

Crankshaft Peak Firing Pressure Bearing Capability Enhancement

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ABSTRACT

To uprate a 6-Cylinder In-line engine from 123 kW to 165 kW in power and upgrade the emission from Euro-2 to Euro-3 it was required to go for higher peak-firing pressures (PFP). The capability of Engine's Crankshaft to withstand the PFP was increased from 125 bar to 150 bar, maintaining the same cylinder centre distance. A crank-train model was used to achieve the required crankshaft strength for infinite fatigue life. The three aspects of crankshaft design, namely, crank strength, bearing selection, journal-pin lubrication and torsional vibration were considered during the design stage.

The strength to withstand 150 bar PFP was achieved by increasing the crank web-thickness. To maintain the same cylinder centre distance, crankpin and main-journal lengths were reduced. Increased throw stiffness was achieved by increasing the crankpin diameter to improve crankshaft torsional behaviour. Fatigue factor of safety for the webs were calculated as per the FVV (Forschungsvereinigung Verbrennungsmotoren) method. Bearings were modeled as hydrodynamic bearings and the calculations were carried out with Sommerfeld number as per Butenschoen procedure. Torsional vibration calculations were carried out as per the procedure in BICERA (British Internal Combustion Engine Research Association) handbook and the maximum shaft twist was controlled using an appropriate damper.

The new crankshaft passed the fatigue test in a test-rig for 150 bar PFP. Crankshaft lubrication and bearing life were validated by an engine endurance cycle to accelerate the failure. Torsional vibration measurement was carried out and the maximum free-end torsional deflection was within 0.2 deg. The results from crank-train simulation agreed well with the experimental results. The new crankshaft design is now in series production and has been proven in field for over a million km.

INTRODUCTION

The trend in Engine Development is going towards higher power rating to suit different applications and to meet the emission legislation norms. To meet these requirements, engine manufacturers increase the peak firing pressure. A higher peak firing pressure means better combustion and better efficiency. But at the same time, all the power train components have to be upgraded to withstand the high combustion pressure.

The crankshaft is one of the most critically loaded components in engine power train as it experiences cyclic loads in the form of bending and torsion during its service life. Fatigue is a major consideration in the design and performance evaluation of materials, components and structures since 90% of all mechanical failures are attributed to fatigue. The failure could be significantly reduced by using proper and efficient design and manufacturing. Focus should also be given on weight reduction due to need of higher payloads and reduced emissions.

The present work mainly focused on the three aspects of crankshaft design, namely, the crank strength, the bearing selection, the journal-pin lubrication and the torsional vibration. For meeting the torque requirement shown in [Figure 1](#), the capability of the existing crankshaft was enhanced to withstand PFP of 150 bar, keeping the cylinder centre distance the same. This achieved by increasing the crank web thickness, the crank pin diameter and the main journal diameter.

The 3D model of the crankshaft is prepared. Then the simulation for strength, bearing and torsion are carried out. FEA software is used to determine the crankweb stiffness. Finally the crankshaft is tested at a rig for estimating the fatigue strength. And after it passes all the tests, it is released

for series production and has been running in field successfully.

(See Figure 1 after last section of paper.)

MODEL DESCRIPTION

The simulation software comprises of three modules-strength, bearing and torsion. The circuit diagram for the simulation on the graphic user interface is shown in Figure 2. The crank train is modelled as a piston connected to connecting rod by a piston-pin and small-end bush. The connecting rod is connected to the crankpin by a hydrodynamic bearing connection. The crankshaft is modeled as a chain of a damper, journals, webs, crankpins and a flywheel. The journals are connected to the engine block by hydrodynamic bearings. The six pistons, the connecting rods, small-end, big-end and journal bearings are assumed to be identical in geometry and mass for simplification.

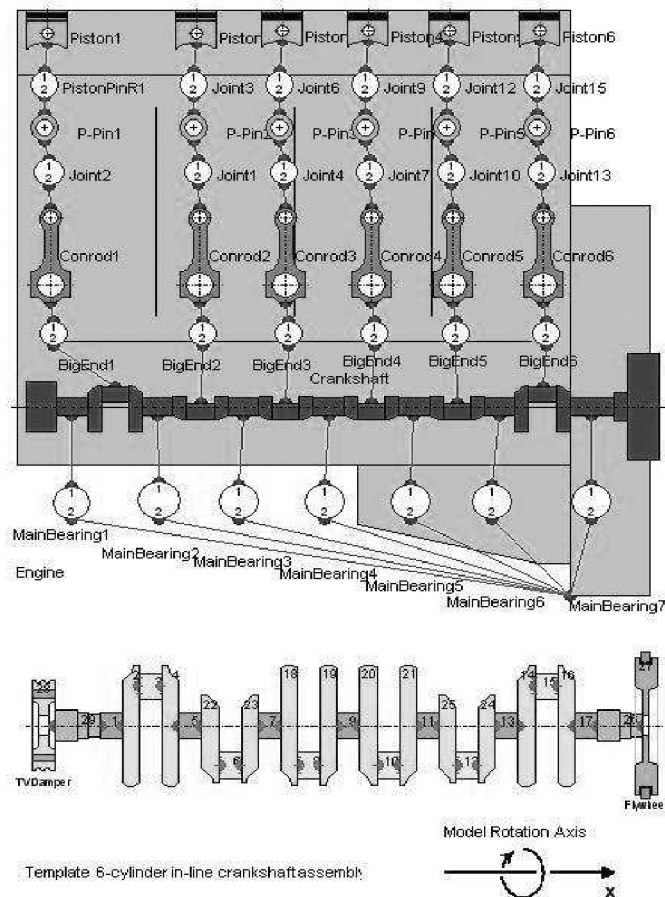


Figure 2. Schematic Diagram of 165kW Euro-3 engine crankshaft modeled in AVL Excite Designer

CRANK-TRAIN MODELLING

STRENGTH MODULE

The crankshaft is designed to last theoretically an infinite number of cycles under normal operating conditions. The forces applied to the crankshaft, mainly, are alternating, compression to tension, gas force to inertia force. The cyclic loading due to the periodicity of the load makes the crankshaft design a typical fatigue problem.

In line with the traditional design practice the crankshaft fatigue strength should be greater than the stress experienced by a factor equal to the Fatigue Factor of Safety (FFOS). The old crankshaft designed for 125 bar had a FFOS of 1.1. To avoid having an unnecessarily heavy crankshaft, the target of FFOS for new crankshaft was also set to 1.1.

The nominal stress in the crank web is related to the projected area of the web critical section (section across, crankpin fillet to journal fillet). From the bending moment on web neutral axis and section modulus of the area of projected web section the nominal stress in bending is evaluated. Considering the gas forces according to cylinder pressure diagram, reciprocating inertia forces due to piston and conrod reciprocating portion, centrifugal forces due to the conrod rotating portion, crankpin, crank webs and due to the counterweights the radial force in the webs is calculated as follows-

$$\sigma_{BN} = \frac{M_B}{Z} ; \sigma_{QN} = \frac{F}{A}$$

Considering the crankpin and journals as solid shafts with uniform cross-sections transmitting torque, the torsional stress is evaluated as follows-

$$\tau_N = \frac{T}{J}$$

The stress in the fillet is greater than the nominal stress by a factor equal to stress concentration factor (SCF) for fillets. SCF for the present crankshaft is determined as per the procedure published by FVV [1] and the actual fillet stresses in bending and torsion are obtained as follows-

Crankpin Fillet (Bending): $\sigma_{cB} = \alpha \sigma_{BN}$

Main Journal Fillet (Bending): $\sigma_{jB} = \beta_B \sigma_{BN} + \beta_Q \sigma_{QN}$

Crankpin Fillet (Torsion): $\sigma_{cT} = \alpha_T \tau_N$

Main Journal Fillet (Torsion): $\sigma_{jT} = \beta_T \tau_N$

The stresses in the fillets are due to a combination of bending and torsional loads. The combined stresses are calculated according to the energy distortion theory as follows-

$$\sigma_{comb} = \sqrt{\sigma_a^2 + 3\sigma_T^2}$$

The maximum (σ_{max}) and minimum (σ_{min}) combined stresses in fillets are calculated for a complete cycle assuming that the maximum and minimum bending stresses occur simultaneously with the maximum and minimum torsional stresses. And the fillet stress amplitudes and mean fillet stresses are calculated as follows:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}, \quad \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

The FFOS is determined as the ratio of the local fillet fatigue strength (σ_f) to the maximum fillet stress amplitude (σ_a)

$$FFOS = \frac{\sigma_f}{\sigma_a}$$

BEARING MODULE

The hydrodynamic bearing calculation is carried out by simplifying the Navier-Stokes equation assuming that the inertial forces are negligible compared to the frictional forces and the force due to gravity. The momentum equation is simplified assuming that the pressure gradient along the radial direction of the bearing is constant.

Then the equations of motion are found based on no-slip conditions. The Reynold's equation is obtained using the principle of conservation of mass and the equation of motion.

$$\frac{\partial}{\partial \phi} \left((1 + \varepsilon \cos \phi)^3 \frac{\partial \Pi}{\partial \phi} \right) + \left(\frac{D}{BR} \right)^2 \frac{\partial}{\partial z} \left((1 + \varepsilon \cos \phi)^3 \frac{\partial \Pi}{\partial \phi} \right) = -6 \left(\varepsilon \sin(\phi - \partial) - \frac{2\varepsilon}{\omega^*} \frac{d\partial}{dt} \sin(\phi - \partial) - \frac{2}{\omega^*} - \frac{d\varepsilon}{dt} \cos(\phi - \partial) \right)$$

To solve the Reynold's equation, the motion of the shaft is divided into plain rotary motion and plain displacement. The equations for both the motions are separately solved. Hydro-dynamically effective angular velocity defined by Fraenkel [2] is used to reduce the non-steady load to steady-load. In the steadily loaded cylindrical slider bearing, the journal pulls viscous oil due to its own rotation hence the load is balanced by the resultant of a pressure distribution in the oilfilm. In the state of equilibrium, there is no radial motion of the journal and the angular position of the minimum oilfilm thickness is always constant. The supporting power, eccentricity and the minimum oil film thickness are a function of bearing shell diameter, width, relative clearance of the bearings, dynamic oil viscosity and absolute journal angular velocity.

The Sommerfeld number is calculated as per the procedure suggested by Butenschoen. Sommerfeld numbers for Rotational and Displacement supporting power are calculated by the following equation.

$$S_{OD} = 0.5 \left(\frac{\varepsilon}{(1 - \varepsilon^2)^2} \left(\frac{D}{BR} \right)^2 \right) \sqrt{\pi^2 (1 - \varepsilon^2) + 16\varepsilon^2} a_1 \frac{(\varepsilon - 1)}{a_2 \varepsilon}$$

$$S_{OR} = \left(\frac{\frac{4}{\left(\frac{D}{BR} \right)^2}}{(1 - \varepsilon^2)^{2.5}} \right) \left(\left(\frac{\pi}{2} - 0.5\pi \cos(\varepsilon) \right) (1 + 2\varepsilon^2) + 1.5\varepsilon \sqrt{1 - \varepsilon^2} \right) a_1 \frac{1 - \varepsilon}{a_2 - \varepsilon}$$

TORSION MODULE

The torsion module calculates the natural frequency and critical orders, shear stress in crankpin, flywheel speed irregularity, damper heat dissipation and angular twist in each section of crankshaft. The inputs required for the above-calculated variables are-

- Crankpin, journal and web dimensions.
- Mass, moment of inertia and stiffness of Crank-webs
- Moment of Inertia of flywheel
- Moment of inertia of damper housing, damper hub and damper-inertia-ring.
- Damping constant of damper material and stiffness of damping element (rubber ring/ viscous liquid)

(See [Figure 3](#) after last section of paper.)

[Figure 3](#) shows how the crankshaft is modeled as a spring mass system. Each crankpin and journal is considered as a disc having moment of inertia, separated by crank webs having torsional stiffness acting as springs. The torsional vibration damper is modeled as a spring mass system having

two inertial masses of the hub and inertia ring separated by a rubber or viscous element having torsional stiffness. The natural frequency of the system is then calculated based on Holzer's Frequency Tabulation method.

CHANGE IN WEB DIMENSIONS

Since the crankshaft was to be used in the same base engine, a constraint was placed on the cylinder bore centre distance. With the increase in peak firing pressure, the force experienced by the crank web also increases. As a result the web critical cross-section was required to be strengthened to withstand greater bending. To achieve the same, web thickness was to be increased in the new crankshaft. The increase in the web thickness was compensated with a reduction in the crankpin and the journal lengths. This modification allowed maintaining the same bore centre distance. The crank throw stiffness was increased to improve the crankshaft torsional behaviour. An increased overlap dimension was achieved by increasing the crankpin diameter.

DISCUSSION

CRANK-THROW INERTIA

The new crankshaft was modelled in 3D. The inertia of throws with and without the counter-weights was calculated in Pro-E using the crankshaft 3D Model. [Table 1](#) shows the increase in crank-throw inertia for the new crankshaft.

(See [Table 1](#) after last section of paper.)

To sustain 150 Bar PFP, modifications were also made in the piston and the connecting rod. The moment of inertia of the new crankshaft partially increased due to the increased mass of the newly developed connecting rod and piston.

CRANK-THROW STIFFNESS

The torsional stiffness of cranks webs was determined by finite element analysis (FEA). The crank-throw was meshed with tetrahedral elements. The face of one journal was fully constrained in all six degrees of freedom and a torsional load of 1000 Nm was applied about the crank-throw rotation axis. The maximum angular twist was obtained as a result of FEA. The throw stiffness was then calculated as the torque required to produce unit angular twist. [Table 2](#) shows the increase in torsional stiffness for the new crankshaft. A FEA based approach for stiffness estimation was chosen to improve the accuracy of the calculation. The conventional method using empirical relationships as presented in [3], [9] can also be used for preliminary stiffness calculation.

(See [Table 2](#) after last section of paper.)

(See [Figure 4](#) after last section of paper.)

(See [Figure 5](#) after last section of paper.)

CHANGE IN BEARING DIMENSIONS

To withstand higher combustion loads and the change in dimensions of the crankpin and journal, the conrod big-end bearing and the journal bearing was to be redesigned. Based on the surface roughness $R_{zDIN} = 2.5$ ($R_p=1.25$), it was required to maintain a minimum oil-film thickness (MOFT) of greater than 1.3 microns for both-the crankpin and journal bearing to avoid surface-to-surface contact. For constant bearing-shaft geometry, MOFT is a function of the supply pressure of the oil to the bearing and the load. At lower speeds, the oil-pump speed is lower; consequently the oil pressure in the main-oil gallery is lower. Hence, the MOFT is critical at lower speeds at full load conditions. To calculate the loads on the bearings, the effect of the gas forces and the reciprocating inertia forces is taken into account. For main bearings, the effect of bending moment is taken into account. This was necessary as the 6-cylinder crankshaft was designed with eight counter-weights (CW) rather than the ideal design with twelve CWs. This was a design decision taken to reduce the material cost of the crankshaft in production.

(See [Figure 6](#) after last section of paper.)

(See [Figure 7](#) after last section of paper.)

[Figure 6,7,8,9](#) show the results of bearing simulation as obtained for the final configuration. It can be observed that the MOFT is critical for the lower shell for the journals bearings and the upper shell for the conrod bearings.

(See [Figure 8](#) after last section of paper.)

(See [Figure 9](#) after last section of paper.)

An appropriate bearing geometry was arrived at by several iterations to obtain MOFT greater than 1.3 microns at 1000 rpm, which was the lowest high torque speed. The unit load on crankpin and journal bearing was calculated by simulation. The bearing material used for the old crankshaft was found to be on the margin for the allowable limit. A material was chosen with an allowable load limit of upto 40 N/mm². [Figure 10](#) shows the unit load on main bearings at full load speeds. The maximum unit load is 32 N/mm², which is below the load allowable limit. The position of oil-groove in the bearing shell and the oil hole for oil-supply were fixed based on the load and the journal eccentricity.

(See [Figure 10](#) after last section of paper.)

TORSIONAL ANALYSIS

The natural frequency, critical orders and angular twist were calculated based on the torsional model. [Figure 11](#) and [12](#)

show the natural frequency modes and critical speeds for the new crankshaft. To reduce the angular twist for 4.5, 6 and 7.5 orders a suitable viscous damper was chosen. The maximum angular deflection with the viscous damper was found to be below 0.2 deg at crankshaft hub, which is well within the design target.

The properties of the silicon fluid in the viscous dampers are a function of the heat dissipated by the damper. The damper surface area should be sufficient to transfer the heat produced during operation for assuring the life of the damper [4]. The heat dissipated by the viscous damper was calculated and the existing damper area was found to be sufficient to dissipate the heat produced for an ambient temperature of 90 °C.

(See [Figure 11](#) after last section of paper.)

(See [Figure 12](#) after last section of paper.)

The minimum flywheel moment of inertia was calculated to restrict the maximum cyclic speed irregularity at flywheel below 10 % and maximum angular acceleration below 2000 rad/s².

(See [Figure 13](#) after last section of paper.)

CRANKSHAFT FATIGUE TESTING

Fatigue testing of the new crankshaft was carried out to validate the design at 150 bar. The aim of the fatigue tests was to derive an expectation value for the component fatigue strength. The crankshaft was tested on a bending rig test. The fail-safe level calculated based on the testing and was found to be greater than 150 bar. Beyond 150 bar test load, cracks were found on the ground surface of the crankpin and not in the crankpin fillets. The nature of the cracks suggested the cause to be surface notch effect. A potential increase in fatigue strength can be achieved with an improvement of the surface finish of the crankpin.

In terms of fatigue failures of the crankshaft, bending is considered to be the most critical loading. Therefore only bending tests were performed. The most critical areas are the crankpin fillets. At the test-rig, the crankshaft was clamped at a crankpin and a main journal and the bending load was applied at another main journal.

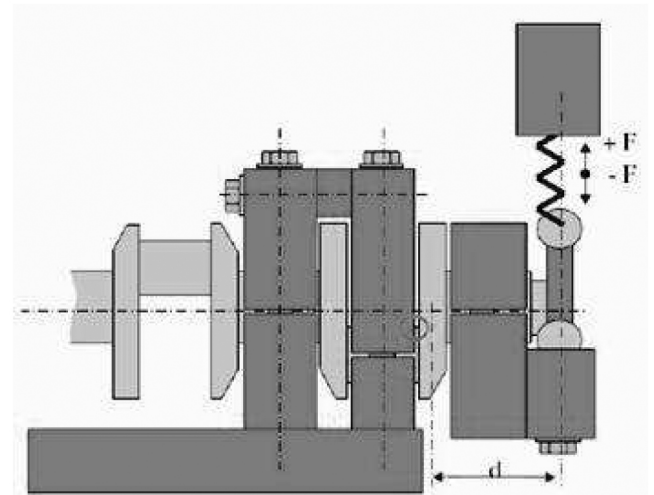


Figure 14. Sketch of Crankshaft Fatigue Test Setup

The following parameters were used:

- Ultimate number of load cycles: 5 million
- Wave form: Sinusoidal
- Test Frequency: 35 Hz
- Constant load amplitude and load ratio ($R = -0.14$)

With the given mass forces and an assumed PFP of 150bar, a nominal bending moment of 900N, was calculated with a load ratio of -0.14 .

(See [Table 3](#) after last section of paper.)

Based on the listed results a statistical evaluation was done. A fatigue limit model was used to account for the deviation of a fail-safe stress level from the measured fatigue data considering a Log-Gauss distribution. The result of the model is a fail-safe level at 970 Nm with a survival probability of 95%. Since the maximum calculated load under engine operation is 900Nm the crankshafts are expected to sustain a PFP of 150 bar with a safety factor of 1.08.

(See [Figure 15](#) after last section of paper.)

The crankshaft was further tested for combined torsional and bending strength on an actual running engine in a 1000 h overload (equivalent to 300,000 km on road) engine endurance cycle with a PFP of 140 bar. The bearings and the crankshaft dimensions taken after endurance showed no signs of excessive wear or a need of regrinding. Over 500 crankshafts have since been proven in field for various engine ratings varying from 123 kW to 165 kW validating the robustness of the design.