



Experimental Investigation of the Transient Performance of a Turbocharged Compression Ignition Engine at Different Operating Conditions

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Abstract: In this work, an experimental investigation was conducted in order to study the transient performance of turbocharged compression ignition engine at different operating conditions. The experimental work was performed on a test rig comprising a four stroke 5.67 liters water-cooled turbocharged compression ignition engine and a Heenan hydraulic dynamometer. Instrumentation included devices for measuring engine speed, load, exhaust smoke level, exhaust temperature, fuel pump rack travel, turbocharger speed, and inlet air manifold pressure. The test facility was provided with a computer controlled data acquisition system to enable fast and accurate measuring and evaluating the engine response to sudden demand of increased load or speed. Two groups of tests were carried out. In the first group, the transient response of the engine to sudden acceleration demands was measured. During these tests, the dynamometer water flow rate was kept constant at pre-adjusted values namely 0, 10, 20, 30 and 50 % of the engine full load then the engine was accelerated from the initial speed of 1000 rpm to the target speed by quickly moving the fuel pump control lever forward to its maximum travel position. Measurement of the engine response to the sudden increase in dynamometer load was performed in the second group of tests. The fuel pump control lever was locked at its initial setting at initial speed of 2000 rpm and initial load of 20, 30, 40 and 50% from the engine full load then the load was increased suddenly to its maximum value. It is shown that the response time at which the engine speed reach to the desired value correspond to full fuel pump rack position increases with the increase of the initial brake torque. Very slow response was noticed at initial loading of 30 and 50% this due to the increase of resisting load quickly to a higher value which suppresses the engine speed increase due to the decrease of the surplus torque (engine minus resisting torques). During load acceptance tests it was shown that as the initial load decreases, 'harder' turbocharger lag period, lower air-fuel ratio, higher crankshaft deceleration and lower engine speed were obtained, Which initiate larger governor displacement which increase the amount of fuel injected to increase the fuel load to the desired full load conditions.

Keywords: Diesel engine, turbocharger, transient performance, load acceptance- acceleration

1. Introduction

One of the most important challenges facing the modern engine development engineers is to improve vehicle economy and reduce the levels of green house gas emissions. During the last three decades, turbocharging has largely contributed in developing diesel engines with higher specific power output, higher efficiency and reduced harmful emissions.

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Although turbocharging the diesel engine is beneficial, it is the turbocharged diesel engine that suffers way more than its naturally aspirated counterpart from poor transient response.

The most notable characteristic of transient turbocharged diesel engine operation is the turbocharger lag, which is caused by the lack of mechanical connection between turbocharger compressor and engine crankshaft. Consequently, the power delivered to the turbine must first accelerate the turbocharger shaft in order for the compressor to be able to produce increased boost pressure. The way the energy is transferred through the exhaust manifold, the use of fuel limiters, and the heat losses to the cylinder and exhaust manifold walls are few of the reasons for delaying transient response.

Transient response, especially of turbocharged diesel engines, forms a significant part of their operation and is often characterized by short but serious off-design functions [1]. Typical transient cases experienced by compression ignition engines, lasting from a few seconds up to several minutes, are load acceptance (change) with constant governor setting, speed change under constant load, simultaneous speed and load change and cold or hot start. In this paper, the term transient will be used to describe only changes in load at constant speed (load acceptance and speed change at constant load (acceleration)).

The importance of fast engine response to load and speed changes is highlighted by its reflection on the driver convenience, economy and environmental conditions. Typical applications suffering from turbocharger poor performance under changing load or speed conditions are:

- Diesel electric generating sets where very rapid loading of the alternator or generator can occur.
- Marine engines, during maneuvering.
- Turbocharged trucks during acceleration.

Traditionally, the study of internal combustion engines operation has focused on the steady-state performance, with minor, if any, attention paid to the unsteady state or more accurately termed transient operation. However, the majority of daily driving schedule involves transient conditions. In fact, only a very small portion of a vehicle's operating pattern is true steady-state. Historically, however, the research on transient diesel engine operation was initiated from the observation by engine manufacturers in the 1960s that when highly-rated, medium-speed diesel engines are employed in sudden 0–100% step load changes, severe difficulties are encountered, even leading to engine stall [1]. In recent years, it is the global concern about environmental pollution that has intensified the respective studies; particulate, gaseous and noise emissions typically go way beyond their acceptable values following the extreme, non-linear and non steady-state conditions experienced during dynamic engine operation.

Considerable work has been done dealing with the transient characteristics and influencing factors for the diesel engine with a turbocharger such as acceleration performance, starting stability, sudden large load, and the speed range of the engine. Most of previous papers have described the simulation studies of the matching of turbocharger and transient performance of turbocharged diesel engine response. Different techniques use to simulate turbocharged diesel engine under transient conditions. The models used were; linear or sampled data modeling [2,3], Quasi-linear models [4,5]. Filling and emptying models [6:9] and wave action model [10,11]. Experimental studies were used to measure the transient response of turbocharged diesel engines, validate the developed models [12:16], others to evaluate different techniques used to improve engine response [17:22].

The purpose of this work was to experimentally investigate the transient performance of the turbocharged diesel engine during rapid acceleration and applying sudden load regimes at different operating conditions.

2. Experimental Setup

The experimental work was conducted on a complete rig (available in the laboratory of mechanical power and energy at the Military technical college for testing naturally aspirated as well as turbocharged diesel engines. The test rig includes the engine and all the instrumentation necessary for measuring and recording the operating parameters. An on-line data acquisition system is furnished to improve the speed and accuracy of data collection and recording. A transport diesel engine of type Mercedes-Benz with an open chamber is used. This is a four stroke 6-cylinder with 97 mm bore, 128 mm stroke, and 17: 1 compression ratio. Detailed engine specifications are given in Tables (1-2)

Engine external loading was carried out by an ELZE /Heenan hydraulic dynamometer. The fluid used was water with which the maximum breaking power could reach 170 kW at 4000 rpm. The engine and dynamometer shafts were directly coupled through a cardan shaft.

Table (1) Engine Technical Data

Model	Mercedes 352 series, Natural aspirated Diesel engine
Compression ratio	17:1
No. of Strokes	4
No. of cylinders	6
Arranging	In-line
Cooling	Water
Bore	97 mm
Stroke	128 mm
Combustion chamber	Open type, Direct Injection
Cam shaft	Sided
Speed range	800-2800 rpm
Maximum power	120 HP at 2800 rpm
Maximum torque	28 kp.m at 1600 rpm
Static injection	23 CABTDC
Firing order	1536241
Min. compression pressure	20 bar at 150-200 rpm
Injector opening pressure	200 bar

Table (2) Turbocharger Data

Type	UK PAT, Air research Turbochargers Division
Compressor Data:	
Inlet diameter	62 mm
Outlet diameter	40 mm
Impeller type	Backward with splitter vanes
Number of vanes	16
Impeller eye diameter	44 mm
Turbine Data:	
Inlet diameter	64 mm
Number of entry	Double
Number of vanes	14

The test rig is fully instrumented in order to acquire experimental data as well as to monitor the engine operating conditions. Figure (1) gives a general scheme of the complete test rig showing numbered locations where important pickups and transducers are positioned. A list of these locations and the corresponding measured parameters is given in Table (3).

**Table (3): Measurement locations and the parameters
(Relevant to Figure (1))**

Location	Measured Parameter	Measuring device
1	Engine speed	Shaft encoder
2	Brake torque	Hydraulic dynamometer equipped with load cell
3	Boost pressure	Pressure transducer
4	Turbocharger speed	Photo-electric sensor
5	Injection pump rack position	Linear voltage displacement transducer (LVDT)
6	Smoke meter	Photo-electric technique
7	Exhaust temperature	Thermo couple

The Data Acquisition System consists of three main elements, national instrumentation BNC-2120 connector panel, A national instrumentation PCI-MIO-16E-1 data acquisition card and a PC Computer with data acquisition software (Lab VIEW). . The data acquisition card is used to collect the measured data from the measuring instruments through the BNC connector panel either analog or digital directly or after amplification and converts the analog inputs to digital data which is then recorded by the computer under soft ware control.

3. Test procedure

The performance of the turbocharged diesel engine under transient operation was experimentally evaluated. Two groups of tests were carried out. In the first group, the transient response of the engine to sudden acceleration demands was measured. Measured parameters shown in Table (1) were fed to the data acquisition system which is operated and controlled via the computer facility. Lab View software was used in order to acquire and record the historical variation of each measured parameter during the transient operation of the engine. Data were thus recorded for a period of approximately 30 seconds from the beginning of each test.

3.1 Acceleration Tests

The engine is warmed up for a sufficient period of time until normal operating conditions are reached. Engine speed is then adjusted to 1000 rpm at the required initial dynamometer reading, namely 0, 8, 16, 24 and 40 which corresponds to 0, 10, 20, 30, and 50 % of engine maximum load. When the start test signal is given, the engine is accelerated by quickly pushing the fuel pump control lever forward. At the same instant, the measurement procedure is triggered in order to start data logging and recording. When data acquisition is finished, recorded values are reviewed and then saved for later analysis.

3.2 Load Acceptance Tests

After warming up, the engine speed is adjusted to 2000 rpm with the dynamometer reading fixed at the required initial value, namely 16,24,32 and 40 which correspond to 20, 30, 40, or 50 % of the engine maximum load. When the start test is signaled, the dynamometer load is

suddenly increased by quickly opening the dynamometer water flow tap. At the same instant, the data acquisition system is triggered, recorded data during each test, are saved in a separate file for analysis and comparisons as discussed in the following sections.

4. Results and Discussions

The fundamental difference between acceleration and a load acceptance transient event lies in the load torque profile and the speed governor behavior [1]. During load acceptance transient tests, resistance torque increases abruptly; the significant torque deficit that is established during the first cycles of the transient event is responsible for the initial drop in engine speed as well as for the response of the speed governor, engine fueling and torque up to the final equilibrium at the same or lower engine speed. On the other hand, during acceleration, load torque remains practically constant or increases moderately following engine speed, while the governor reaches almost instantly its maximum fueling position, remaining there for most of the transient event; the fuel limiting function determining the exact fueling response. The function of fuel pump rack limiter which is sensitive to boost pressure is to control fuel input in relation to air flow during engine acceleration to limit exhaust smoke emission by preventing air to fuel ratio from becoming excessively rich.

It should be mentioned that due to the type of the dynamometer used (hydraulic type), no control over the brake torque could be exercised. In other words, the dynamometer brake load increase due to the increased hydraulic eddies during the engine acceleration period. The resulting extra brake torque unfavorably impedes the engine acceleration ability. This problem is aggravated at faster engine response, which makes the comparative study presented in this work more difficult. Employing eddy current (electrical, relatively easy to control) dynamometer, gives better capabilities in fixing the transient brake torque and therefore obtaining more vivid conclusions. However, increasing the resisting load during acceleration is found in many practical situations such as vehicles which are derived by turbocharged diesel engines.

4.1 Acceleration Test Results

Figure (2) shows the results of acceleration tests. Prior to each test, the engine speed was set to 1000 rpm at the pre-required brake load of 54 N.m which is corresponding to 20% from the engine full load. At the initial, steady-state operating conditions, engine and load torques are in equilibrium at (57% from full fuel pump rack travel as shown in figure 4.1a). After the accelerator pedal was pushed, the governor control lever reaches almost immediately its maximum position corresponding to full-fueling (100% from fuel pump rack travel) in almost 0.7 sec as shown in figure (2.a). The significant surplus of net (engine minus load) torque aids rapid build-up of engine speed, boost pressure and turbocharger speed. The displayed results show that the engine accelerates from 1000 to 2800 rpm in about 13 seconds as shown in figure (2.b).

During this period the turbocharger speed sharply increases from about 12000 to 50000 rpm in 8 sec due to the increase of fueling with much sharper rate than air-flow then the equivalence ratio reaches its maximum value. After that the turbocharger speed drops to the final steady value of 42000 rpm after 10 sec, as shown in figure (2.c), as the fuel pump rack limiter control the amount of fuel injected with the consequent decrease in fuel to air ratio. Variation of the intake manifold pressure is closely connected to the changes in the turbocharger speed as shown in figure (2.d).

In the same time the resistance torque increases moderately, load torque increases moderately to its maximum value of about 180 N.m in about 3 sec owing to the increase in engine speed,

hence, hydraulic resistance of the rotating part in the hydraulic dynamometer. Then decreases to about 80 N.m due to decrease of the fuel air ratio. As a result, the final operating condition at 2800 rpm, achieved after 18 sec, corresponds to a higher loading than the initial one as shown in figure (2.e)

High amounts of smoke emissions are noticed during the early cycles of the acceleration (from about 5% to 75%) as shown in figure (2.f), owing to the initially low air–fuel ratio as fueling has increased much more rapidly than air-supply. In fact, full-fueling has been established after 0.7 s, whereas boost pressure needs at least 10 to reach its maximum value. During much of this period the engine practically operates as naturally aspirated.

Figures (3 to 8) show the acceleration tests results at different initial brake torque namely 0, 27, 54, 81, and 135 N.m corresponds to 0, 10, 20, 30 and 50 % of the engine full load.

Figure (3) shows the development of the rack position with time for different initial brake torques. Accelerations tests were carried out by pushing the governor control lever manually. The acceleration time is a factor which may affect the transient performance of turbocharged diesel engine so the acceleration time should be kept constant at different test which is clear in the figure from the parallel line representing the rack position during acceleration.

The development of engine speed with time for different initial brake torques is shown in figure (4). It is shown that the response time at which the engine speed reach to the desired value correspond to full fuel pump rack position increases with the increase of the initial brake torque. This can be attributed to the deficit of the surplus engine torque relative to the load resisting torque. Very slow response was noticed at initial loading of 30 and 50% this due to the increase of resisting load quickly to a higher value, which suppress the engine speed increase due to the decrease of the surplus torque (engine minus resisting torques).

Figure (5) shows the development of the resisting torque with time when accelerating from 1500 rpm to the target of 2800 rpm. It was found as the initial load increase the torque increases to a higher value of load this may be due to the increase of the amount of fuel injected at higher load and then before the aneroid limit the fuel to air ratio. This also increases the turbocharger speed to a higher maximum value before it drops again to its steady state value corresponding to the final speed and load as shown in figure (6). Variation of the intake manifold pressure is closely connected to the changes in the turbocharger speed as shown in figure (7).

The smoke level is shown to increase suddenly to a peak value at initial loading of 0, 10 and 20 %. As shown in figure (8) owing to the initially low air–fuel ratio as fueling has increased much more rapidly than air-supply and then drops to its steady state value due to the action of the fuel pump rack limiter which limit the increase of fuelling until sufficient air is supplied. At higher initial loading the smoke level increases continuously until reach its steady state value this can be attributed to the increase of the amount of fuel at initial loading and small travel of the rack position and then the effect of fuel pump limiter.

The key parameters in the above results is that affect the whole engine behavior during acceleration test are the resisting load characteristics and the fuel pump rack limiter characteristics as well as the design parameter mentioned before such as total mass moment of inertia of the whole engine and turbocharger, manifold volume, and the valves timing which are kept constant during tests.

4.2 Load Acceptance Test Results

Figure (9) displays the results of load acceptance test when initial speed was set at 2000 rpm, and initial brake torque was set to 75 N.m which corresponds to 30% of the engine full load conditions.

At the initial conditions, the engine and load (resistance) torque are equal and the air–fuel ratio is relatively high due to the low loading. As soon as the new higher load is applied (this is accomplished in 8 s as shown in figure 4.9a), there is a significant deficit in the net (engine minus load) torque, since the engine torque cannot instantly match its increased load counterpart, and hence the engine speed drops as shown in figure (9.b). This is sensed by the governor sensing element, which, in turn, shifts the fuel pump rack towards a position of increased fueling. As the engine torque increases and starts to overcome the brake load, the fuel rack is pulled back to decrease the fuelling and limit the engine developed torque. The turbocharger speed increases with the increase of fuelling and then decreases to its steady state value as shown in figure (9.c). The intake manifold pressure increases as the turbocharger speed increases as shown in figure (9.d).

At the same time, the air–fuel ratio decreases since the air–supply cannot instantly match the higher fueling owing to the delayed response of the turbocharger in building-up the required delivery pressure. The smoke level increases from about 30% to 80% as shown in figure (9.e). At the same time, the increased gas temperature owing to the low air–fuel ratios is also noticed as shown in figure (9.f).

Figures (10 to 15) show the load acceptance test results at different initial brake torque namely 54, 81, 108 and 135 N.m corresponds to 20, 30, 40 and 50 % of the engine full load. The magnitude of the applied load change plays a primary role in engine response, mainly as regards maximum and final speed droop (Figure (10)). At the initial steady-state conditions, the engine and load torques are equal and the air–fuel ratio is high due to the low-loading. As soon as the new, higher load is applied, a significant deficit is observed in the net (engine minus resistance) torque, since the engine torque cannot instantly match its increased load counterpart, so that engine speed drops. Obviously, the lower the initial load (The final load is constant for all cases), the higher this torque deficit during the early cycles of the transient event. This, in turn leads to a ‘harder’ turbocharger lag period, lower air–fuel ratio, higher crankshaft deceleration and lower engine speed. This is clear in figure (10) which shows that the engine speed drops from 2000 rpm to 1200 rpm in about 10 seconds when the engine initial load is 20% from its full load (Brake torque of 54 N.m) while it takes about 4 seconds to drop from 2000 rpm to 1600 rpm when the initial load is 50% from its full load condition.

Figure (11) shows that the engine load reach its full load condition when the initial load was preset at 20, 30, 40 and 50% fro its maximum load. However the time taken to reach its maximum load increases as the initial load decreases owing to increases of the turbocharger lag.

As the initial load decreases the speed drop increases which initiate larger governor displacement which increase the amount of fuel injected to increase the fuel load to the desired full load conditions. Then the turbocharger speed increases from 25000 rpm to 35000 at initial load of 20% while it increases from 34000 to 37000 when the initial load is preset to 50% from the engine full load conditions shown in figure (12). The boost pressure is shown to follow the behavior of the turbocharger speed as shown in figure (13).

The smoke level reaches a steady state value of about 80% at all initial loading starting from 15, 30, 40 and 60% at 20, 30, 40 and 50 % initial load respectively. The time duration to

reach steady state is almost the same for all conditions which reveals higher rate of smoke level increase at lower initial loading as show in figure (14). This due to the more fuel injected at small loading for the engine to reach the engine full load condition.

Figure (15) shows the development of exhaust temperature for different initial loading. The time needed to reach steady state value is longer at low initial loading of the engine and this explain why the turbocharger lag increase at low initial loading.

The key parameters in the above results is that affect the whole engine behavior during the load acceptance tests are the specific governor characteristics as well as the design parameter mentioned before such as total mass moment of inertia of the whole engine and turbocharger, manifold volume, and the valves timing which are kept constant during tests.

Conclusions

In general, turbocharger lag, the torque and speed response of a turbocharged diesel engine, is a complex phenomenon that is influenced by a variety of dynamic, thermodynamic and design parameters, e.g., engine dynamics (moment of inertia, manifolds volume), governor characteristics, turbocharger dynamics, match, configuration and turbine nozzle area, engine fueling and valve timing; So it is very difficult to compare results quantitatively with respect to response time during any change in engine speed and load even they have the same power rating at the same operating conditions.

The key parameters in the above results concerning acceleration test are the resisting load characteristics and the fuel pump rack limiter characteristics which are affected dramatically by the initial load at which the engine was preset. It was found that the response time at which the engine parameters reach to the desired value correspond to full fuel pump rack position increases with the increase of the initial brake torque. This can be attributed to the deficit of the surplus engine torque relative to the load resisting torque. Very slow response was noticed at initial loading of 30 and 50%. Also, at higher initial loading the smoke level increases continuously until reach its steady state value due to the increase of the amount of fuel at initial loading and small travel of the rack position and then the effect of fuel pump limiter.

The key parameter in results concerning load acceptance tests is the specific governor characteristic which is affected dramatically with step increase on load. The lower the initial load (The final load is held constant for all cases), the higher the torque deficit (engine torque minus resisting torque) during the early cycles of the transient event. This, in turn leads to a 'harder' turbocharger lag period, lower air-fuel ratio, higher crankshaft deceleration and lower engine speed. A higher rate of smoke level increase at lower initial loading is noticed. This due to the more fuel injected at small loading for the engine to reach the engine full load condition.

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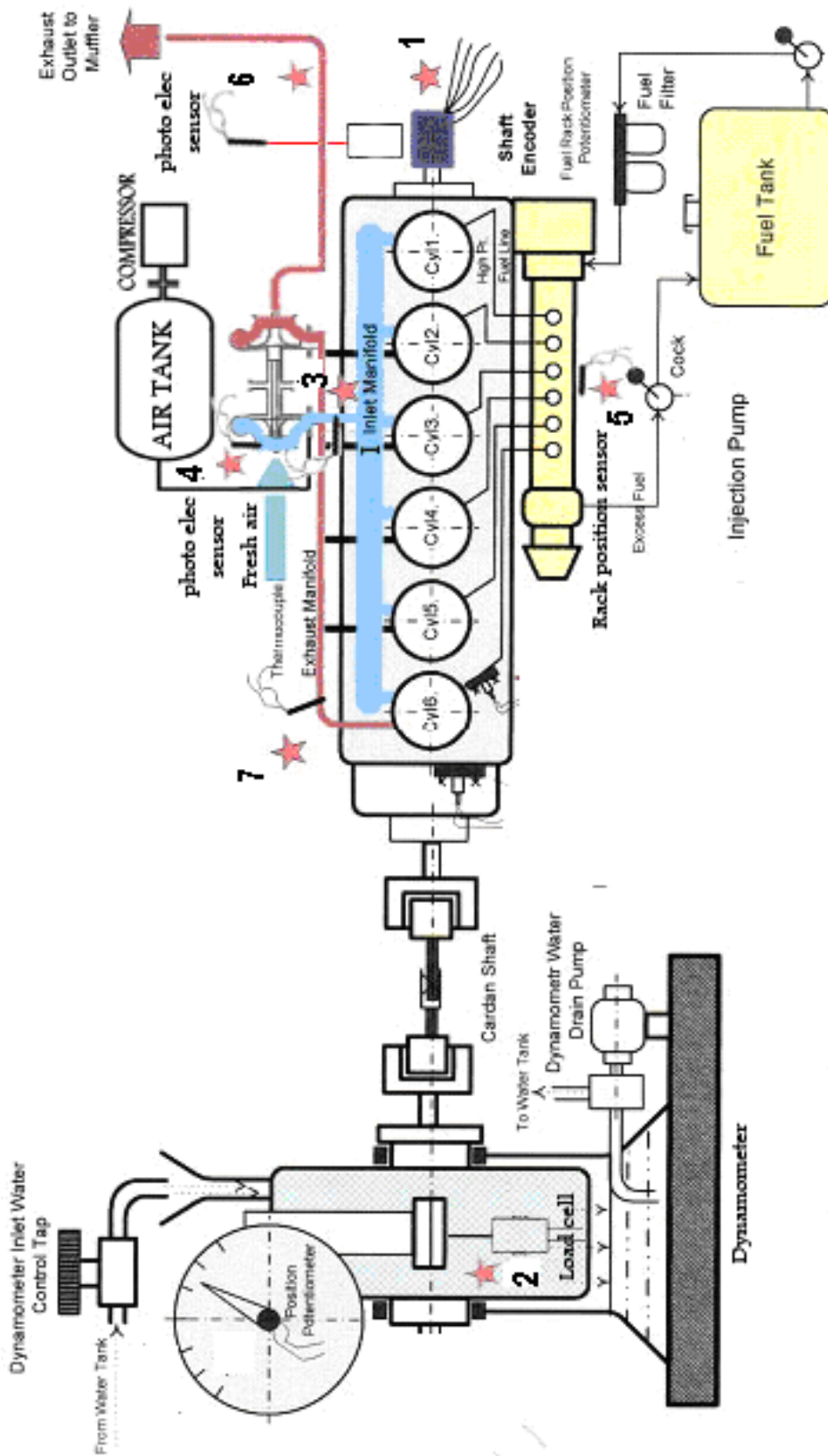
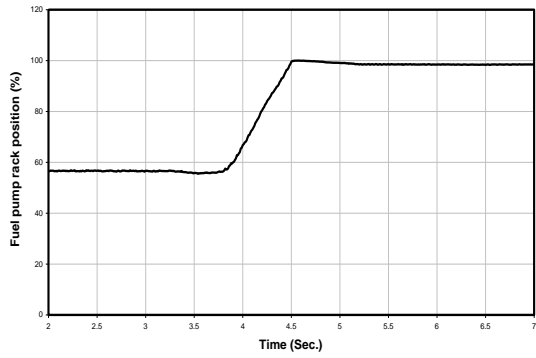
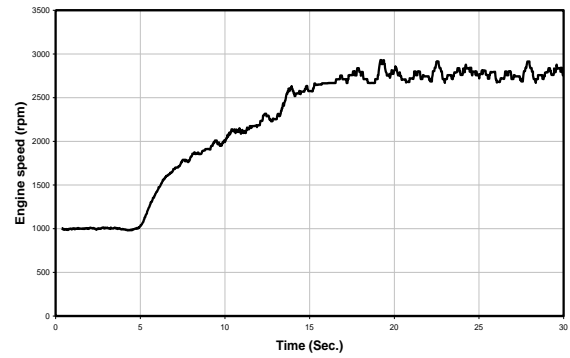


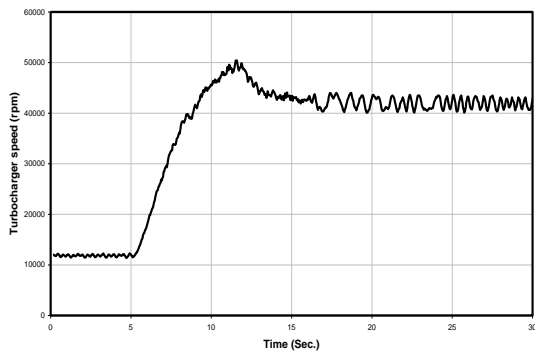
Figure (1) Schematic of the Test Rig,
(Numbers indicate the points of measurement of the i^{th} parameter)



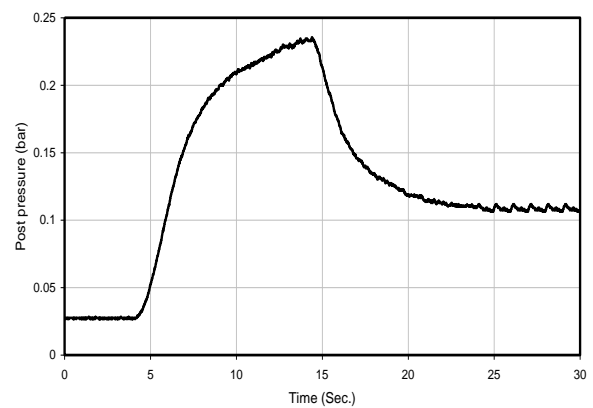
(a): Fuel pump rack position



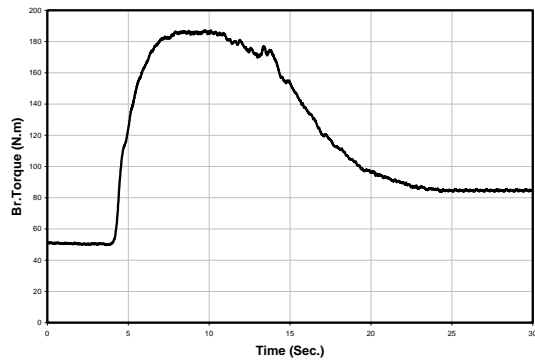
(b): Engine speed



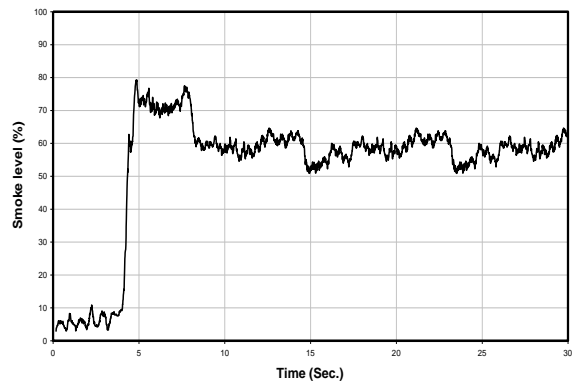
(c): Turbocharger speed



(d): Boost pressure

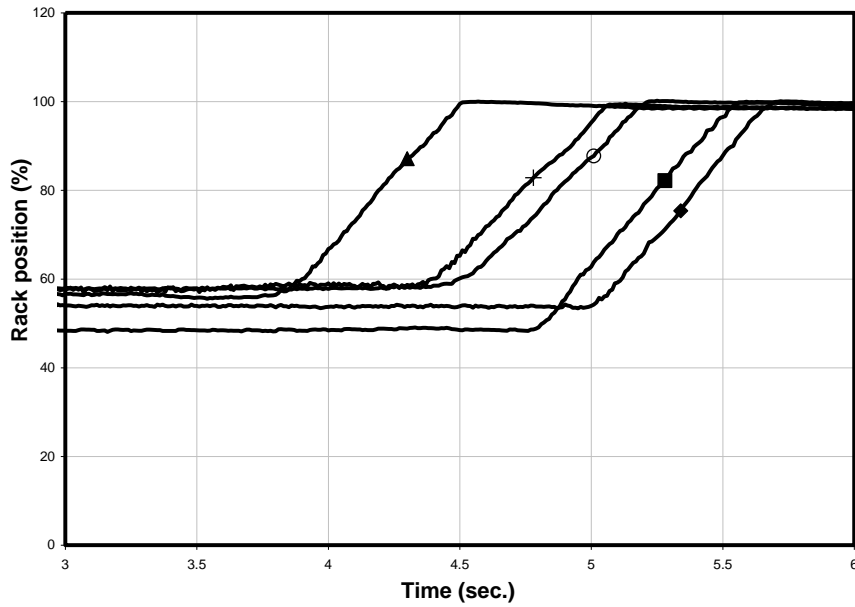


(e): Brake torque



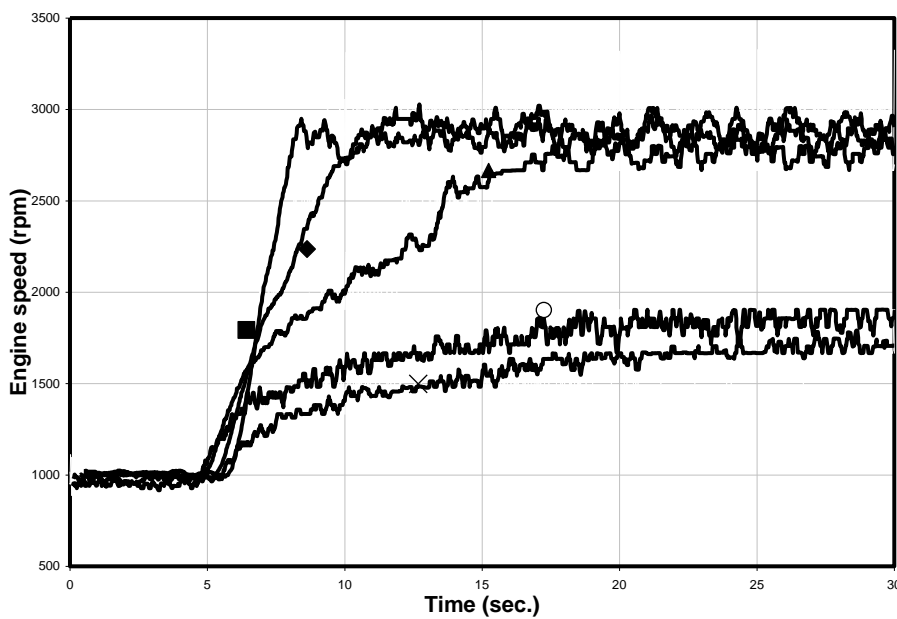
(f): Exhaust smoke level

Figure (2): Development of the turbocharged diesel engine performance during a sudden acceleration from 1000 to 2800 rpm at initial load of 20% from the engine full load



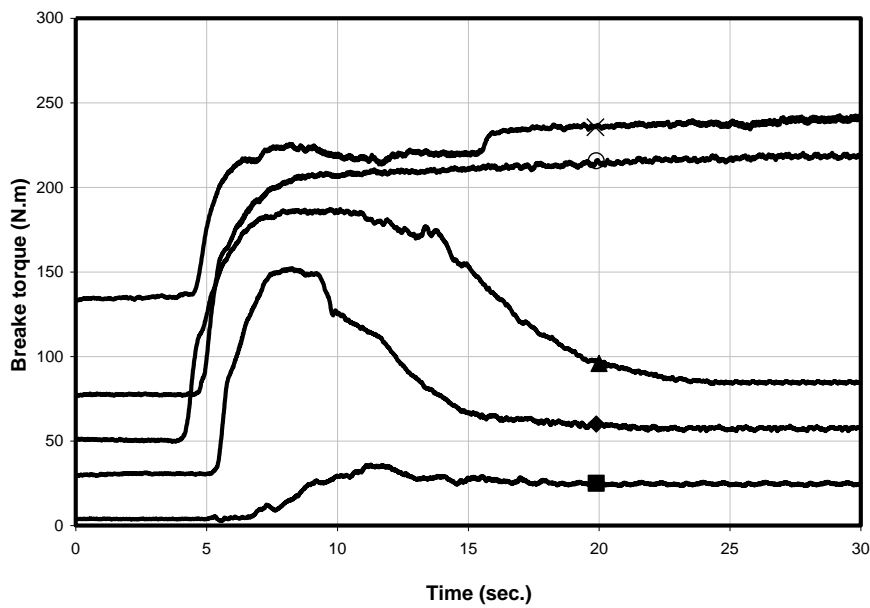
■ 0% initial load ◆ 10% initial load ▲ 20% initial load ○ 30% initial load × 50% initial load

Figure (3) Development of the rack position with time for different initial brake torques during a sudden acceleration.



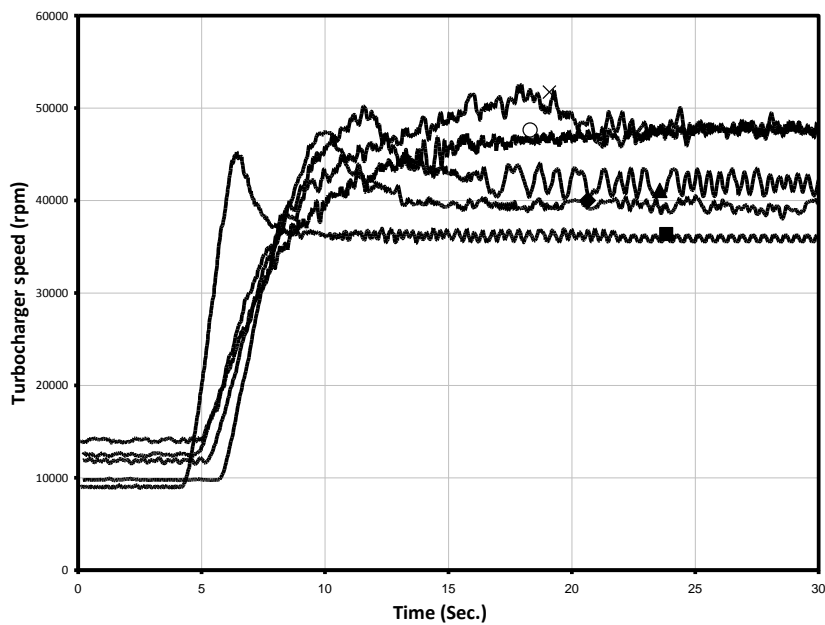
■ 0% initial load ◆ 10% initial load ▲ 20% initial load ○ 30% initial load × 50% initial load

Figure (4) Development of engine speed with time for different initial brake torques during a sudden acceleration



■ 0% initial load ◆ 10% initial load ▲ 20% initial load ○ 30% initial load × 50% initial load

Figure (5) Development of the resisting torque with time when accelerating from 1500 rpm to the target of 2800 rpm



■ 0% initial load ◆ 10% initial load ▲ 20% initial load ○ 30% initial load × 50% initial load

Figure (6) Development of the turbocharger speed with time for different initial brake torques during a sudden acceleration

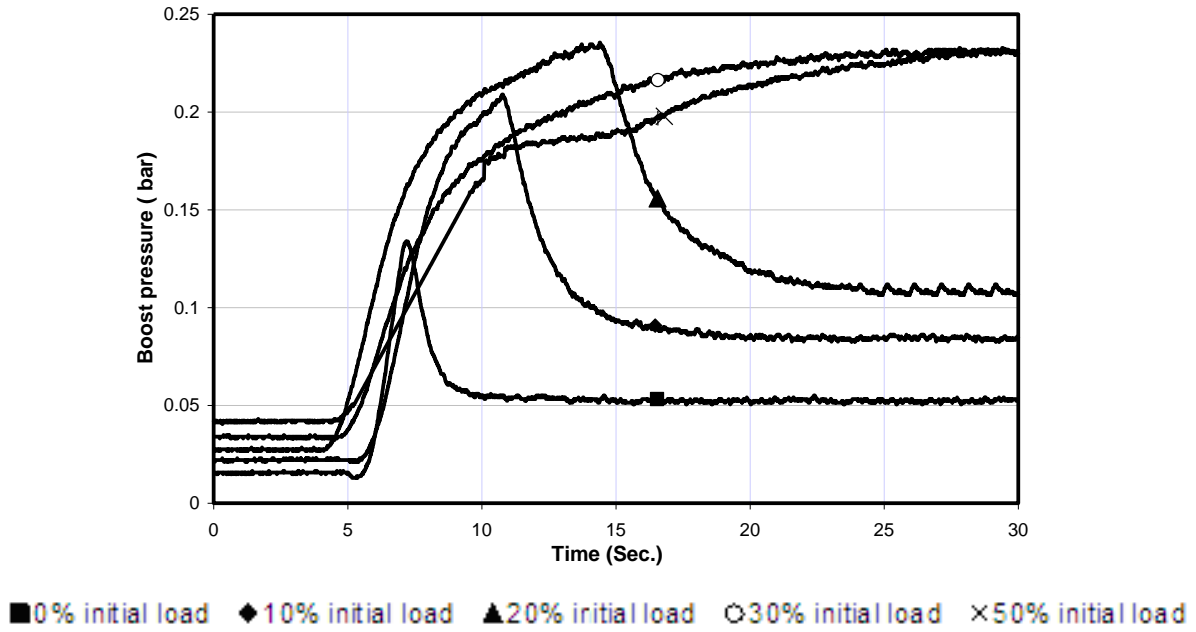


Figure (7) Development of the boost pressure with time for different initial brake torques during a sudden acceleration

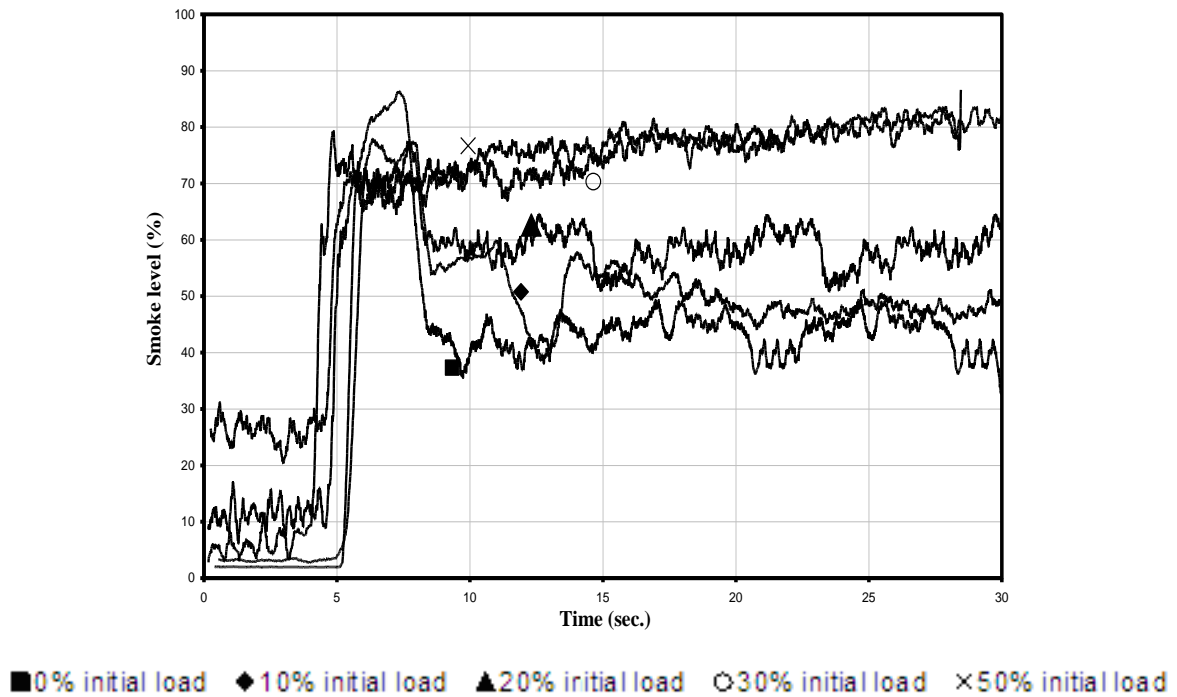
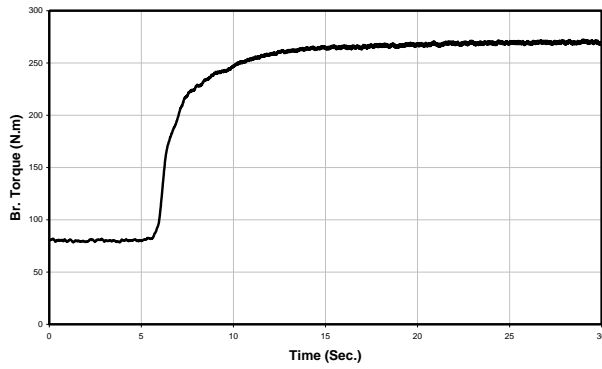
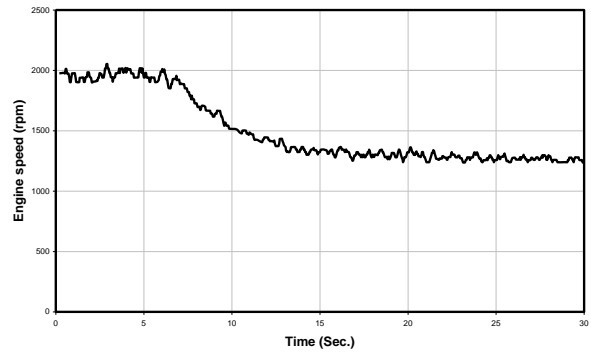


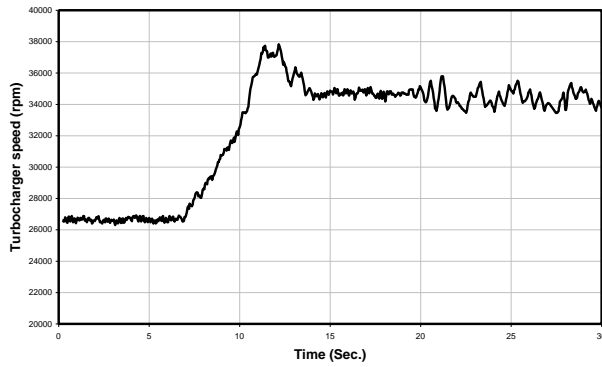
Figure (8) Development of the smoke level with time for different initial brake torques during a sudden acceleration



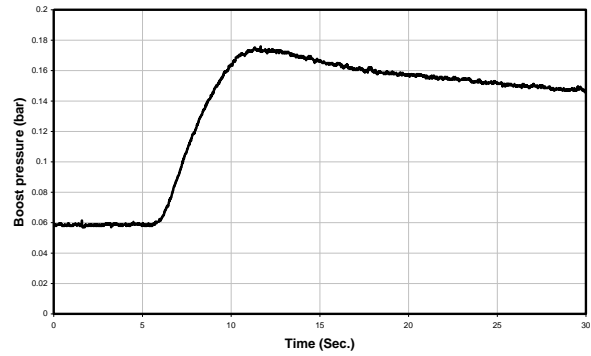
(a): Brake torque



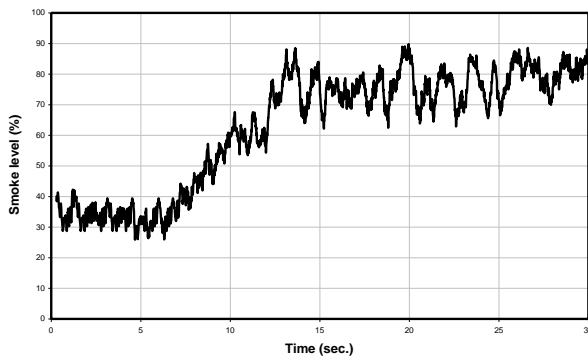
(b): Engine speed



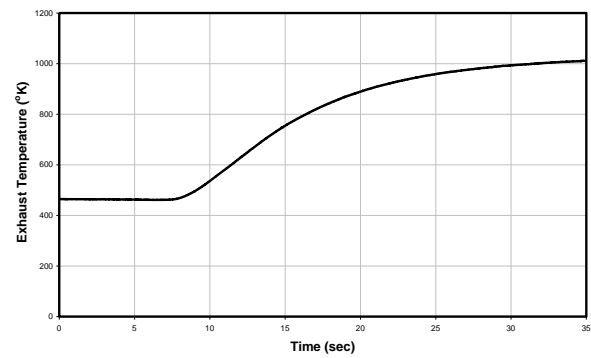
(c): Turbocharger speed



(d): Boost pressure



(e): Exhaust smoke level



(f): Exhaust temperature

Figure (9): Development of the turbocharged diesel engine performance during load acceptance test from initial brake torque of 81 N.m to engine full load condition

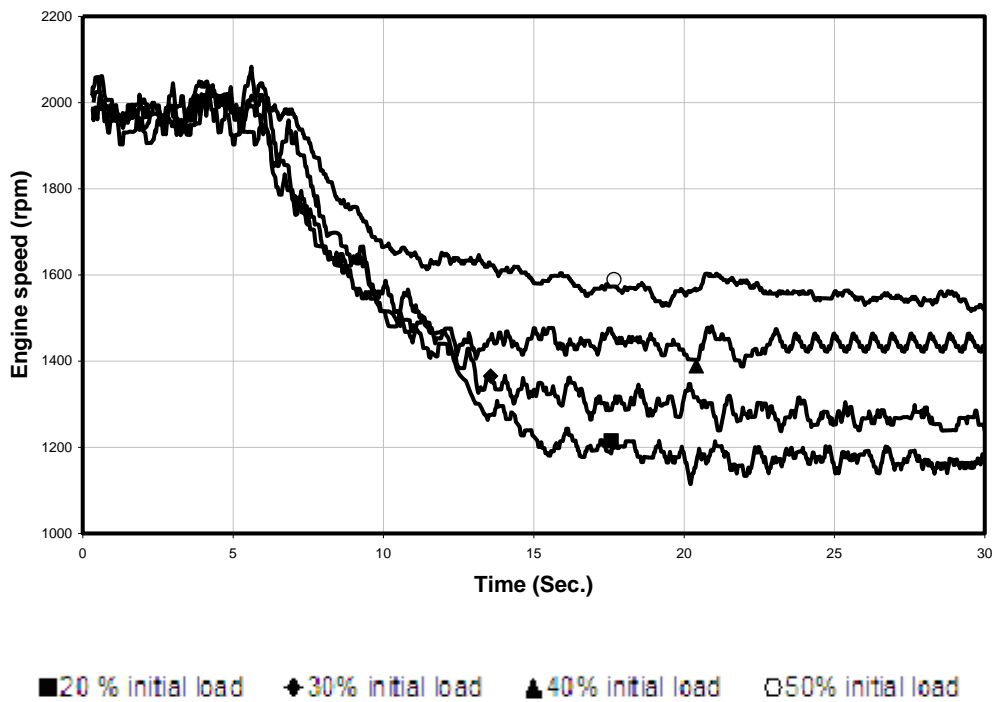


Figure (10) Development of engine speed with time during a load acceptance test at different initial brake torques

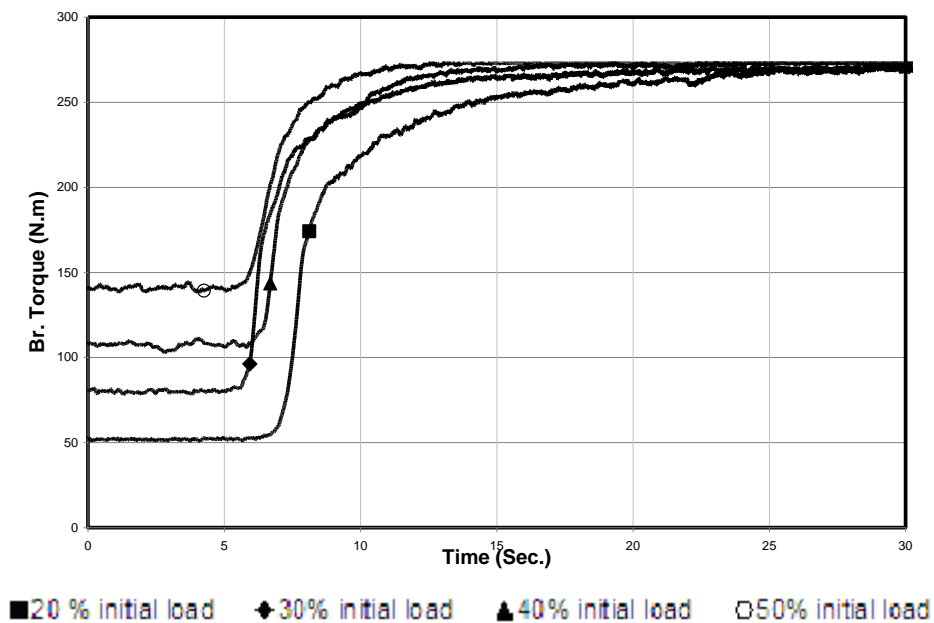


Figure (11) Development of engine brake torque with time during a load acceptance test at different initial brake torques

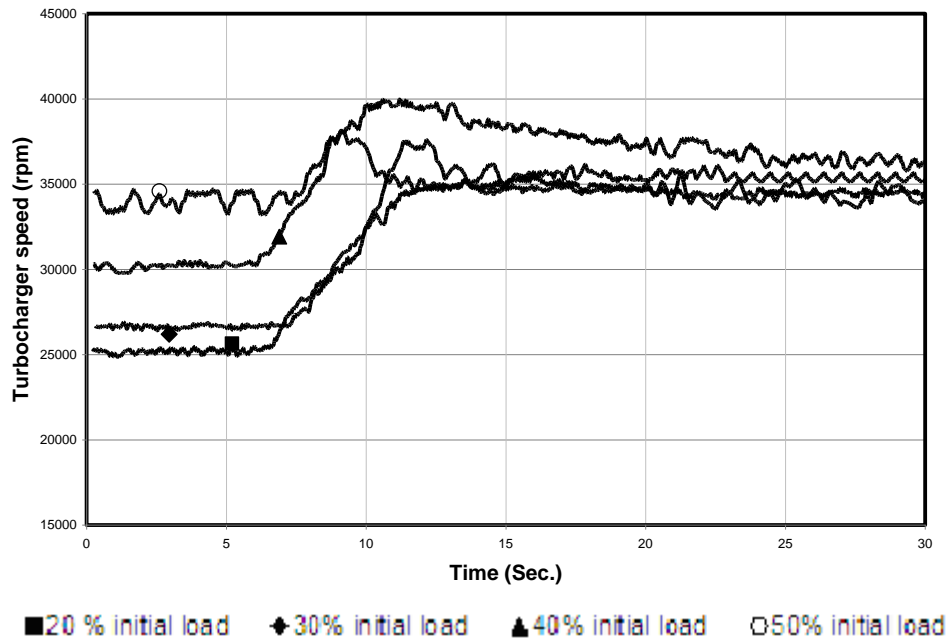


Figure (12) Development of turbocharger speed with time during a load acceptance test at different initial brake torques

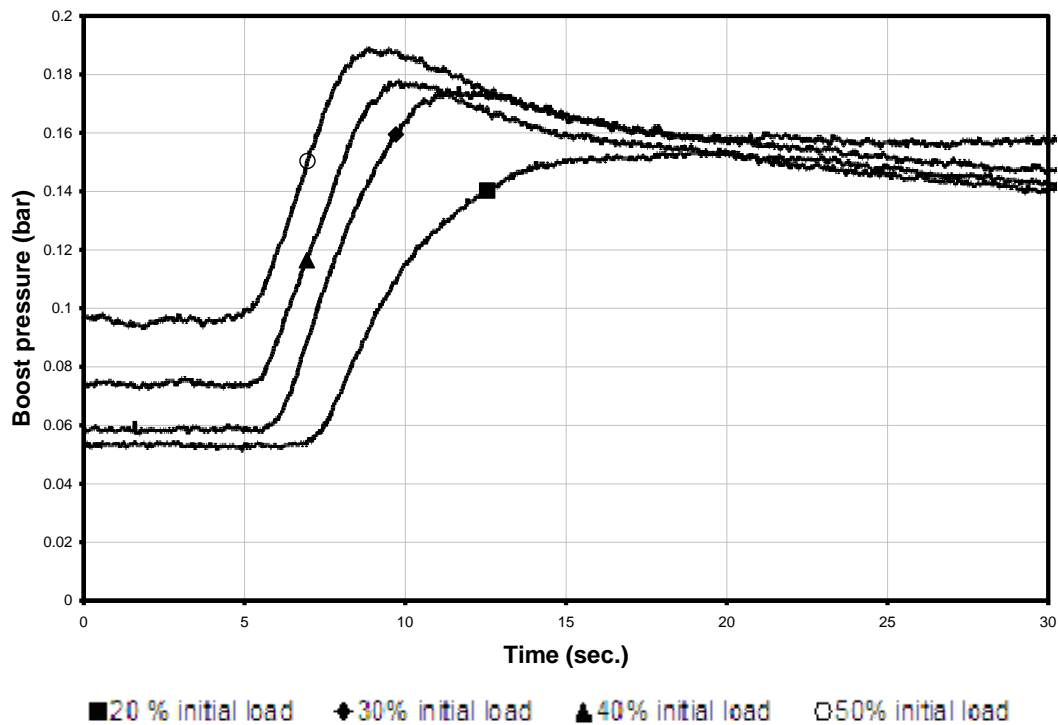


Figure (13) Development boost pressure with time during a load acceptance test at different initial brake torques

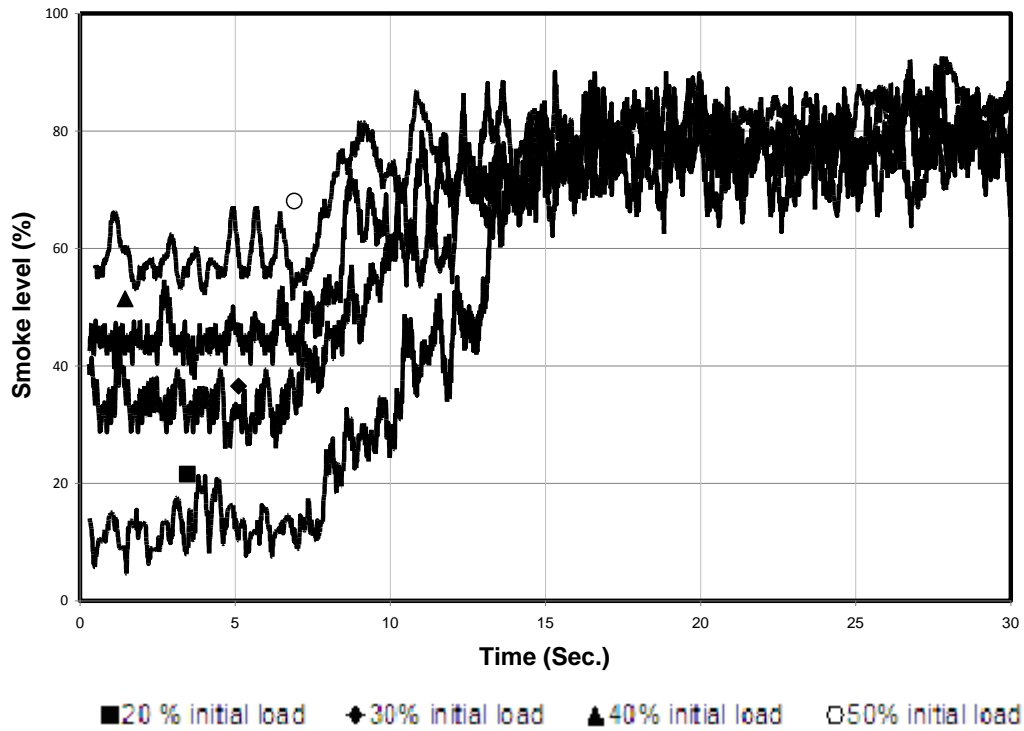


Figure (14) Development exhaust smoke level with time during a load acceptance test at different initial brake torques

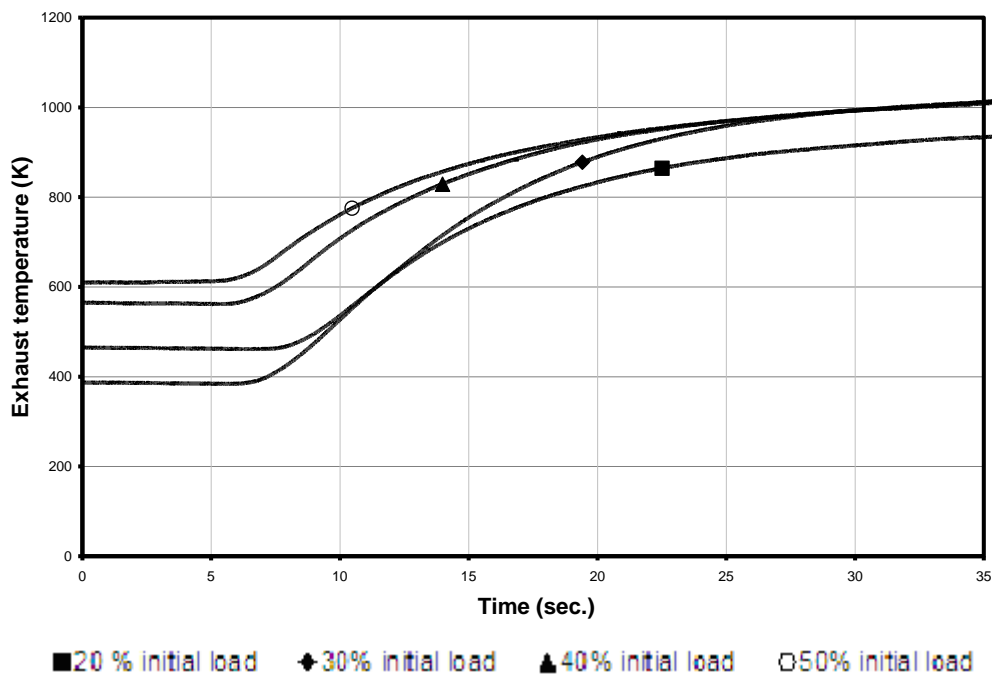


Figure (15) Development exhaust temperature with time during a load acceptance test at different initial brake torques