NVH Characteristics of Air Assisted Direct Injected (DI) Spark Ignition Four Stroke Engines

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SYNOPSIS

The ability to run unthrottled or with partial throttling at part load combined with stratified combustion potentially gives direct injection (DI) gasoline engines significant fuel consumption reductions. These combustion changes alter the timing, amplitude and frequency content of the cylinder pressures. This paper examines the NVH changes on four stroke gasoline engines due to DI with an emphasis on Orbital's air assisted system. A combustion noise model which calculates direct and indirect noise from cylinder pressure data is discussed. The application of DI to four stroke engines requires specific subsystems and the effects of these are also reviewed.

1. INTRODUCTION

Environmental pressures including the need to reduce greenhouse gases has caused unprecedented interest in the application of direct injected four stroke (DI4S) technology to conventional gasoline automotive engines. Direct injection (DI) is regarded as the next major step forward in the evolution of automotive powerplants and it can provide significant advantages in fuel economy, emissions and performance. In particular gasoline DI allows engines to run stratified combustion and minimal throttling or no throttling at part load. This results in very lean operation and reduced levels of pumping work losses.

The two basic type of gasoline DI systems are:

- Orbital Combustion Process (OCP) low pressure, air assisted DI system, and
- high pressure, single fluid systems (HPDI).

This paper examines the noise vibration and harshness (NVH) changes on a generic four stroke engine due to the application of direct gasoline injection with a particular emphasis on

Orbital's air assisted DI system. This paper provides an overview of Orbital's Combustion Process (OCP) and shows that the NVH effects are known, predictable and manageable.

The application of an empirical noise model of a generic four stroke engine that predicts the effect of combustion characteristics on overall engine noise is described. This model permits an overall engine noise comparison of combustion systems including the relative combustion and mechanical noise contributions. The overall engine noise for an OCP DI process is compared directly to that of the generic four stroke engine using the empirical model developed.

The application of DI to four stroke engines requires specific subsystems and the noise and vibration aspects of OCP system components are reviewed, e.g. air intake, air/fuel injectors and air compressor.

2. BRIEF OCP SYSTEM DESCRIPTION

The Orbital DI combustion process has been developed and refined over a period of 12 years, and has been applied to two stroke applications for automotive, motorcycle and marine use.

At the centre of the OCP system is a solenoid actuated air/fuel charge injector that delivers a finely atomised cloud of gasoline directly into the combustion chamber. The fuel is metered and delivered to the air injector by a conventional manifold fuel injector. Figure 1 illustrates a typical development installation for an automotive four stroke application.

The fuel injector first injects a precisely controlled quantity of fuel into an air chamber that is at the top of the air / charge injector. The air injector is then used to deliver the air and fuel mixture (typically at 6.5 bar) directly in to the combustion chamber as a finely atomised cloud that is immediately ready for combustion.

The compressed air required for the injection process is supplied by a small (30 cc to 40 cc for a four cylinder engine) conventional single cylinder reciprocating compressor that is driven off the engine accessories drive belt. The fuel is supplied by conventional fuel pumps.

3. REVIEW OF ENGINE NOISE MECHANISMS

3.1 Overall

Radiated engine noise due to engine structural vibration can generally be considered to be comprised of combustion noise, primary mechanical noise and secondary mechanical noise. Ancillary components can also be significant noise sources at some operating conditions.

Combustion noise, often termed direct combustion noise, results from the cylinder pressure exciting the engine structure directly causing it to vibrate and radiate noise (1, 3, and 5). Mechanical noise is the noise due to parts moving across a clearance causing an impact. Mechanical noise associated with impacts due to the crank mechanism including pistons, connecting rods and the crankshaft is termed primary mechanical noise. Primary mechanical noise is controlled by the balance of gas forces and inertia forces acting on components as they move across the clearance and impact. Primary mechanical noise is load and speed dependent. The gas force component of primary mechanical noise is frequently referred to as indirect combustion noise or combustion controlled mechanical noise. The inertia component is referred to as inertia controlled mechanical noise.

Secondary mechanical noise is considered to be noise not associated directly with the crank mechanism. Examples are noise due to injectors, water pump, valve mechanisms and timing gears. Secondary mechanical sources are generally independent of engine load and sources other than those integral to the OCP fuel system are not considered in this paper.

The overall noise of an engine is the sum of all the base engine and ancillary noise sources. The noise balance of the engine gives the relative contributions of all noise sources to the overall noise at a given operating condition. The noise balance is engine specific and depends on the engine hardware and the combustion process.

3.2 Primary Mechanical Noise

The major primary mechanical noise sources that are affected by combustion forcing are typically piston slap and bearing impacts. Piston slap results from the horizontal component of the connecting rod reaction force to the combined gas and inertia forces acting on the piston (2 and 3). This horizontal force acting on the piston changes with crank-angle and a reversal in the side force acting on the piston causes the piston to move across the cylinder resulting in an impact. There are several reversals of this piston side force over the cycle and therefore several slap events. As a minimum, slap events occur at piston top dead centre (TDC) and piston bottom dead centre (BDC) during each cycle. The piston side force is usually greatest for an impact occurring near TDC during the combustion stroke, hence this is usually the most severe impact. For the control of piston slap, it is the balance of gas forces and inertia forces in the region where the most significant piston slap occurs that is most critical.

The effect of cylinder pressure on the magnitude of the primary mechanical noise has been studied at the Automotive Design Advisory Unit (ADAU) of Southampton University over a number of years (3 and 4). The majority of the work has been on diesel engines. Parametric studies and experimental work have demonstrated that the general relationship between cylinder pressure and primary mechanical noise is the noise increases in proportion to twenty times the logarithm of the net force on the piston near TDC. Note that the net force is the resultant of the gas force and the inertia force.

$$SPL \propto 20 \times Log(Net\_Force@TDC)$$

The above relationship was derived initially for piston slap noise but is also applicable to bearing impacts. Crankshaft bearing impacts result from the reversals of the force acting on the crankshaft journals. The derived relationship is considered valid for primary mechanical noise resulting from piston slap and crankshaft bearing impacts.
The relationship indicates that doubling cylinder pressure at TDC at low speed (when inertia forces are low) will result in an increase of 6 dB in combustion controlled primary mechanical noise. The resulting increase in overall engine noise is dependent on the relative significance of the combustion controlled mechanical noise to the total noise balance.

3.3 Direct Combustion Noise

Direct combustion noise is due to the rapidly fluctuating cylinder pressure force and is a combination of the combustion forcing and the attenuation characteristics of the engine block. The combustion forcing is dependent on the form, magnitude and repetition rate of the cylinder pressure and consequently any change to the combustion process that affects the cylinder pressure form can affect combustion noise. The magnitude of the lower frequency harmonics (approximately below 300 Hz) of the spectra of the cylinder pressure forcing function are governed by the magnitude of the peak pressure and are largely independent of the shape of the time trace. The form of the cylinder pressure time trace determines the magnitude of the mid and higher frequency harmonics. Cylinder pressure developments with very rapid increases have very high levels of high frequency energy. Smooth cylinder pressure developments have low levels of mid and high frequency energy.

4. DI VERSUS MANIFOLD INJECTED

4.1 Combustion

The major fuel economy benefits of gasoline DI are achieved by unthrottled engine operation. In practice, the process is partially throttled but cylinder pressures remain high and this results in high peak cylinder pressures at no load and light load relative to a standard manifold injected gasoline engine.

An understanding of the DI process and its impact on cylinder pressure can be achieved by comparing the cylinder pressure of an engine with an OCP direct injected system to a typical manifold injected system. It should be noted that the OCP system allows for a broad range of combustion characteristics and the data presented below is one example only, selected to highlight potential differences between manifold injected and DI and there is considerable scope for optimisation within the OCP DI system.

A comparison of cylinder pressure at idle between a standard manifold injected engine and an OCP DI equipped version is given in Figure 2. This graph shows that the OCP DI process results in significantly higher peak pressures that occur closer to TDC compared to a manifold injected engine at idle. This trend is typical for DI cylinder pressure in the low / part load region.

Figures 3 and 4 show the cylinder pressure spectra at minimum load relative to full load for the manifold injected engine and the OCP DI processes respectively. This data demonstrates a typical trend for DI combustion processes where DI results in higher direct combustion forcing than does the manifold injected engine.

At high load, DI processes commonly aim for homogenous combustion similar to that of the standard manifold injected engine. In practice, the full load combustion characteristics of the OCP DI process do differ from the manifold injected process due to engine hardware differences and calibration and operational strategies employed for optimal part load emissions / economy and full load performance. These include strategies selected for effective stratified operation at low load; the valve timings selected for NOx emission control at part load; the in-cylinder motion selected to support optimal part load stratification and ignition strategies; and compression ratio increases to improve full load performance.

4.2 Low frequency Powertrain Vibration

The primary base engine low frequency vibrations of an I4 are the resultant of the reciprocating out of balance of the crank mechanism and the combined inertia and combustion torque fluctuations (7). A comparison of low frequency engine vibration between a standard multipoint manifold injected engine and an OCP DI engine showed that peak pressure had a significant influence on the low frequency vibration and a development vehicle fitted with an OCP DI engine showed higher levels of second order idle vibration than the multipoint manifold injected vehicle.

4.3 Torsional Excitation

High levels of combustion forcing at part load will increase the level of torsional excitation to the engine and its components. This may result in an increase at low speed in gearbox noise, timing gear noise and valve train noise (7). On a demonstrator vehicle fitted with the OCP DI system, a change clutch type has eliminated a second order gear rattle.

4.4 Linearity

Combustion noise linearity is important for driver feed back because as the engine load increases, a driver expects a progressive increase in the engine noise. The lack of linearity with speed and load for rapid burn petrol engines has been reported as a significant refinement issue (9) . Linearity when combustion controlled mechanical noise dominates depends on a progressive increase in peak cylinder pressure near TDC as load or speed increases. Linearity when direct combustion noise dominates depends on a progressive increase in the spectral content of the cylinder pressure as load or speed increases.

The cylinder pressure data set for the generic 1.8 litre multipoint manifold injected engine shows linearity with load and speed for peak cylinder pressures, cylinder pressure rise rates and acceleration rates of cylinder pressure. A similar comparison for OCP DI data does not show this general linearity.

4.5 Combustion Stability

Combustion stability affects the subjective noise quality. This combustion noise stability is affected by combustion cycle to cycle variations and it is influenced by the magnitude of the cylinder pressure, the timing of combustion and the spectral content (6 and 8). The OCP DI process has demonstrated combustion stability comparable to a standard manifold injected process.

4.6 Subjective Noise Issues

A third octave comparison of the spectral content of the cylinder pressures shows that the OCP DI process has higher levels of high frequency forcing at part load – Figures 3 and 4. This high frequency combustion characteristic will be reflected in the radiated engine noise and combined with the changes in primary mechanical noise, will result in sound quality differences.

5. COMBUSTION NOISE PREDICTION MODEL

5.1 Model Construction

To understand the effect on overall engine noise of the implementation of a DI combustion process to a manifold injected engine, an empirical model for engine noise was developed for a generic 1.8 litre four cylinder four stroke 16 valve engine.

The base engine noise was modelled as the sum of the impact mechanical excitation noise and direct combustion excitation noise. The model was based upon simple impact theory and assumed a constant structural attenuation for both mechanical and combustion forcing. Constants of proportionality were empirically derived by correlating the model with the measured surface vibration data (from experimental work with manifold injected and DI combustion) and one metre noise data.

Engine Noise = Primary Mechanical Noise + Direct Combustion Noise

$$\text{Engine Noise} = K(2.F_B + F_T + F_{TC}).N^3 + H.C$$

where:

- K, H: Constants of proportionality
- F_B: Inertia Force on Piston at BDC
- F_T: Inertia Force on Piston at TDC
- F_{TC}: Net force on piston at TDC during combustion
- N: Engine speed
- C: Third octave band combustion forcing level at 1 kHz

The model predicts direct combustion noise, primary mechanical noise, and overall engine noise for a given set of combustion characteristics as described by the cylinder pressure at TDC and the spectral level of the cylinder pressure at the 1 kHz third octave band. The level of the cylinder pressure spectra at 1 kHz was selected as a single number measure of direct combustion forcing. Providing the characteristics of the spectra are smooth over the mid frequency region, then the 1 kHz level provides a good single number estimate of combustion forcing (10). Note that the process is not limited to utilising the level at 1 kHz and alternative parameters may be required for alternative combustion processes. A non-linear effect of peak pressure on noise at low speed as determined from the experimental data was included.

Secondary mechanical noise, ancillary noise, noise quality issues and the general spectral content of the noise have not been considered in the empirical model. The model did not consider secondary effects of the combustion forcing such as torsional vibration and crankshaft bending. The model assumed the base engine noise was controlled by the primary mechanical noise and direct combustion noise. Correlation between the empirical model characteristics and the data base noise validated this assumption.

5.2 Empirical Model Predictions

The model has been used to predict engine noise levels under all load and speed conditions for the corresponding cylinder pressure characteristics. A generic comparison of the different combustion processes can be obtained by considering the engine noise over a derived “road load” curve. This road load curve ranges from low speed and load to high speed and load with power being proportional to speed cubed and maximum power occurring at 5500 rpm.

Figure 5 gives a comparison of the combustion and mechanical noise contributions to overall noise for the multipoint manifold injected engine over the road load curve. The data shows that mechanical noise dominates the overall noise over the speed range except in the high speed region. Combustion noise does not dominate the overall noise at any speed.

Figure 6 gives a similar comparison of the combustion and mechanical noise contributions to overall noise for a sample OCP DI calibration / engine over the road load curve. The data shows that mechanical noise dominates the overall noise in the low speed region for this calibration. Combustion noise is significant from the mid speed region and is approximately equivalent to the mechanical contribution above 3500 rpm.

A comparison of the predicted noise difference over the road load curve between the multipoint manifold injected engine and the selected OCP DI calibration is given in Figure 7. The data predicts that the selected OCP DI calibration would increase noise over the whole road load curve. The graph shows an additional road load calibration (OCP DI -high) to demonstrate the effect of calibration on the predicted road load noise.

5.3 Modelling - Effect of Cylinder Pressure Amplitude at TDC on Road Load Noise

The model was used to assess the sensitivity of noise over the road load curve to cylinder pressure amplitude at TDC. For the purposes of this exercise, upper and lower limits of the pressure amplitude at TDC over the road load curve were selected from test data while the 1 kHz cylinder pressure spectrum level was kept constant. While the two parameters are not totally independent, there is considerable ability to vary primarily the cylinder pressure amplitude at TDC by the timings of the injection and the ignition. Figure 8 shows the results of the modelling and that the overall noise in the low speed and mid speed region is very sensitive to the magnitude of the cylinder pressure at TDC.

5.4 Modelling - Effect of Cylinder Pressure Spectrum on Road Load Noise

The model was used to assess the sensitivity of noise over the road load curve to the 1 kHz cylinder pressure spectral level. For the purposes of this exercise, the cylinder pressure spectrum was varied by +/- 3 dB from the initial road load data while the cylinder pressure amplitude at TDC was kept constant. Again while the two parameters are not totally independent, the cylinder pressure spectrum at 1 kHz is more a function of cylinder pressure rise rate than of amplitude at TDC. Figure 9 shows the predicted impact of varying the cylinder pressure spectrum and that the impact of higher cylinder pressure spectrum levels is most significant in the mid and high speed regions. At 5000 rpm, a 6 dB increase in the 1 kHz spectral level results in a predicted 3.5 dB increase in noise.

5.5 Development of Cylinder Pressure Guidelines

The empirical model results demonstrated the significance of knowledge of the noise balance of an engine for setting guidelines on cylinder pressure development. The model showed that noise benefits are achieved by reducing the forcing function of significant noise source. For example, if engine noise at a given condition is dominated by combustion controlled mechanical noise then overall noise will be reduced by limiting the combustion pressure near TDC.

At operating conditions where combustion noise and mechanical noise are both significant there are an infinite number of combinations of pressure at TDC and combustion spectrum levels that may combine to give a certain level of overall engine noise. The empirical model can be used to derive a contour graph of the relationship between the magnitude of the cylinder pressure at TDC and the 1 kHz spectral level for a given noise target at a given speed / load condition.

The model predicts mechanical noise reductions by reducing the cylinder pressure at TDC. This suggests that a change in combustion timing that results in lower pressure at TDC will give noise reductions. This is true providing the change in timing is sufficient to reduce the pressure on the piston during the piston slap impact event. The model approximates this time as TDC whereas the actual timing and duration of the impact on a crank-angle basis depends on engine speed and is affected by piston geometry although it is near TDC.

6. DI SPECIFIC SUBSYSTEMS

6.1 Air Compressor

From an NVH perspective the OCP air compressor can be regarded as a typical belt driven engine accessory with the usual issues of mechanical noise, residual imbalances, and drive torque pulses.

The mechanical noise consists of reed valve impacts, piston slap and rod / crank mechanism impacts. This has been satisfactorily treated with attention to clearances, bearing selection, reed valve design and acoustic volumes. Intake noise from the OCP air compressor typically consists of engine order pulses. Attenuation of these pulses may require an additional simple expansion chamber which can be separate or may be integrated into the airbox. The noise levels of the compressor have been demonstrated as being significantly below no load engine noise with no sound quality issues.

Vehicle testing has shown that the effects of residual imbalances and drive torque pulses are not discernible provided that the rotational speed of the compressor does not coincide with any powertrain resonant frequencies.

6.2 Air / charge Injector

The OCP DI system uses on a dual fluid injector to inject fuel into the cylinder. The noise from the injectors is generated by the opening and closing impacts and consists of very short duration impacts repeating at firing frequency. The requirement to support high pressure operation and construction for durability in a combustion chamber environment results in injector noise that is higher than for a typical manifold fuel injector.

The impact energy radiates directly as noise from the injectors and also is transmitted to the cylinder head and fuel rail through the injector mounting. This transmitted energy is radiated as noise from the engine structure. The direct injector noise is effectively attenuated with covers. The indirectly radiated noise can be reduced by isolation of the injectors and / or structural changes to reduce the radiation efficiency of the engine components in the critical frequency range. The mounting system for the injectors needs to seal and locate into the combustion chamber against combustion temperatures and pressures.

6.3 Engine Air Intake System

The air intake system may require some minor changes. Reduced throttling at low loads results in higher airflow and may require additional noise suppression. Similarly the “open” throttle provides a more direct path for engine / intake noise to escape.

7. CONCLUSIONS

7.1 Overall

The NVH effects of implementing the OCP DI system are known, predictable, and manageable. Additionally, the OCP DI system's flexibility and wide operating range allows for considerable optimisation for NVH while maintaining the inherent benefits of DI Combustion.

7.2 Combustion Effects

All DI four stroke gasoline combustion systems with stratified combustion and minimal throttling at part loads will have similar levels of combustion forcing and this will be higher than for conventional manifold injected engines. This is due to the higher peak cylinder pressures and faster burn rates. For all DI systems, noise and refinement must be considered during the development of calibration and combustion strategies to minimise the disadvantages compared to manifold injection. The major refinement differences between the DI systems will occur from calibration strategies and mechanisms used.

For an engine where the noise balance has low levels of secondary mechanical and ancillary noise sources in overall noise level terms, we can expect the following changes:

- Idle noise and low speed / low load base engine noise will increase primarily due to the higher cylinder pressures at TDC.
- Mid speed / mid load noise will increase due to a combination of higher cylinder pressure spectral levels and higher cylinder pressures at TDC.
- Full load noise may increase due to higher cylinder pressure spectral levels where strategies have been employed to improve full load performance.
- The inherently high combustion pressures will result in higher levels of low frequency vibration at idle.
- Extra intake noise attenuation may be required due to higher airflow at low speeds.

7.3 Combustion Noise Prediction Model

The combustion noise model has proved to be a useful tool for allowing early noise evaluation of engine calibrations and hardware. Similarly it can also be used to generate low noise calibration guidelines for use during power and emissions development.

7.4 DI Specific Subsystems

- The air compressor need not reduce refinement levels.
- The air / charge injectors are noisier than conventional gasoline manifold fuel injectors.
- Air Intake noise may increase due to unthrottled or minimal throttling at low loads.

8. ACKNOWLEDGEMENTS

This report was prepared with input and assistance from Orbital Engine Company Australia.

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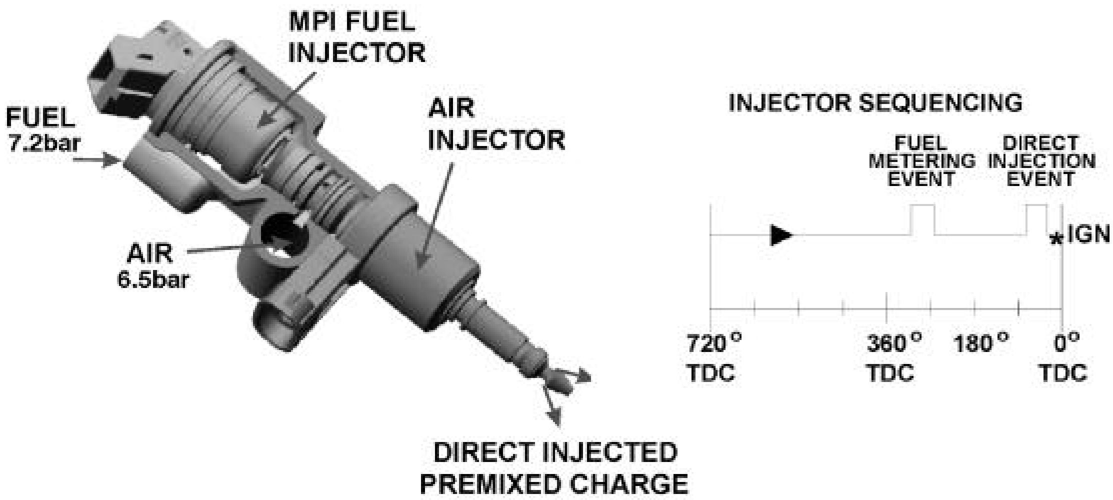
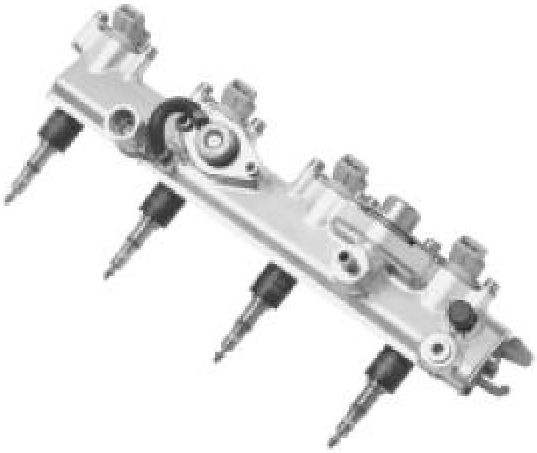
Figure 1: OCP Direct Injection Components and Development Installation.



Schematic of Air Injector Assembly

Development Vehicle Installation



Development of Fuel System Assembly

Figure 2: Cylinder Pressure at Idle - OCP DI versus Manifold Injected.

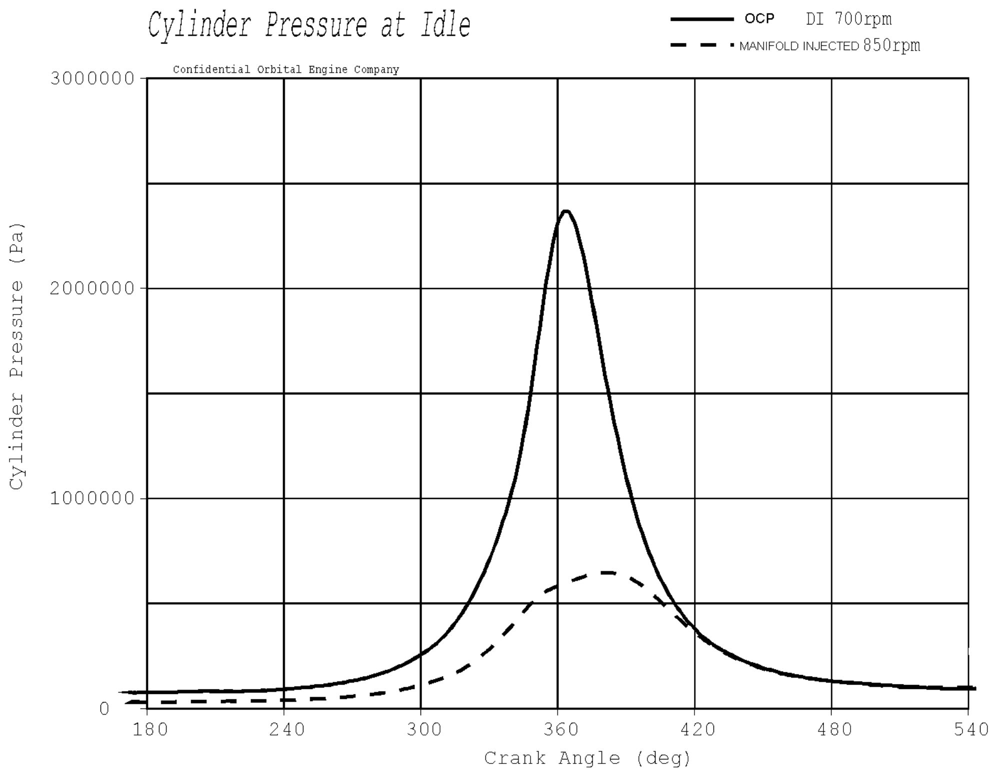


Figure 3: Cylinder Pressure Spectrum @ 3000 rpm - Manifold Injected Gasoline, Minimum Load versus Maximum Load.

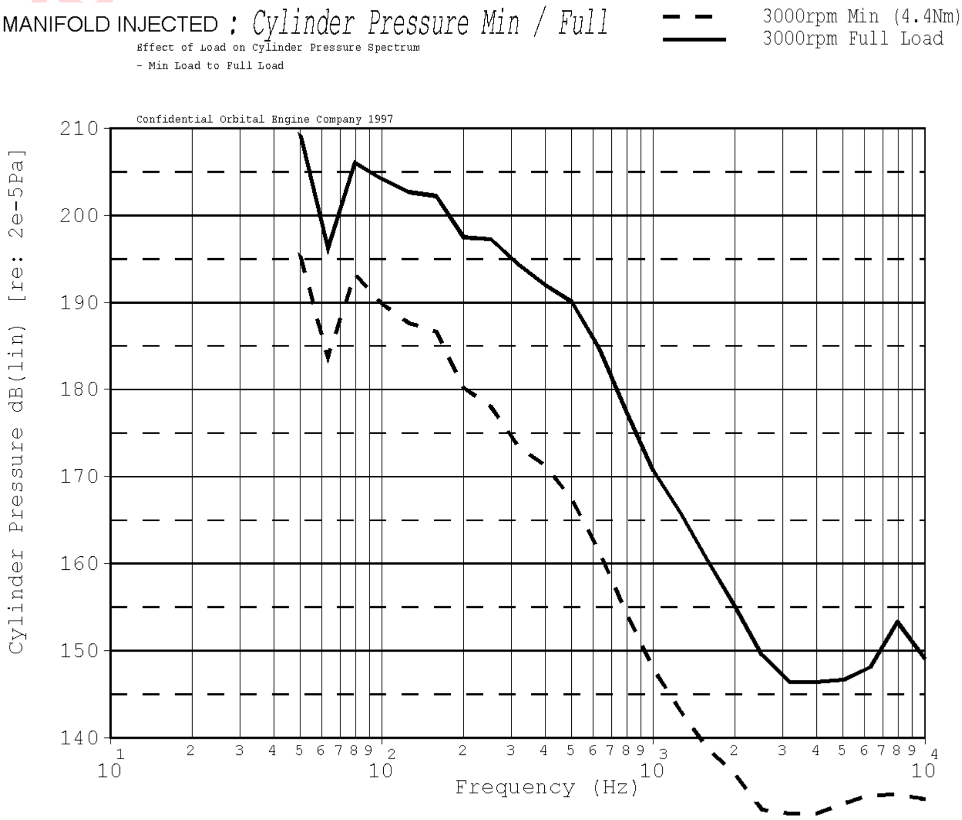


Figure 4: Cylinder Pressure Spectrum @ 3000 rpm - OCP DI Minimum Load versus Maximum Load.

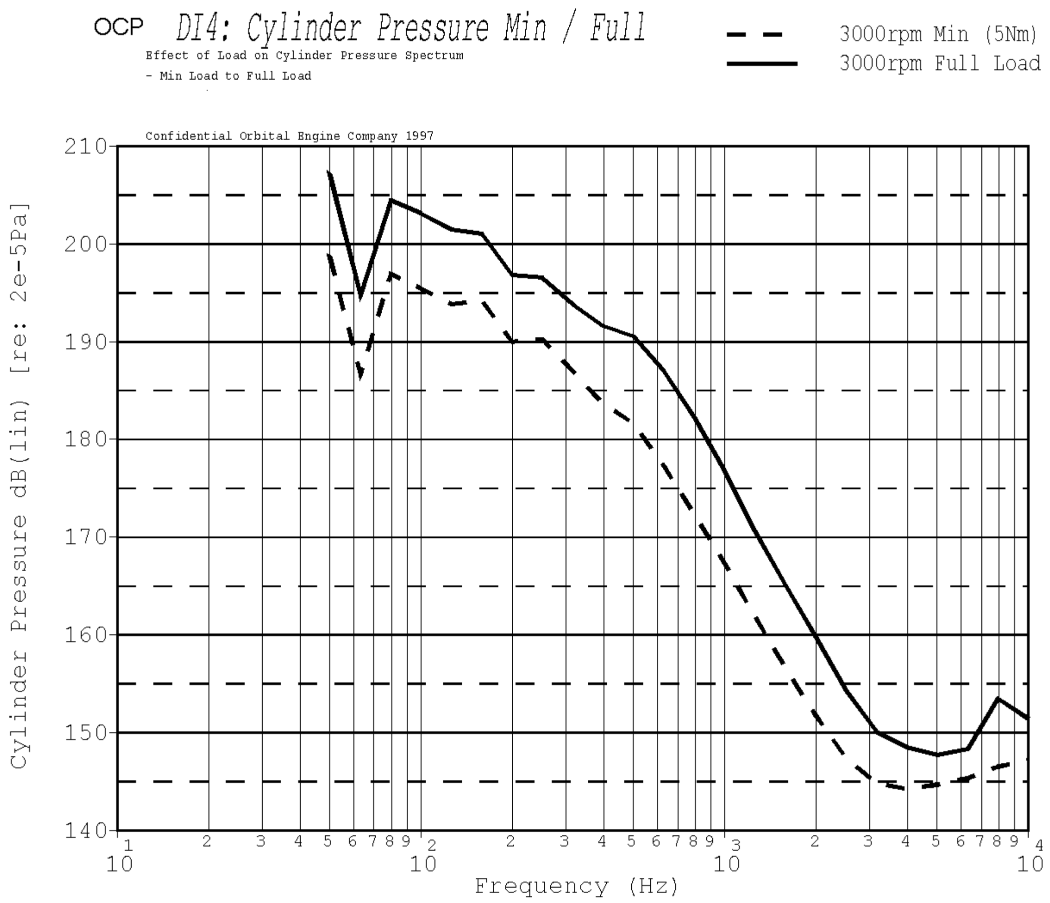


Figure 5: Noise Prediction Model for Manifold Injected Gasoline Engine, on Road Load Curve.

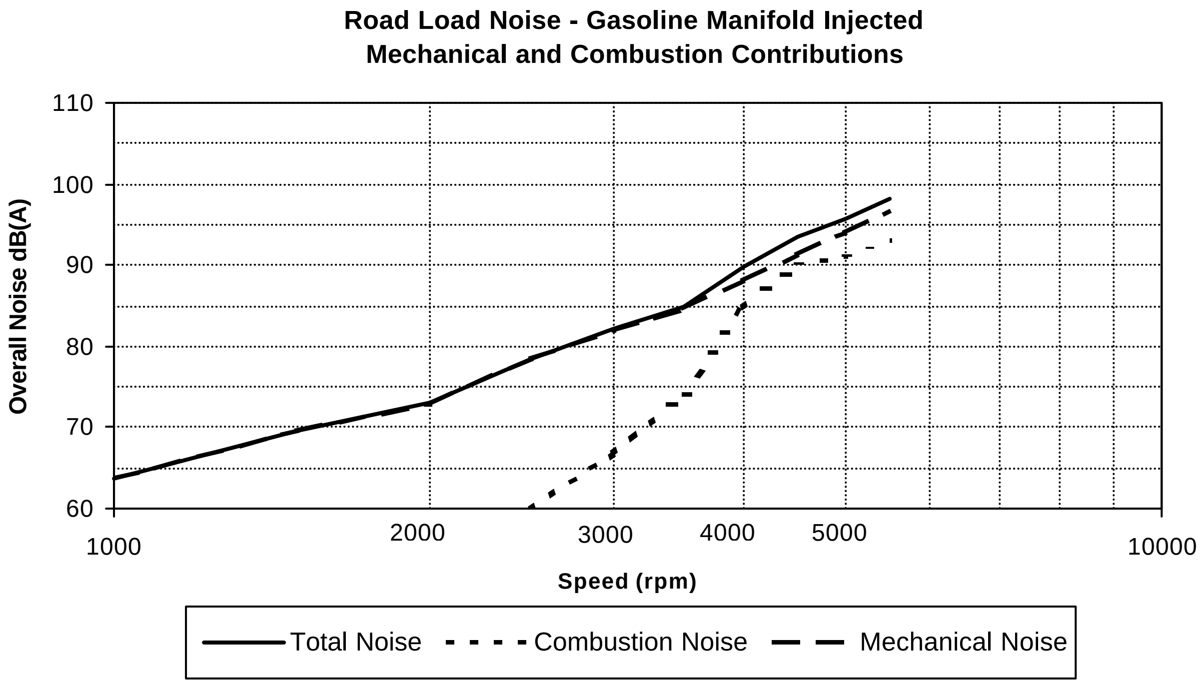


Figure 6: Noise Prediction Model for OCP DI Engine, on Road Load Curve.

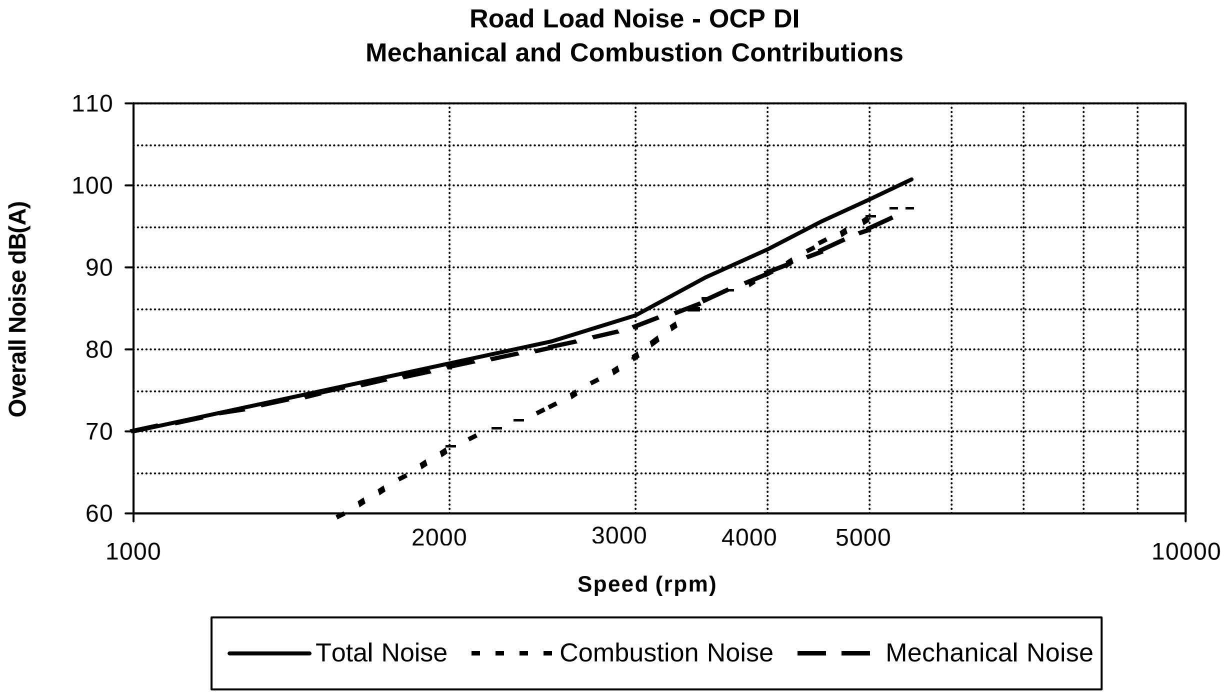


Figure 7: Noise Prediction Model, Overall Noise Comparison for Manifold Injected Gasoline Engine and Two OCP DI Calibrations on Road Load Curve.

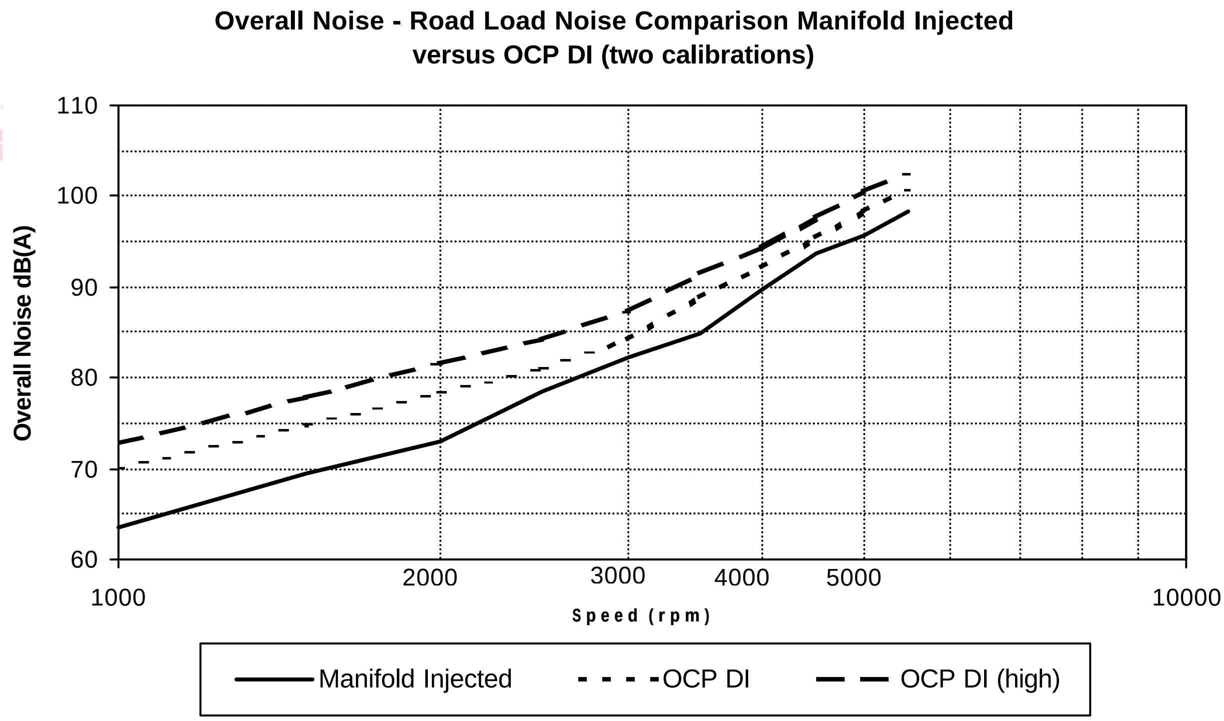


Figure 8: Noise Prediction Model for DI, Effect of TDC Cylinder Pressure on Overall Noise Levels, on Road Load Curve.

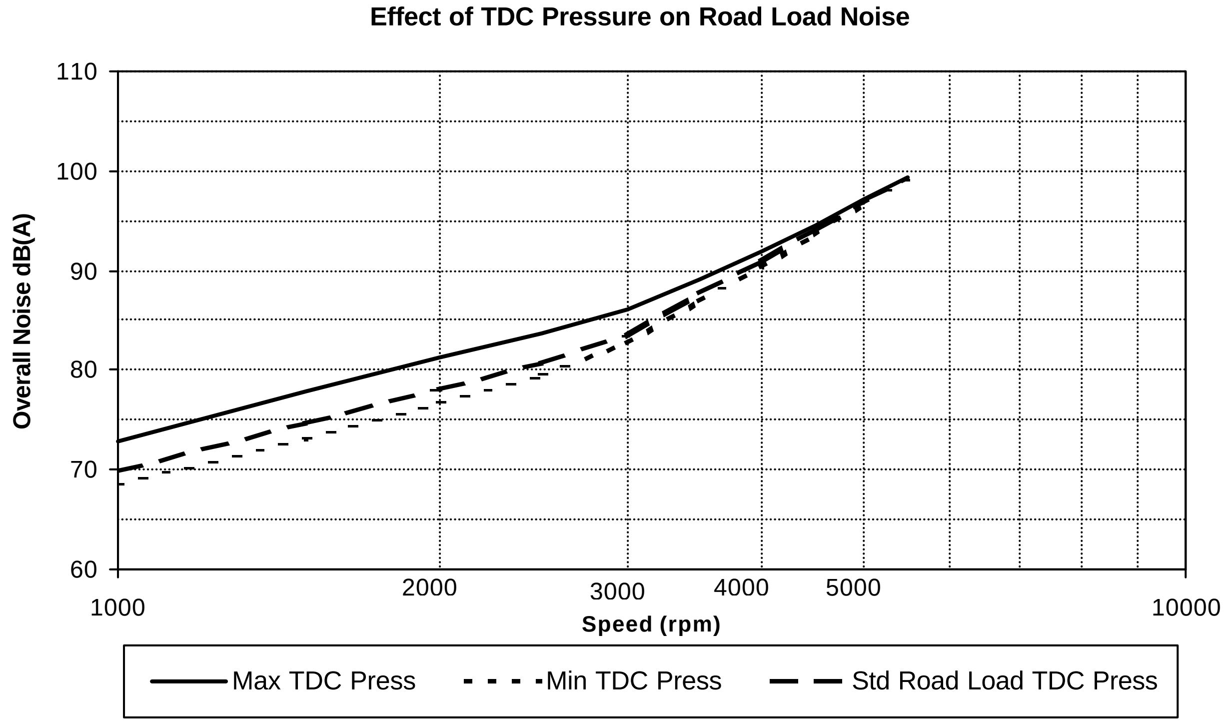


Figure 9: Noise Prediction Model for DI, Effect of Cylinder Pressure Spectrum on Overall Noise Levels, on Road Load Curve.