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ABSTRACT

FORD OTOSAN has developed a new heavy-duty diesel engine, ECOTORQ, for the new Ford Cargo Trucks whose production started in September 2003. The engine is 7.3 liters, 6-cylinder in-line, with common rail fuel injection system and overhead camshaft design having 4 valves per cylinder. The engine meets the Euro-III exhaust emissions limits, which were in effect when it was introduced, and the engine incorporates the potential to meet Euro-IV. Modern computation and simulation methods were used and extensive experimental studies were made during the design and development stages, which helped reach the targets of compactness, modular structure, low fuel consumption, low noise level and low emissions.

INTRODUCTION

FORD OTOSAN, an affiliate of Ford Motor Company and Koc Holding, has been producing Cargo trucks since 1982 mainly for domestic market. In 1999, it was decided to start a program to develop a totally new heavy-duty diesel engine to replace Dovertech, the predecessor engine of the vehicle, in order to;

- comply with the market demands of higher power, higher durability, lower cost of ownership and lower noise level
- meet future exhaust emission legislations
- offer a competitive product

The design work started with a clean sheet of paper and concept of the new engine was developed depending on the market research, benchmarking and field data evaluation studies. The first prototype ran in October 2000. Following the long period devoted to engine base calibration, mechanical development, design verification, vehicle calibration and product validation phases, production started in July 2003. The first truck powered by an ECOTORQ engine was introduced into the market in September 2003.

In this paper, the design features of the engine are described and special measures to reach engineering targets are outlined. Verification methods including the test cycles that simulate the specific loading conditions of Turkey heavy-duty truck field, and the calibration approach that focuses on low fuel consumption over field duty cycles are mentioned. Features for low noise level, and achievements brought by these features are presented. Special measures applied to the engine for low oil consumption are described, and because it is the dominant parameter for oil change interval in modern heavy-duty diesel engines, soot in oil rate of the engine is presented. Effects of high sulfur content in the fuel on oil change interval are discussed and finally, special engine management software functions are listed, and their purposes are explained.

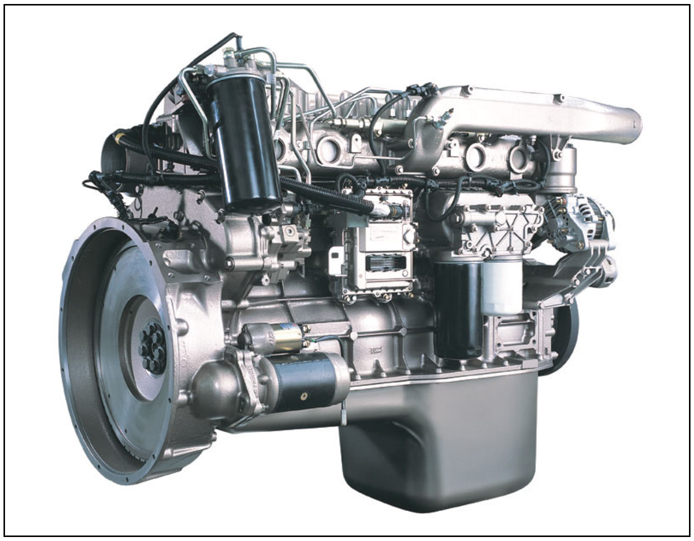

Figure 1 – ECOTORQ Engine

ENGINE SUMMARY

ECOTORQ is an in-line 6-cylinder, 24 valve, common rail, and electronically controlled heavy-duty diesel engine. The engine is suitable to be used on medium-duty and heavy-duty trucks. It can also be very convenient for bus, marine and industrial applications.

An image of the engine is shown on Figure 1. Brief technical specifications are given on Table 1. Power and torque curves of the engine are shown on Figure 2 and Figure 3 for 177 kw and 221 kw versions, respectively.

Table 1 Engine Specification

Specification		240 PS	300 PS
Capacity	Liters	7,33	
Bore	mm	112	
Stroke	mm	124	
Compression Ratio		17,4:1	
Oil Pan Capacity	Liters	28	
Weight (*)	kg	708	
	Performance		
Exhaust Brake	kw	140	140
Power	kw	177	221
Rated Speed	rpm	2400	2400
Max. Torque	N-m	840	1100
Max. Torque Speed	rpm	1200-1800	1400-1800
Noise Level (**)	dB(A)	93,4	93,4
Soot in Oil Rate	g/hr	<0,25	
Oil Consumption	g/kw-hr	<1,0	
Expected Engine Life (B_{10})	km	400,000	
Expected Engine Life (B_{50})	km	750,000	
Max. Altitude w/o de-rating	m	2500	
Cold Start Ability	°C	-25 (with aid)	
		-15 (w/o aid)	
Exhaust Emissions		Euro-III	
(*) Weight of the version used on Cargo trucks, dry weight according to DIN 70020 (**) See Table 4 for details			

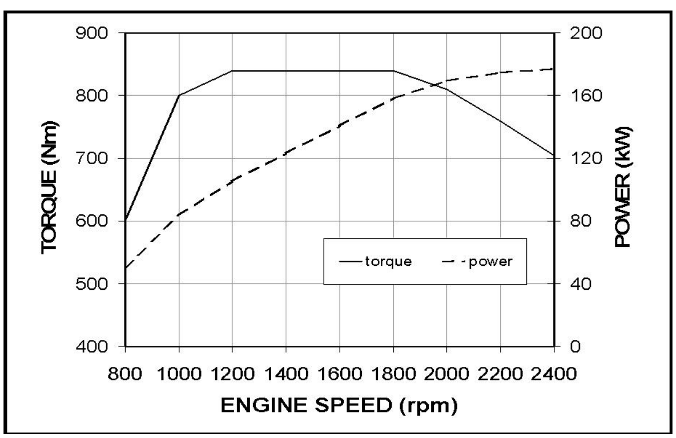

Figure 2 – Torque & Power Curves for 177 kw version

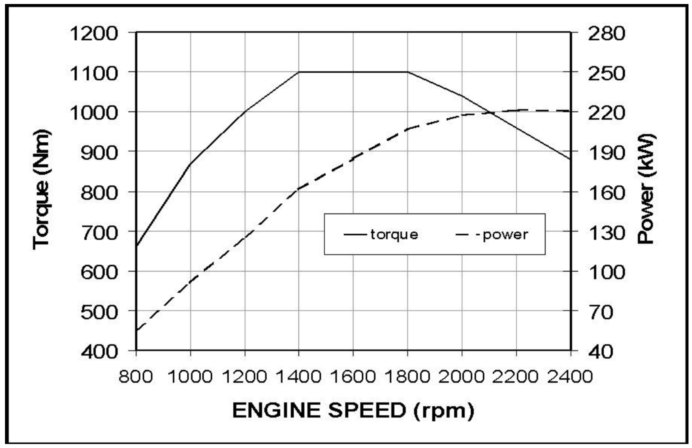

Figure 3 – Torque & Power Curves for 221 kw version

ENGINE CONCEPT AND DESIGN FEATURES

ENGINE CONCEPT

The project started with the following three studies to define the concept.

- Market research: Expectations and demands of the market were searched to understand the customer priorities, which would then to be used as the key information in determining the compromises between conflicting engine attributes.
- Field data evaluation: 384 heavy-duty diesel engines that had completed their lives were collected and torn down. The aims of the study were to analyze their major failure modes and to identify the noise factors and challenging conditions of the field resulting from road conditions, overloading, customer habits and driving behavior.
- Benchmarking: A number of competitor engines with the latest technology were examined and trends in the heavy-duty technology were investigated. Advantages and disadvantages of different concepts for critical systems, sub-systems and parts were evaluated with assistance from suppliers and consultant companies.

The market research work conducted showed that the trend of customers' higher power demand would continue for the segment of Cargo trucks in the future. This was giving a direction of high power density and indicating the need of provision to be provided for higher ratings. It was decided to keep the power density of the engine around 30 kw/liters for the high rating variant, considering the challenging field conditions of the truck field in Turkey. The target value for the future rating upgrade, on the other hand, was chosen as 258 kw (350 PS) with a 8.0 L total swept volume and thus with a power density of 32 kw/litres.

The engine was designed for Euro-III exhaust emission limits; however, in order to incorporate the potential for Euro-IV, peak fire pressure capability of 180 bar was targeted, although this value would not be needed to fulfill Euro-III requirements.

Modular design features were applied wherever possible for compactness and in order to be ready for potential future regulations and/or market demands. For example, as shown in the Figure 4, different designs of oil, coolant and fuel filtering modules using either spin-on or cartridge type fully combustible filter elements are provided while keeping the base parts of the engine as the same.

Right hand side of the engine was determined to be the "service side". ECU and all filter elements are placed on this side. Left hand side was designed as "hot side". The turbocharger was placed on this side, in order to be far enough from ECU and filter elements.

ENGINE SUB-SYSTEMS

Base Engine Cast Components - Cylinder block and head are made from laminar gray cast iron. It is a well-known fact that the local strength values on a cast iron component depend also on the thickness of the particular location and cooling behavior of the part during the casting process [1]. For the cylinder blocks of 6 cylinder in-line engines of large sizes, the main bearing walls close to center (number 3, 4 and 5 where front wall and rear wall is called as 1 and 7, respectively) suffer from this phenomenon and material strength is lower at these regions compared to the front and rear walls, since these walls stay in a larger core block during casting and thus cool down more slowly. Control parameters such as shakeout time, types and amounts of alloying elements and inoculants used also influence variations in material strength values. As shown on Table 2, process improvement by optimizing these parameters resulted in increased material strength values at the center walls and reduced difference between end and center walls.

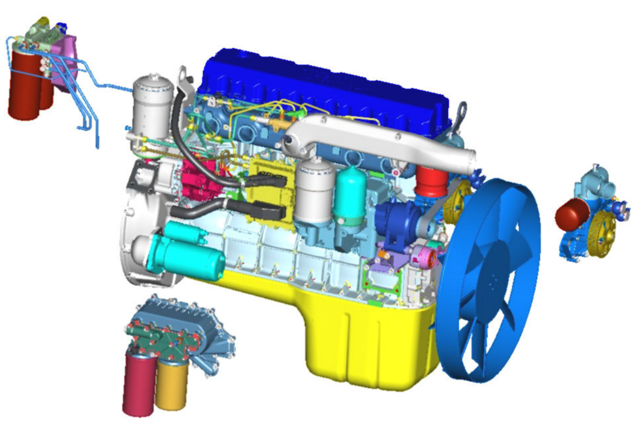

Figure 4 – Example to Modular Concept

Table 2 Cylinder Block Casting Process Improvements

	Side Walls (no 1&7)	Center wall (no3)
Ultimate tensile strength - baseline (MPa) (*)	210	170
Ultimate tensile strength - improved (MPa) (**)	225	215

(*) Powder inoculants of 60 gram, 5 hours of shakeout time
(**) Composition and process differences are as the following:
- Composition: 0.20% Mo and 0.10% Cr added to baseline chemical composition.
- Inoculants: The same chemical composition with the baseline inoculants, but insert type instead of powder. Amount is increased to 200 gr.
- Shakeout time: 3.5 hours instead of 5 hours.

Note: Values are average of tensile tests on the specimens that are taken from the particular locations indicated

The cylinder block is symmetric and has a deep skirt. Outer and internal ribs are optimized for low noise, and acceptable bore distortion for both assembly and fired engine conditions. The shape of the skirt is designed considering the option of engine capacity increase. Cylinder bore walls are optimized for both homogenous cooling of cylinders and minimized cylinder bore deformations. The bottom flange of the cylinder block is tailored to the high capacity oil pan and implementation of a ladder frame (Figure 5).

Main bearing fixation is from sides, following the current practice of the manufacturing plant. Width grading on saddles and caps ensures maximum of 60-micron interference limit determined by CAE analysis (Figure 6).

Number, arrangement and size of the coolant passages on the cylinder block top deck and cylinder head fire deck are provided to not only produce efficient and homogenous cooling to cylinder head, but also to allow efficient paths for cleaning core sand from between the cylinders and in the ports.

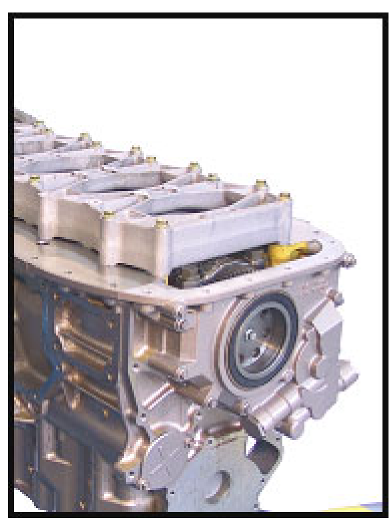

Figure 5 – Cylinder block and ladder frame

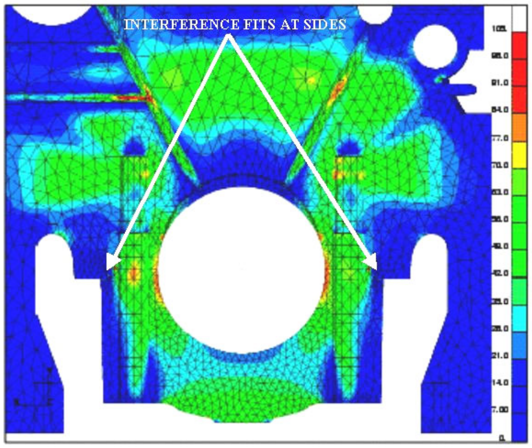

Figure 6 – Main bearing fixation concept

Flanges for air compressor and high-pressure fuel pump are incorporated to the rear end of the cylinder block to reduce number of parts and cost.

The cylinder head has four valves per cylinder design and injectors are placed at the center of the cylinders. Cylinder water jacket cores are made of two pieces to enable assembling of the port cores (Figure 7).

Intake manifold is incorporated to the cylinder head for compactness, and the single core of the all intake ports maintains dimensional stability (Figure-7).

Water jacket side of cylinder head fire deck has ribs for increased stiffness against high peak fire pressures.

Injector is placed in a copper sleeve located in the coolant jacket. Stepped tip geometry and assembly process developed ensures robustness in gas sealing (Figure 8).

<u>Power Cylinder</u> – High durability aluminum silicon alloy pistons are equipped with three rings. Full cutback top land profile provides increased scuff resistance and reduced liner wear while ensuring robustness against usage of fuel having high sulfur content.

Top land geometry is a critical control parameter in ensuring the robustness against hard carbon built up at top land region between piston and liner. High sulfur content fuel promotes soot generation. During the development stage, different top land geometries have been studied. Two main approaches were;

- “tight design” where the gap is very small and does not allow carbon deposit build up between cylinder liner and piston at the piston top deck region
- full cutback geometry where the gap is large enough to prevent carbon deposits hardened by hammering effect, but let the soot burn and/or thrown away.

Advantages of the tight designs were better fuel consumption (up to 1,5 %), and better particulate emissions, while its disadvantages were identified as low scuff resistance in case of overheating and slightly higher liner wear. Full cutback designs were found as very robust as far as scuff resistance was concerned. Considering the field conditions, full cutback approach was preferred for the engine.

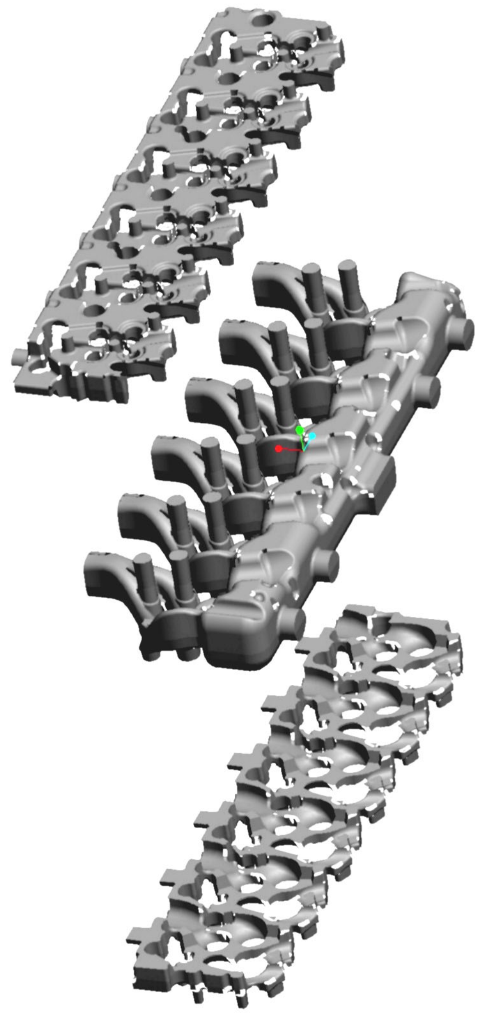

Figure 7- 3D models of water jacket core

Pistons are oil gallery cooled. Optimized inlet and outlet hole sizes of the gallery helps in satisfying required cooling rates with reduced oil flow rates. Because the flow rates are low enough to be compensated, the piston-cooling jets have no one-way valves.

Cast iron dry liners are fit to cylinder block with interference.

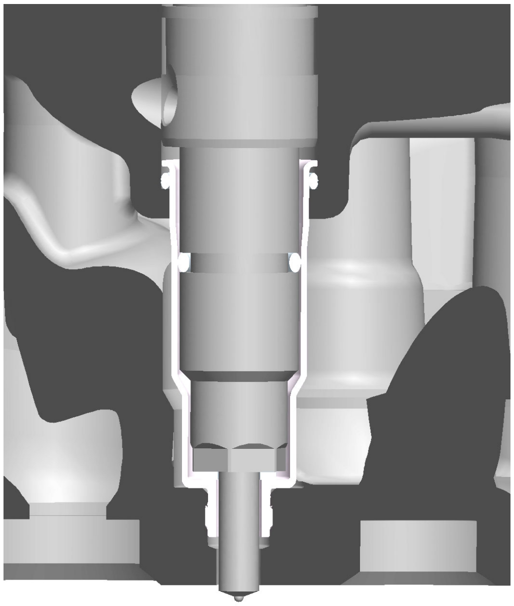

Figure 8 – Copper Sleeve as injector seat

Crank Train and Front End Auxiliary Drive (FEAD) System – Fillet induction hardened, micro-alloyed steel, untwisted crankshaft has 8 integrated counterweights. Fracture split, micro-alloyed, forged steel connecting rods have tapered small ends. Connecting rod bearings are from steel-aluminum composite material, and composite main bearings have electroplated sliding layer with an intermediate nickel layer and tin-flash. A viscous torsional vibration damper is employed and tension of the 8-groove timing belt is controlled by an auto-tensioner.

Lubrication System – Lubricating oil is supplied by a gerotor type oil pump integrated to the aluminum engine front cover. Pressure regulating valve (PRV) governs the oil pressure in the main oil gallery while an additional safety valve limits the pump outlet pressure. Pressure regulating valve is sensing directly from the main oil gallery, unlike systems where this signal comes from pump outlet. The main oil gallery pressure is therefore not affected by pressure drop values varying from influences such as dirt load on the filter element. The PRV returns excess oil to the suction side of the pump, while the safety valve opens directly to the oil pan.

Two different oil modules are available as mentioned previously. The first alternative includes a full flow cartridge type, fully combustible oil filter element and a centrifugal oil cleaner, while the other has a spin-on type full flow oil filter assembly and a very fine spin-on type by-pass oil filter element that replaces centrifugal oil cleaner. Oil cooler is new generation aluminum type. The cooler is isolated from the engine body by non-conductive gaskets to decelerate electrolytic corrosion.

Laminated steel deep drawn oil pan has 28 liters oil capacity. Sump geometry with 358 mm depth and small draft angles maximize the oil capacity in the defined packaging constraints.

Engine Sealing – PTFE (Polytetrafluoroethylene) seals are used as the crankshaft seals. Front crankshaft seal is cassette type, which is more convenient in dusty environments. Due to cost benefit compared to using gaskets, anaerobic liquid sealants are used at the joints between cylinder block and parts that have wide but rigid flanges such as flywheel housing, front cover and coolant pump. NBR (nitrile rubber – Butadiene Acrylonitrile Copolymer) or FKM (fluorocarbon rubber) coated spring steel gaskets are used where the sealing area is relatively small and rigidity of the flanges is relatively low (e.g. oil module, oil cooler, T/C oil feed and return pipe flanges).

Both multi-layer and single layer steel cylinder head gaskets have been developed for different applications having different peak fire pressure distributions over the load and speed.

Cooling System – Coolant module consists of coolant pump, integrated thermostat housing and coolant filter. Coolant flow-rate tuned to 221 kW-Euro-III version can be increased by just modifying the impeller size and body machining while the cast body design is being kept unchanged.

A broad erosion and rust problem in the water jackets of cylinder blocks and cylinder heads was identified during the previously mentioned field data evaluation study, where a number of engines from the field were inspected. Considering that the possible cause can be the use of tap water instead of specified coolant, which is very common in the field, coolant filter with a built-in chemical was integrated to the coolant module.

The chemical protects the parts in contact with the coolant even if tap water is used as the coolant, while the filter elements trap the dirt in the coolant.

Valve Train System – A chilled cast iron camshaft placed in the cylinder block actuates the valve train system. Every rocker arm actuates a bridge, which pushes 2 valves (Figure 9). There is a spherical joint at the valve side tip of the each rocker arm.

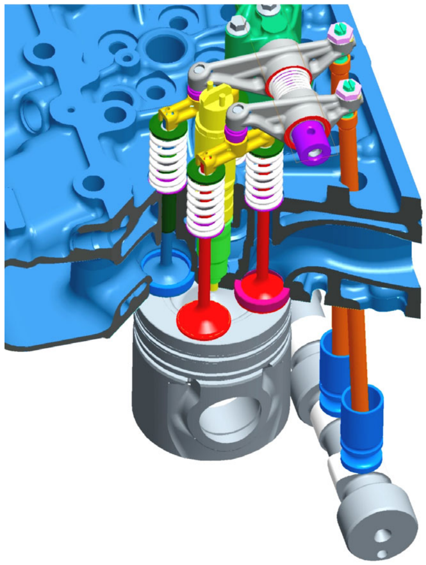

Figure 9 – Valve train

SPECIAL MEASURES TO MEET ENGINEERING TARGETS

ROAD DUTY-CYCLE STUDIES

Thorough understanding of the local field conditions was crucial in the development process. The road duty-cycle was necessary because fuel consumption as felt by the customer and the durability of the engine is directly related to the actual cycles on the field.

Extensive data was collected on the test vehicles with different engine versions along various routes. The main variables in focus were the engine speed, fuel load, and vehicle speed.

The road duty-cycle represents the frequencies of the operating states. The state of the engine can be defined as a combination of engine speed and fuel load. In order to make over-the-cycle estimations, state distributions are sufficient. On the other hand, when actual load cycles are desired to be simulated on the test bed, transient characteristics also gain importance. The transient characteristics in the road-cycles can be captured by a statistical approach that generates a model expressing the state transitions during the cycle. The data collected on the test vehicles with ECOTORQ engines are also processed to build such a model to generate finite-step dynamometer cycles. Software routines are developed for model building and dynamometer cycle generation.

ENGINE MODELING

In order to aid the calibration work on the dynamometer and to evaluate cycle-dependent metrics much more easily, an engine modeling approach was employed in the development process. The primary aim of the model is to predict the engine outputs (torque, emissions, specific fuel consumption) at any operating point (engine speed, fuel quantity) and injection parameters (injection timing and rail pressure). This model represents the response of the hardware system consisting of the fuel injection equipment, power cylinder, intake and exhaust components, turbocharger, etc. Hence, the model is useful in generating new calibrations and evaluating existing ones with the knowledge coming from the road duty-cycle analysis.

The estimate for engine output is denoted by $\hat{y}_i$, $i = 1,...,M$. Here, M is the number of engine outputs of interest, which can be the torque, specific fuel consumption, peak firing pressure, or any emission type. The estimate is evaluated using a function of the form,

$$\hat{y}_i = f_i(\alpha, P, \beta_i) \tag{1}$$

Functions f_i are typically n-th order polynomials with the independent variables of start of injection (SOI), α, and rail pressure, P, and parameter vector, $\boldsymbol{\beta_i}$, whose elements are the parameters of the polynomial. The parameter vector is not constant but predicted according to the operating point of the engine, which is defined by the engine speed, N, and the fuel injection quantity, q.

The modeling process starts with identifying the location of the training nodes according to the density function of the actual duty-cycle data. For each training node (N_j,q_j), a set of dynamometer experiments are made to obtain the engine outputs y_{ijk} corresponding to different SOI and rail pressure values (α_{jk}, P_{jk}), $j = 1, ..., J$ and $k = 1,..., K_j$. Here, J is the number of nodes and K_j is the number of tests made at node j. Parameter vectors $\boldsymbol{B}_{ij}$ at the nodes are found by least squares fitting of (1) to the data. Parameter vectors β_i at an intermediate point (N,q) are evaluated using $\boldsymbol{B}_i$ vectors evaluated for the closest three nodes using interpolation.

A set of software tools is developed for the purposes of data cleaning, function fitting, and estimation. Table 3 shows the difference between the model output and the actual measurement results at four distinct operating points. The model is formed using data points that do not coincide with the speed and quantity values used in comparison data. It is observed that the accuracy is high enough to use the model in offline calibration approaches. The model is especially useful in predicting the loading as a result of the firing pressures in the cylinder and the cycle-SFC (specific fuel consumption).

Table 3. Comparison of model estimates and dynamometer results.

Inputs					
Engine Speed (rpm)		1498	1498	1698	1800
Quantity (mg / stroke)		97.5	97.5	72.5	97.4
Start of Injection (degree crank)		-12.0	-8.5	-9.5	-10.5
Rail Pressure (MPa)		109	90	93	135
Outputs					
Torque (N-m)	Estimate	853.2	844.2	609.2	847.8
	measured	848.7	836.3	606.5	849.8
	% difference	**0.5**	**0.9**	**0.4**	**-0.2**
SFC (g/kw-hr)	Estimate	196.4	198.5	204.6	197.5
	measured	197.0	199.4	204.3	196.8
	% difference	**-0.3**	**-0.5**	**0.1**	**0.4**
Peak Fire Pressure (bar)	Estimate	150.1	128.8	115.9	150.6
	measured	150.0	128.0	114.0	151.0
	% difference	**0.1**	**0.6**	**1.7**	**-0.3**

DESIGN FEATURES FOR LOW NOISE LEVEL

Low noise features have been studied starting from the concept development stage. Acoustic analysis of the engine directed the design and optimization of the size and places of the ribs on the structural components.

Several low noise features of the engine are as the following:

- Gear train placed at flywheel side.
- Sound deadened, laminated sheet steel, deep drawn oil pan (damping properties of the material is optimized for 100 °C)
- Ladder-frame (stiffener at engine block skirt) application (Figure 5)
- De-coupled aluminum rocker cover
- Cast recesses on the both sides of the main bearing saddles
- Noise optimized symmetric cylinder block
- De-coupled ECU cooling plate, which is also utilized as a sound barrier at the intake side of the engine
- Integrated air intake manifold

Table 4- Noise Level Measurements

Noise level in dB(A) (Average SPL @ 1 m.)	Version A (*)	Combustion Noise (Version A)	Version B (**)
Rated power (2400 rpm)	**93.4**	**84.4**	**92.7**
Full load, 2000 rpm	91.2	82.0	90.6
Full load, 1500 rpm	86.5	79.0	85.9
Full load, 1000 rpm	85.2	84.1	84.4
Low idle	73.1	70.0	72.0

(*) **Version A:** Hardware configuration and engine calibration currently used on Cargo trucks
(**) **Version B:** The same hardware configuration and calibration with Version A except from additionally mounted rail cover.

The sound pressure levels of the version used on the Cargo trucks are shown on Table 4. Another version with a rail cover has given better results, as shown on the same table. Dominating source was identified as the air intake tube at intake side and this shows that the engine had a potential of further noise reduction for the applications where the intake tube could be de-coupled and/or placed far enough from the cylinder head.

Noise quality of the engine has also been evaluated during NVH development studies. AVL Noise Annoyance Index was used in evaluation of noise annoyance [2-4] and its four components loudness, sharpness, periodicity and impulsiveness were compared for baseline condition, final condition and a few of selected noise reduction features. As presented on Figure 10, overall noise quality differs for different hardware configurations mainly as the result of changes in loudness and sharpness. Version 39 has rail cover added to baseline engine. Version 40 uses sound absorbent lining inside the rail cover. Version 41 is obtained by attaching air compressor and PAS pump assembly to Version 40. Version 45 differs from V41 by oil pan material, ordinary steel instead of laminated steel. Therefore, it can be concluded that rail cover has a diverse effect on overall sound quality where the components attached to the cylinder block and laminated steel oil pan result in better quality.

LOW OIL CONSUMPTION

The market survey results had shown that truck owners had given high importance not topping up oil levels, which would require low oil consumption. Low oil consumption was considered as a key issue in emission development for Euro-4 and US2007 legislations and these led a challenging oil consumption target, <0.1 g/kW-hr at rated power conditions. High efficiency breather system, plate honing applied on the cylinder block, LLR (low leak rate) valve stem seals and well-matched liner and ring pack contributed in achieving the target.

Contribution of the assembly loads on cylinder bore deformation was minimized and parent bore distortion on the cold assembled engine could be kept in ±4 microns by the application of the plate honing process.

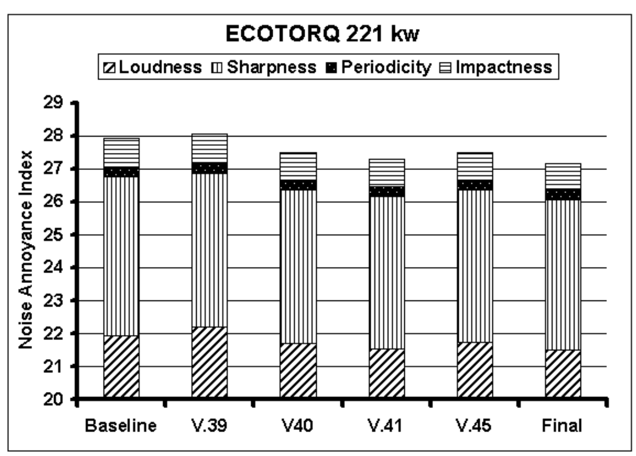

Figure 10 – Noise Annoyance Test Results with Different Versions.

ENGINE LIFE

Total of more than 40.000 hours of dynamometer testing and over 5 million km of vehicle testing were completed by the end of design verification stage.

Both liner wear and ring wear are very critical for the engine life [5]. Long duration dynamometer and vehicle durability tests ended up with satisfying results. As an example, typical results from a dynamometer durability test are shown in Figure 11 and in Figure 12.

EXTENDED OIL DRAIN INTERVAL

In order to ensure long oil drain interval, which generally depends on the field and service conditions, it is very important to limit the "soot in oil" rate since the total soot content in the oil is one of the dominant parameters that are reducing the lubricating oil life. Injection timing parameters including pre-, main and post injection strategy, piston top surface geometry and in-cylinder airflows were studied experimentally to minimize the rate. Achieved level is shown on Figure 13. These are the results without centrifugal oil cleaner or by-pass oil filter. Such a rate contributes to engine durability, as well; since increased soot content in lubricating oil accelerates wear of engine components [6-7].

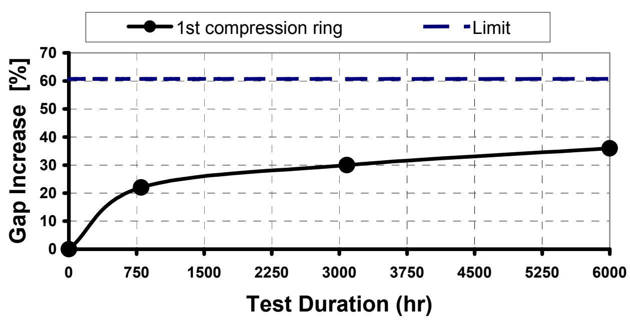

Figure 11 – Typical liner wear figures from a test engine

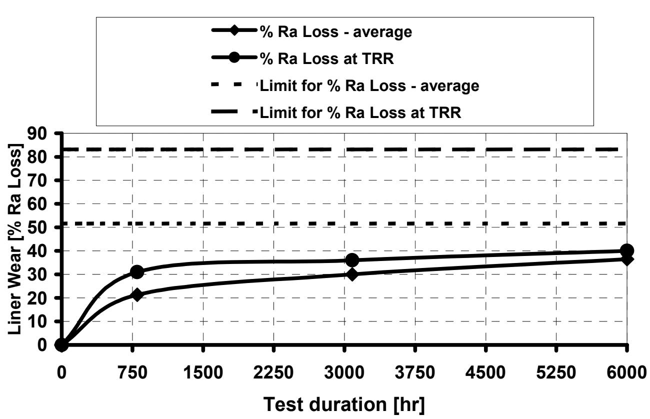

Figure 12 – Typical first compression ring wear figures from a test engine

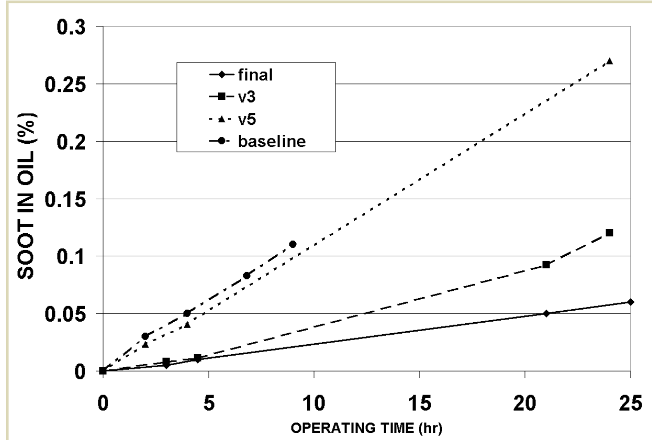

Figure 13 - Soot in oil rate tests (with the same and reduced amount of oil in the engine, 10.000 gr)

Figure 14 shows the difference in change of TBN (Total Base Number) with sulfur content of the fuel used. TBN almost stayed constant through 180 hours when diesel fuel that has sulfur content less then 500 ppm was used. When the same test was repeated with the fuel whose sulfur content was around 5000 ppm, TBN values reduced 50% in 125 hours.

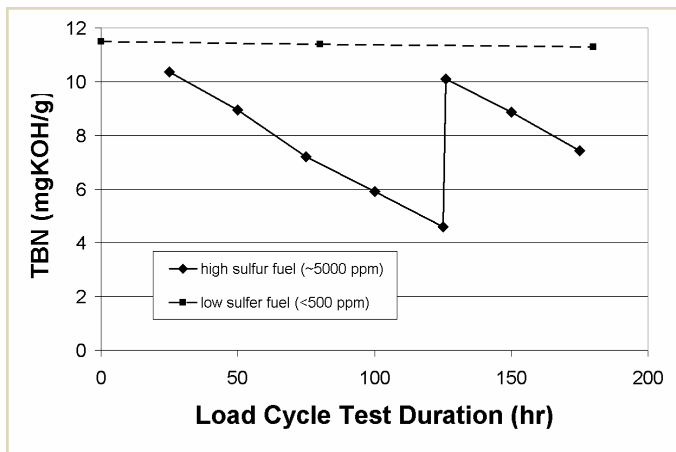

Figure 14- Change of TBN with low-sulfur and high-sulfur fuels.

Oil samples collected from test vehicles driven by customers have also been analyzed during the development stage. Figure 14 and Figure 16 includes data from field, where fuel with high sulfur content (up to 7000 ppm) have been used. It can be deduced from Figure 15 by a rough calculation that the limit of 3% soot content in lubricating oil will not be passed in 90.000 km mileage. On the other hand, as it can be seen from Figure 16, average life of the oil can be estimated as 45000 to 50000 km. Thus, as far as the oil drain interval of ECOTORQ engine for the particular field is concerned, change of TBN is more critical than soot amount accumulated in the lubricating oil because of the high sulfur content in the typical fuel used by the customers.

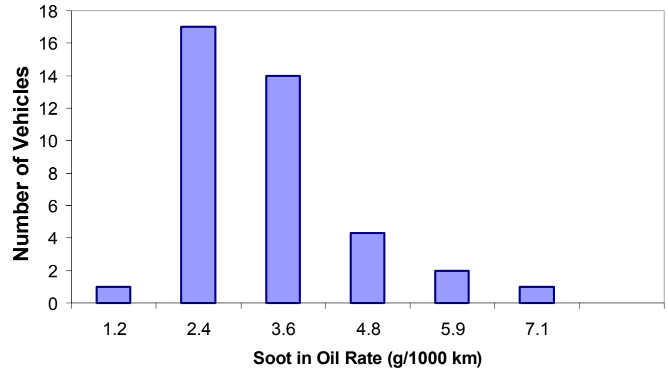

Figure 15- Soot in oil rate results from field. (Engines run with 28 liters oil in average)

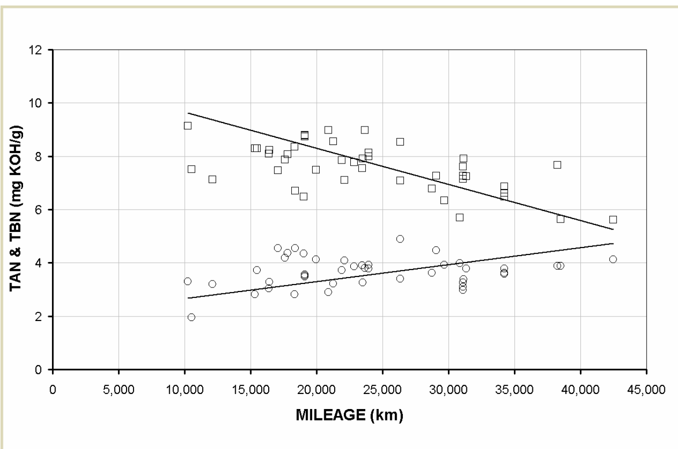

Figure 16- TBN and TAN (Total Acid Number) data from field.

ENGINE MANAGEMENT

Robert Bosch common rail fuel injection system is driven and controlled by an electronic control unit, EDC7. In addition to the standard engine management functions, some new functions were developed for the ECOTORQ engine including

- Storage of the duty-cycle of the engine
- Monitoring of the high-pressure circuit for faults in the injection system.
- Oil contamination prediction and oil change function.

The ECU can be diagnosed with Ford WDS (Worldwide Diagnostic System) and FORD OTOSAN EDS (ECOTORQ Diagnostic System) tools. EDS software is developed by FORD OTOSAN for the service shops. Its main features are trouble-code retrieval, data-logging, flash programming of the ECU, performing specific checks, problem-based diagnostics and help information, integrated service manual, spare-parts catalogue and wiring diagram.

SUMMARY

FORD OTOSAN is offering a new fully electronically controlled, common rail, 24 valve, in-line 6-cylinder, 7.33 liters heavy-duty diesel engine, ECOTORQ. The engine has satisfied challenging engineering targets that ensure high durability and low cost of ownership.

The design, the manufacturing techniques and the diagnostic tools developed all collaborate to satisfy diverse customer requirements.

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