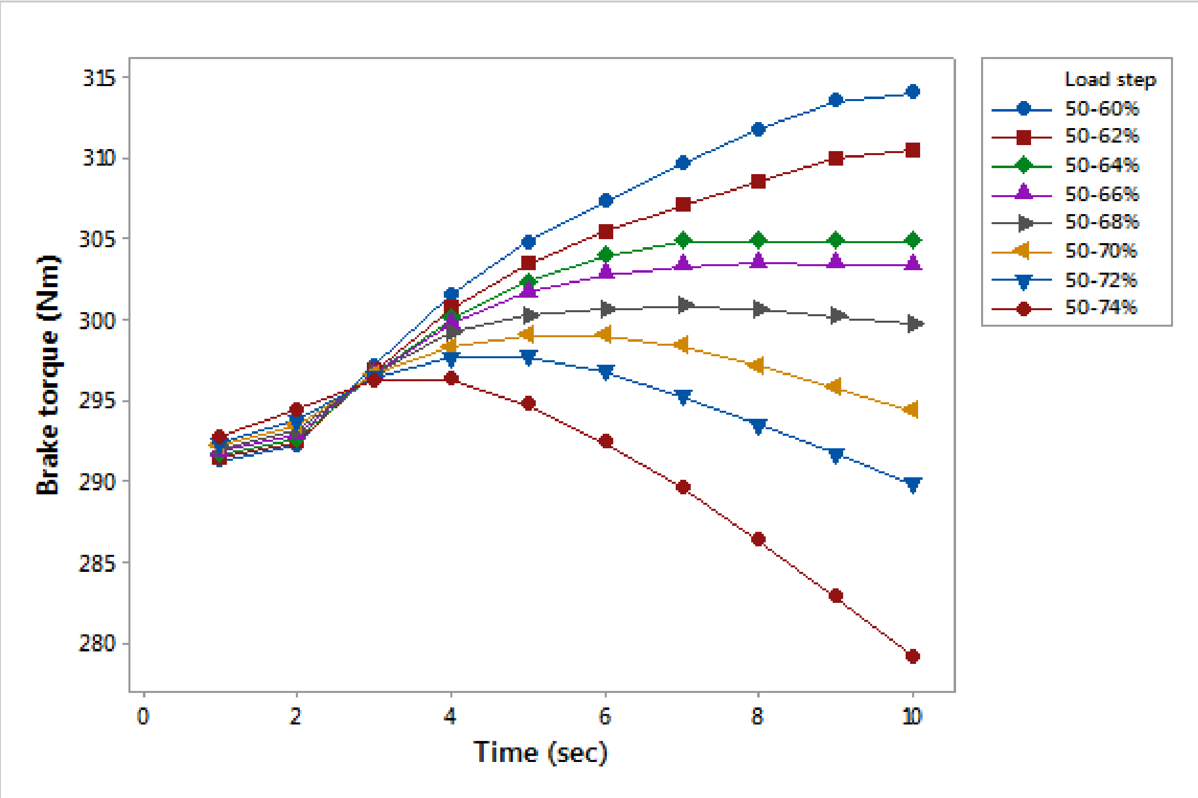

Figure 3.10: Variation of brake torque with different load steps in 2 sec, initial load 50%

However, to observe the effect of load schedule, that load step is chosen that is common to both 1 sec and 2 sec and which brings maximum turbo lag reduction in 1 sec. So, the effect of air injection under that load step in 2 sec will be investigated in chapter 5 and the results will be compared with that of 1 sec.

3.6 Measurement uncertainty

The measured parameters generally show some dispersion from the mean values during engine testing. This is defined in terms of measurement error which shows the variation between measurements of same quantity on the same test. To obtain measurement error the same test needs to be repeated many times [100]. According to [101], precision index can be written as

$$S = \sqrt{\frac{\sum_{i=1}^{N}(X_{\mathrm{i}} - \bar{X})^2}{N-1}} \tag{3.19}$$

To report the uncertainty for the average value of N measurements, standard error is used which can be written as

$$\sigma_{\mathrm{i}} = \frac{S}{\sqrt{N}} \tag{3.20}$$

where S is the precision index as defined in equation 3.19 and σ_{i} is the standard error. The measurement repeated N times can then be typically written as [102]

$$X = \bar{X} \pm \sigma_{\mathrm{i}} \tag{3.21}$$

where $\bar{X}$ is the mean of the dataset and σ_{i} is the standard error as defined in equation 3.20.

For the estimation of measurement uncertainty, the tests were repeated 5 times and the mean values were calculated for analysis. Brake torque, brake power and brake specific fuel consumption were the parameters that were investigated for accuracy. The standard error in this study is represented by ± symbol as shown in Table 5.1, chapter 5.

3.7 Summary of the chapter

The focus of the chapter was on the simulation software used to meet the overall objectives of the study. Simulation models were developed and engine test bed was described for validation studies of the simulation model which will be covered in Chapter 5. Test procedures for implementing the various technologies of improving the transient response of the turbocharged diesel engine have been developed. Simulation strategies and various transient operating conditions were discussed in detail under the test procedures. Control systems were constructed for modelling the techniques.

4

ELECTRIC TORQUE ASSISTANCE AND REDUCTION IN INERTIA OF COMPRESSOR WHEEL

Torque assistance method and reduction of inertia of compressor wheel are highlighted in the literature review as turbocharger specific measures to improve quality of transient response of turbocharged diesel engine. To meet objectives 1 and 2, these two methods are simulated in the present chapter and parametric study has been undertaken to analyze the quality indicators such as outlet pressure of the compressor and the compressor speed. The turbo lag is quantified to obtain the close to optimal transient response of turbocharged diesel engine. The Lotus Engine Simulation (LES) software is utilised which can predict full and part load performance under steady state and transient operating conditions.

The block diagram for the methodology is shown in Figure 4.1. It highlights the steps involved in the construction of simulation model as well as the control system for the improvement of transient response of the turbocharged diesel engine. The figure shows the system design as the initial step for simulation based on engine specification and operating parameters. A simulation strategy is then adopted to model the transient event. Turbo lag quantification is done for both torque assistance and inertia reduction method and based on turbo lag reduction, the transient response is reported.

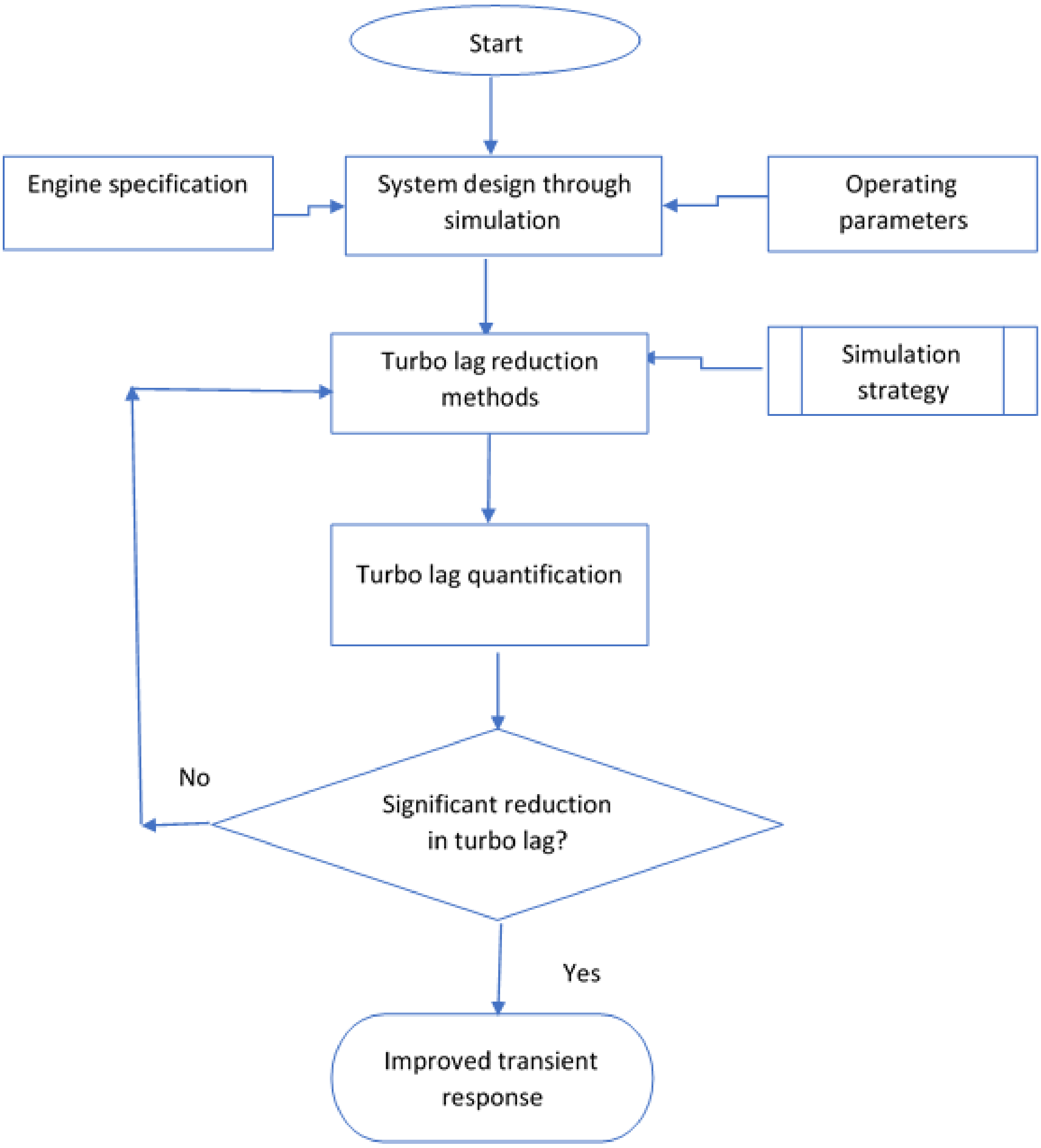

Figure 4.1: Block diagram for the proposed system

The length of the transient is taken as 6 seconds. The parameters for the quantification of turbo lag are listed in Table 4.1. To observe the output response of the turbocharger, compressor outlet pressure is taken as the major parameter for investigation under transient conditions. In addition, to analyse the turbocharger performance the effect of torque assistance is also investigated on turbocharger speed. To further analyze the transient period and to obtain quantitative estimates the rate of change of the parameters have been investigated. The quality factors listed in Table 4.1 are selected as transient response measure of the turbocharged engine.

Table 4.1: Quality factors for transient test

S.No	Quality factors
1	Compressor exit pressure
2	Rate of change of compressor exit pressure
3	Compressor speed
4	Rate of change of compressor speed

4.1 Results and discussion

To meet the objectives for torque assistance and inertia reduction method, the simulation results of the proposed system are presented here. The modeling of the real system is considered by choosing the parameters from the literature. The simulations are performed using LES software for engine performance parameters as well as various turbocharging parameters. Since LES is well utilised by world wide clients and validated thoroughly at LOTUS [39], the simulation model in the present study is validated against the "Multi-cylinder turbocharged model" of Lotus engine simulation.

4.1.1 Steady state simulations

To study the engine performance during steady state operation, the variation of the engine parameters are presented in Figure 4.2 - 4.4. These figures show the engine performance characteristics with variation in engine parameters as a function of engine speed. The plots show the variation in BSFC (brake specific fuel consumption), brake torque, brake power and BMEP (brake mean effective pressure) which are the important engine performance indicators.

Figure 4.2 shows the torque rises rapidly to a maximum of around 178 Nm at 3000 rpm and then gradually decreases with the increase in engine speed. The brake mean effective pressure increases up to 11 bar at 3000 rpm then gradually decreases with engine speed as shown in Figure 4.3 At lower speeds i.e. below 3000 rpm BMEP drops off due to heat losses. While at higher speeds (above 3000 rpm) BMEP decreases because it becomes difficult to ingest a full charge of air.

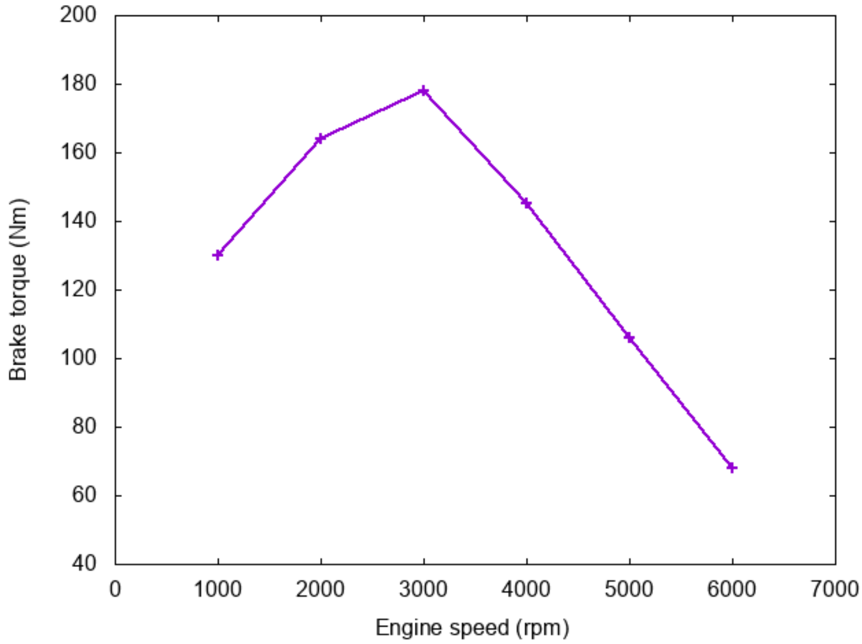

Figure 4.2: Brake torque

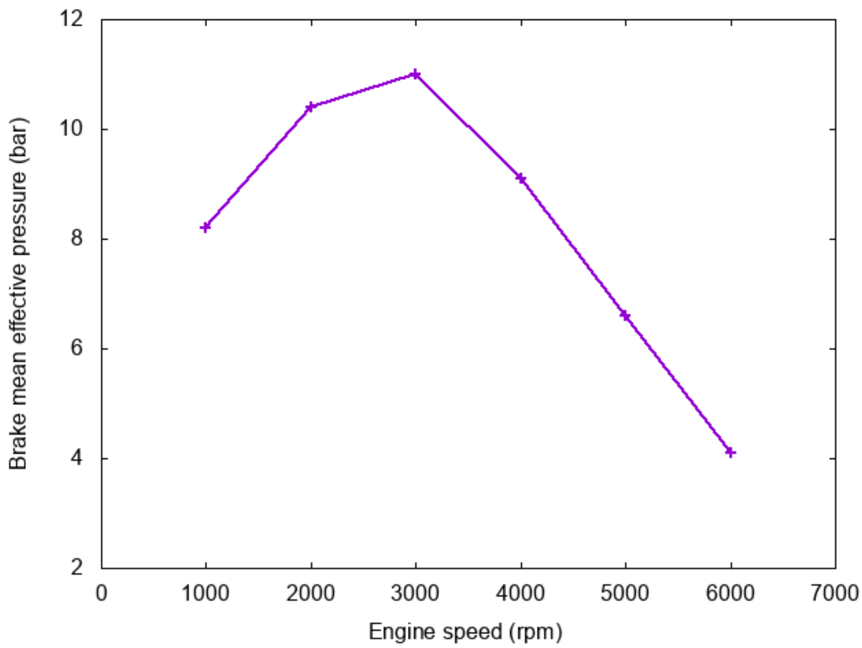

Figure 4.3: Brake mean effective pressure

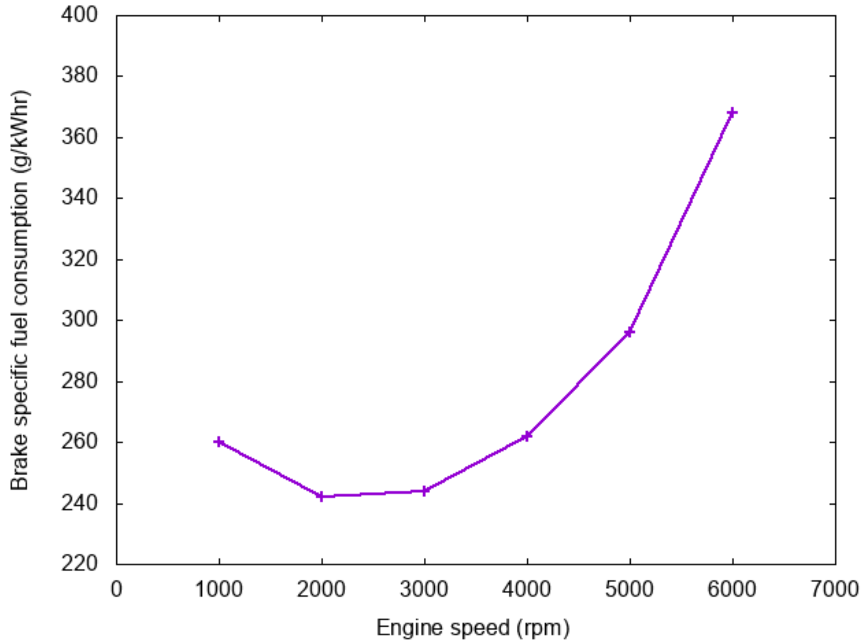

Figure 4.4: Brake specific fuel consumption

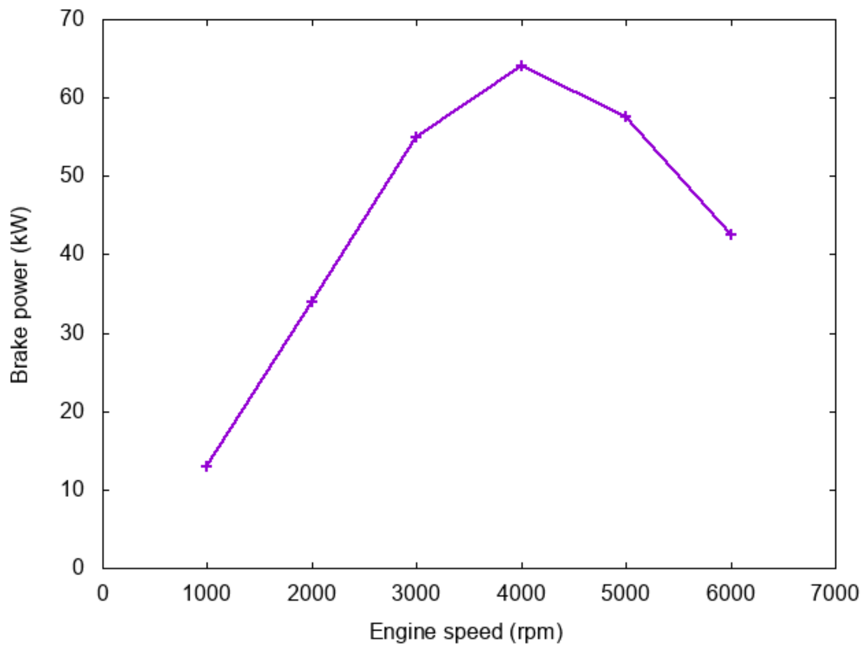

Figure 4.5: Brake power

BSFC reflects the efficiency of a combustion engine to produce power by burning the fuel. Figure 4.4 shows that the lowest value of specific fuel consumption is around 2000 rpm. At lower engine speed the fuel consumption increases due to increased time for heat losses from the gas to cylinder and piston wall. As the speed increases the fuel consumption also increases and hence the combustion efficiency decreases. The maximum power produced by the engine is about 64 kW at around 4000 rpm. Figure 4.5 shows the variation of Brake power with respect to engine speed. Initially it increases and then after 4000 rpm it starts decreasing.

As compared to steady state operation, intake air delay occurs during transient operation. Due to the deterioration of combustion boundary conditions during transient operations, the steady state operation findings cannot be applied directly and completely to transient operation. Nevertheless, transient operation studies should be based on steady state operation [103]. The steady state results presented in this section are comparable with the literature [39]. So, the information provided in this section related to steady state testing is useful. In the next section, emphasis is made on the transient operation as it is the event during which the engine performance deteriorates.

4.1.2 Transient simulation

In this section the results are presented for the effect of torque assistance and inertia reduction method under the event of fueling rate change as discussed in Chapter 3. Different values of torque assist are taken in the range of 0-0.32 Nm and investigations are performed to obtain the optimum value of torque assist. The value of the additional torque is motivated by the studies performed by Ibaraki et al. [61]. For this purpose, four different values are taken in the selected range as 0.08 Nm, 0.10 Nm, 0.16 Nm and 0.32 Nm. The response of compressor exit pressure for the different torque assist values is shown in Figure 4.6 The figure shows that initially by increasing torque assistance, the response is improved i.e.0.10 Nm torque assist offers more benefits than 0.08 Nm assist but further increase in torque to a value of 0.16 Nm gives almost similar but more stable response. Further increase in torque to a value of 0.32 Nm also gives almost same response. So the optimum value of torque assist is observed to be 0.16 Nm. This is consistent with the value reported in the literature [61].

Based on the above investigations 0.16 Nm torque is provided as an additional torque to the turbocharger shaft in the Lotus engine simulation environment for further evaluation. The Simulation is performed with and without electric torque assistance. The effect of torque assistance is noticeable in figure 4.7 Initially the compressor outlet pressure shows an initial delay in response before reaching the steady state. As shown in

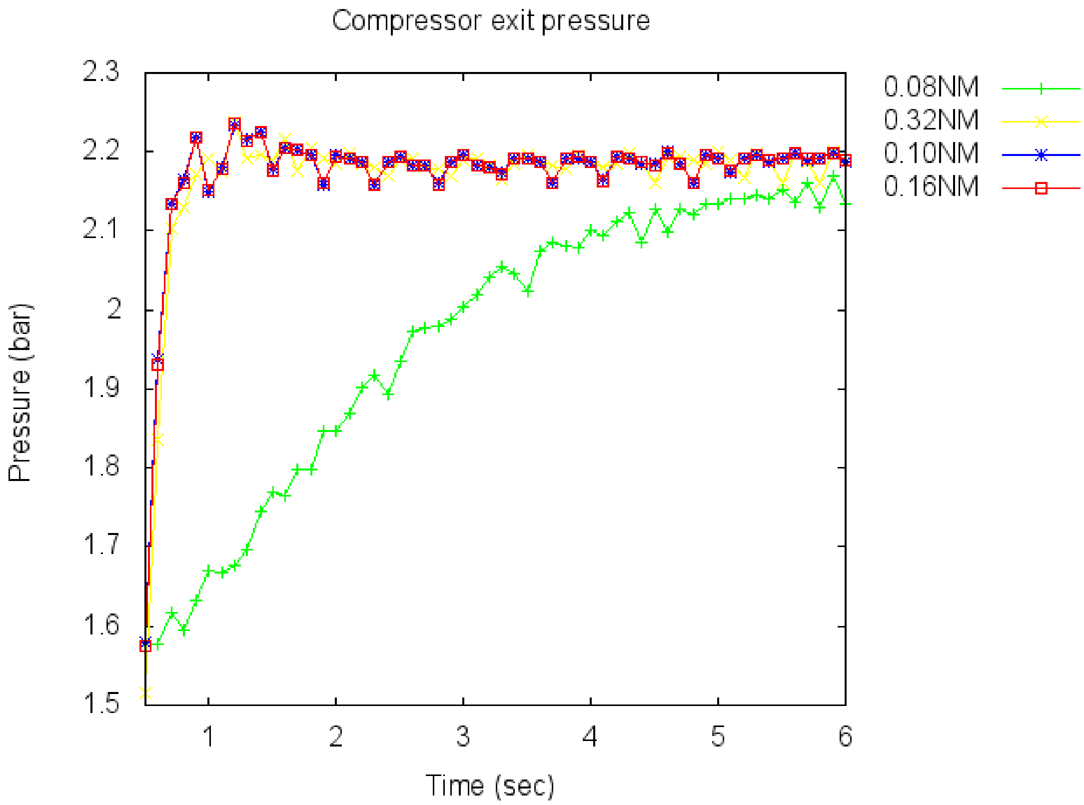

Figure 4.6: Compressor exit pressure response for different value of torque assist

figure 4.7 that the compressor pressure reaches the steady state condition in 4.8 seconds with no torque assistance. The compressor outlet pressure response is significantly improved with torque assistance and the compressor pressure reaches the steady state in 1.8 second. The rate of change of compressor exit pressure response is plotted in figure 4.8. It shows much greater and faster rate of change for torque assist system as compared to without torque assist systems. The response time is reduced by 3 seconds with torque assist.

Figure 4.9 shows the effect of torque assistance on compressor speed response. When compared with no assist systems, it is clear that with torque assist the response time is improved by 3 seconds. This difference is more significant when rate of change of compressor speed response is plotted as illustrated in figure 4.10. In case of torque assistance, greater and quicker response is observed and the difference in time to reach the equilibrium is found to be 3 seconds.

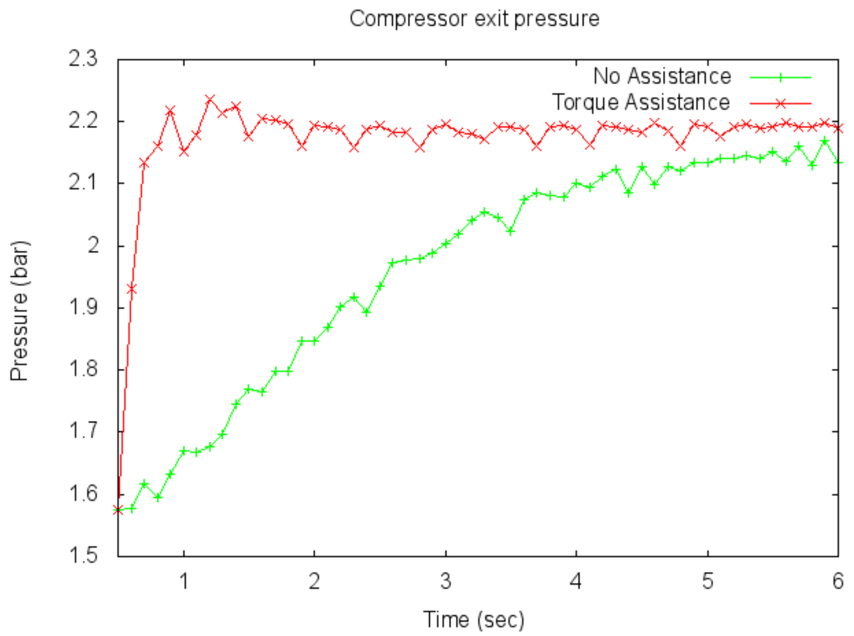

Figure 4.7: Compressor exit pressure with and without torque assist

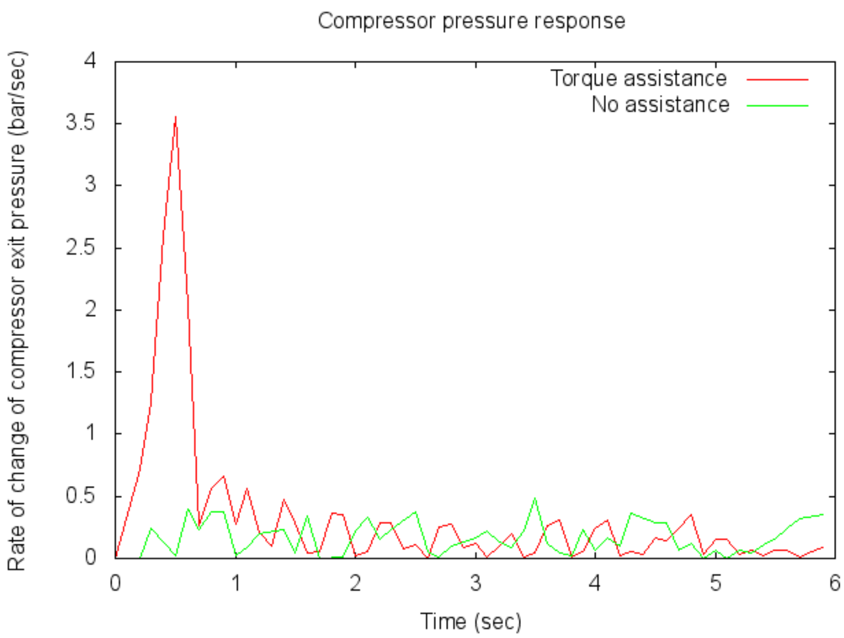

Figure 4.8: Rate of change of Compressor outlet pressure response

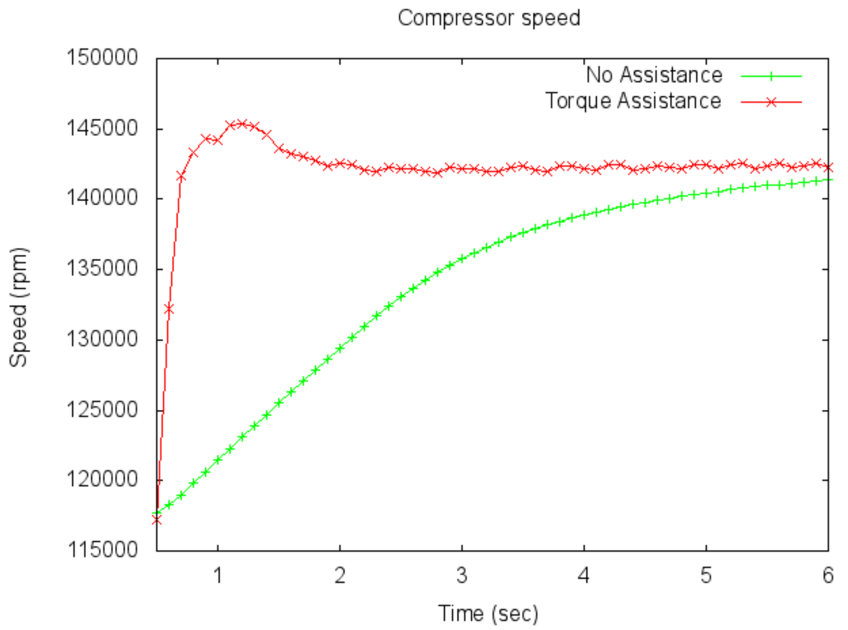

Figure 4.9: Compressor speed response with torque assist and without torque assist

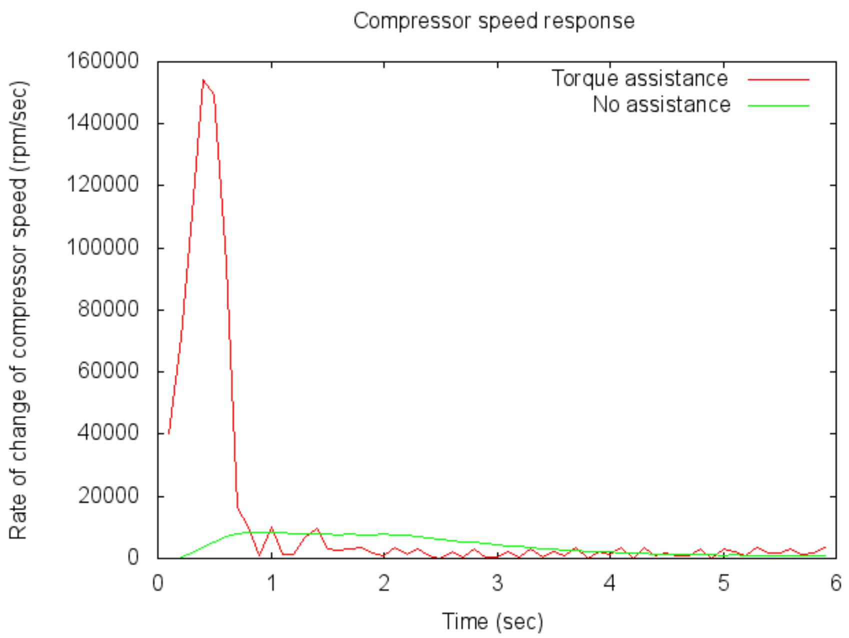

Figure 4.10: Rate of change of compressor speed response

Table 4.2: Comparison of the results obtained by torque assist method. TL: Turbo lag

Indicator	Torque Assist			
	No Assist		Torque Assist	
	TL	Indicated Value	TL	Indicated Value
CEP	4.8	2.13	1.8	2.2
RCEP	3.5	0.39	0.5	3.5
CS	5.6	141350	1.8	145354
RCS	1.0	8310	0.5	154200

The quantitative analysis of torque assist method is presented in Table 4.2. The quality indicators are compared for the case of torque assist and no torque assist. The analysis shows that the turbo lag is significantly reduced by the application of torque assistance. The compressor exit pressure (CEP) reaches the steady state condition in 4.8 seconds with no torque assistance. The compressor outlet pressure response is significantly improved with torque assistance and the compressor pressure reaches the steady state in 1.8 second. The rate of change of compressor exit pressure (RCEP) response shows much greater and faster rate of change for torque assist system as compared to without torque assist systems. The response time is reduced by 3 seconds with torque assist. For compressor speed (CS) the response time is 5.6 seconds without torque assist while with the torque assist the response time is 1.8 seconds. So the response time is improved by 3.8 seconds. Rate of change of compressor speed shows faster response with torque assist system. Moreover the results also show improvement in the indicated value for all the indicators with torque assist.

As discussed in chapter 1 and 2, the transient response can also be improved by reducing the inertia of the compressor. For inertia reduction, the values under consideration are shown in Table 4.3. It shows the reduction in compressor inertia whereas the initial value is 6e-6 kgm^2 and the reductions under consideration are -10%, -25% and -40%.

Table 4.3: Reduction in compressor inertia

Original	6.0e-6 kgm^2
-10%	5.4e-6 kgm^2
-25%	4.5e-6 kgm^2
-40%	2.4e-6 kgm^2

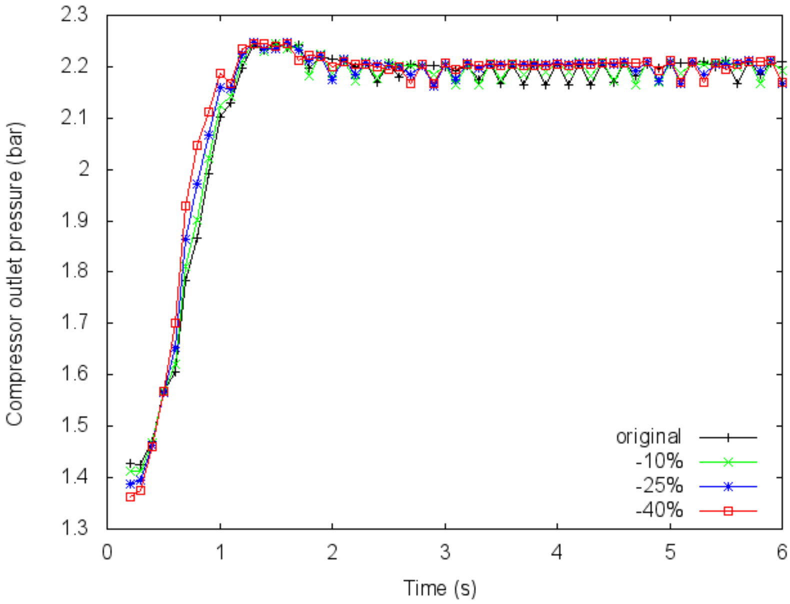

Figure 4.11: Compressor exit pressure with reduction in inertia values

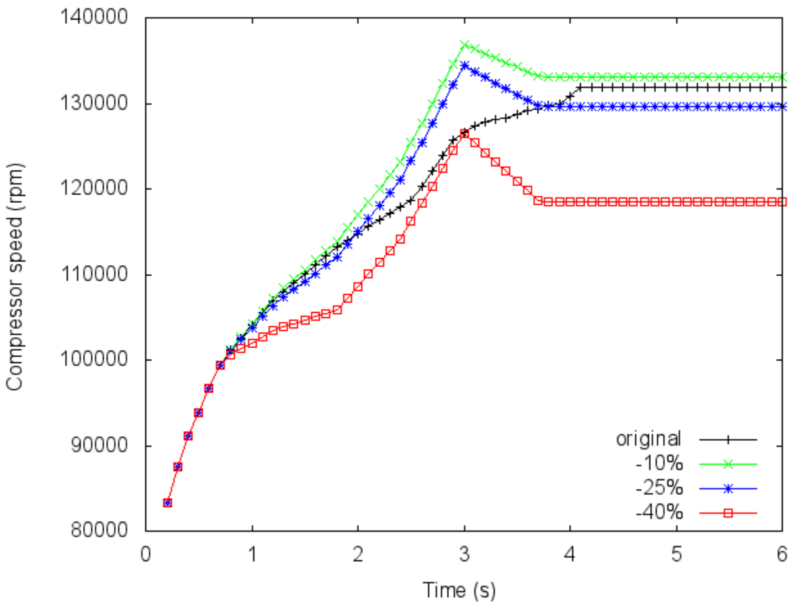

Figure 4.12: Compressor Speed with reduction in inertia values

Figure 4.11 shows the outlet pressure response of a turbocharger with reduction in inertia by 10%, 25% and 40%. The response of the turbocharger with inertia reduction is compared with that of original inertia. It is depicted from the figure that although similar response is observed for different inertia reduction, the delay in the response of the compressor exit pressure is reduced by inertia reduction. However the reduction of delay is seen to be very small. This is in accordance with the previous studies [63]. The analysis will be more clear when turbo lag reduction will be quantified in the coming paragraphs.

To further analyze the effect of reducing the inertia on the transient response, compressor speed is plotted in Figure 4.12 with different values of inertia as described above. From the figure, it can be depicted that reducing the turbocharger inertia by 10% increases the compressor speed which is in accordance with equation 1.1 discussed in chapter 1. However, increasing the magnitude of inertia reduction doesn't bring improvement in compressor speed, both in terms of turbo lag and indicated value.

Table 4.4: Comparison of the results obtained by inertia reduction method. TL: Turbo lag

Inertia Reduction	CEP		CS	
	TL	Indicated Value	TL	Indicated Value
Original	1.20	2.13	4	131901
-10%	1.10	2.14	3.8	133090
-25%	1.10	2.15	3.8	129576
-40%	1.10	2.15	3.8	118482

Table 4.4 shows the quantification of turbo lag for compressor exit pressure and compressor speed for the reduction in inertia of the compressor wheel. From the table it is clear that the improvement brought by this method is seen to be very small as the turbo lag is reduced by only fraction of seconds (0.2 sec) for both compressor exit pressure (CEP) and compressor speed (CS). It is worth noting that there is no benefit of decreasing the inertia below -10%.

4.2 Summary on electric torque assist and reduction in inertia of compressor wheel

1. Turbo lag decreases for compressor exit pressure with torque assistance of 0.16 Nm and the response time is reduced by 3 seconds.

2. Turbo lag decreases for compressor speed with torque assistance of 0.16 Nm and

the response time is improved by 3.8 seconds.

3. Rate of change of compressor exit pressure as well as rate of change of compressor speed shows much greater and faster response for torque assist system as compared to without torque assist systems.

4. Improvement in transient response by reduction in inertia is marginal and -10% reduction is the optimum value.

Here we achieved objective 1 and 2 by simulating passive methods of improving the transient response using Lotus engine simulation software. Further studies will be performed in the next chapter to investigate the results of active method of air injection for the improvement of the transient response of turbocharged diesel engine.

5

Air injection strategies

As per the literature review performed in chapter 2, three methods are highlighted that improve the transient response of turbocharged diesel engine. In the previous chapter, inertia reduction and torque assist method were found to improve the transient response of the engine. The findings of literature review indicate that air injection into the intake manifold is a highly promising measure to mitigate the drawback of poor transient performance of turbocharged diesel engine. Investigations are carried out in this chapter to show the effect of air injection at intake manifold on the transient response of the engine using modelling. Ricardo wave engine model is designed with the engine variable and operating parameters.

Figure 5.1 shows the pictorial representation of the different strategies used in the present study to simulate the transient response of the diesel engine. To find the optimum technology, the results of the three methods are compared by choosing common parameters and same simulation strategy. The effect of selected technology is then simulated for acceleration and load tests.

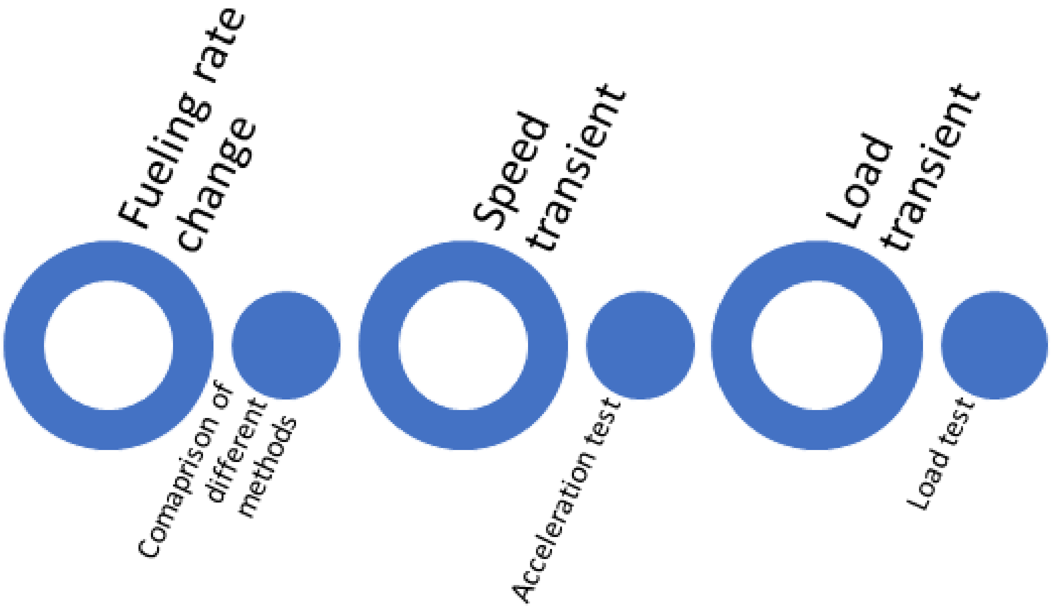

Figure 5.1: Strategies to model the transient response of diesel engine

5.1 Detailed investigation on air injection systems

The simulation results of the proposed system are presented here. Real engine is simulated using Ricardo wave for steady state and transient conditions. Based on the experimental data for steady state, the Ricardo wave model is validated for engine performance parameters such as brake torque, brake specific fuel consumption and brake power. The wave model is then modified to include air injection system at intake manifold. The control system is formulated using the co-simulation of Ricardo wave with Simulink [104].

5.1.1 Validation with real engine

Steady state testing is a useful tool for cross checking the available measurements in addition to the verification of experimental setup. The test results for the steady state are collected from the real CI engine and these results are used to validate the Ricardo wave model. It is assumed that the 1D gas dynamic code used in Ricardo wave has sufficient prediction capabilities to yield the reliable results. The nominal working range of the engine is 1000 - 2200 rpm. Engine performance parameters are chosen for validation of Ricardo wave model with the real data. These are:

1. Brake Torque (BT)
2. Brake specific fuel consumption (BSFC)
3. Brake power (BP)

The validation of different engine performance parameters as a function of engine speed is shown in Figure 5.2, 5.3 and 5.4. Figure 5.2 depicts that initially torque rises at low engine speeds and attains the maximum value of 476 Nm at 1200 rpm and then gradually decreases with the increase in engine speed. It can be clearly observed from the Figure that the predicted values of brake torque by Ricardo wave model are in good agreement with the experimental data.

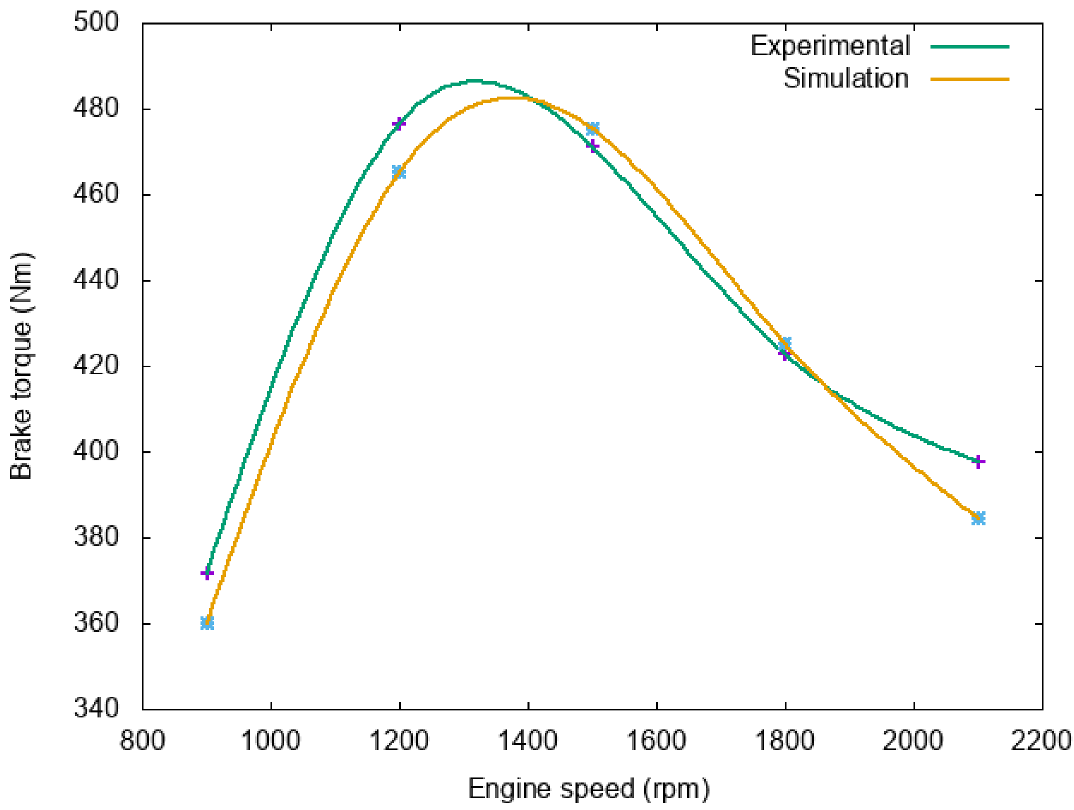

Figure 5.2: Validation of simulated values of brake torque with experimental data

Calculated values of BSFC were also satisfactory as shown in Figure 5.3. The validation of Ricardo wave with experiment shows a good agreement between the simulation values and experimental data for BSFC. The increase in fuel consumption at lower engine speed is due to the increased time for heat losses from the gas to cylinder and piston wall.The lowest value of specific fuel consumption is 0.201 kg/kWhr at 1800 rpm. Engine speed plays an important role for the fuel consumption in the engine. As the speed increases, the fuel consumption also increases due to increase in friction power and hence the efficiency decreases.

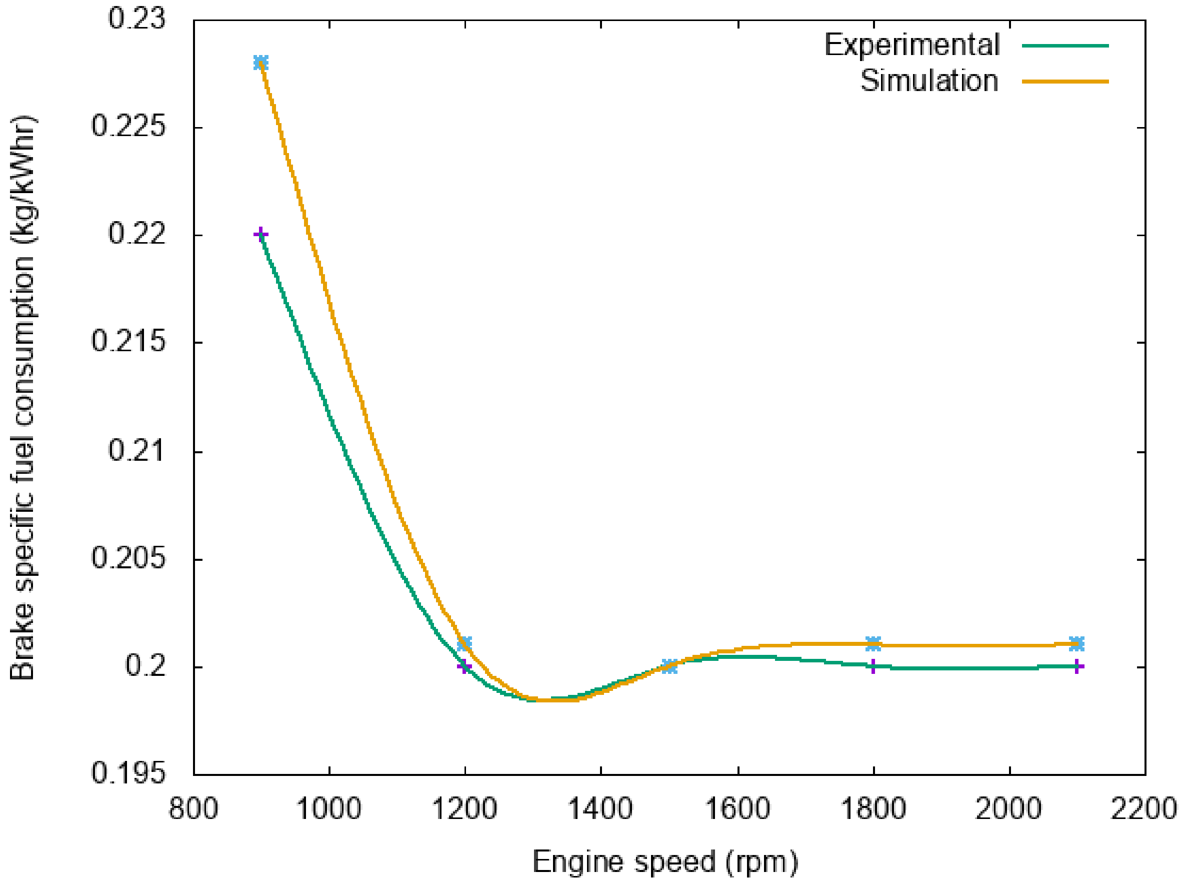

Figure 5.3: Validation of simulated values of brake specific fuel consumption with experimental data

Figure 5.4 shows the variation of brake power with respect to engine speed. Good agreement is obtained for brake power in the simulated and measured values. It is observed that the brake power increases with an increase in engine speed. The maximum power produced by the engine is about 79.58 kW at around 1800 rpm.

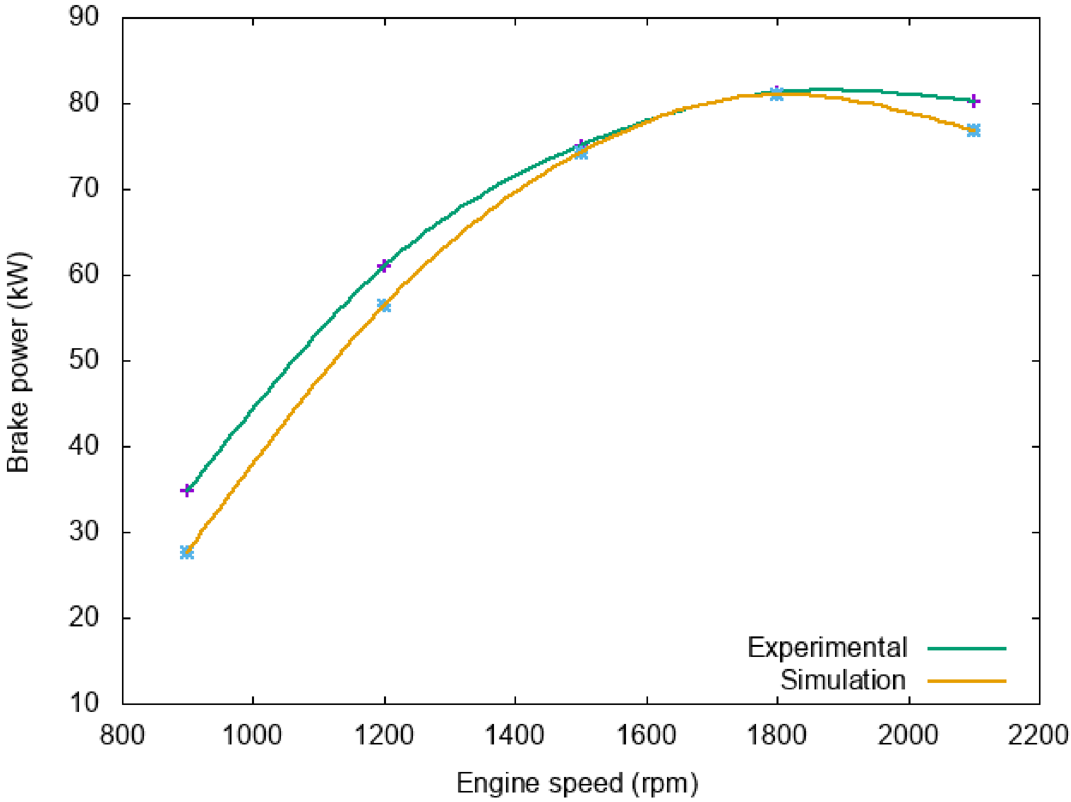

Figure 5.4: Validation of simulated values of brake power with experimental data

Table 5.1: Comparison of Ricardo wave results with experimental data for performance parameters

Engine speed (rpm)	Brake power (kW)		Brake torque (Nm)		BSFC (kg/kWhr)	
	Experiment	Ricardo	Experiment	Ricardo	Experiment	Ricardo
900	34.72 ± 0.22	27.6	371.75± 1.89	360.1	0.22 ± 0.004	0.228
1200	60.92 ± 0.74	56.4	476.45 ± 1.33	471.1	0.20 ± 0.002	0.201
1500	74.97 ± 0.30	74.2	471.05 ± 1.68	475.3	0.20 ± 0.0	0.200
1800	81.25 ± 0.29	81.0	422.70 ± 1.17	425.2	0.20 ± 0.0005	0.201
2100	80.20 ± 0.20	76.8	397.72 ± 1.03	394.49	0.20 ± 0.001	0.201

The quantitative measure of the agreement between the experiment and the simulation is presented in Table 5.1. It can be noted that Ricardo wave satisfactorily reproduces the experimental data for all the chosen performance parameters. The average difference between the experimental and the simulation data for brake power, brake torque and BSFC is 6%, 2% and 1% respectively. As discussed in section 3.2, the difference can be ascribed to the usage of default TCMAP in the simulation.

Henceforth, the calibrated wave model is expected to be able to predict the transient behavior of the chosen heavy duty compression ignition (CI) engine since the Ricardo wave has sufficient prediction capabilities to yield the reliable results for transient simulations.

5.2 Air injection system and its comparison with other methods of turbo lag reduction

Torque assist method and reduction in inertia method is investigated in sub section 4.1.2 to achieve objective 1 and 2 respectively. In this section, same simulation strategy is opted as described in section 3.5.1.1 to investigate the effect of air injection on the transient response of turbocharged diesel engine, covering objective 3. To know the effect of injection pressure, Figure 5.5 and Figure 5.6 are plotted. These figures show that the engine performance factors such as brake torque and brake power have noticed significant improvement with injection pressure up to 2.5 bar. Increasing the injection pressure after 2.5 bar doesn't bring significant improvement in these engine performance factors. So the injection pressure of 1 bar, 2 bar and 2.5 bar are chosen to perform an analysis of the effect of air injection on the response characteristics of the turbocharger.

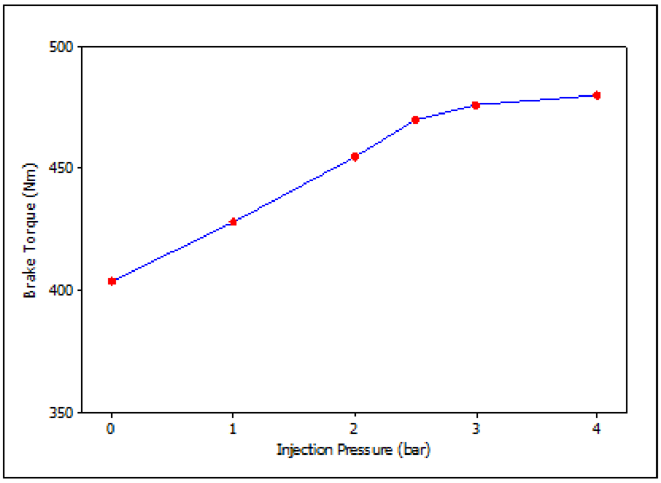

Figure 5.5: Variation of brake torque with injection pressure

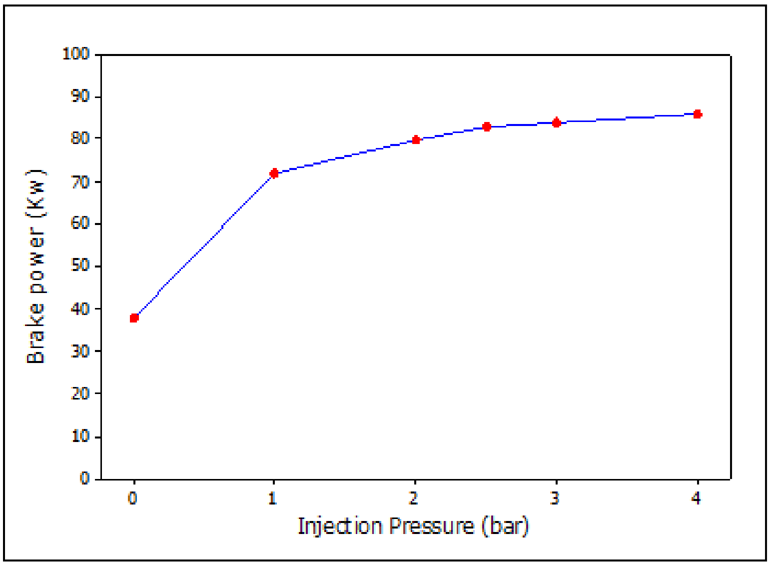

Figure 5.6: Variation of brake power with injection pressure

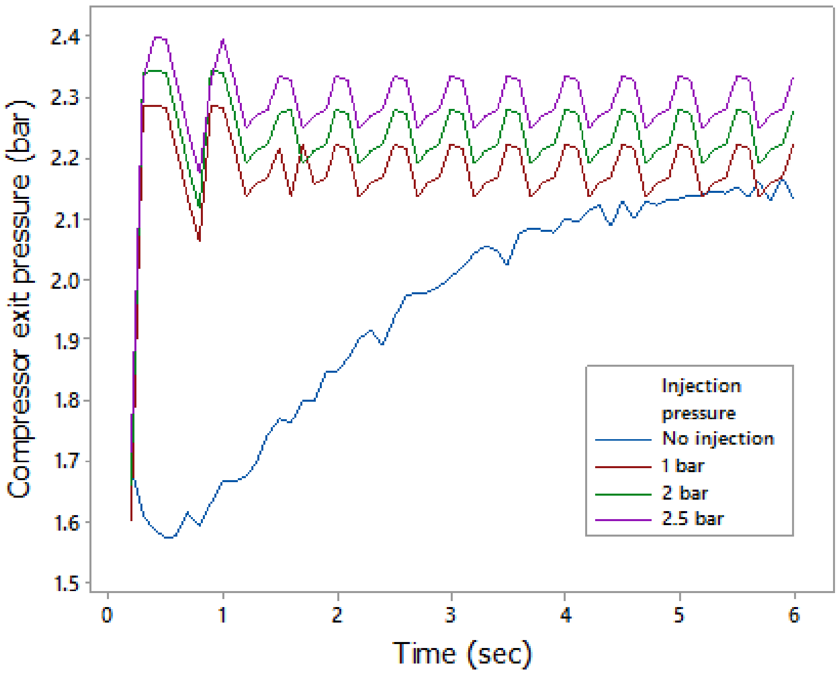

Figure 5.7: Effect of air injection pressures on compressor exit pressure

Air injection with 1 bar, 2 bar and 2.5 bar injection pressure is then simulated and the results are presented in Figure 5.7 and Figure 5.8. The results show the effect of air injection on compressor exit pressure (CEP) and compressor speed (C_Speed) respectively which are the same parameters as chosen in torque assist and inertia reduction methods. As shown in Figure 5.7 and Figure 5.8, with the application of air injection, turbo lag is reduced significantly for both the parameters for 1 bar, 2 bar and 2.5 bar when compared with no air injection. 2.5 bar air injection brings maximum improvement in the transient response of the turbocharger.

A comparison is now made for the assessment of relative improvement in transient response brought by the three methods. A value of 0.16 Nm is used for torque assist and inertia is reduced by 10% as per the conclusion from the investigations carried out in chapter 4. For air injection, 2.5 bar pressure is considered as it brings maximum improvement in transient response. It is clear from Figure 5.9 that when none of these methods is applied, compressor exit pressure and compressor speed rises rapidly but takes significant time to reach to the steady state, yielding greater turbo lag.

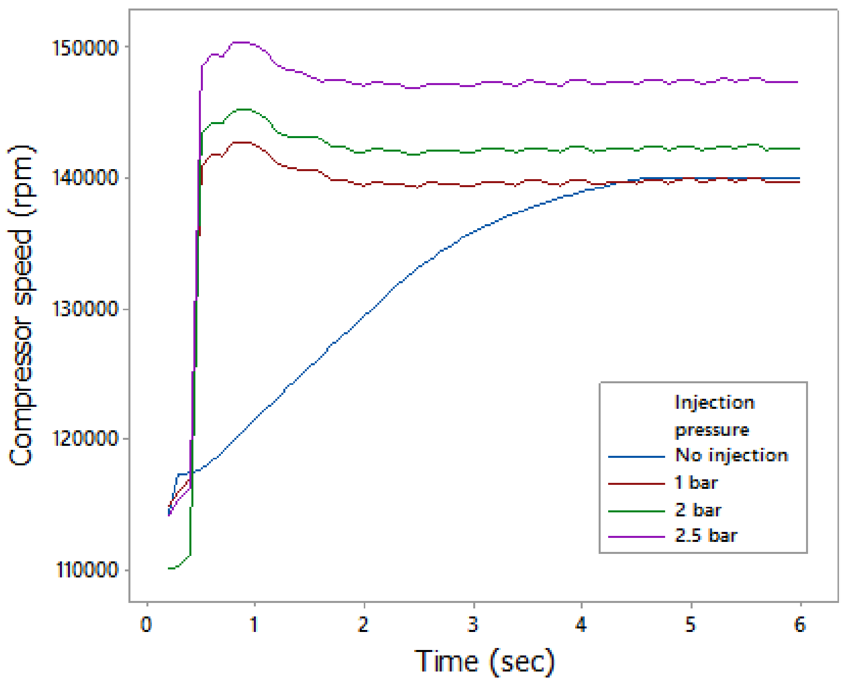

Figure 5.8: Effect of air injection pressures on compressor speed

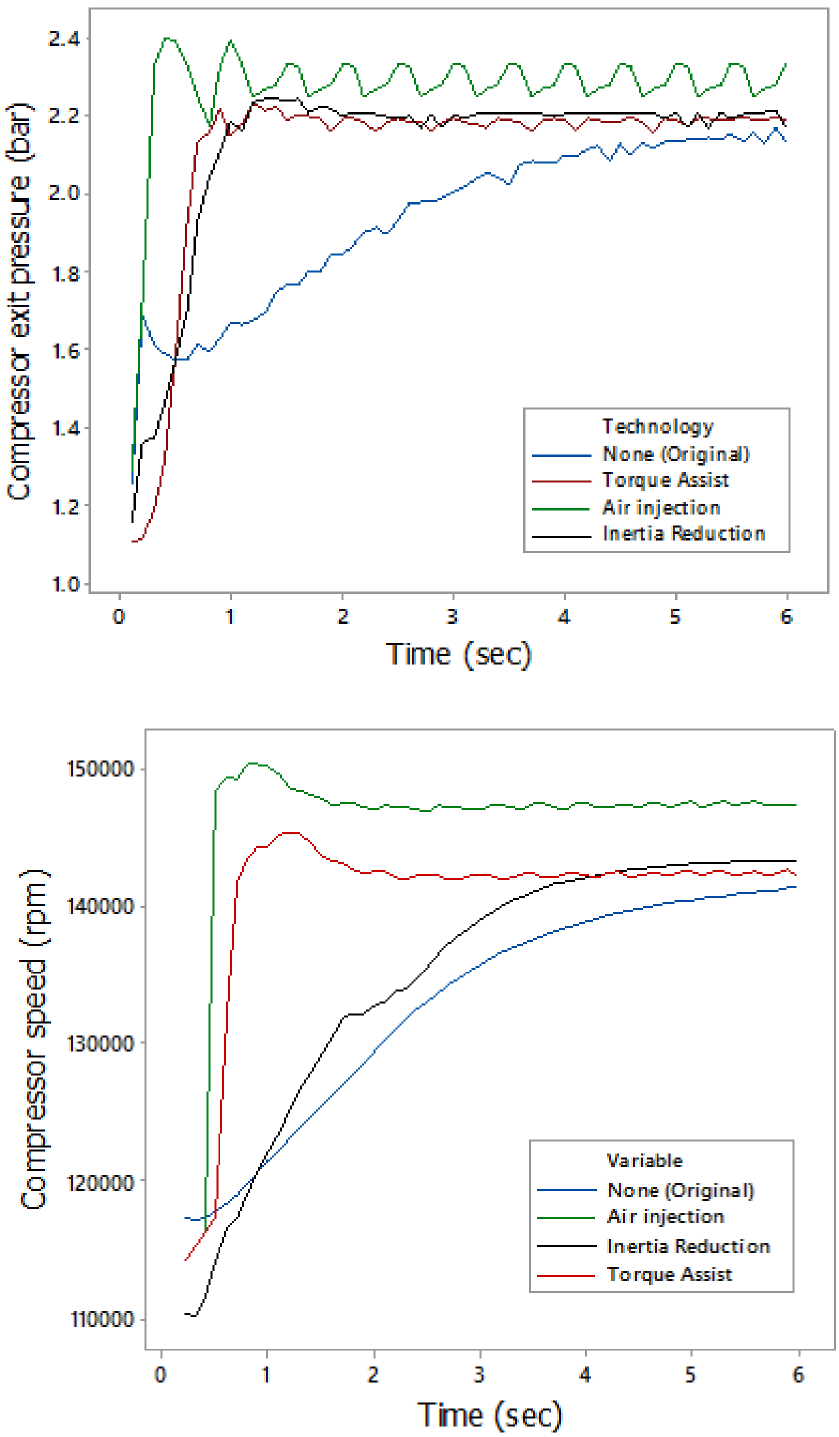

Figure 5.9: Comparison of compressor exit pressure response (top) and compressor speed response (bottom) with torque assist method, air injection method, reduction in inertia method and original (no method applied)

The maximum achievable improvement in the transient response by the three methods is evaluated based on compressor exit pressure and compressor speed. The resulting turbo lag for compressor exit pressure and compressor speed is quantified as presented in Table 5.2. With the methods applied for transient response improvement, turbo lag is significantly reduced. However, the air injection method shows to outperform the other two methods, yielding the minimum turbo lag. The next section includes the detailed investigations on the effect of air injection system with different transient performance factors.

Table 5.2: Comparison of turbo lag observed for compressor exit pressure (CEP) and compressor speed (C_Speed) response by different methods

Method	Turbo lag (sec)	
	CEP	C_Speed
Original	5.1	5.5
Inertia reduction	2.2	4.9
Torque assist	1.5	1.9
Air injection	1.2	1.6

5.3 Evaluation of air injection system under acceleration test

Acceleration tests represents the transient even at low engine speed and sudden acceleration. Simulations are performed for the acceleration tests to evaluate the effect of air injection systems on the transient response of the turbocharged diesel engine. Acceleration time and air injection pressure are varied under three tests as discussed in section 3.5.1.2 to observe the turbocharger response with the application of air injection at intake manifold. Air injection is simulated as discussed in section 3.3.2. These investigations will cover objective 5.

From Figure 5.2, it can be noted that the torque performance of the turbocharged diesel engine is poor at 1000 rpm engine speed. For these reasons simulations are performed at 1000 rpm engine speed with and without air injection during the transient conditions of rapid acceleration. During this time the engine speed is increased from 1000 rpm to 1700 rpm.

The time to acceleration is taken as 1 sec and 2 sec as they correspond to the case of rapid acceleration. For air injection, as discussed in section 5.2, 1 bar, 2 bar and

2.5 bar injection pressure are considered. To analyze the effect of varying the injection pressure, the simulation of air injection is performed to establish its effect first on the engine performance factors.

The major parameters investigated under acceleration tests are the following turbocharger response parameters:

- Compressor speed (CS) - The speed at which the compressor is capable for continuous operation is known as compressor speed.

- Compressor exit pressure (CEP) - The outward pressure available at the exit of the compressor is known as compressor exit pressure.

- Turbine inlet pressure (TIP) - The exhaust gas pressure available at the exit of the engine cylinder which impinges into the turbine blade is known as turbine inlet pressure.

- Turbine inlet temperature (TIT) - It is the temperature of the exhaust gases that enter the turbine unit.

These parameters indicate the output response of the turbocharger. So these are taken as a measure of transient response of the turbocharged engine.

5.3.1 Test 1

In the first test, the turbocharger response parameters are observed for the acceleration times of 1 sec and 2 sec for the speed change from 1000 rpm to 1700 rpm respectively. The effect of acceleration time on the response parameters of the turbocharger such as compressor exit pressure (CEP), turbine inlet pressure (TIP) and turbine inlet temperature (TIT) is depicted in Figure 5.10, 5.11 and 5.12 respectively. Similar trends are observed for turbo lag for all the chosen parameters of the turbocharger. For 1 sec acceleration time, the turbo lag values are found to be 6.4 sec, 6.8 sec and 8.0 sec for CEP, TIP and TIT respectively. On the other hand, the turbo lag values are 6.0 sec, 6.0 sec and 6.8 sec for CEP, TIP and TIT respectively for 2 sec acceleration time.

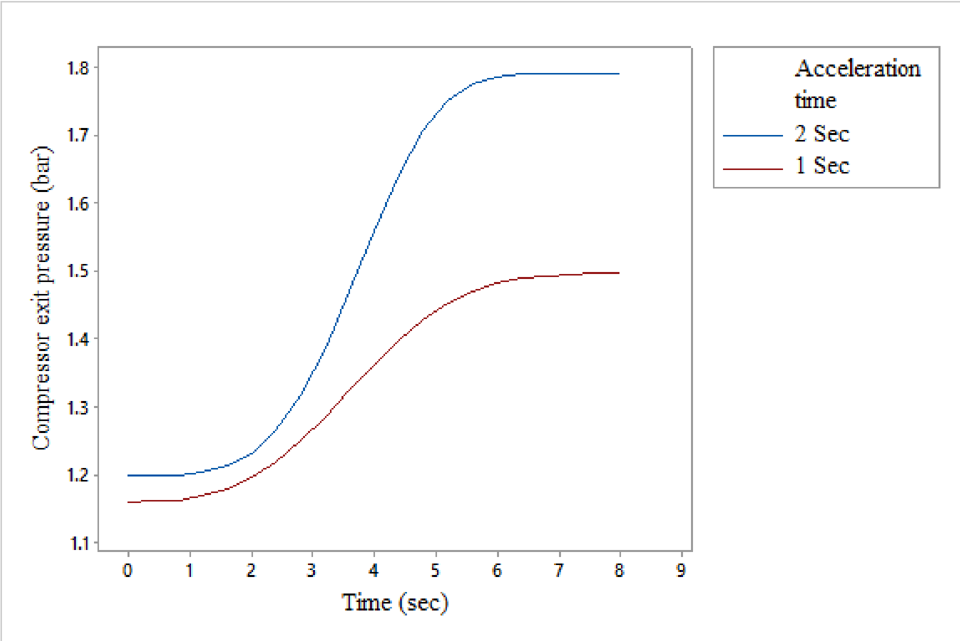

Figure 5.10: Effect of acceleration time on compressor exit pressure (CEP) response

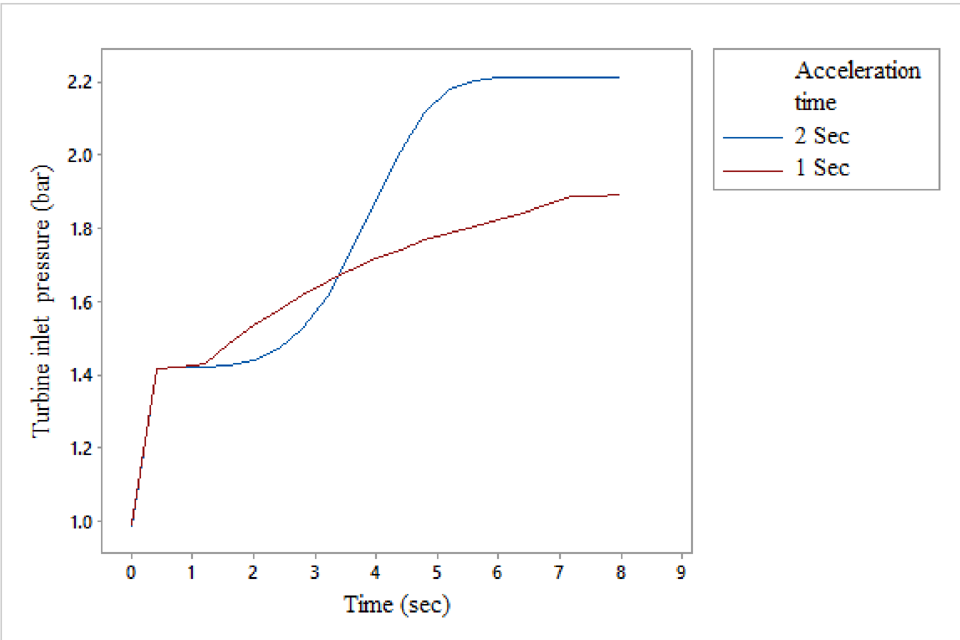

Figure 5.11: Effect of acceleration time on turbine inlet pressure (TIP) response

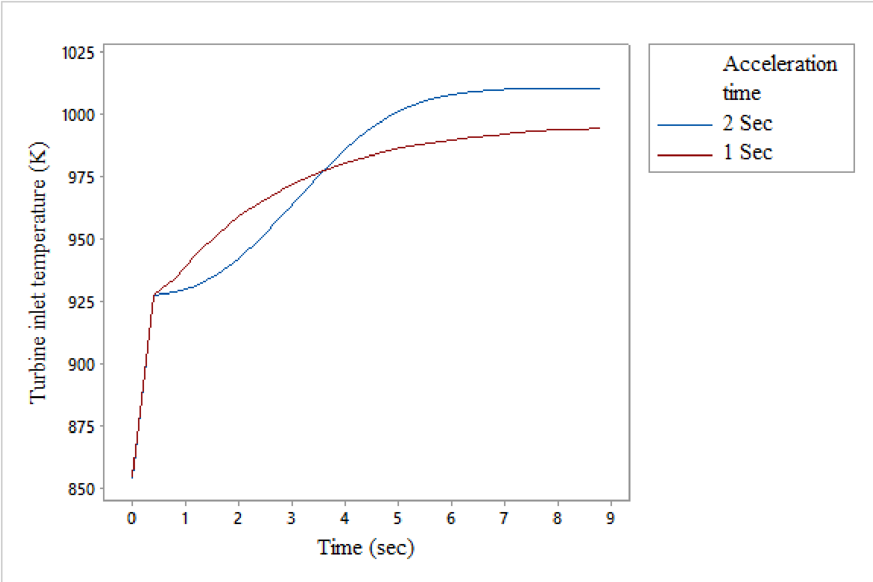

Figure 5.12: Effect of acceleration time on turbine inlet temperature (TIT) response

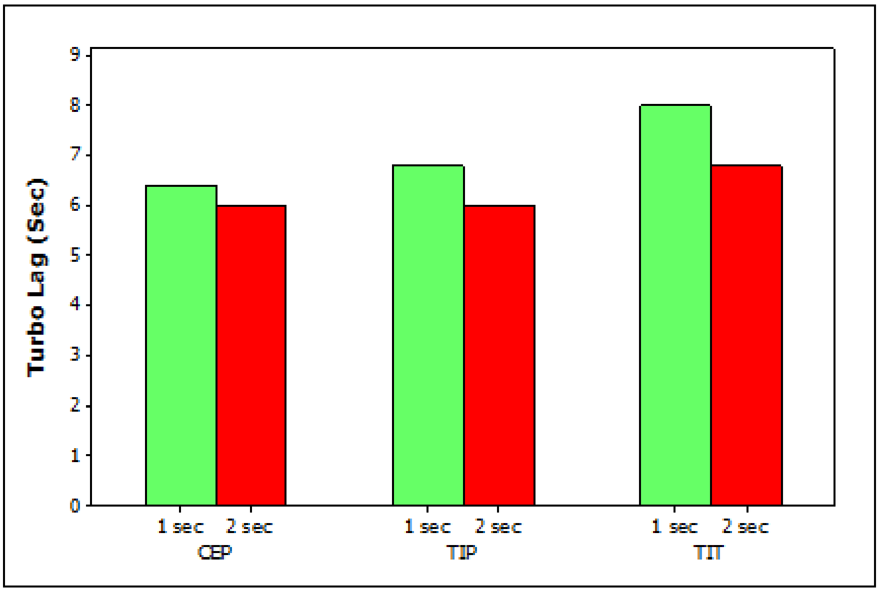

Figure 5.13: Variation of response parameters (CEP, TIP and TIT) with acceleration time

For comparative analysis, the values are presented in Figure 5.13. It is clear from the figure that a decrease in time duration for acceleration from 2 sec to 1 sec increases the turbo lag and hence 1 sec corresponds to more rapid acceleration. The finding is consistent with the fact that turbo lag arises due to delayed response of the turbocharger and the faster is the transient event, comparatively slower will be the response. Based on this finding, 1 sec time duration for acceleration will be considered as the time duration for rapid acceleration for test 2.

5.3.2 Test 2

As per the investigations carried out in Test1, it should be noted that 1 sec acceleration time corresponds to rapid acceleration for which the improvement in transient performance needs to be measured. So in this test: the time for acceleration is taken as 1 sec and the effect of air injection pressure is measured on turbocharger response parameters such as CEP, TIP and TIT. The cases of additional air injection are compared with that of no injection. In the present investigation, the injection pressures considered are 1 bar, 2 bar and 2.5 bar to observe the optimum improvement in transient response.

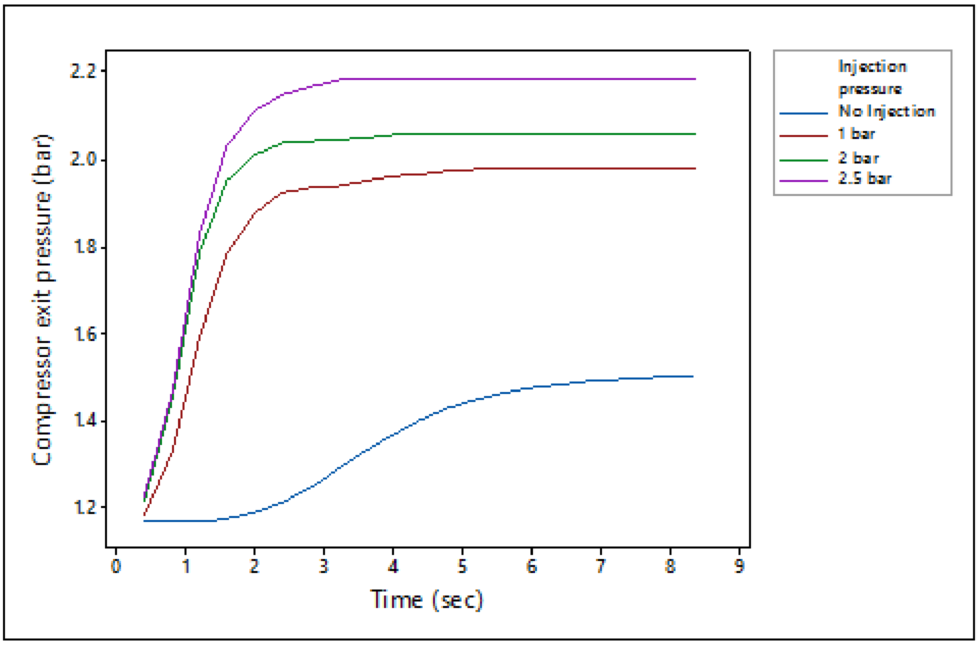

Figure 5.14: Effect of air injection pressures on compressor exit pressure response

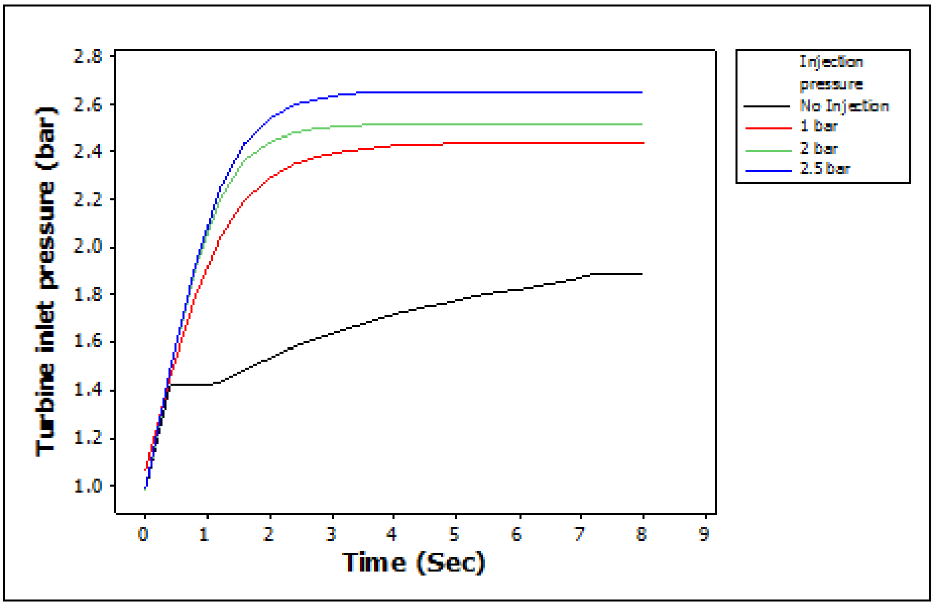

Figure 5.15: Effect of air injection pressures on turbine inlet pressure response

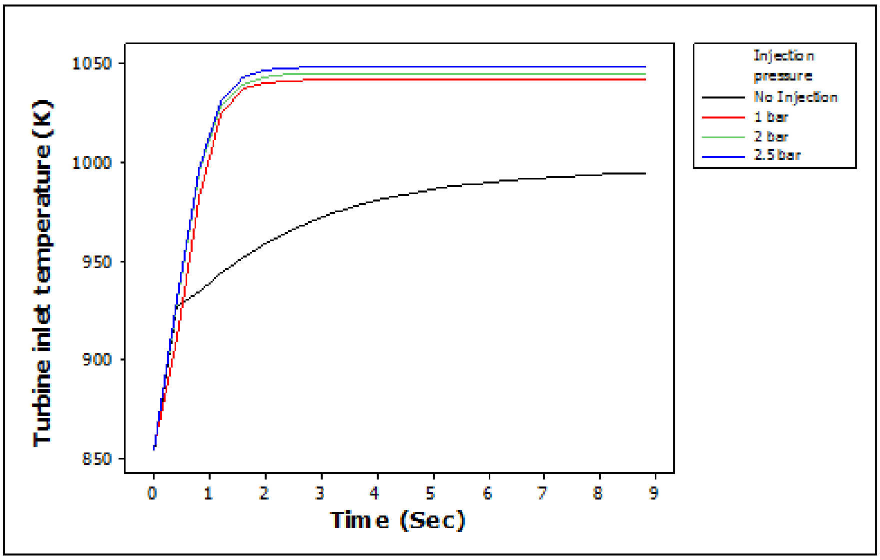

Figure 5.16: Effect of air injection pressures on turbine inlet temperature response

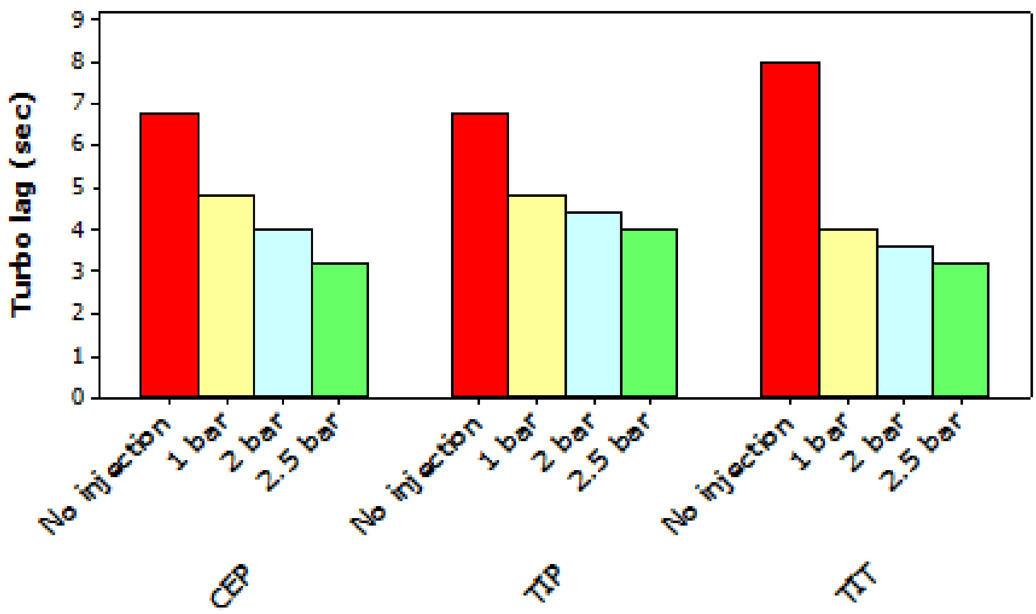

Figure 5.17: Variation of response parameters with air injection pressures for turbo lag

When additional air is injected at intake manifold, more air is supplied to the cylinder that optimizes the combustion processes. The consequence is the increase in exhaust energy which in turn increases turbine inlet pressure. This causes the turbocharger to rotate faster yielding improved turbocharger response [48]. The effect of air injection is noticeable as shown in Figure 5.14, 5.15 and 5.16. Initially the parameters show an initial delay in response before reaching the steady state. The recovery time with air injection is significantly reduced as compared to that of no air injection for all the parameters. Turbo lag decreases and the final steady state values are also improved with the increase in air injection pressure. The comparative reduction in turbo lag with varied air injection pressure for each response parameter is shown in Figure 5.17.

Although each of the selected injection pressure is sufficient to improve the transient performance as shown in Figure 5.14, 5.15 and 5.16, however, it is important to find the optimum value of the injection pressure. So in addition to the performance improvement evaluation with air injection, it is worth calculating that how much energy is imparted with the air injection. Turbo lag reduction with minimum energy required by air injection will be an additional source of information for selecting the optimum injection pressure. The energy supplied per sec (E) through air injection can be calculated as:

$$E = \frac{P_1 \dot{m}}{\rho} \tag{5.1}$$

where P_1 is the pressure before the orifice, $\dot{m}$ is the mass flow rate of the air passing through the orifice and ρ is the air density. Mass flow rate of the air passing through the orifice can be calculated from equation 3.16, described in chapter 3.

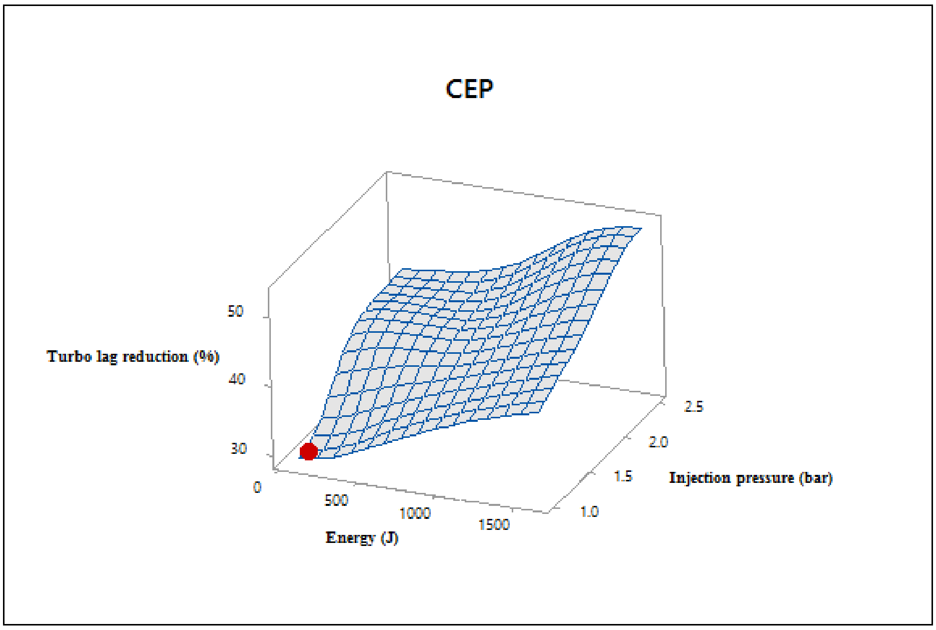

Figure 5.18: Dependency of turbo lag reduction on air supply parameters for compressor exit pressure (CEP)

Figure 5.18, 5.19 and 5.20 show the reliance of turbo lag reduction on air assist pressure and in turn the energy required for each of the selected performance parameter. Higher the injection pressure, more the energy required to significantly reduce the turbo lag. For comparative analysis, the turbo lag reduction per unit energy corresponding to the injection pressure is presented in Table 5.3. It can be concluded from the table that the maximum improvement in percent turbo lag reduction/energy is brought by air injection corresponding to 1 bar pressure. Hence, 1 bar air injection is the optimum injection pressure with percent turbo lag reduction per unit energy as 0.290 per joule for compressor exit pressure, 0.392 per joule turbine inlet pressure and 0.555 per joule for turbine inlet temperature (TIT). The optimum points are highlighted with red colour in the figures.

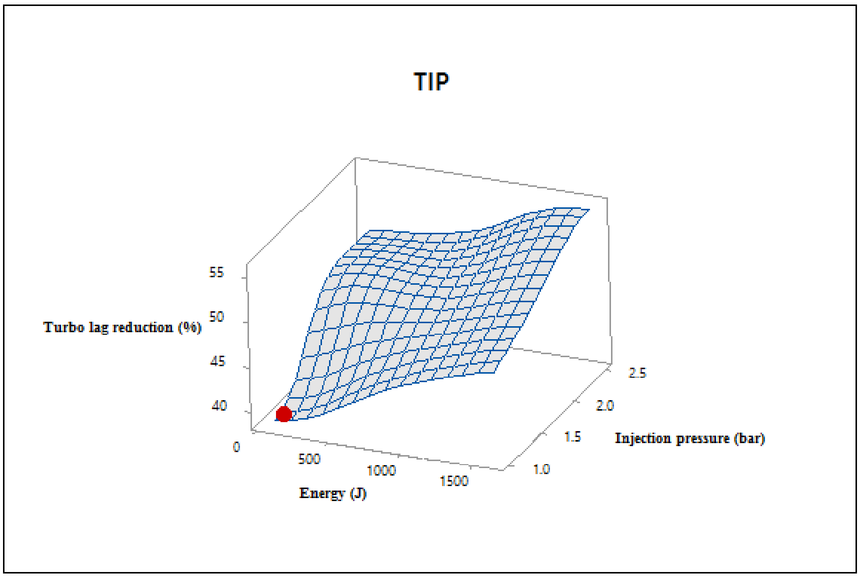

Figure 5.19: Dependency of turbo lag reduction on air supply parameters for turbine inlet pressure (TIP)

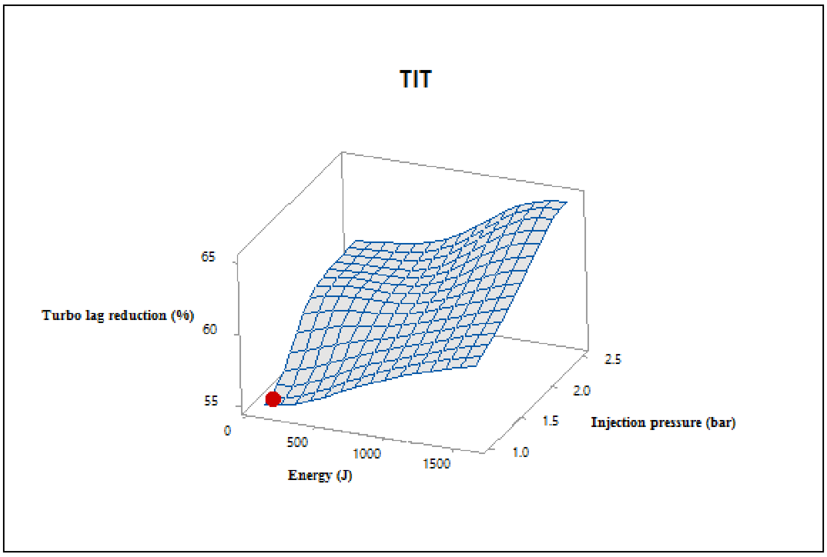

Figure 5.20: Dependency of turbo lag reduction on air supply parameters for turbine inlet temperature (TIT)

Table 5.3: Turbo lag reduction per unit energy corresponding to air injection pressure for turbocharger response parameters (CEP, TIP and TIT)

Injection Pressure (bar)	Turbo lag reduction per unit energy		
	CEP	TIP	TIT
1	0.290	0.392	0.555
2	0.102	0.122	0.146
2.5	0.032	0.034	0.040

5.3.3 Test 3

In this test 1 bar is taken as the optimum injection pressure as found in test 2. The response characteristics are observed for acceleration time of 1 sec and 2 sec respectively for the speed change from 1000 rpm to 1700 rpm as shown in Figure 5.21. As per the comparative analysis illustrated in the figure, the turbo lag quantification for the air injection of 1 bar highlights a remarkable reduction in turbo lag for 1 sec transient as compared to that of 2 sec. This difference is more perceptible when the rate of change of response parameters is plotted as shown in Figure 5.22, 5.23 and 5.24.

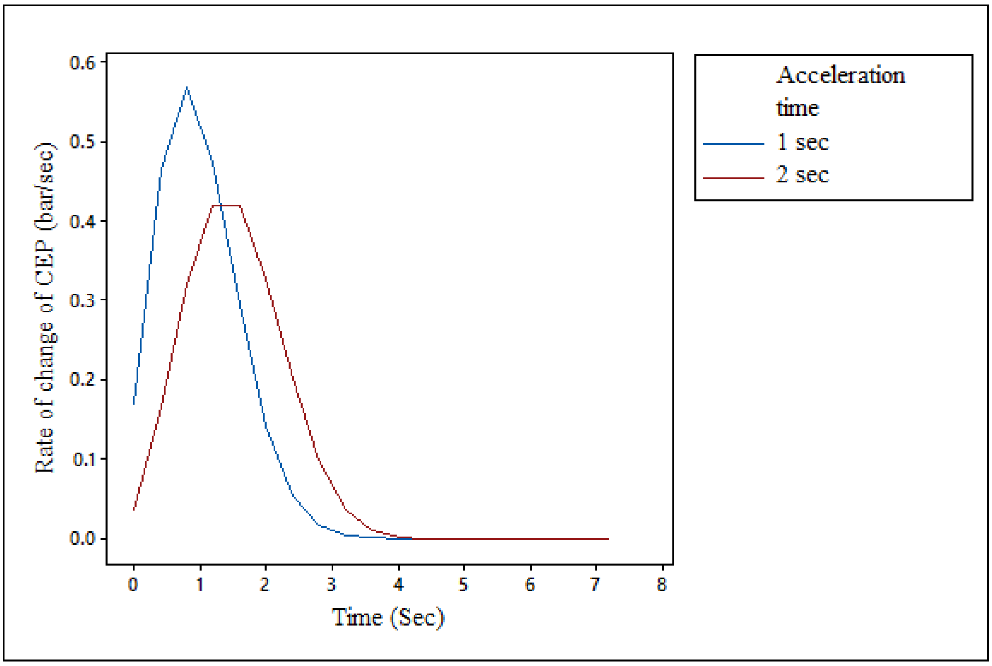

Figure 5.22: Rate of change of compressor exit pressure

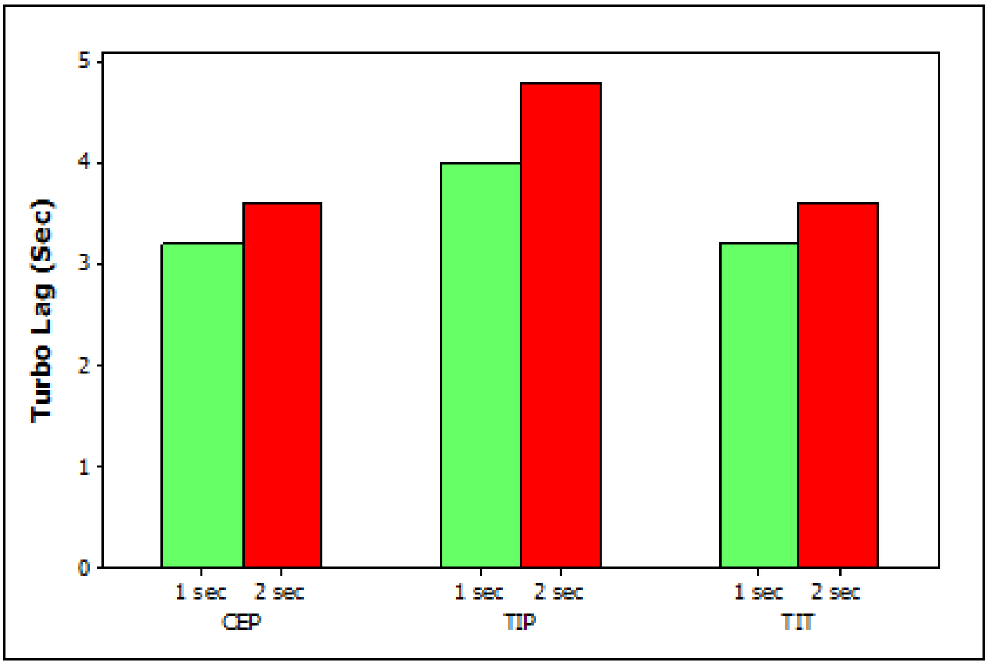

Figure 5.21: Effect of air injection on response parameters with variable acceleration time

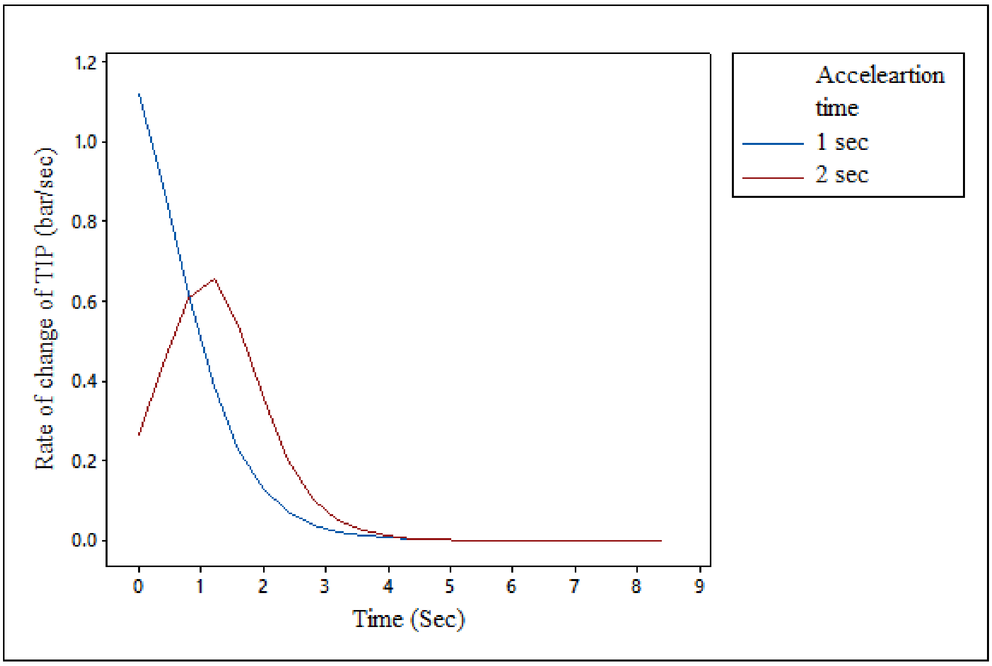

Figure 5.23: Rate of change of turbine inlet pressure

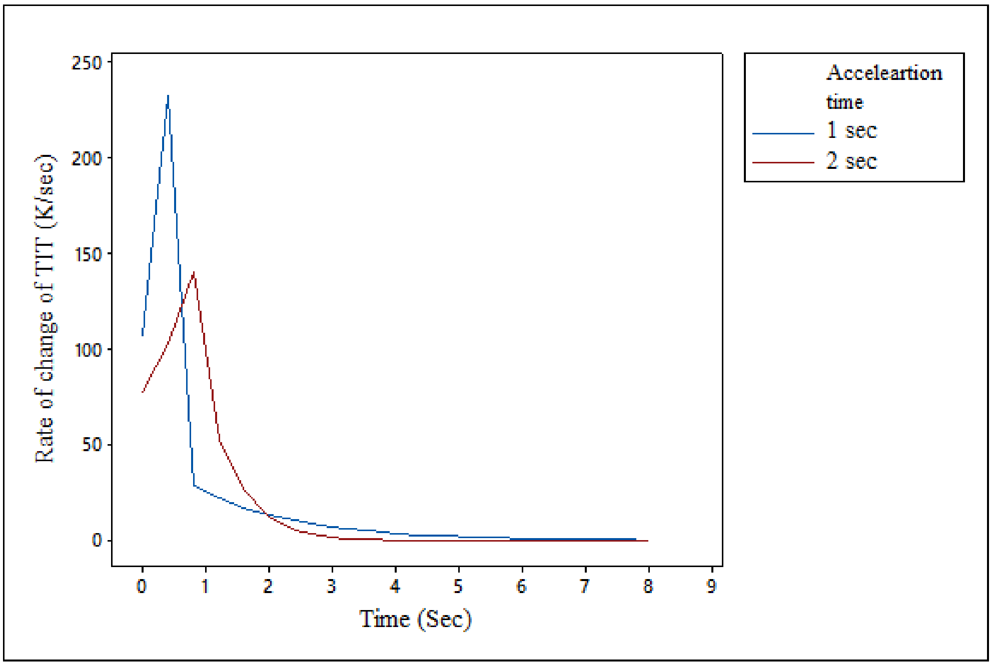

Figure 5.24: Rate of change of turbine inlet temperature

The figures show that faster response is observed for 1 sec when air injection is applied. Hence more improvement is achieved where the turbo lag is more prominent (that is for 1 sec acceleration time as reported in Test1). This ensures that the air injection at intake manifold greatly improves the transient response of turbocharged diesel engine for the event of rapid acceleration.

5.4 Evaluation of air injection system under load test

Load step demand of the engine is another important event that limits the transient response of the turbocharger. In urban roads, the engine speed and the load varies suddenly and frequently, resulting in increased exhaust emissions. In such operations, the effect of air injection technique to access the transient response of the engine is of great interest. To this aim, investigations under speed transient are already presented in the previous section and the effect on exhaust emissions will be covered in the next chapter. In this section, response of the turbocharged diesel engine is investigated for different load conditions. The method of analysis and the overall tests are summarized in Figure 5.25. Three cases of load transients are considered: constant load, load magnitude variation and load scheduling. Air injection technique is simulated and after optimization of injection pressure, its effect on the transient response is presented.

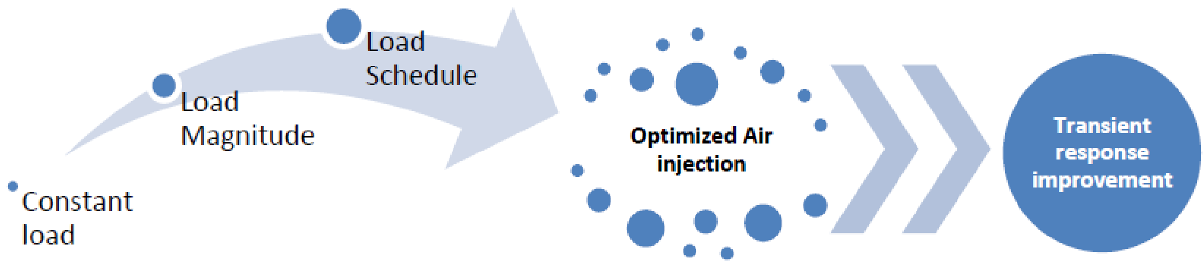

Figure 5.25: Schematic of load tests

5.4.1 Turbocharger response for constant load step

The transient performance of the turbocharged diesel engine is investigated in terms of turbocharger response parameters. As discussed in the test procedures in Chapter 3, the load is kept constant with initial load 80% and the final load 100%. Five engine speeds are chosen within the operating range of the engine. Optimization performed in acceleration test (Test 2) for air injection is referred and based on that, the effect of 1 bar air injection is observed on turbocharger response parameters.

Results are presented in two types of diagrams. The first ones (Figure 5.26, Figure 5.27 and Figure 5.28) show the effect of 1 bar air injection on compressor exit pressure (CEP), turbine inlet pressure (TIP) and turbine inlet temperature (TIT) for engine speeds 1000 and 1800 rpm under the constant load to show the effect of air injection at lower and higher engine speed respectively. The second ones (Figure 5.29, Figure 5.30 and Figure 5.31) show the dependence of turbo lag reduction (%) and steady state value ratio on engine speed. By the term steady state value ratio (SS Value ratio), it is meant the ratio of the final steady state value of the response parameter for air injection to that of no air injection. In addition to turbo lag reduction, SS Value ratio is used as an indicator for the effect of air injection on the equilibrium state value of the response parameter.

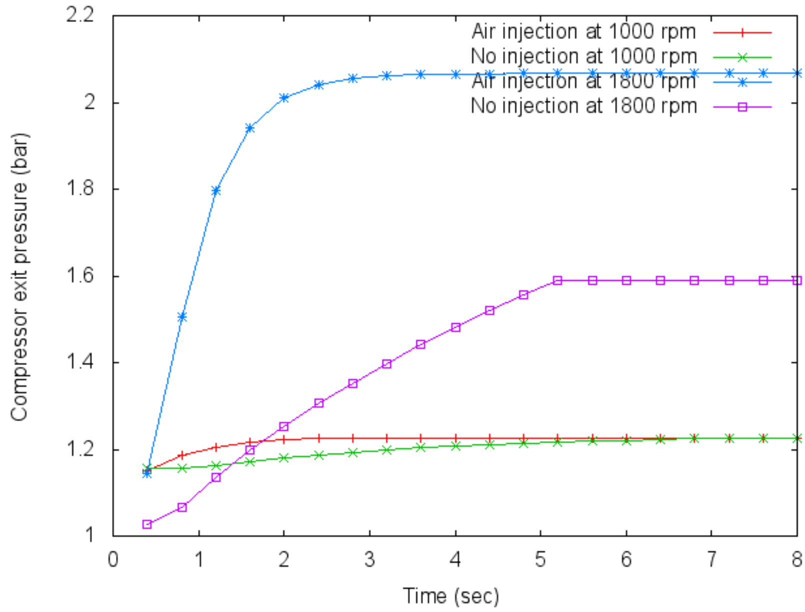

Figure 5.26: Effect of air injection on compressor exit pressure response during a load change from 80% to 100% at different engine speeds

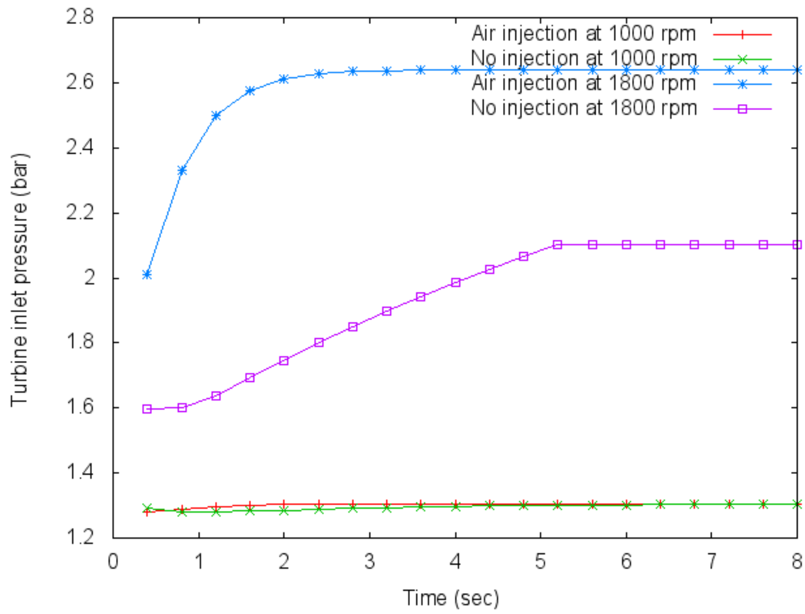

Figure 5.27: Effect of air injection on turbine inlet pressure response during a load change 80% to 100% at different engine speeds

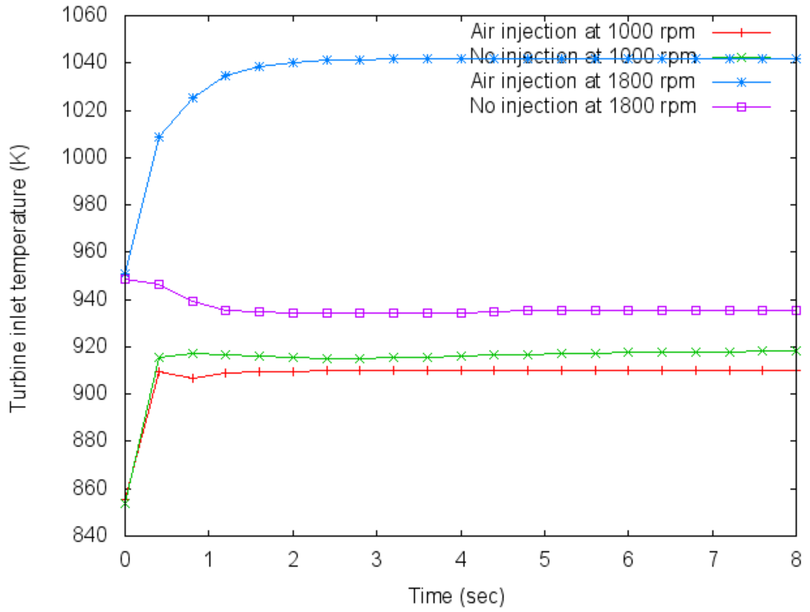

Figure 5.28: Effect of air injection on turbine inlet temperature response during a load change 80% to 100% at different engine speeds

Figure 5.26, Figure 5.27 and Figure 5.28 compare the turbocharger response parameters with air injection and without air injection when the engine is subjected to a load step from 80% to 100% in 1 sec under constant engine speeds of 1000 and 1800 rpm. Figure 5.26 and Figure 5.27 show that as the speed increases CEP and TIP also increases that supplies more energy to the turbocharger. Figure 5.28 shows that the maximum attainable value of TIT is lower when air injection is applied at lower engine speed (1000 rpm). While at higher engine speed (1800 rpm) air injection brings same improvement in TIT also.

To further analyse the results obtained above, steady state value ratio of the response parameter under consideration and turbo lag reduction (%) are plotted with respect to the engine speed. Figure 5.29, Figure 5.30 and Figure 5.31 show the dependency of steady state value ratio of the response parameter under consideration and turbo lag reduction (%) on the engine speed. It can be seen from these figures that in terms of SS value ratio, the effect of air injection is more at higher engine speeds (1600-1800 rpm) as greater torque and power are produced while at lower engine speeds (1000-1400 rpm) the technique brings far less improvement.

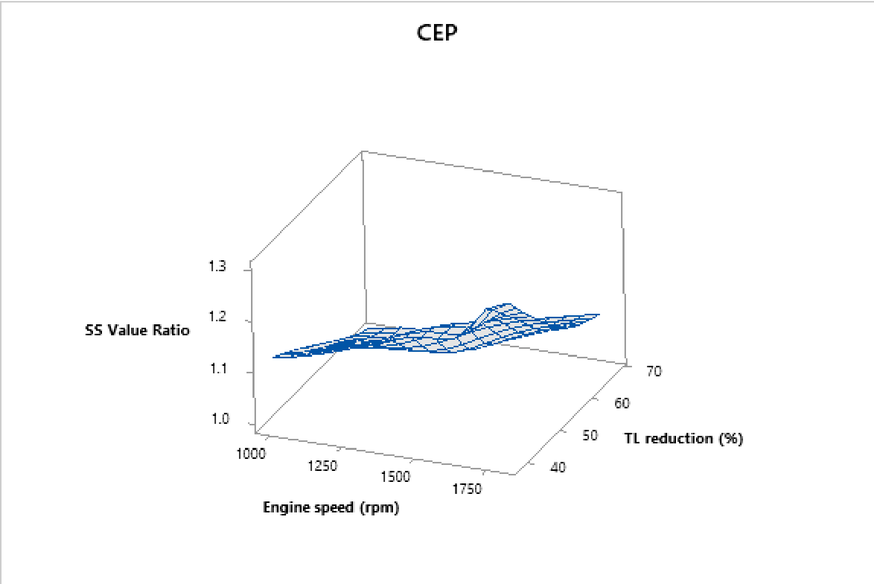

Figure 5.29: Dependency of turbo lag and SS value ratio on engine speed for compressor exit pressure (CEP)

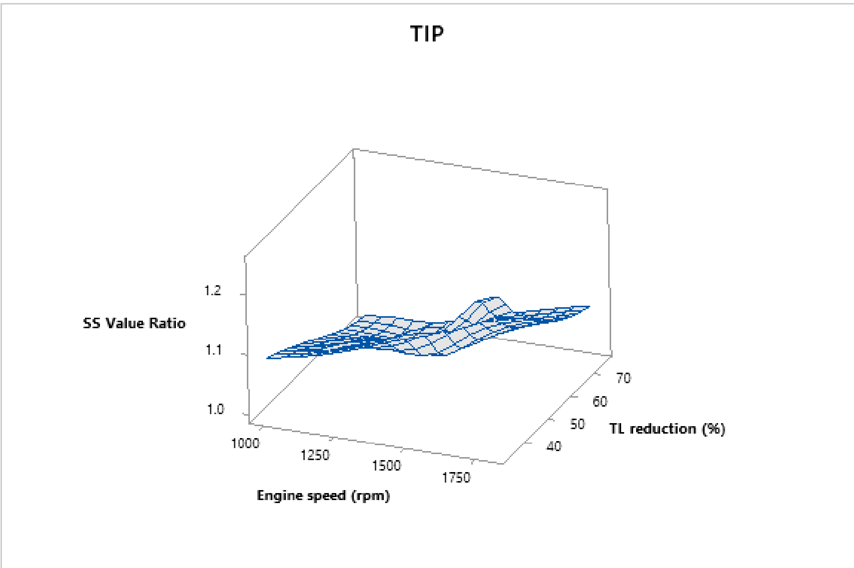

Figure 5.30: Dependency of turbo lag and SS value ratio on engine speed for turbine inlet pressure (TIP)

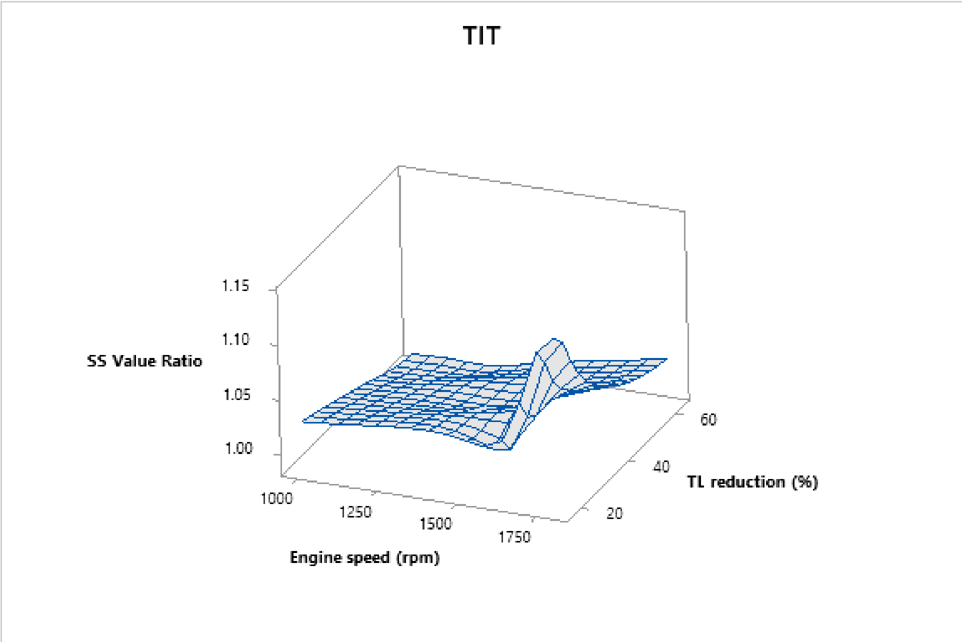

Figure 5.31: Dependency of turbo lag and SS value ratio on engine speed for turbine inlet temperature (TIT)

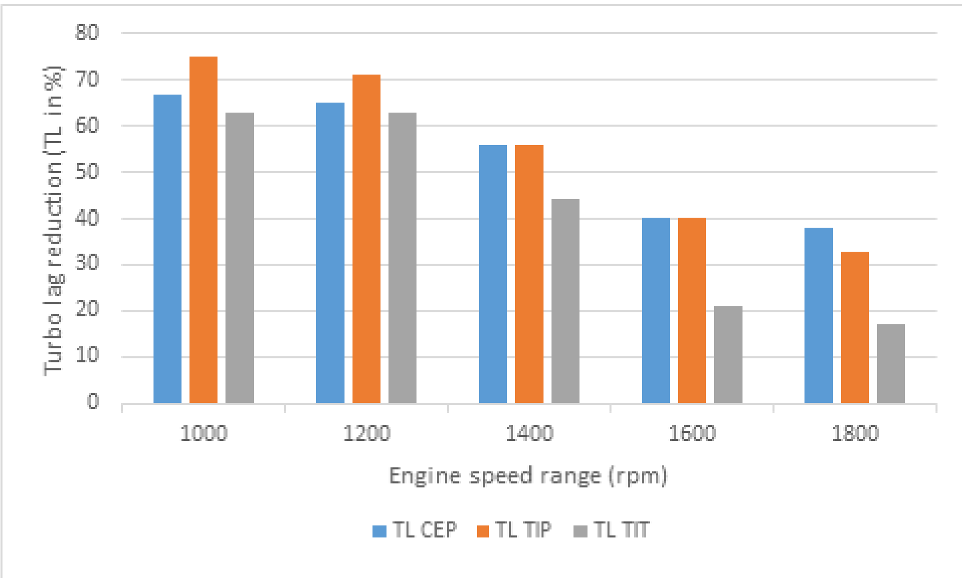

Figure 5.32: Dependency of transient response improvement on engine speed in terms of turbo lag reduction (%)

To access the effect of air injection on the transient performance of the diesel engine under these operating conditions of constant load and speed variation, Figure 5.32 is plotted. From the figure it can be seen that significant improvement in transient performance is observed for CEP, TIP and TIT at 1000-1400 rpm engine speed in terms of turbo lag reduction (%) while at 1600-1800 rpm engine speed, air injection yielded lesser improvement in the transient performance.

5.4.2 Load magnitude variations

Initial engine speed is 1000 rpm and load magnitude is varied in 1 sec. The key performance indicators (KPI) are chosen based on engine and turbocharger response. Various load steps of increasing magnitude are investigated and the effect is observed on these KPI:

- Engine speed
- Compressor exit pressure (CEP)
- Turbine inlet pressure (TIP)
- Turbine inlet temperature (TIT)

5.4.2.1 Effect of air injection for varying load magnitude

Identification of load step is an important aspect in the load transients to consider those load steps whose application can adversely affect the engine performance. As discussed in section 3.5.1.3.2, the results of L1 to L6 load tests are analysed and discussed for the effect of air injection. A series of demonstrative diagrams are provided to analyze the effect of changing the mass flow of the injected air on the injection pressure. This will provide optimization of air injection followed by the transient response improvement analysis for the effect of optimized air injection.

With the application of load step in 1 sec, the engine speed decreases for L1 to L4 load tests (Initial load 10%) as shown by "No Injection" curves in Figure 5.33. Higher the load magnitude, greater is the speed drop. To improve the engine response, air injection is applied at intake manifold. Minimum air injection pressure, required to prevent zero engine speed are 1.5 bar, 2.6 bar, 4.4 bar and 6 bar for L1, L2, L3 and L4 load tests respectively. However, it is noted from the figure that these injection pressure are not sufficient to prevent the engine from stalling which is observed for both, no injection as well as air injection for the given load tests.

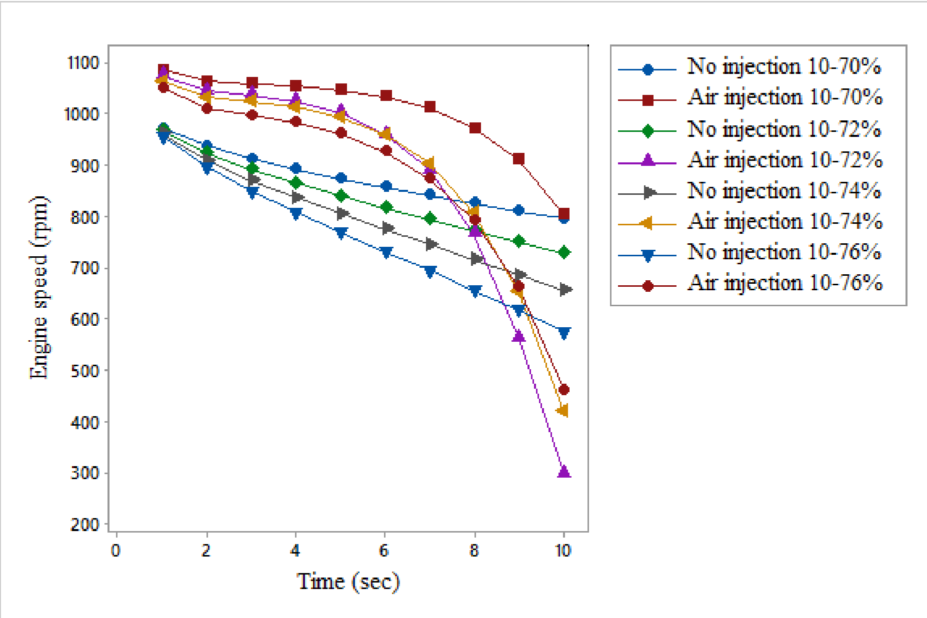

Figure 5.33: Effect of air injection on engine speed with different load steps, initial load 10%

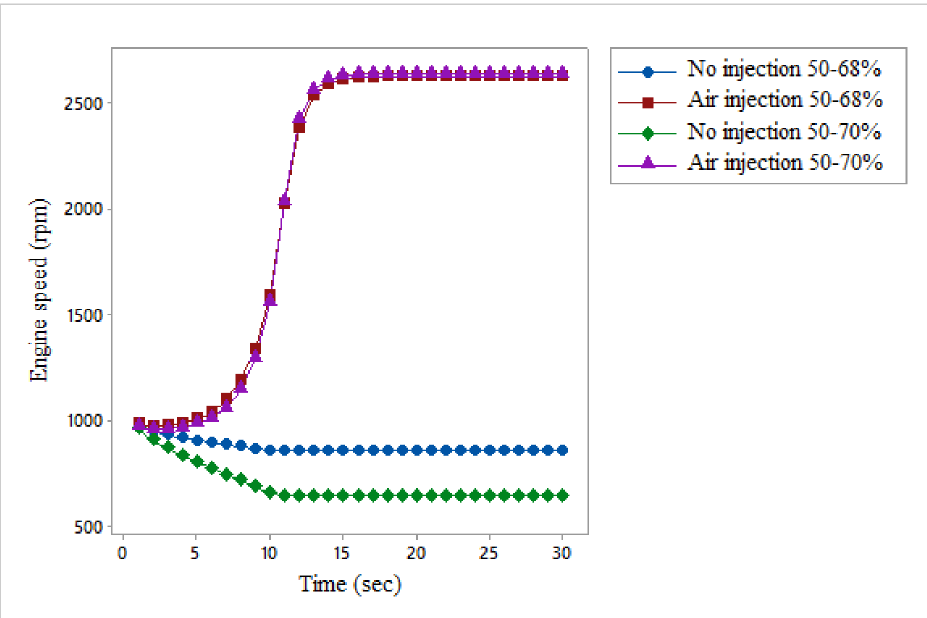

Figure 5.34: Effect of air injection on engine speed with different load steps, initial load 50%

Similarly, in case of 50% initial load, L5 and L6 load tests need investigation. When there is no air injection, the speed decreases as shown in Figure 5.34. With the application of air injection with minimum injection pressure, speed gradually increases but takes too long to reach the steady state. It takes 19 sec with 2.4 bar air injection for L5 load test and 18 sec with 4.2 bar air injection for L6 load test.

So, based on the engine speed response for air injection under two tests, it is worth considering higher injection pressure to obtain the equilibrium state of the engine speed for the load tests L1 to L6. But injection pressure can not be increased randomly as it may cause surging [99]. Optimization of injection pressure is hence essential to avoid this situation. In the next section, optimization of injection pressure is done based on the mass flow rate of the injected air which depends upon the orifice diameter as discussed in Chapter 3 (equation 3.16).

5.4.2.2 Optimization of injection pressure

In the investigations performed on speed transient, optimum injection pressure is obtained based on the energy imparted per second. From equation 3.16 and equation 5.1 it can be depicted that energy imparted depends upon the orifice diameter which regulates the mass flow of the injected air. So, rather than calculating the energy imparted per second for each load step, orifice diameter can be chosen as a good choice of parameter for the purpose of optimization of injection pressure bringing novelty in the optimization process.

From the different load steps for which the engine speed deteriorates, following load steps are considered to observe the effect of increasing the load magnitude:

1. Initial load 10%, final load increased from 70% to 76% (Load tests L1 and L4).

2. Initial load 50%, final load increased from 68% to 70% (Load tests L5 and L6).

For the optimization of air injection, L1 test is chosen and the results are presented in Figure 5.35. The figure shows the effect of varying the orifice diameter on the engine speed for load step L1. It is observed from the results that there is a correlation between the orifice diameter, injection pressure and turbo lag value.

Top left plot of Figure 5.35 for orifice diameter 5 mm shows the application of higher injection pressures to overcome stalling. It shows that higher the injection pressure, better the transient response in terms of turbo lag. Minimum required pressure to overcome zero speed is 1.8 bar and the turbo lag is 15 sec. Increasing the injection pressure decreases the turbo lag and 4 bar yields the minimum turbo lag of 13 sec. So, for 5 mm orifice diameter, the optimum injection pressure is 4 bar with turbo lag 13 sec. Further, optimum injection pressure is observed for 10 mm, 15 mm and 20 mm orifice diameter. In the same manner, optimum injection pressure and the turbo lag value is noted for different orifice diameter for L4, L5 and L6 load tests. The quantification is presented in Table 5.4.

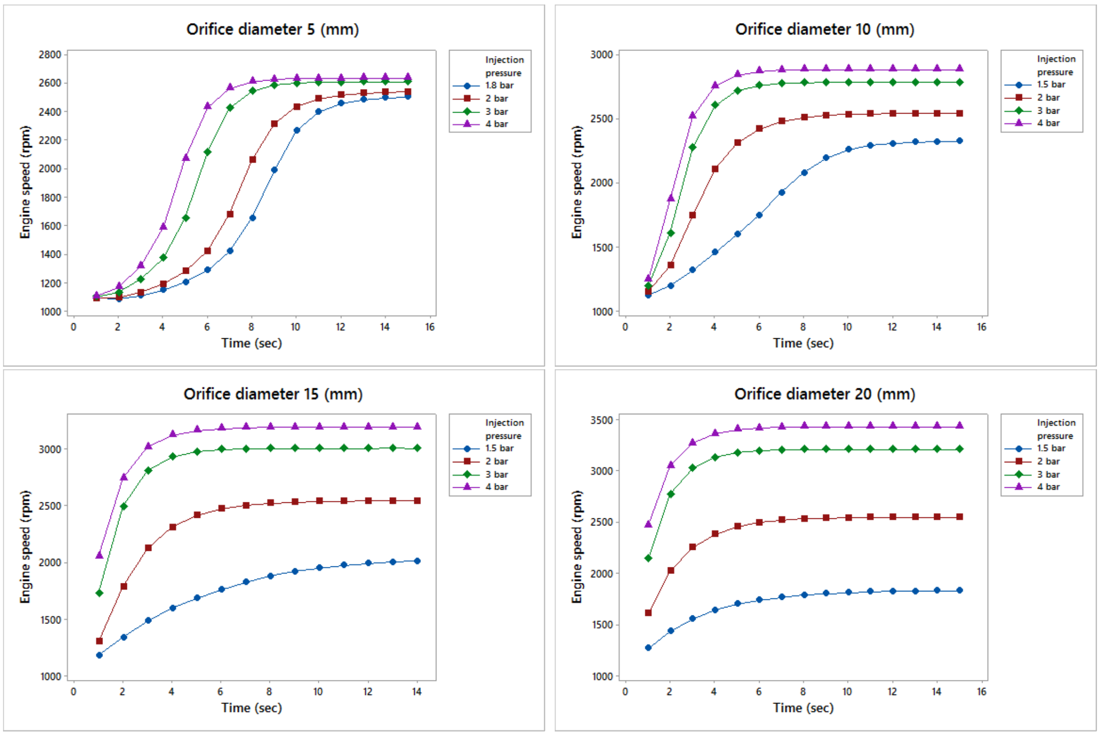

Figure 5.35: Engine speed with varying orifice diameter and injection pressures for load test L1

Table 5.4: Dependency of injection pressure (IP in bar) and turbo lag (TL in sec) on orifice diameter(mm) for the different load steps

OD	Load tests							
	L1		L4		L5		L6	
	IP	TL	IP	TL	IP	TL	IP	TL
5	4	13	8	12	5	12	6	12
10	3	10	4	10	4	10	3	10
15	3	9	3	9	3	10	3	9
20	3	10	3	10	4	11	3	10

It can be noted from Table 5.4 that orifice diameter considerably affects the injection pressure and turbo lag. This can be explained from equation 3.16 which indicates that the mass flow rate of air is directly proportional to the area of the orifice. So increasing the orifice diameter increases the mass flow rate of air which in turn increases the air fuel ratio. This leads to better combustion and hence turbo lag is reduced.

This dependency of injection pressure and turbo lag on orifice diameter for each test is demonstrated in Figure 5.36 to Figure 5.39. For each test, it can be noted from the figures that at 5 mm orifice diameter, turbo lag is considerably high (13 sec for L1 and 12 sec for L4, L5 and L6 each). Also higher injection pressure is required (4 bar for L1, 8 bar for L4, 5 bar for L5 and 6 bar for L6) at 5 mm orifice diameter.

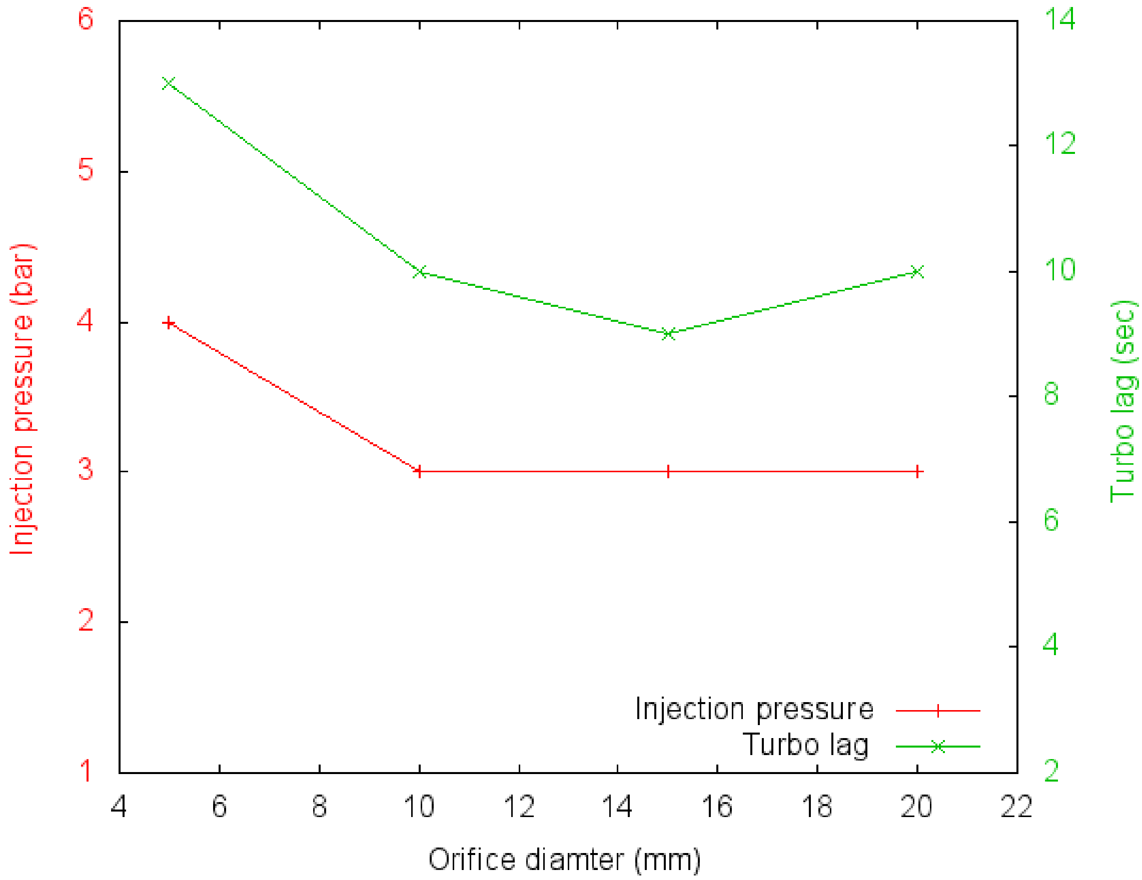

Figure 5.36: Correlation between turbo lag, injection pressure and orifice diameter for L1 test

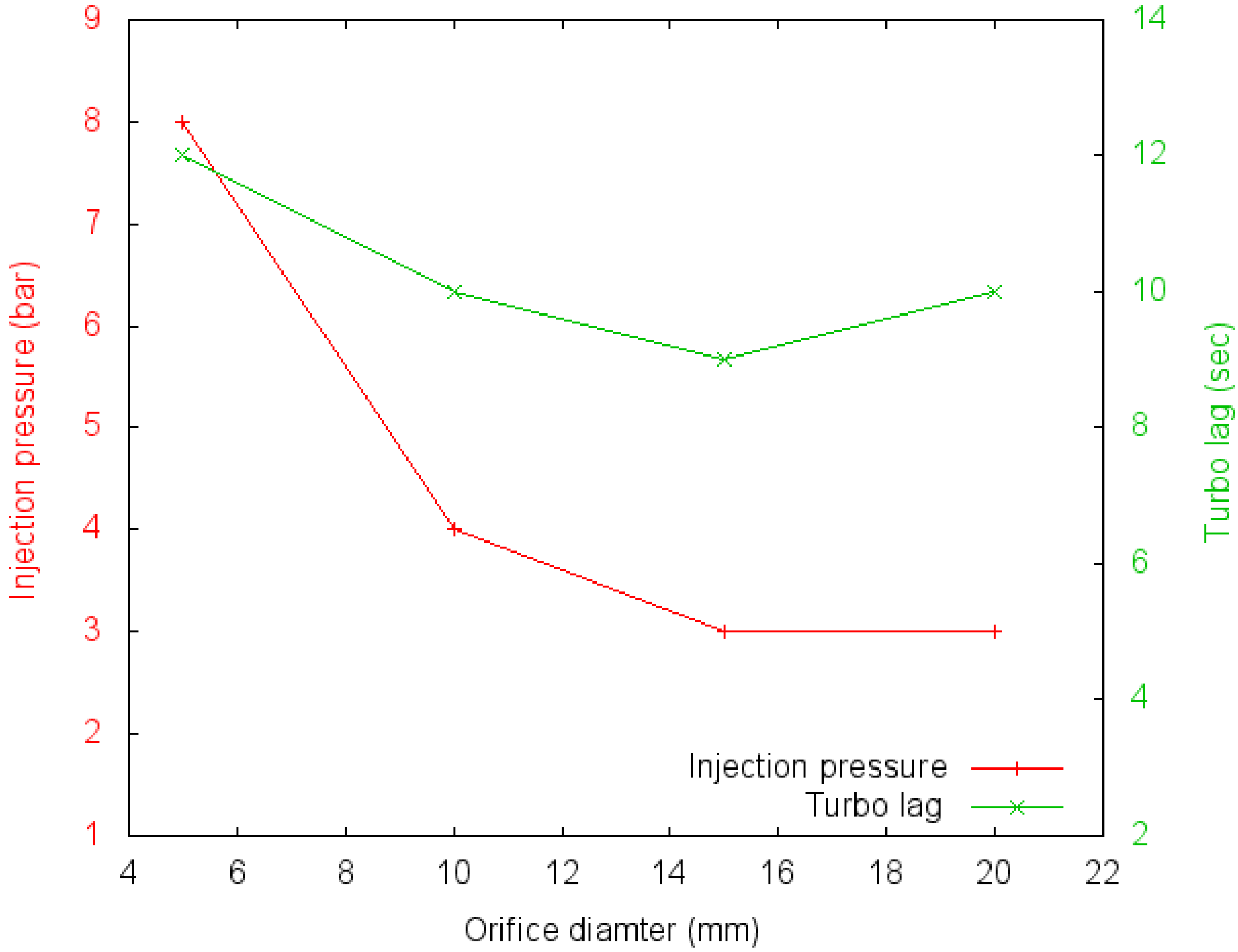

Figure 5.37: Correlation between turbo lag, injection pressure and orifice diameter for L4 test

It can be analyzed from the figures that increasing the orifice diameter from 5 mm to 10 mm drastically reduces the injection pressure and turbo lag for the different load steps. Increasing the orifice diameter to 15 mm yields 3 bar injection pressure with minimum turbo lag. Nevertheless, increasing the orifice diameter beyond 15 mm doesn't bring significant improvement in turbo lag. A dip in turbo lag value at 15 mm orifice diameter for all the chosen tests (L1, L4, L5 and L6) indicates the peculiarity of these tests. This means that further increasing the mass flow rate of air by increasing the orifice diameter may cause the compressor to surge as discussed in section 5.4.2.1. Due to this reason turbo lag increases after 15 mm orifice diameter.

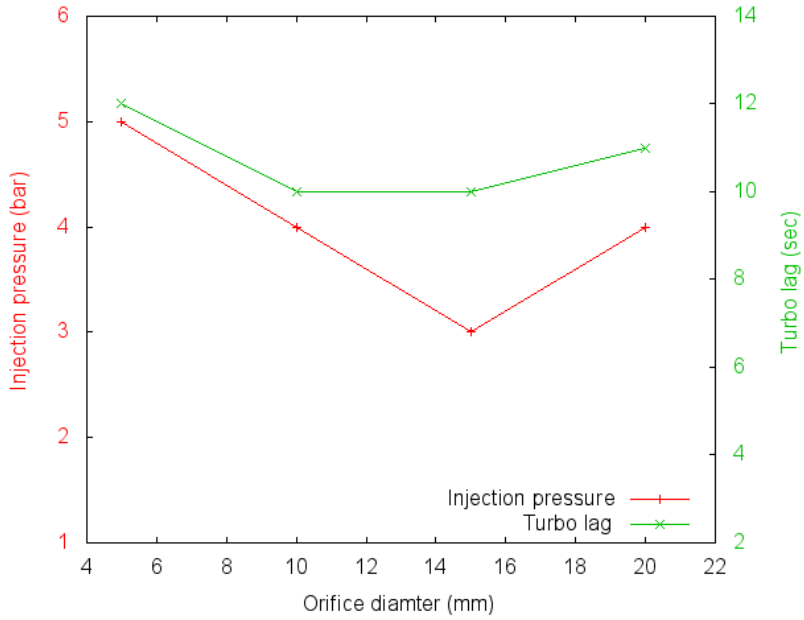

Figure 5.38: Correlation between turbo lag, injection pressure and orifice diameter for L5 test

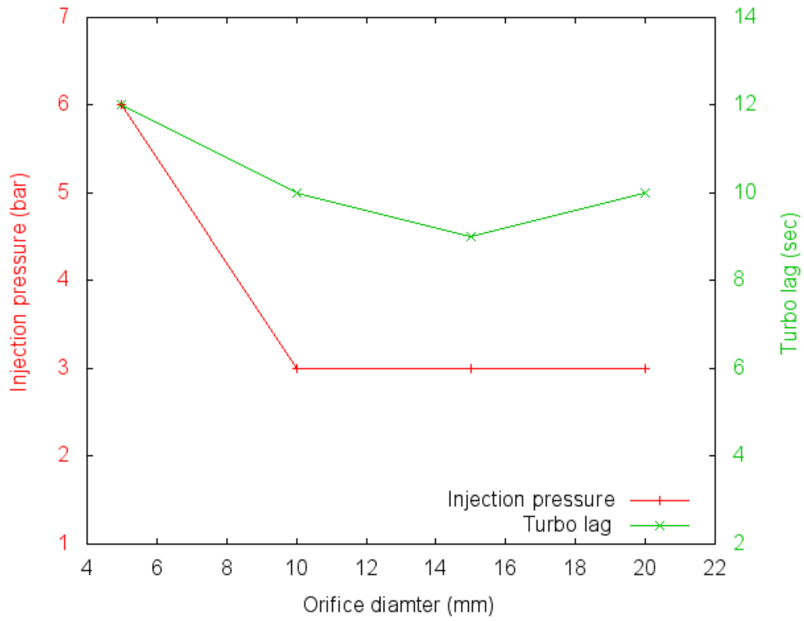

Figure 5.39: Correlation between turbo lag, injection pressure and orifice diameter for L6 test

So, for the purpose of optimization, orifice diameter will be taken as 15 mm and the effect of 3 bar air injection for the different load steps will be investigated on other key performance parameters.

5.4.2.3 Effect of air injection on turbocharger response parameters

As concluded in the previous section, orifice diameter is taken as 15 mm and the effect of 3 bar air injection for the two increasing load steps (L1 to L4 and L5 to L6) will be investigated in this section on the turbocharger response parameters: CEP, TIP and TIT. As shown in Table 5.5, the TL reduction (%) is noted for CEP, TIP and TIT after the application of air injection.

Table 5.5: Turbo lag reduction (%) for the turbocharger response parameters under different load tests

Load	TL reduction (%)		
step	CEP	TIP	TIT
L1	33	50	45
L4	27	26	25
L5	25	42	42
L6	45	50	45

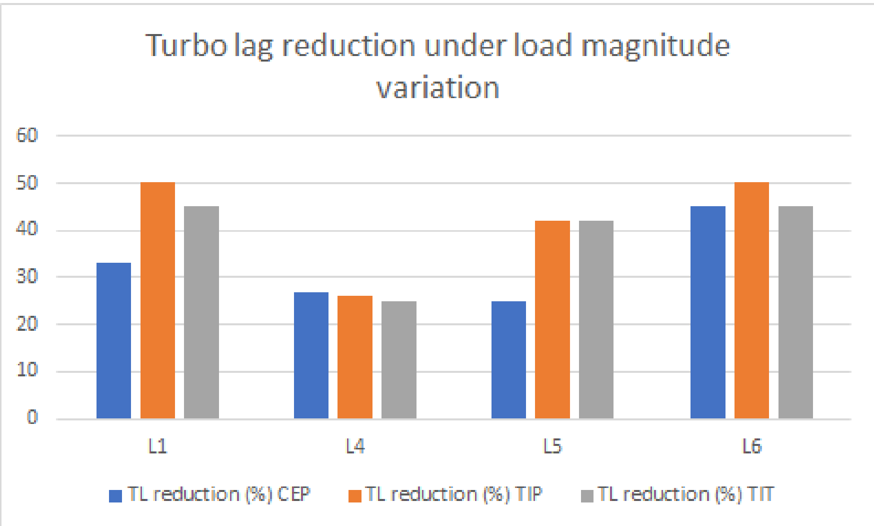

Figure 5.40: Effect of air injection on turbocharger response parameters for different load magnitude

Figure 5.40 shows that air injection significantly reduces the turbo lag for the turbocharger response parameters under the effect of different load steps. Nevertheless, as per the quantification presented in Table 5.5, increasing the magnitude of stronger load (Initial load 10%) decreases the turbo lag reduction. This is because of the fact that it is difficult for the engine to cope with high loading instantly [16]. Whereas, increasing the magnitude of lighter load (Initial load 50%) increases the turbo lag reduction. Maximum improvement in turbo lag reduction is observed for 50-70% load step (Load test L6).

5.4.3 Load schedule

Based on the results of the previous section, air injection under Test L6 brings maximum turbo lag reduction in 1 sec. So, for the purpose of observing the effect of load scheduling, the effect of air injection for the same load step (Test L6) in 2 sec will be investigated in the present section and the results will be compared with that of 1 sec.

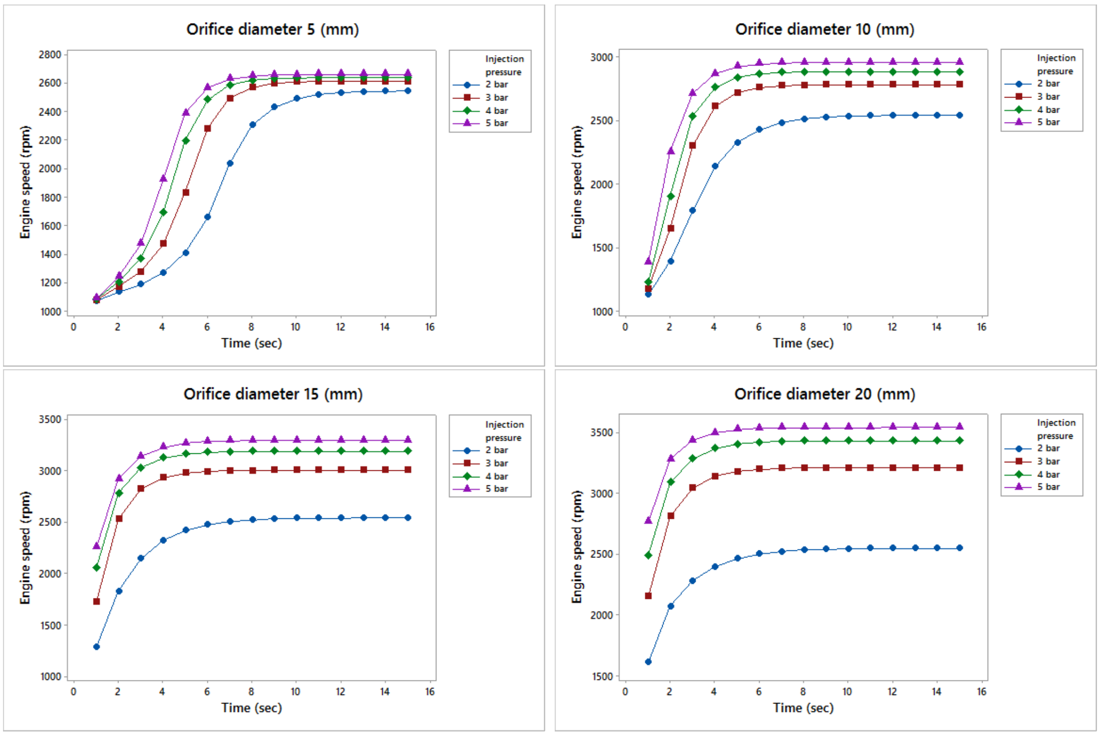

Figure 5.41: Engine speed with varying orifice diameter and injection pressures for load step 50-70% in 2 sec

Before modeling the effect of air injection, again optimization of injection pressure based on orifice diameter is done as shown in Figure 5.41. From the analysis of the figure, optimum value of air injection is found to be 3 bar at 15 mm orifice diameter. So, the effect of 3 bar air injection at 15 mm orifice diameter is investigated on turbocharger response parameters.

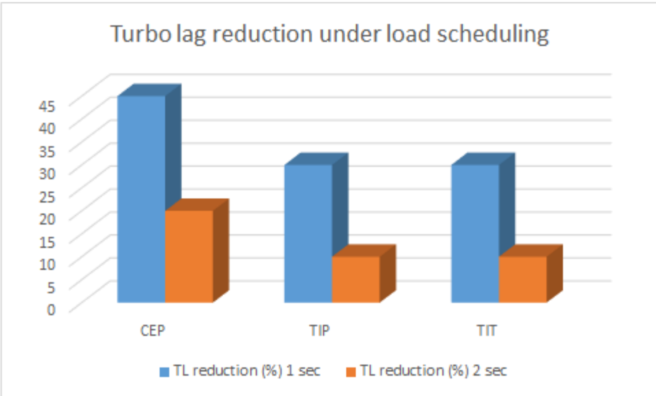

Figure 5.42: Effect of air injection on turbocharger response parameters for different load schedules for L6 load test

Figure 5.42 shows the effect of air injection on CEP, TIP and TIT under L6 load test for 1 sec and 2 sec load schedules. Significant reduction in turbo lag is observed for these three parameters, however, the turbo lag reduction (%) is more for 1 sec than for 2 sec. So, air injection technique is more effective for 1 sec load transient than for 2 sec. This behaviour is similar to that of acceleration tests summarizing that the air injection technique is more effective for 1 sec transients as compared to 2 sec.

5.5 Summary of the chapter

In this chapter, real engine is modeled using Ricardo wave simulation software and after the validation using the steady state simulations, the transient performance of the turbocharged diesel engine is investigated. Air injection at intake manifold is simulated to improve the transient response. The transient performance of the diesel engine is investigated under various acceleration tests. Finally, the effect of air injection on transient response of the turbocharged diesel engine is simulated under different load conditions. Based on the results obtained from this study the following conclusions are drawn for this specific heavy duty diesel engine:

1. Engine performance parameters are tested for steady state and Ricardo wave successfully reproduces the experimental data for the chosen performance parameters

i.e. brake torque, brake specific fuel consumption and brake power.

2. Comparison is made between the three methods of transient response improvement and of the three methods considered, air injection is identified as the most efficient technique with respect to turbo lag reduction.

3. The turbocharger response parameters such as compressor exit pressure (CEP), turbine inlet pressure (TIP) and turbine inlet temperature (TIT) increases with the increase in air injection pressure and 2.5 bar injection pressure brings the maximum improvement in the transient response.

4. The amount of energy imparted for air injection should not be neglected while reducing the turbo lag for the improvement in transient response. Greater reduction in turbo lag can be achieved by applying higher assist pressure up to 2.5 bar but at the cost of compromised energy imparted per second.

5. 1 bar is the optimum injection pressure with turbo lag reduction per unit energy as 0.290, 0.392 and 0.555 per joule for compressor exit pressure, turbine inlet pressure and turbine inlet temperature (TIT) respectively.

6. Turbo lag is more prominent when the acceleration time is 1 sec than that for 2 sec. With the application of air injection, faster recovery time is observed for 1 sec rapid acceleration than that of 2 sec.

7. Under constant load step, both the turbo lag reduction as well as the value of the maximum attainable parameter depends upon the engine speed. Air injection brings maximum improvement in turbo lag reduction at lower speed range (1000-1400 rpm) while its effect on maximum attainable value is observed at higher speed range (1600-1800 rpm).

8. 3 bar air injection at 15 mm orifice diameter is found to be the optimum value of air injection and its effect on turbocharger response is investigated under load magnitude variations. The results show that with air injection, although the turbo lag reduces under all the load steps, the improvement is more noticeable for lighter load variation than for stronger load. Maximum improvement under load magnitude increment is observed for 50-70% load step.

9. The effect of changing the load schedule from 1 sec to 2 sec is observed on turbocharger response parameters. Optimization of injection pressure shows 3 bar air injection at 15 mm orifice diameter to be more effective for 1 sec load schedule than for 2 sec indicating more benefit in case of faster load application.

6

EFFECT OF AIR INJECTION ON EMISSION CHARACTERISTICS OF A CI ENGINE DURING TRANSIENT OPERATION

Today, environment protection has become a topic of central concern. Adverse effects of diesel fuel on human health and environment have led the governments to put forward the regulations for permissible exhaust emissions. It is expected that emission legislation will be further highlighted to achieve 'near zero' emissions by 2030 [105]. Euro standards are developed by Europe lowering the emissions requirement continuously from Euro I to Euro VI since 1993. Euro VI legislation demands that CO, HC and NO_x emissions should not exceed 1.5 g/kWh, 0.13 g/kWh and 0.4 g/kWh respectively [66]. Present chapter deals with the effect of air injection technique on exhaust emissions and to ensure that the application of the technology comply with Euro VI standard.

The effect of air injection on the transient response of the turbocharged diesel engine is investigated in Chapter 5. The results of the investigation show that with the application of air injection at intake manifold, the transient performance of the turbocharged diesel engine is improved under the transient events of fueling rate change, speed transient and load transient. This indicates that the delay between fueling and air-supply is reduced by the application of air injection. The consequence would be better air supply to the cylinders that improves combustion. The exhaust emissions that are generated due to poor combustion following the turbo-lag period should be reduced. To this aim, investigations are carried out in the present chapter to analyze the effect of air injection on exhaust emissions.

For the overall optimization of the system, it is important to ensure that the air injection technique doesn't increase the exhaust emissions. For this purpose, optimization of injection pressure and mass flow of the air through the orifice diameter is performed and exhaust emissions are analysed. This will provide novel contribution as firstly the effect of air injection technique is rarely evaluated for exhaust emissions and secondly optimization of air injection for satisfying Euro 6 standard will be a new and important study due to emission requirement legislations. This will pave way for further research in this area. In the present chapter, investigations are performed to achieve objective 8 and objective 9.

6.1 Emission characteristics under speed transient

As discussed in chapter 5, low speed transient - 1000 rpm to 1700 rpm in 1 sec corresponds to the case of rapid acceleration and air injection is found to improve the transient performance of the engine under this case. To evaluate the effect of air injection on exhaust emissions and to comply with Euro 6 standard, optimization of injection pressure based on mass flow of the air through the orifice diameter is done for the speed transient of 1000 - 1700 rpm in 1 sec. Explicit multiple diagrams are provided to describe the effect of orifice diameter and in turn injection pressure on emissions.

6.1.1 Optimization of air injection for emissions under speed transient

Air injection pressure is varied and its effect on exhaust emissions (CO, HC and NO_2) is observed for 5 mm, 10 mm and 15 mm orifice diameter. The results are presented in Figure 6.1, Figure 6.2 and Figure 6.3.

The variation of CO emission values with injection pressure and orifice diameter is shown in Figure 6.1. It is clear from the figure that in case of CO emissions, 1 bar injection pressure increases the emissions if compared with 2 bar and 3 bar. The emissions also increase with the increase in orifice diameter. 2 bar injection pressure reduces the CO emissions for 5 mm and 10 mm orifice diameters. Further increasing the orifice diameter to 15 mm increases the CO emissions for 2 bar also. Whereas, 3 bar air injection significantly reduces CO emission particularly for 10 mm. There is no benefit of further increasing the injection pressure or orifice diameter and hence 3 bar injection pressure at 10 mm orifice diameter is the optimum value for CO emissions.

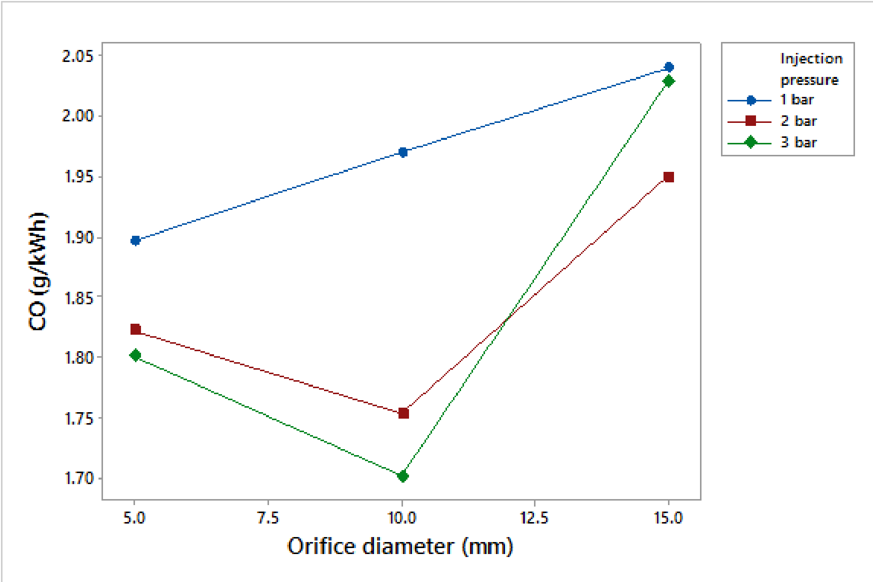

Figure 6.1: CO emissions with varying injection pressures and orifice diameter

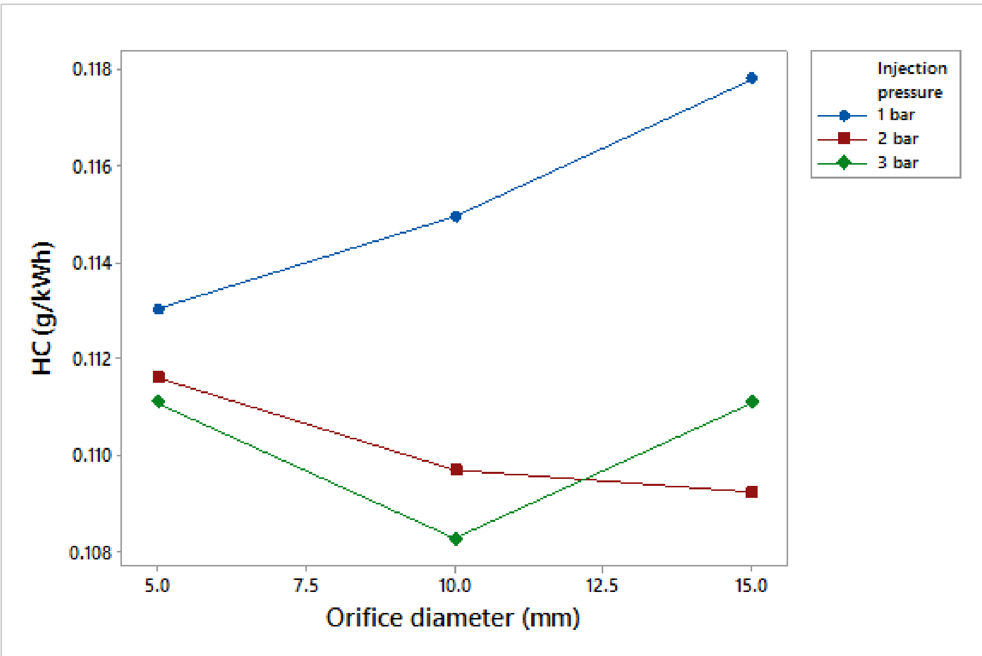

Figure 6.2: HC emissions with varying injection pressures and orifice diameter

Figure 6.2 depicts the HC emissions with varying injection pressure and orifice diameter.The analysis of the figure shows similar trend for CO emissions. 1 bar injection pressure is not sufficient to reduce the HC emissions for 5mm, 10 mm and 15 mm orifice diameter. Increasing the injection pressure upto 3 bar decreases the emissions and minimum HC emissions are observed for 3 bar air injection at 10 mm orifice diameter.

Significant reduction in NO_2 emissions is observed with air injection as shown in Figure 6.3. Increasing the injection pressure doesn't make much difference in the relative improvement in NO_2 emission reduction and hence 1 bar injection pressure at 10 mm is enough to minimize the emission as shown in Figure 6.3.

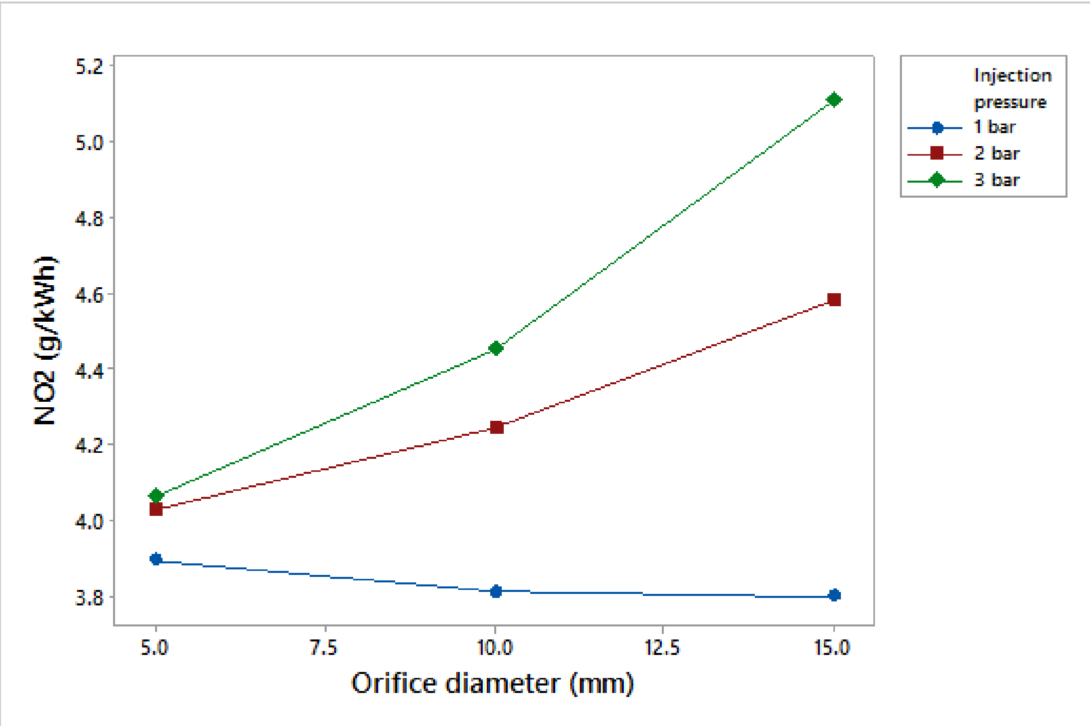

Figure 6.3: NO_2 emissions with varying injection pressures and orifice diameter

Diesel engine emissions typically contains NO and only 5% is the NO_2 [92], so for the purpose of evaluating the emissions against Euro 6 standard, only CO and HC emissions will be considered. Figure 6.4 is plotted for CO and HC emissions for 3 bar air injection and varying orifice diameter. With 3 bar air injection at 10 mm orifice diameter, CO and HC emissions are 1.5 g/kWh and 0.107 g/kWh respectively which satisfies Euro 6 emission standard [66] as shown with green color bar in Figure 6.4. So, the optimum injection pressure for CO and HC emission is 3 bar at 10 mm orifice diameter. After finalizing the optimum injection pressure, its effect on the concentration of CO, HC and NO_x emissions is evaluated in the next section.

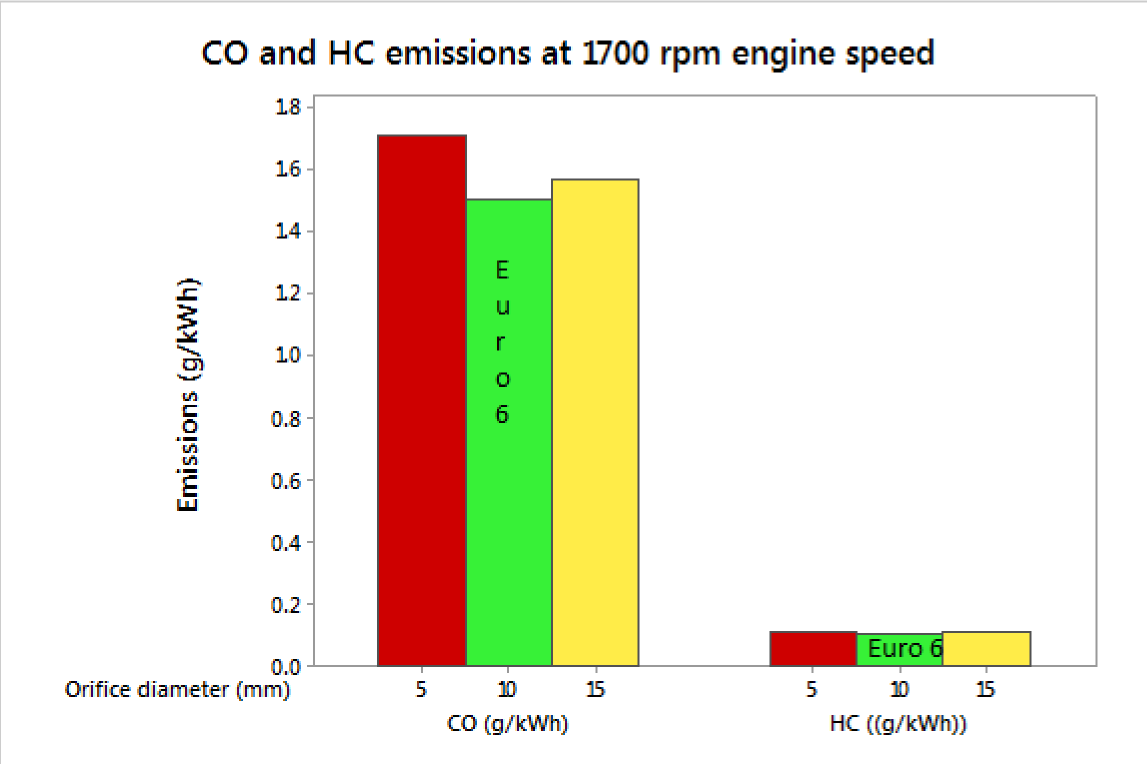

Figure 6.4: Optimization of air injection for CO and HC emissions at 1700 rpm engine speed

6.1.2 Emission reduction under the effect of air injection for speed transient

In this section, the effect of 3 bar air injection on emissions for 10 mm orifice diameter is compared with no air injection and the percent reduction is quantified for comparison. The concentration of the emissions are noted in parts per million (ppm). CO, HC and NO_x emissions are plotted as shown in Figure 6.5. It can be concluded from the figure that the air injection doesn't increase the emissions. Slight reduction is observed for CO (5%) and HC (0.4%) emissions while significant reduction (75%) is observed on NO_x emissions.

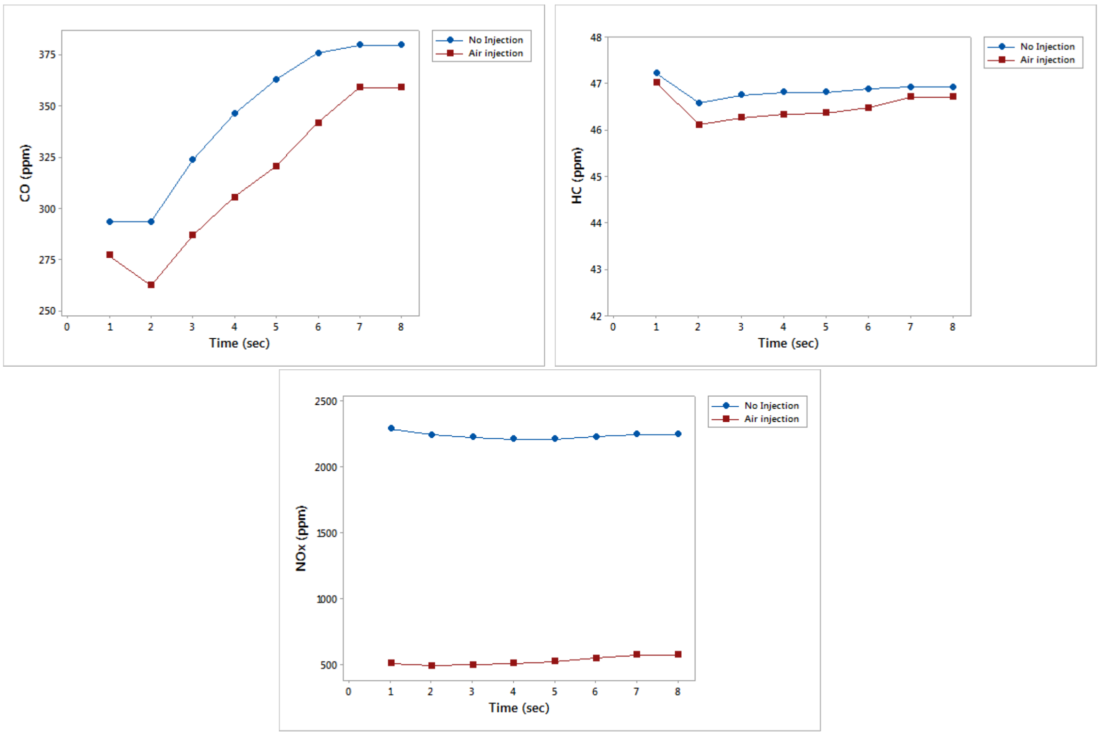

Figure 6.5: Effect of air injection on emission concentration under speed transient: CO emission (left), HC emission (right) and NO_x emission (bottom)

6.2 Emission characteristics under load transient

Effectiveness of air injection technique under load transient is investigated in chapter 5. 3 bar air injection at 15 mm orifice diameter is found to be most effective under 50-70% load step in 1 sec for improving the turbocharger response. In the present section, emission characteristics of the turbocharged diesel engine are investigated under 50-70% load step in 1 sec. CO, HC and NO_2 emissions are analyzed. Optimization of injection pressure and orifice diameter is done for these emissions.

6.2.1 Optimization of air injection for emissions under load transient

Air injection pressure and orifice diameter are varied and CO, HC and NO_2 emissions are plotted as shown in Figure 6.6, Figure 6.7 and Figure 6.8. Figure 6.6 shows that increasing the injection pressure increases CO emissions. For 1 bar air injection, CO emissions are minimum for all the chosen orifice diameters. Minimum value of CO emissions are observed for 15 mm orifice diameter with 1 bar air injection. Figure 6.7

and Figure 6.8 show that for HC and NO_2 emissions, 3 bar air injection at 10 mm orifice diameter and 2 bar air injection at 15 mm orifice diameter respectively yields minimum emissions.

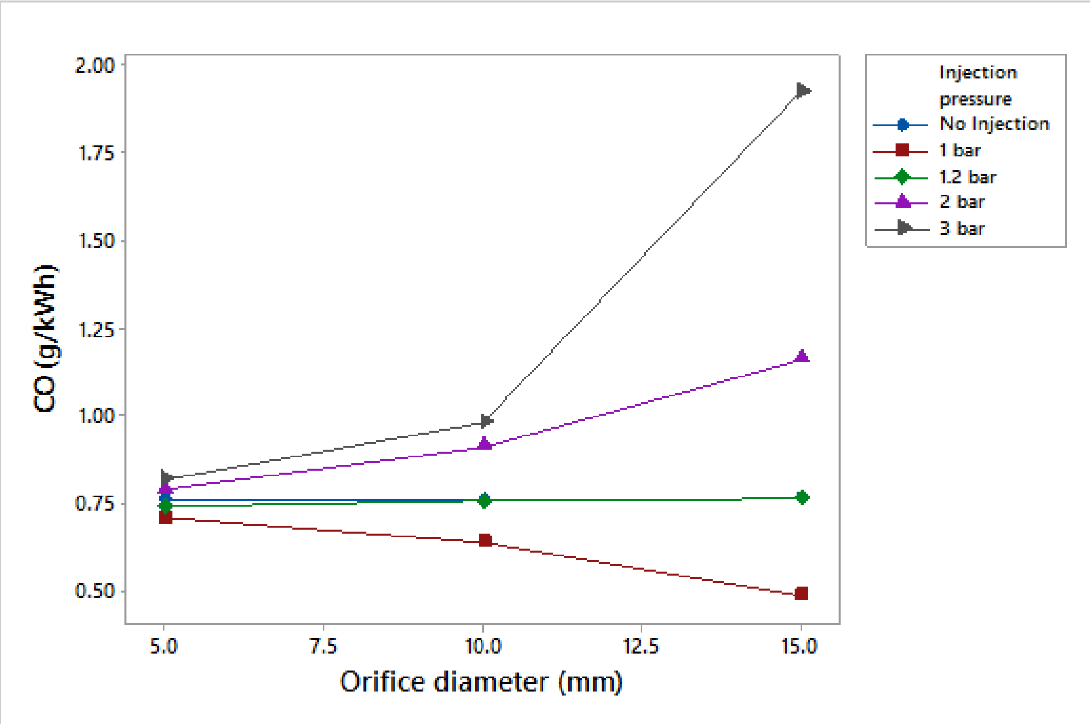

Figure 6.6: CO emissions with varying injection pressures and orifice diameter

However, it is noted that in case of load transient, emissions are below Euro 6 standard for all the chosen injection pressures and orifice diameters. Hence, optimization is done based on the injection pressure and orifice diameter for which all the emissions with air injection are less than no air injection. For this purpose, quantification of emissions are done for each pressure and orifice diameter as presented in Table 6.1, Table 6.2 and Table 6.3. In order to get reduction in all the three emissions when applying air injection, 1.2 bar air injection at 10 mm needs to be considered as it brings reduction in all the three emissions.

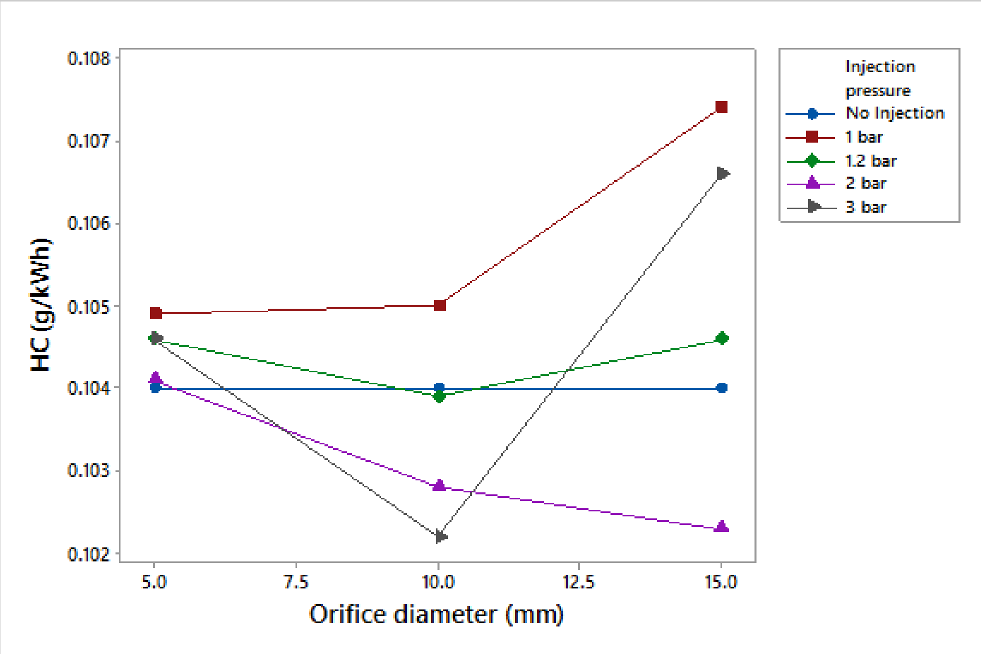

Figure 6.7: HC emissions with varying injection pressures and orifice diameter

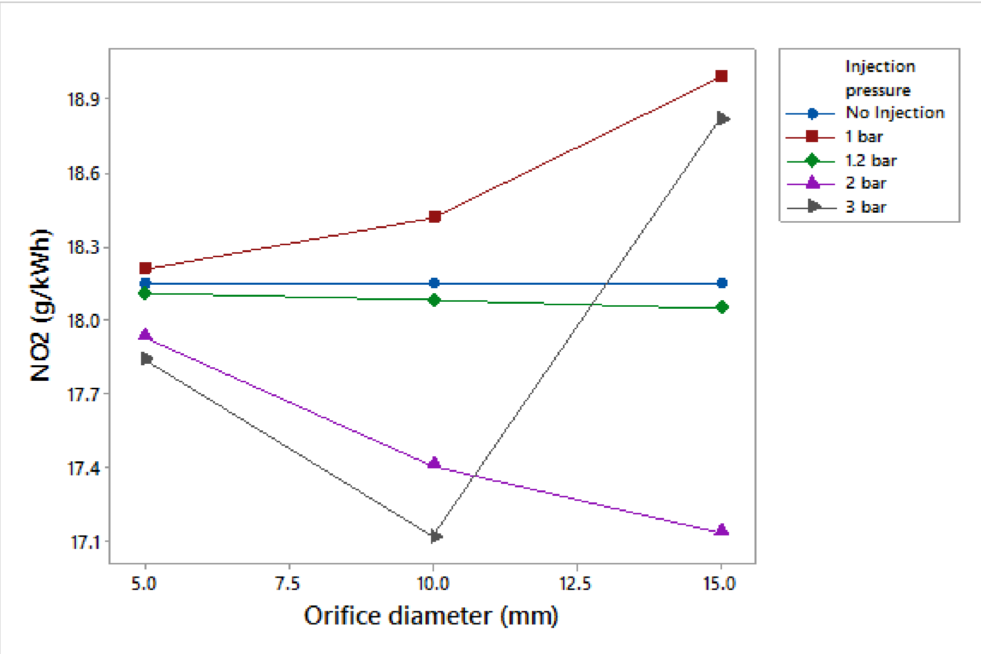

Figure 6.8: NO_2 emissions with varying injection pressures and orifice diameter

Table 6.1: Dependency of CO emissions on injection pressure and on orifice diameter

OD	Injection pressure				
(mm)	No Injection	1 bar	1.2 bar	2 bar	3 bar
5	0.76	0.71	0.74	0.79	0.82
10	0.76	0.64	0.757	0.91	0.98
15	0.764	0.491	0.7642	1.16	1.92

Table 6.2: Dependency of HC emissions on injection pressure and on orifice diameter

OD	Injection pressure				
(mm)	No Injection	1 bar	1.2 bar	2 bar	3 bar
5	0.104	0.1049	0.1046	0.1041	0.1046
10	0.104	0.105	0.1039	0.1028	0.1022
15	0.104	0.1074	0.1046	0.1023	0.1066

Table 6.3: Dependency of NO_2 emissions on injection pressure and on orifice diameter

OD	Injection pressure				
(mm)	No Injection	1 bar	1.2 bar	2 bar	3 bar
5	18.15	18.21	18.11	17.93	17.84
10	18.15	18.42	18.08	17.41	17.12
15	18.15	18.99	18.05	17.14	18.82

6.2.2 Emission reduction under the effect of air injection for load transient

The effect of 1.2 bar air injection at 10 mm on the concentration of emissions under load step 50-70% in 1 sec is then compared with no air injection and the percent reduction is quantified. As shown in Figure 6.9, CO, HC and NO_x emissions are observed in ppm. Percent reduction by the application of air injection is 0.8%, 0.01% and 0.4% for CO, HC and NO_x respectively as shown in Figure 6.9. So, the emissions doesn't increase with the application of air injection under the event of load change also.

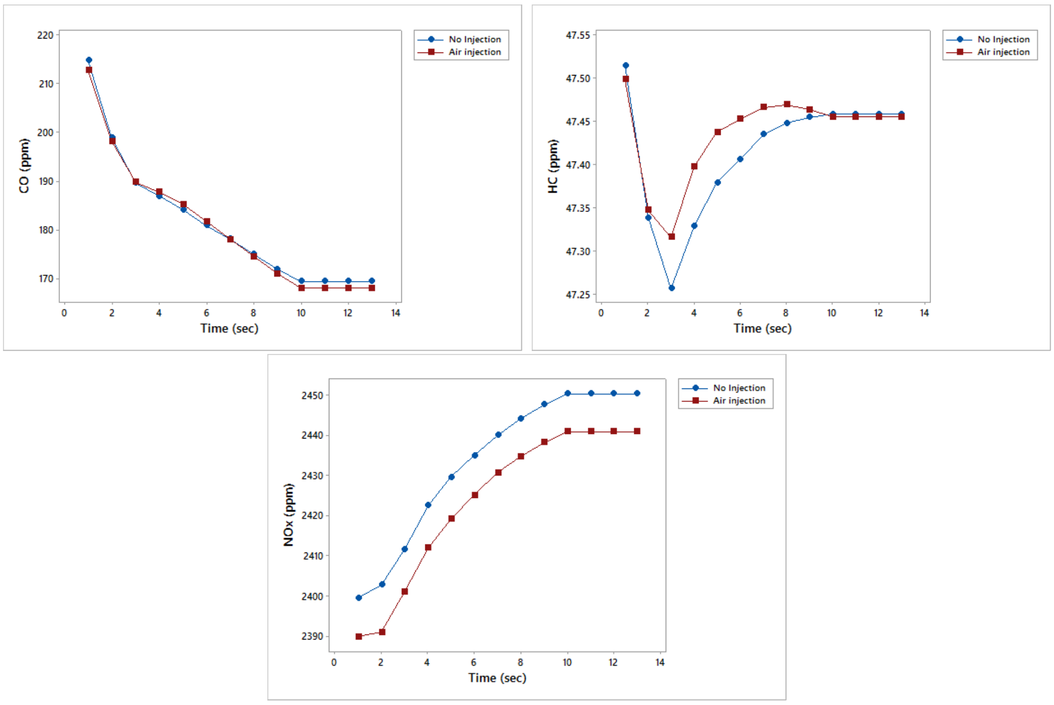

Figure 6.9: Effect of air injection on emissions under load transient

6.3 Overall optimization of air injection system

The effect of air injection on the transient performance of the diesel engine for different types of transient operating conditions is presented and discussed in the previous chapters. For the overall optimization of the system, it is important to consider the environmental performance in addition to the engine performance improvement. The present chapter presented the effect of air injection on exhaust emissions.

Table 6.4: Quantification of the effect of air injection in terms of energy imparted per second. P: injection pressure (bar), OD: Orifice diameter (mm)

Measure	Speed transient		Load transient	
	Air injection	Energy	Air injection	Energy
Performance Characteristics	P: 1 bar OD: 5 mm	99 J	P: 3 bar OD:15 mm	2.5 kJ
Emission Characteristics	P: 3 bar OD: 10 mm	2.7 kJ	P: 1.2 bar OD: 10 mm	446 J

Table 6.4 quantifies the effect of optimum value of air injection on performance and

emission characteristics under speed and load transient operating conditions. For practical considerations, its worth calculating the mass flow rate of the air corresponding to optimum air injection injection reported in chapter 5 and 6 and hence the energy imparted per second. From the table it can be depicted that optimum air injection is 1 bar at 5 mm orifice diameter and 1.2 bar at 10 mm for speed and load transient respectively as these injections correspond to minimum energy imparted per sec.

6.4 Summary of the chapter

The exhaust emissions CO, HC and NO_x are analysed for the effect of air injection technique under speed and load transients. Following conclusions are drawn from the investigations:

1. Optimum injection pressure is 3 bar at 10 mm orifice diameter that satisfies Euro 6 standards for CO and HC emissions under speed transient.
2. Air injection technique under speed transient doesn't increases the concentration of exhaust emissions. Rather, 5% and 0.4% reduction is observed in the concentration of CO and HC emissions respectively.
3. Optimum injection pressure is 1.2 bar at 10 mm orifice diameter for CO, HC and NO_x emissions under load transient.
4. In case of load transient also, air injection doesn't increase the concentration of the exhaust emissions. 0.8%, 0.01% and 0.4% reduction is noted in the concentration of CO, HC and NO_x emissions respectively.
5. In order to achieve overall optimization of the system using air injection, it is found that 1 bar air injection at 5 mm orifice diameter and 1.2 bar air injection at 10 mm are optimum for speed and load transient respectively.

7

CONCLUSIONS

7.1 Major conclusions and thesis contribution

Main conclusions are drawn from the previous chapters and the contributions of the thesis are presented in the following section. The work presented in this thesis focused on the techniques to improve the transient response of turbocharged diesel engine. The techniques were employed through modelling and their impact on the engine performance was investigated. Comparative analysis was performed to observe the relative improvement in transient response brought by these techniques. Based on the analysis, air injection technique was selected and its effect was further investigated for various transient operating conditions and exhaust emissions.

The results obtained in this research are the outcomes of extensive numerical investigations, validated through experiment. Detailed diagrams were shown for speed drop and recovery period of the engine's transient response with respect to different operating parameter. This presents quantification of weightage and effect of each term.

7.1.1 Development of a novel strategy for turbo lag reduction

The transient response of a turbocharged diesel engine was investigated using three main methods: torque assist, reduction in inertia of compressor and air injection. The effect of these methods were investigated over a wide range of steady state and transient operating conditions to quantify interaction of turbocharger with engines through turbo lag phenomena. Fueling rate change representing the transient event was chosen for modelling the effect of these three methods. No such strategy was taken into consideration in the literature for the reduction of turbo lag.

Implementation of torque assistance by simulation had been scarcely reported in the literature. Even though some experimental studies were previously conducted on torque assistance, Lotus Engine Simulation (LES) is used for the first time to investigate its impact on the transient performance. The investigations were performed to reduce the turbo lag using torque assistance and 0.16 Nm torque assistance was found to reduce the turbo lag considerably, thereby improving the transient response of turbocharged diesel engine. Another method of inertia reduction was also simulated using LES which benchmarks the result of the previous study. Marginal improvement was observed in the transient response of diesel engine when -10% inertia reduction is simulated. Finally air injection was simulated for fueling rate change event and a comparison was made between the results of the three methods. Air injection was found to bring maximum improvement in turbo lag reduction.

Due to the immense importance of diesel engine simulation modelling, the simulation based study using two advanced engine simulation software (Lotus Engine Simulation and Ricardo Wave) and the comparison between their results was one of the significant contribution of the present study.

7.1.2 Effect of air injection on the transient response of diesel engine under various operating conditions

Additional air injection at intake manifold significantly reduces the response interval of the turbocharger. This technology has paved way to carry out detailed investigations for a variety of test conditions. Effect of air injection on the response of heavy duty vehicles was scarcely reported. A wide range of operating conditions was considered to investigate the effect of air injection on the response of heavy duty turbocharged diesel engine. The parametric study had been undertaken to observe the response characteristics of the turbocharger under acceleration and load tests.

For acceleration tests, injection pressure was varied and the effect of varying injection pressure was observed on engine performance factors such as brake torque and brake power. These parameters were enhanced with injection pressure up to 2.5 bar. However no significant improvement was noticed after further increasing injection pressure (beyond 2.5 bar). Various tests were formulated to cover different operating conditions under acceleration tests. Optimum injection pressure was found depending upon the energy imparted with air injection. Turbo lag reduction/energy was calculated to find optimum value of air injection pressure for acceleration tests. 1 bar air injection was the optimum injection pressure with turbo lag reduction per unit energy as 0.290 per joule for compressor exit pressure, 0.392 per joule for turbine inlet pressure and 0.555 per joule for turbine inlet temperature (TIT). Quantitative achievement of turbo lag reduction per unit energy in this novel direction was itself a healthy contribution in the area of transient response improvement in turbocharged diesel engine.

For load tests, three different studies were conducted. The performance of the engine was investigated for constant load, load magnitude and load schedule. In the first study, the load was kept constant and a series of engine speeds were chosen based on the engine's operating range. Turbo lag reduction was observed for all the chosen speed range. Air injection brings maximum improvement in turbo lag reduction at lower speed range (1000-1400 rpm) while its effect on maximum attainable value was observed at higher speed range (1600-1800 rpm).

For the second and third study on load tests in Chapter 5, optimization of air injection was done based on the orifice diameter to regulate the mass flow of the air passing through the orifice. In case of load magnitude variation, air injection brings more improvement in lighter load application as compared to stronger load application. 50-70% load magnitude shows maximum improvement by air injection. Further, 3 bar air injection at 15 mm orifice diameter was more effective for 1 sec load schedule than for 2 sec. The results reveal that air injection into the intake manifold improves the turbo-lag under transient condition of load change.

7.1.3 Effect of air injection on environmental performance of the engine

A further noticeable effect of air injection technique was observed on the emission characteristics of the engine. Low speed transient - 1000 to 1700 rpm was taken into consideration and optimization of air injection was done to find the optimum value of air injection. The results of the simulation were evaluated for Euro 6 standard and 3 bar air injection at 10 mm orifice diameter was found to be the optimum injection pressure which brings slight reduction in CO and HC emissions and significant reduction in NO_x emissions. To evaluate the emissions for load tests, 50-70% load step in 1 sec was chosen and optimization was done in the same way as done for speed transient. Emissions were found to be below Euro 6 standard for all the injection pressures and orifice diameter. Quantification of emissions were done and 1.2 bar injection pressure at 10 mm orifice diameter was found to reduce all the three emissions (CO, HC and NO_x) as compared to no air injection.

It can be concluded from the study that although optimum injection pressure was different for different transient operating conditions, overall optimization was necessary. For acceleration tests, 1 bar air injection brings maximum improvement where as for the load tests 3 bar air injection at 15 mm orifice diameter was optimum. Looking at exhaust emissions while applying air injection technique, CO and HC emissions satisfies Euro 6 standard under speed transient with 3 bar air injection at 10 mm orifice diameter while 1.2 bar air injection at 10 mm orifice diameter was the optimum value of air injection for reduced emissions under load transients. For overall optimization of the system, energy imparted per second is calculated for performance and emission characteristics under the transient operating conditions of speed and load changes. Based on this, 1 bar air injection at 5 mm orifice diameter and 1.2 bar air injection at 10 mm are optimum for speed and load transient respectively.

The analysis of the effect of air injection at intake manifold on exhaust emissions clearly indicate that air injection while improving the transient response of turbocharged diesel engine, does not negatively affects the engine emissions in the case of speed or load transients. This indicates that the research has been undertaken in the appropriate direction.

7.2 Limitations and scope for future work

The impact of the technologies discussed in the thesis has been focused mainly on computational basis. The study can provide useful data for designing the real system. It would be worth considering the design of experimental set up based on the outcomes of this research. A dedicated test rig can be designed for the implementation of this technique based on the control strategies proposed in the simulation platform.

Furthermore, since the focus of the thesis was on improving the transient response of diesel engine, hence emissions during transients were observed only to make sure that air injection technique does not need compromise on emissions. Further studies can be performed to evaluate the effect of air injection by simulating a particular driving cycle to meet emission standards. This would allow cost estimation and hence feasibility to implement this technology.

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