Cranktrain Analysis Using ENGDYN and VALDYN

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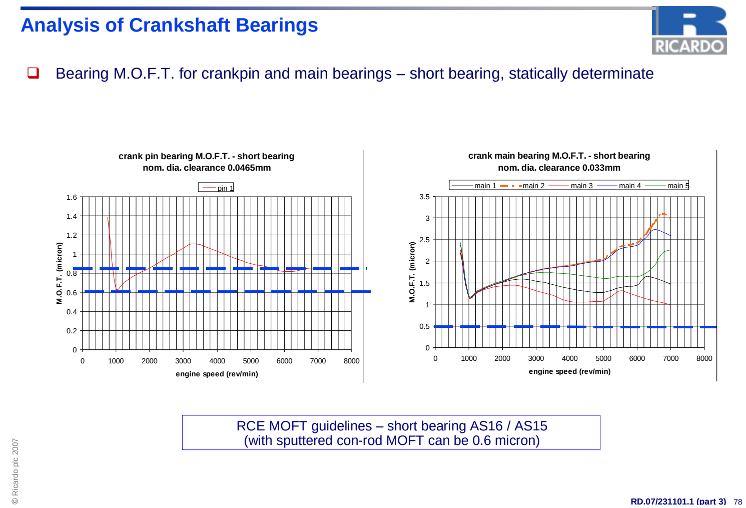
- Concept Analysis
  - Bearings
  - Torsional Vibration
  - Bolted Joints
  - Durability
- Definitive Analysis
  - Durability
  - Bearings

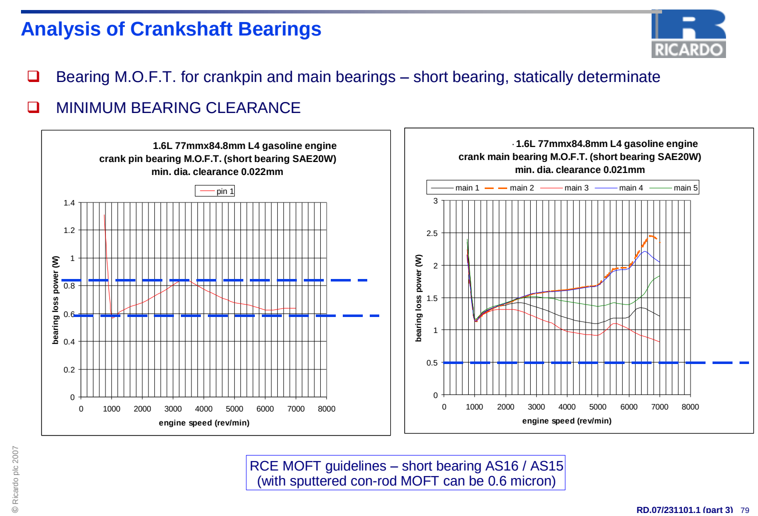
Crankshaft Analysis

	Concept (Non-FE)	Definitive (FEA Based)
Objectives	Bearing Performance Torsional Vibration Durability Assessment Damper Tuning, Flywheel sizing Crankshaft Balancing Basic crankshaft geometry	Axial and Bending Vibration (Flywheel Whirl) Bearing Wear Bearing/Journal Tilt Detailed Durability Assessment (Noise and Vibration assessment)
Models	VALDYN : 1-D Torsional Model (FE stiffnesses) ENGDYN : Rigid lumped-mass model	ENGDYN: 3-D Dynamic Crankshaft Model Cylinder Block Model - Compliant or Dynamic
Analysis	1-D Torsional Vibration Analysis - LFD or Time Domain Static Loads Analysis - Determinate Bearing Loads - Mobility oil film prediction Classical Stress Analysis	Static load prediction - Indeterminate Bearing Loads 3-D Dynamic Analysis - Coupled crank/block/oilfilm solution - Dynamic Bearing loads - Mobility and EHL oil film prediction FEA Stress Analysis

Crankshaft analysis (1) – General comments

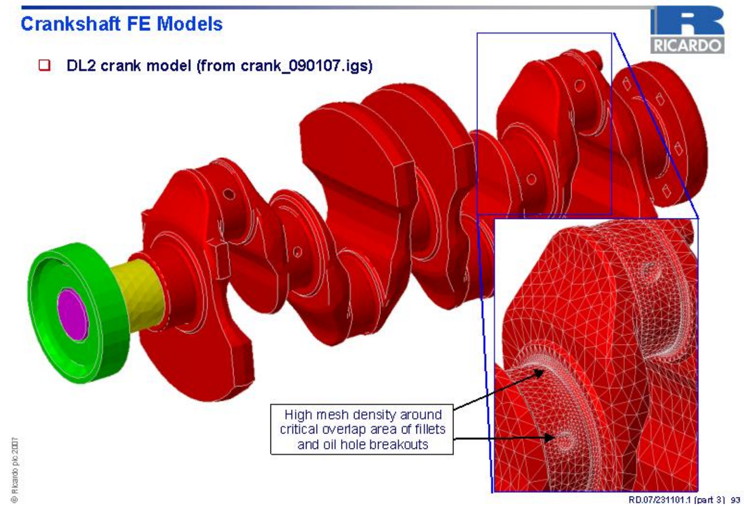
- **Ricardo routinely perform the following analyses on crankshafts during the <u>concept</u> design phase**
  - Calculation of peak specific load and minimum oil film thickness
    - at each bearing (big ends and mains)
    - at full load and no load across the speed range
  - Calculation of torsional vibration
    - crank nose TV displacement
    - twist along the length of the crankshaft
    - TV damper sizing and durability analysis
  - Calculation of stress and safety factor at critical locations using classical analysis technique
    - at oil hole breakouts
    - at all fillets including crank nose fillets
  - Calculation of joint cover factors
    - at crank nose
    - at flywheel bolted joints
  - Calculation of friction

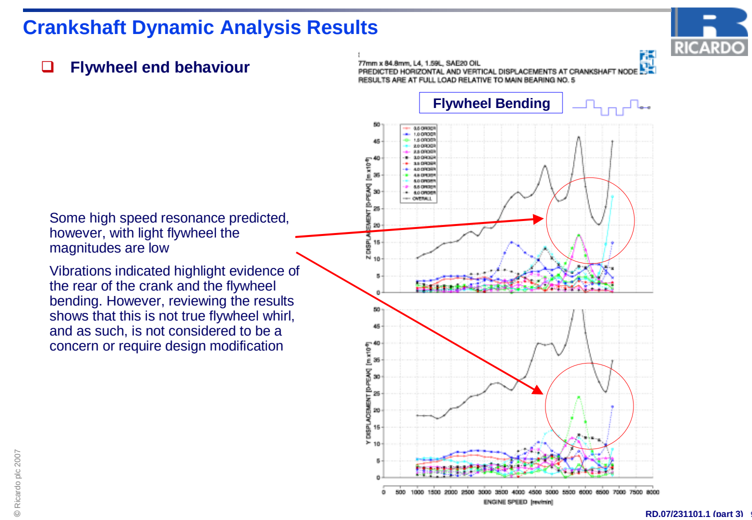




Crankshaft analysis (2) – General comments

- **Ricardo routinely perform the following analyses on crankshafts during <u>definitive</u> design phase**
  - Calculation of stresses and safety factors using finite element technique
  - Calculations of 3D dynamic motion using finite element models
    - Torsional vibration
    - axial vibration
    - bending vibration
    - flywheel whirl modes
  - Calculation of shaft relative tilt angles at each bearing
- **Ricardo have the capability to perform other advanced, non-standard analyses as required**
  - Finite volume hydrodynamic bearing analysis
  - Elastohydrodynamic bearing analysis





Crankshaft analysis (3) – Concept level bearing analysis assumptions

- **For concept studies Ricardo use ENGDYN and normally make the following assumptions for bearing analysis work**
  - Rigid crankshaft and rigid bearing housings
    - Perfectly cylindrical bearing journals and housings
  - Bearing reaction forces calculated by statically determinate loading so each main bearing is influenced by
    - Gas forces from adjacent cylinders only
    - Inertial effects of the pistons and rods from adjacent cylinders only
    - Centrifugal effects of adjacent crank webs only
    - The crank is effectively pin jointed at each main bearing
      - Loads from more distant bays ignored
      - Load sharing by other main bearings ignored
  - Booker Mobility Method used for solution of Reynolds equation
    - Short bearing assumption normally used by Ricardo
    - Method adapted to account for effects of oil holes and grooves
    - Method adapted to account for oil temperature increase due to shearing
    - Mean cold bearing clearance used by Ricardo for calculation of film thickness

Crankshaft analysis (4) – Concept level bearing analysis input data

- **To perform concept analysis of crankshaft bearings the data shown are typically required**

Cylinder pressure curves

Parameter	Value
Full load peak cylinder pressure (bar) @ speed (rpm)	53.4 @ 1000
	59.0 @ 1500
	68.1 @ 2000
	73.6 @ 2500
	77.1 @ 3000
	77.5 @ 3500
	80.0 @ 4000
	83.8 @ 4500
	84.4 @ 5000
	82.7 @ 5500
	79.8 @ 6000
	73.3 @ 6500
Part load peak cylinder pressure (bar) @ speed (rpm)	55.7 @ 6500
No load peak cylinder pressure (bar) @ speed (rpm)	29.1 @ 7000

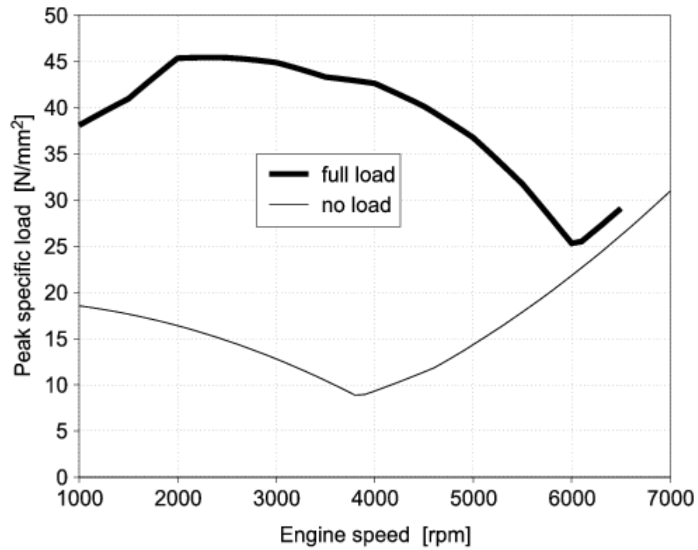
Crankshaft mass properties

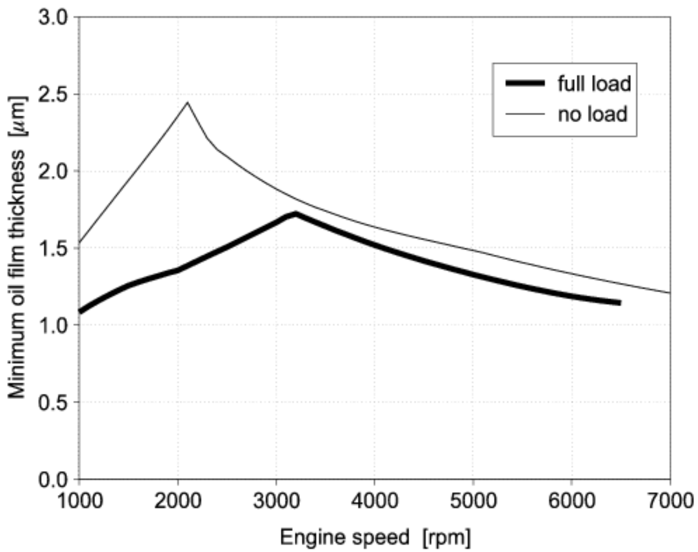
Location	Mass (kg)	X (mm)	Y (mm)	Z (mm)
Web1	0.659	1.46	26.763	0
Web1 c/w	0.794	-0.541	-37.611	0
Crankpin1	0.341	0	-0.1	0
Web2	0.659	-1.46	26.763	0
Web2 c/w	0.794	0.541	-37.611	0
Web3	0.659	1.46	-26.763	0
Web3 c/w	0.794	-0.541	37.611	0
Crankpin2	0.341	0	0.1	0
Web4	0.659	-1.46	-26.763	0
Web4 c/w	0.794	0.541	37.611	0
Web5	0.659	1.46	-26.763	0
Web5 c/w	0.794	-0.541	37.611	0
Crankpin3	0.341	0	0.1	0
Web6	0.659	-1.46	-26.763	0
Web6 c/w	0.794	0.541	37.611	0
Web7	0.659	1.46	26.763	0
Web7 c/w	0.794	-0.541	-37.611	0
Crankpin4	0.341	0	-0.1	0
Web8	0.659	-1.46	26.763	0
Web8 c/w	0.794	0.541	-37.611	0

Parameter	Value
Number of cylinders	4
Bore (mm)	86.0
Stroke (mm)	94.6
Rated speed (rpm)	6000
Firing order	1-3-4-2
Crankshaft rotation	clockwise
Cylinder Offset (mm)	0.0
Pin Offset (mm)	0.8
Connecting rod length (mm)	145.7
Connecting rod mass including bolts and bearing shells (kg)	0.597
Distance between big end and rod centre of mass (mm)	35.73
Connecting rod moment of inertia about centre of mass ($kgmm^2$)	2128
Piston assembly mass including rings and piston pin (kg)	0.4955
Cylinder spacing (mm)	96.0
Main journal spacing (mm)	96.0
Main journal diameter (mm)	55.0
Main journal length (mm)	27.0
Main bearing shell effective length (mm)	18.2
Main bearing shell radial clearance (mm)	0.022
Main bearing shell groove	partial
Main bearing oil feed from	bearing
Groove width (mm)	4
Start of groove (deg)	270
End of groove (deg)	90
Crank pin diameter (mm)	48.0
Crank pin journal length (mm)	24.4
Crank pin shell effective length (mm)	17.8
Crank pin bearing radial clearance (mm)	0.024
No. of pin journal holes	1
Crank pin journal oil hole type	Leading
Crank pin journal oil hole diameter (mm)	5.4
Crank pin journal oil hole angle (deg)	45
Oil grade	5W30
Oil supply pressure (bar)	4.0
Oil supply temperature (°C)	130

Crankshaft analysis (5) – Concept level bearing analysis results

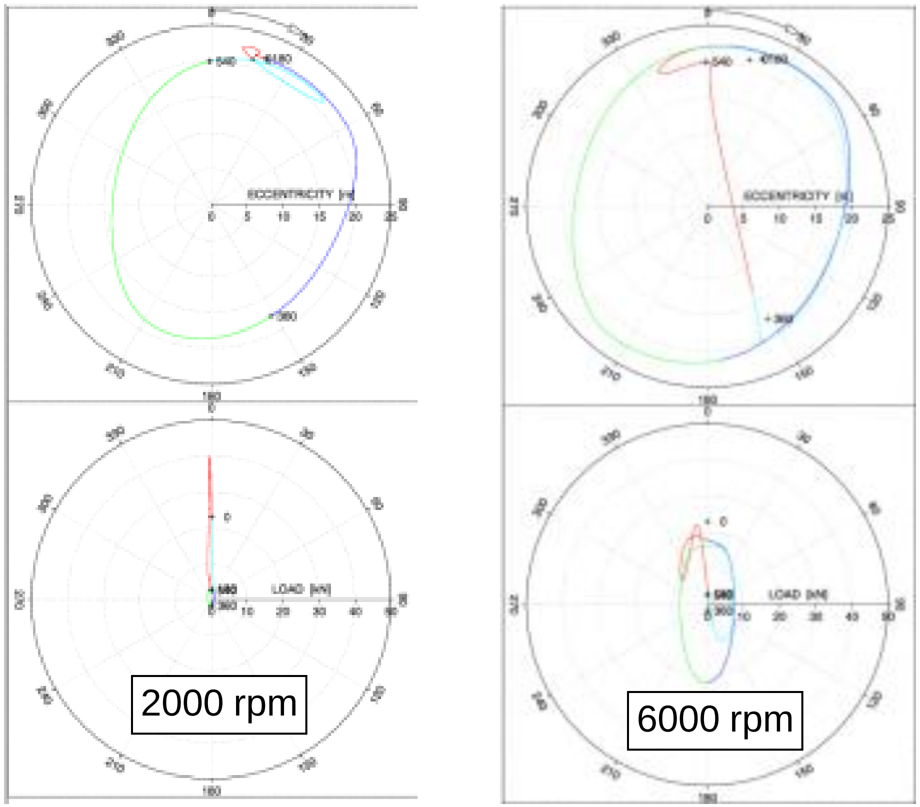
- **Ricardo normally present these results as graphs of**
  - peak specific load against engine speed at full load and no load
  - minimum oil film thickness against engine speed at full load and no load
- **Ricardo have well-developed limits for these values**
  - dependent on bearing type (big end or main)
  - dependent on engine application
  - dependent on bearing material
- **Ricardo normally tabulate key values and compare with limits**
- **Beware**
  - Ricardo limiting values should only be applied when using Ricardo calculation methods
  - Others use different calculation methods and have different limits

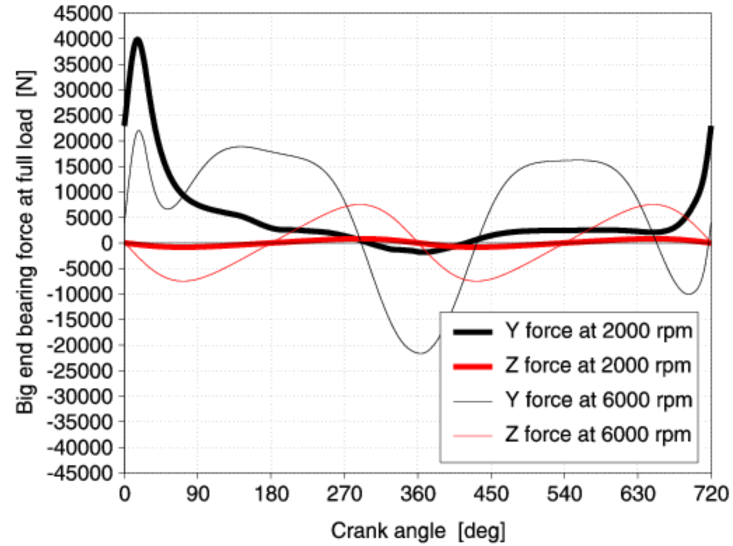


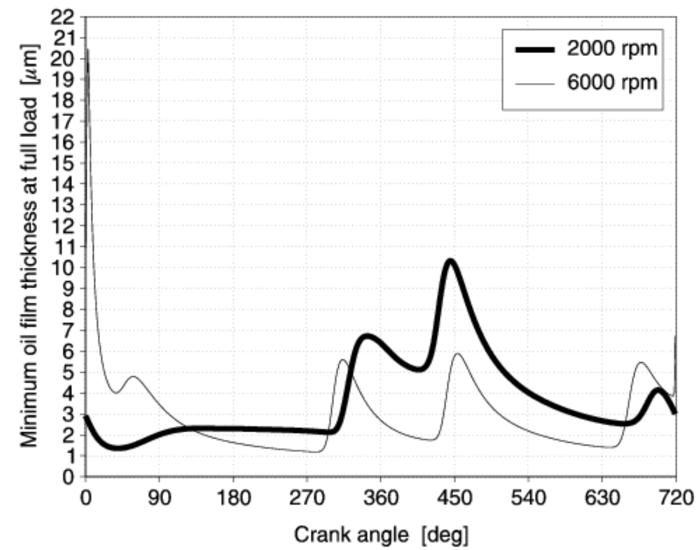


Crankshaft analysis (6) – Concept level bearing analysis results

- **The interplay between gas forces and inertia force at different speeds can be investigated by making plots of**
  - bearing force and film thickness against crank angle
  - bearing journal eccentricity

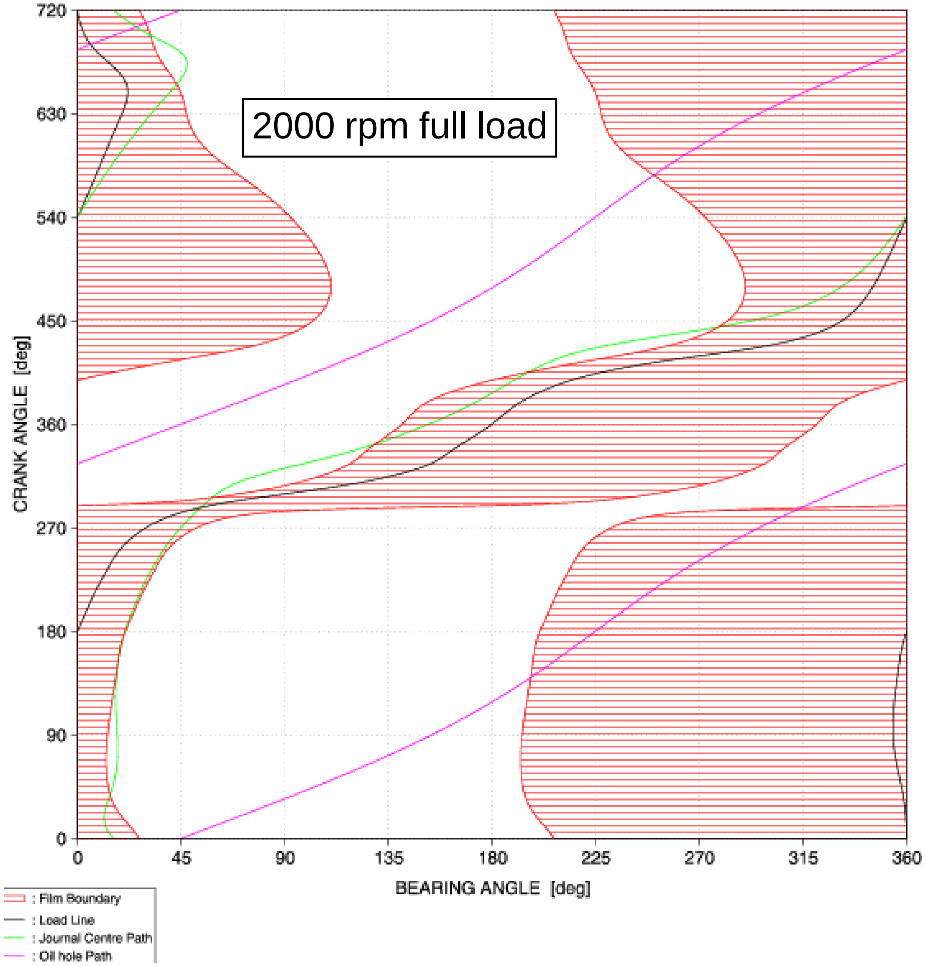


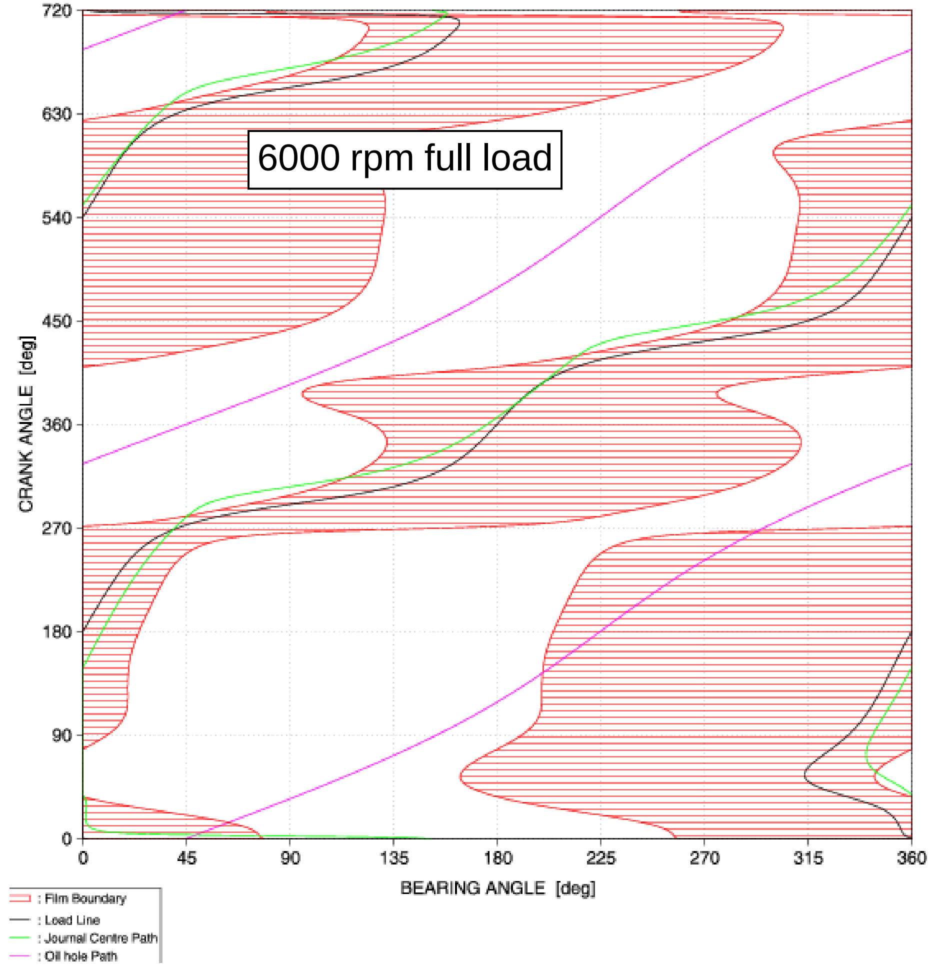




Crankshaft analysis (7) – Concept level bearing analysis results

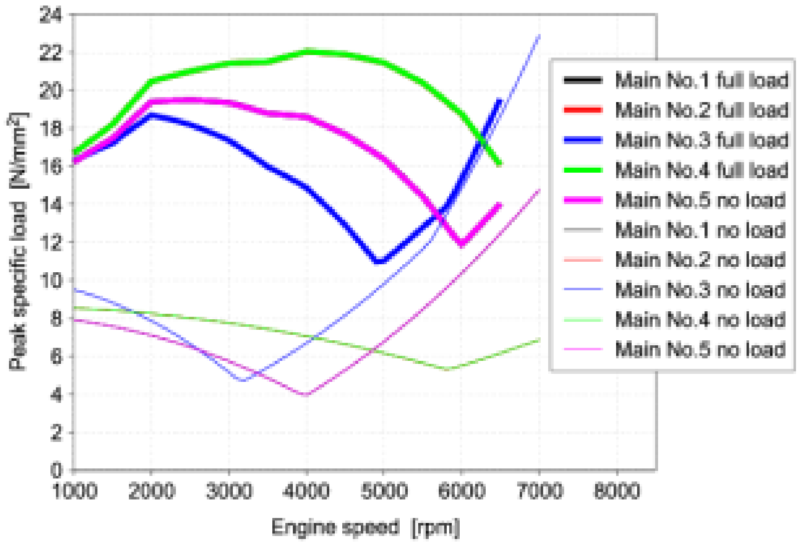
- **Oil film extent maps can be used to visualise**
  - the direction of the load relative to the regions in which a pressurised oil film is present
  - the location of the oil supply hole

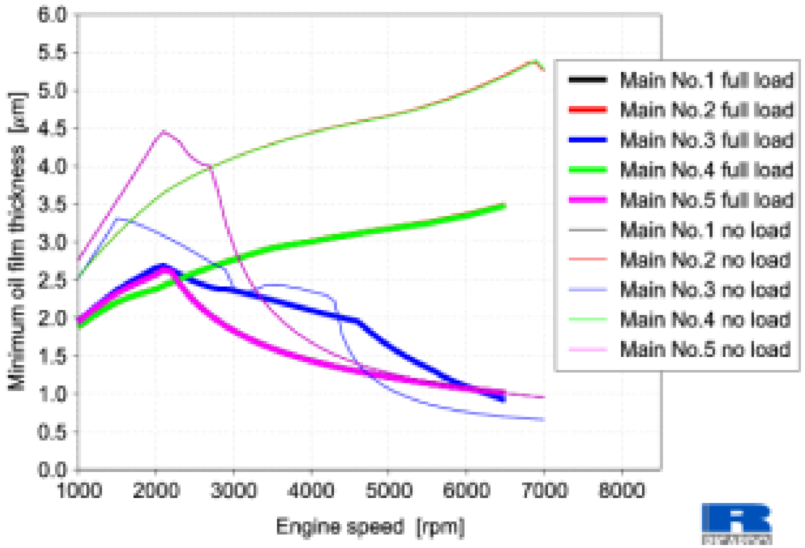




Crankshaft analysis (8) – Concept level bearing analysis results

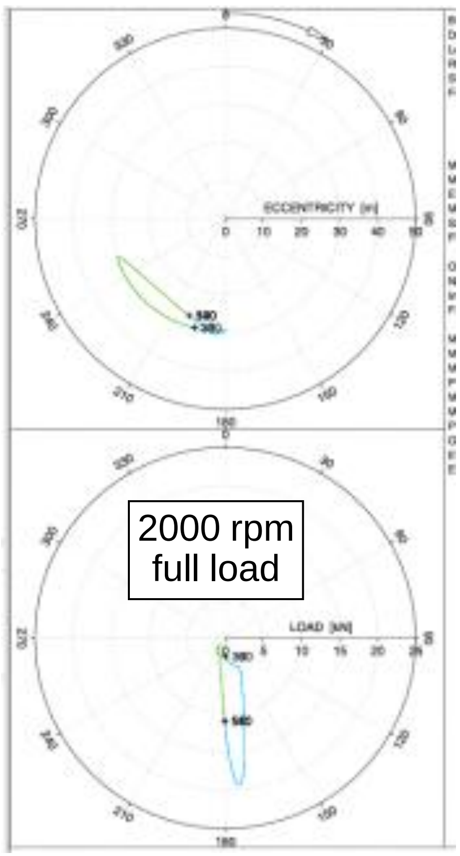
- **Similar analysis results can be calculated at main bearings**
- **Peak forces and minimum film thickness values are similar at**
  - Bearings 1 and 5
  - Bearings 2 and 4
- **Centre main bearing (3) is often the worst case at high speed**
  - No load can be worst case in a I4 engine
  - Dependent on level of counterweighting and value of overspeed used for design

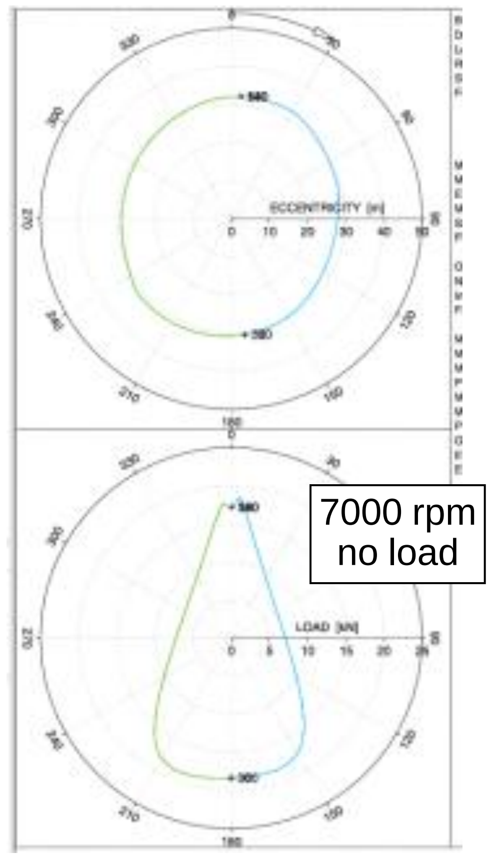


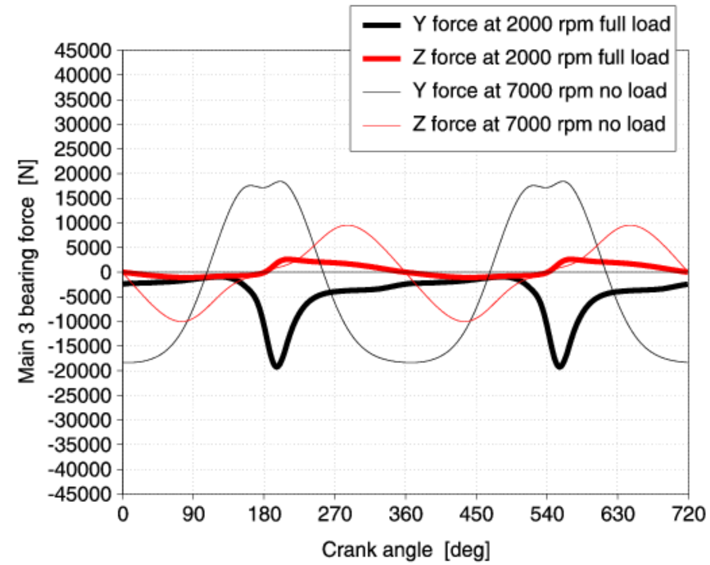


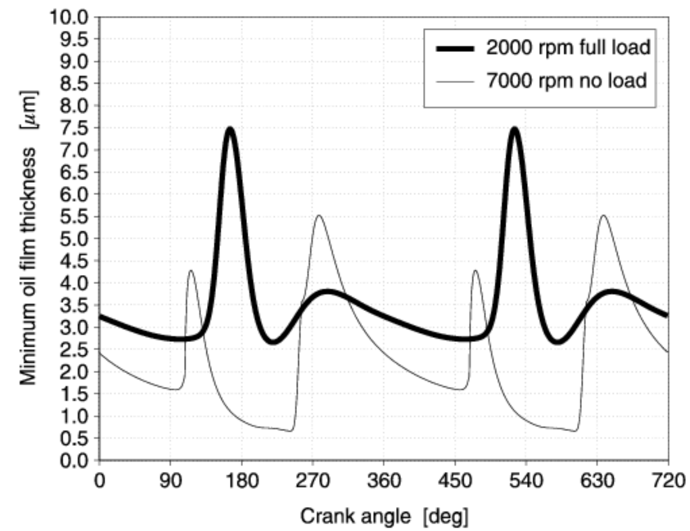
Crankshaft analysis (9) – Concept level bearing analysis results

- **The interplay between gas forces and inertia force at different speeds can be investigated by making plots of**
  - bearing force and film thickness against crank angle
  - bearing journal eccentricity









Crankshaft analysis (10) – Concept level bearing parametric studies

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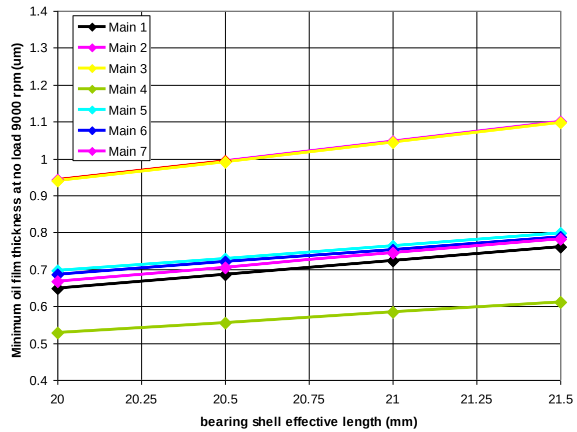
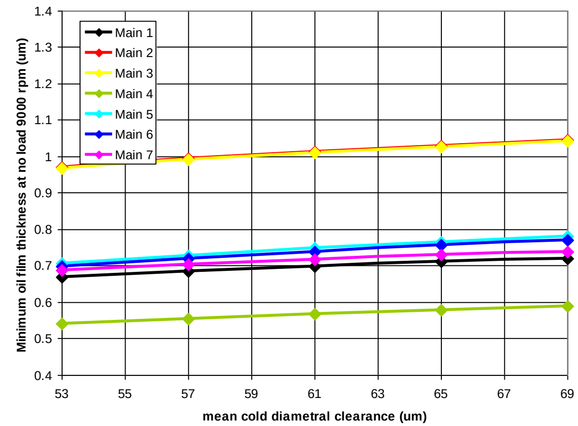
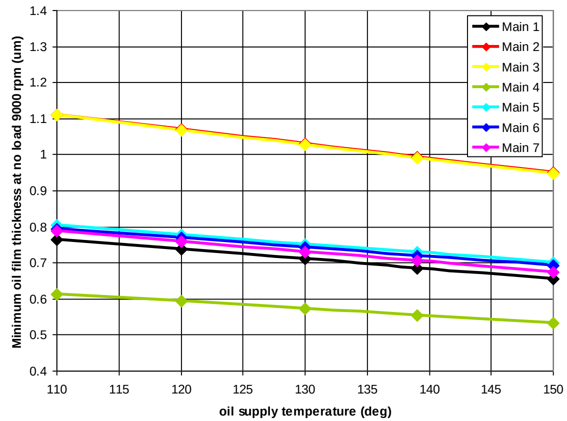
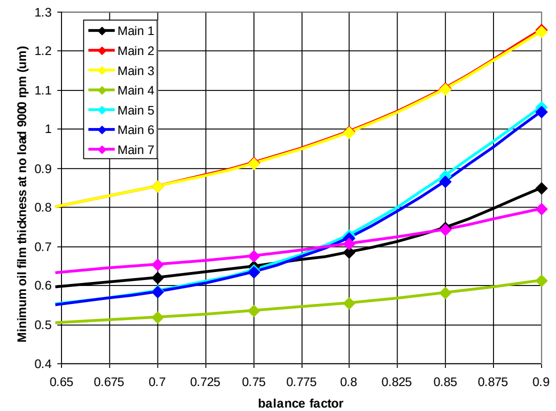
- **Use of these assumptions gives a method that can calculate all results at each bearing across the speed range in 100 rpm intervals in just a few seconds**
- **So this method can be used to make extensive parametric studies of the following on peak specific load, minimum oil film thickness, oil pressure in bearing, oil temperature, oil flow rate**
  - Bearing journal diameter
  - Bearing shell length
  - Bearing clearance
  - Counterweight MR
  - Piston assembly mass
  - Oil supply temperature
  - Oil hole location
  - Oil viscosity grade
  - etc



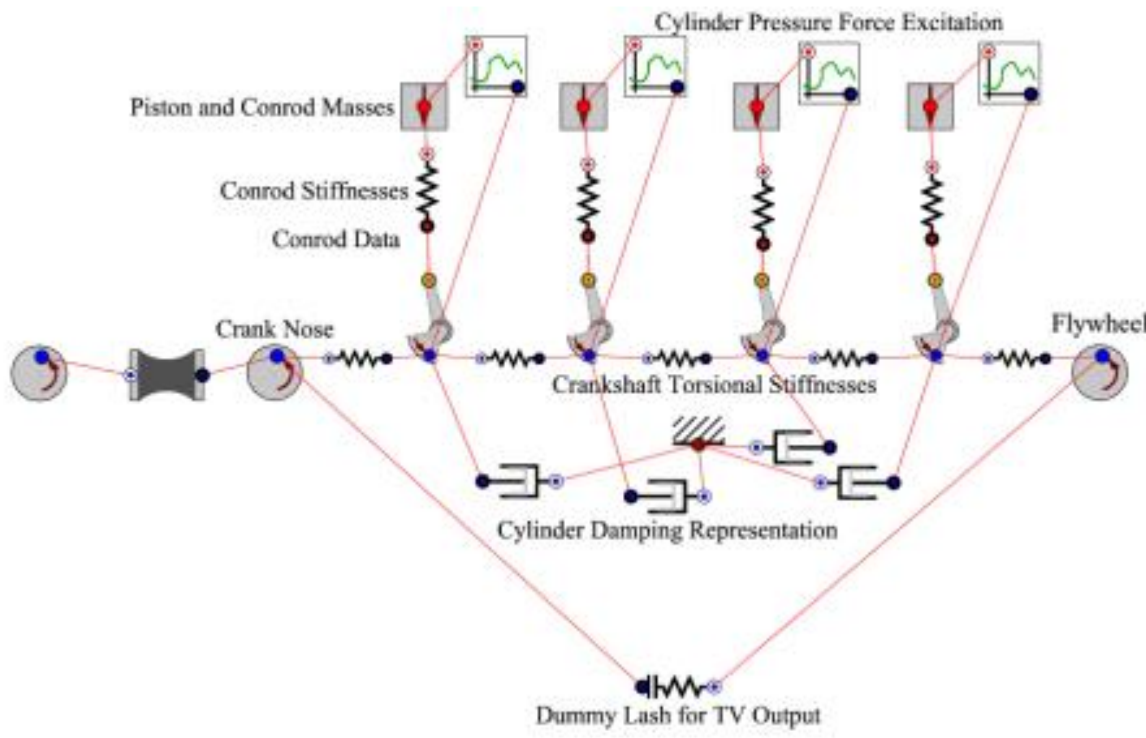






Crankshaft analysis (11) - Concept level TV analysis assumptions

- **For concept studies Ricardo use VALDYN to calculate crankshaft torsional vibration and normally make the following assumptions**
  - Purely rotational dynamics with linear response
  - Analysis in frequency domain
  - Variable inertial effect of pistons and connecting rods modelled as constant inertia applied at each crank pin
  - Inertia of each crank element included and typically “lumped” at cylinders, crank nose and flywheel



  - Torsional stiffness of each crank element used to connect lumped inertia values
  - Torsional damping due to friction, material hysteresis and bearing oil films applied between each crank element and constant speed rotational node

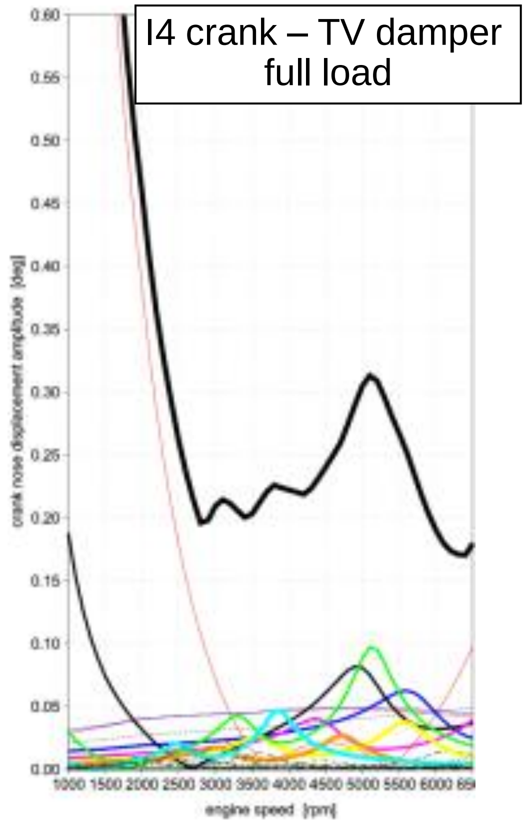
Crankshaft analysis (12) - Concept level TV analysis input data

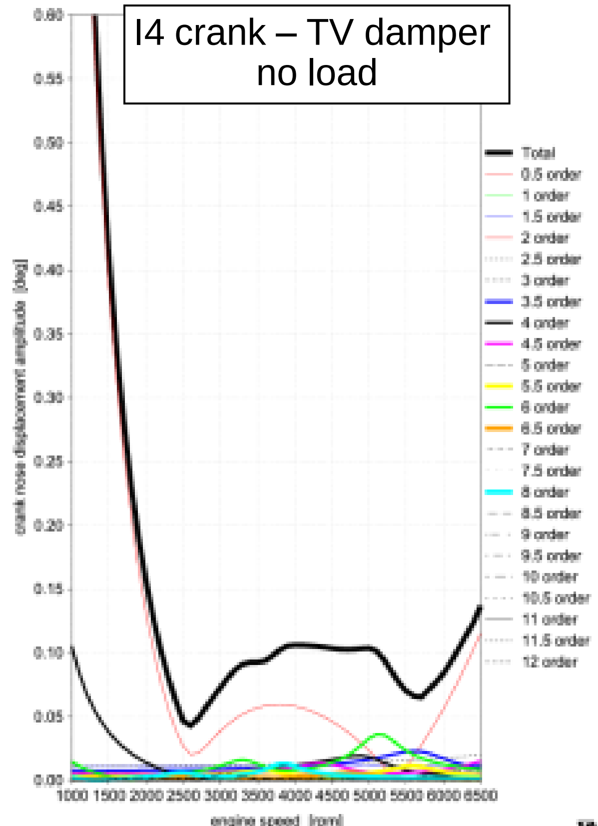
- **Component inertia values are usually obtained from a 3D CAD model**
  - Remember to include effect of pistons and rods
  - Remember to include clutch with single mass flywheel
- **TV damper data (ring inertia, hub inertia, rubber stiffness)**
  - from damper supplier
  - from CAD data
  - use benchmarking and/or package constraints to
- **Crankshaft element torsional stiffness data**
  - from FE model if available
  - or using classical equations to estimate values if not
- **Cylinder pressure data from measurements or from performance simulation model**
- **Crankshaft torsional damping from experience or correlation or model to measured data**

Data	Data
Damper ring inertia (kgm^2)	0.0055
Crank nose inertia (kgm^2)	0.001155
Cylinder 1 inertia (kgm^2)	0.0065732
Cylinder 2 inertia (kgm^2)	0.0065732
Cylinder 3 inertia (kgm^2)	0.0065732
Cylinder 4 inertia (kgm^2)	0.0065732
Flywheel etc (kgm^2) AUTO	0.174743
Flywheel etc (kgm^2) DMF	0.112772
Damper stiffness 1 (Nm/rad)	25500
Crank nose stiffness 2 (Nm/rad)	171035
Crankshaft stiffness (cyl 1 to 2) (Nm/rad)	403300
Crankshaft stiffness (cyl 2 to 3) (Nm/rad)	416625
Crankshaft stiffness (cyl 3 to 4) (Nm/rad)	403300
Crankshaft stiffness (cyl 4 to flywheel) (Nm/rad)	661013
Reciprocating mass (kg)	0.6419
Con rod stiffness (N/mm)	294886
Con rod damping (Ns/m)	5503
Crank throw (mm)	47.3
Connecting rod length (mm)	145.7
Phase angle (cyl 1)	0
Phase angle (cyl 2)	540
Phase angle (cyl 3)	180
Phase angle (cyl 4)	360
Initial position of piston 1 (mm)	0
Initial position of piston 2 (mm)	94.6
Initial position of piston 3 (mm)	94.6
Initial position of piston 4 (mm)	0
Crankshaft damping (Nms/rad)	1.5
Dyno preload (Nm)	0
Cylinder pressure force files	cp.*
Multiplier for force files	580.88

Crankshaft analysis (13) - Concept level TV analysis results

- **Ricardo normally present a graph of crank nose TV amplitude (1/2 peak to peak) against engine speed**
  - at full load
  - at no load
- **This graph typically shows the order content of the crank nose motion as well as the total value**
- **The TV motion plot shows vibration amplitude relative to a node rotating at continuous speed so it is not twist**

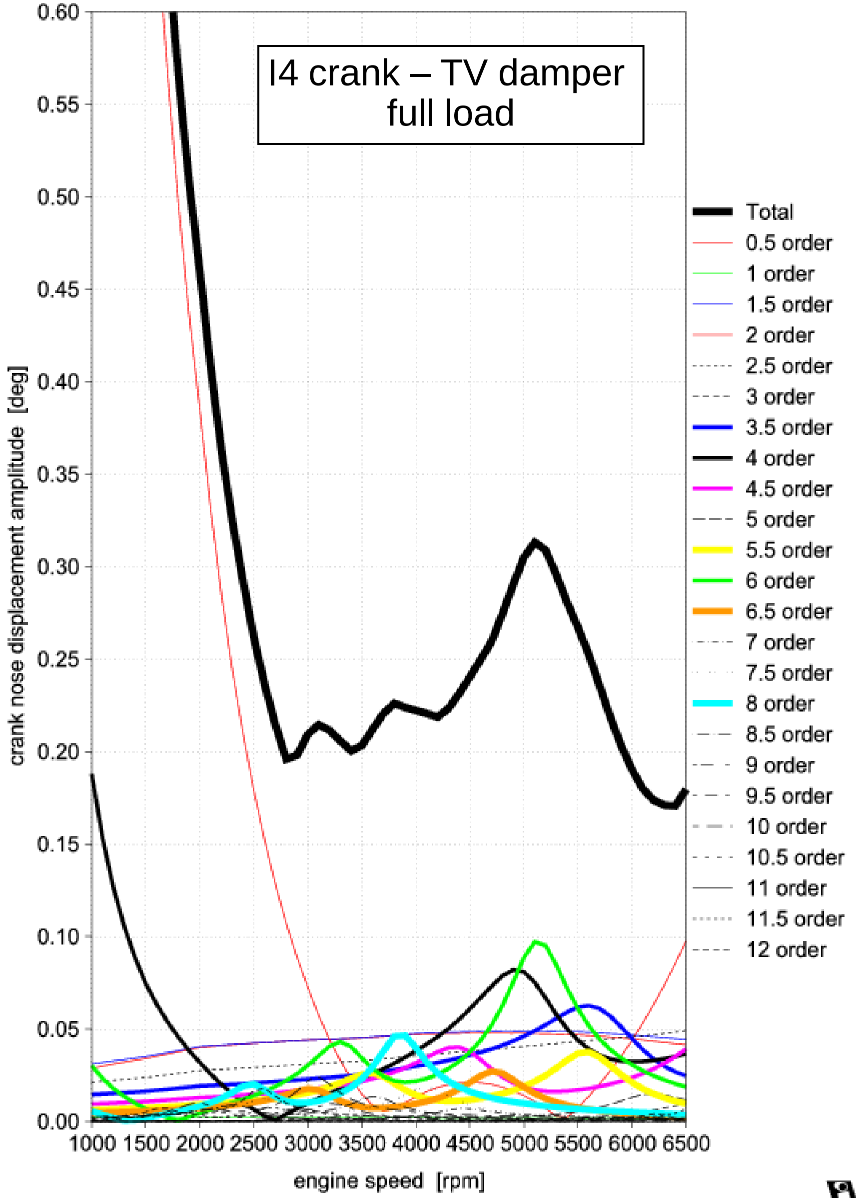




- **These graphs are especially useful because crank nose motion can be measured fairly easily**
  - Comparison between measured and calculated data in the past has led to knowledge of suitable damping values
  - Correlation of crank nose motion is a vital step in the development of a reliable model of 3D crankshaft dynamics

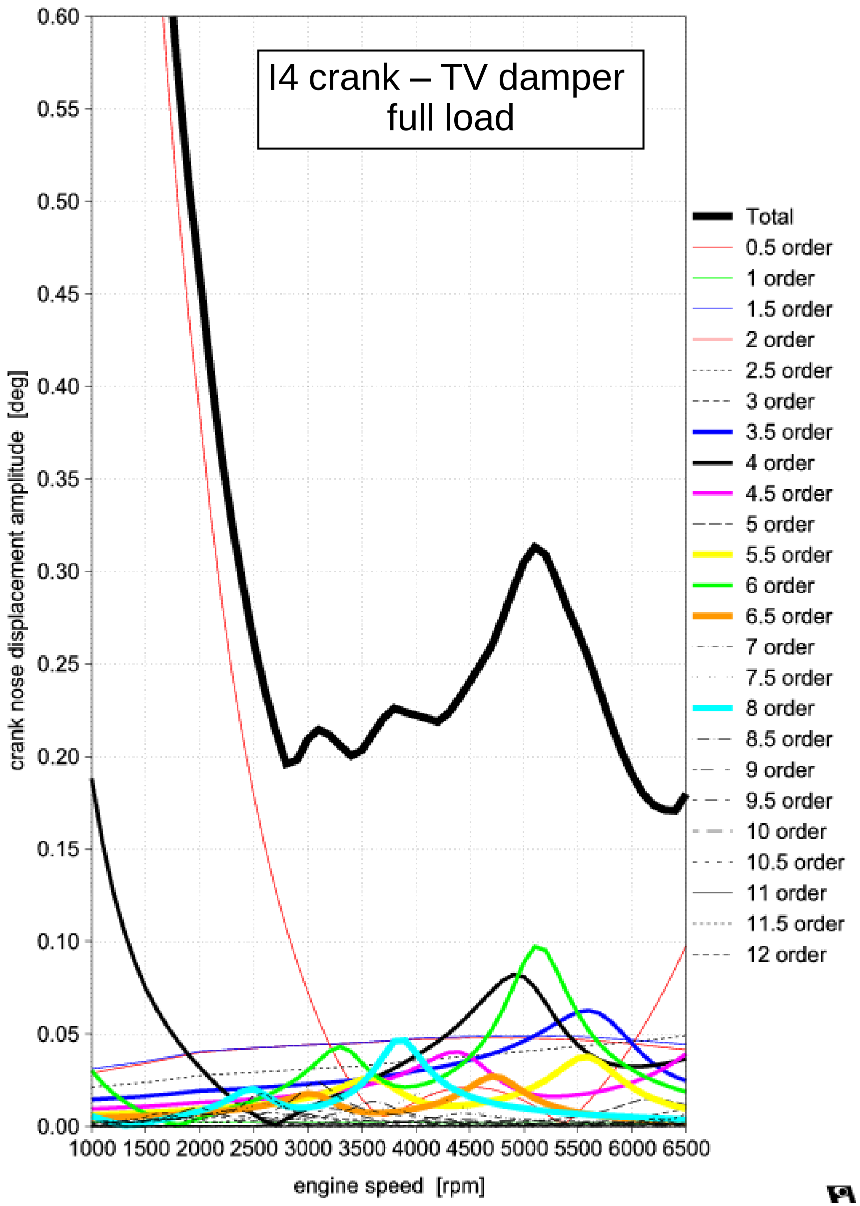
Crankshaft analysis (14) - Concept level TV analysis results

- **At low engine speed and high load the motion is dominated by the "rolling mode"**
  - The whole crankshaft speeds up and slows down as the different cylinders fire
  - For an I4 engine this is mainly 2nd order
  - For a V6 engine this is mainly 3rd order
  - This effect decays rapidly with increasing engine speed
  - the level of rolling mode vibration or cyclic speed variation is set by
    - the flywheel inertia
    - the peak cylinder pressure



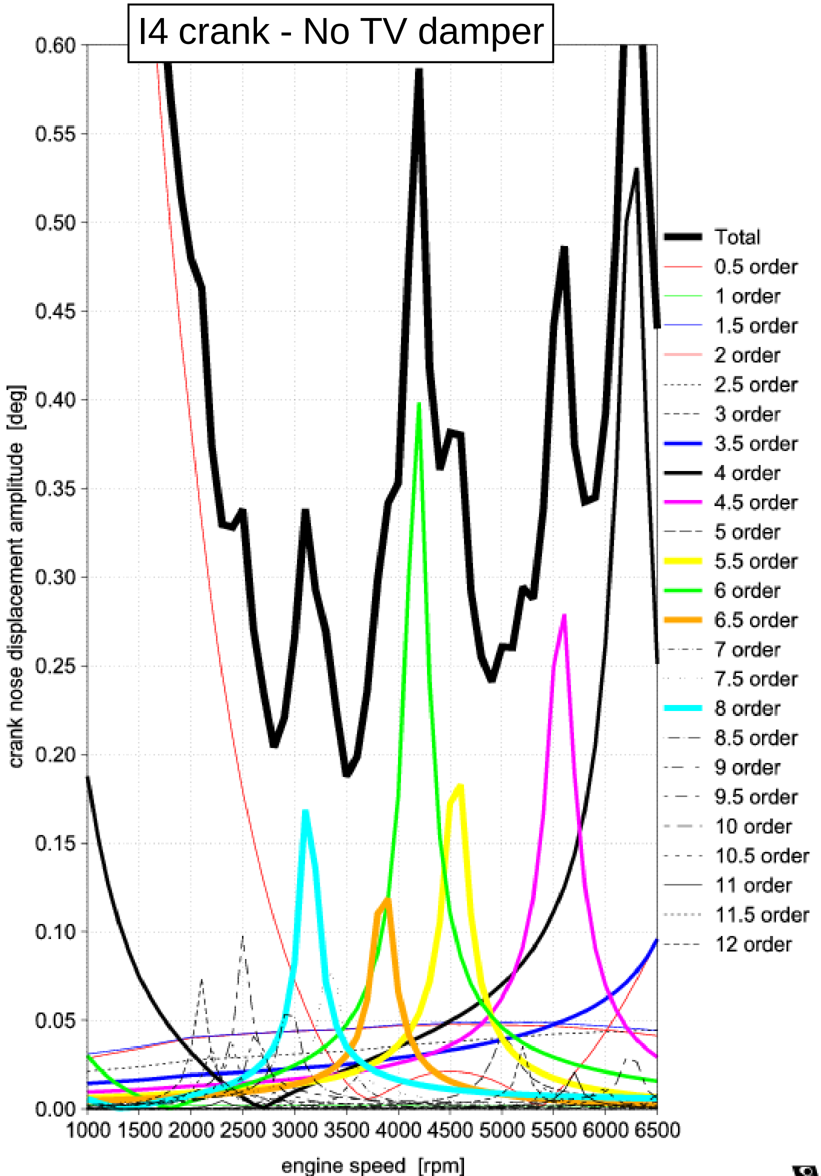
Crankshaft analysis (15) - Concept level TV analysis results

- **At higher speeds various resonant peaks can usually be seen on the plot of crank nose motion**
- **Ricardo have targets for particular orders of vibration for a refined passenger car engine with a rubber TV damper**
  - 4th order for I4 or 6th order for V6 <0.06 deg
  - Higher orders <0.05 deg
- **These resonant peaks occur as the torsional natural frequency of the cranktrain (crankshaft + damper + flywheel) is excited by various harmonics of the applied torques due to gas forces and inertial effects of the reciprocating pistons and rods**
  - The patterns in this data can be grasped more easily if we look at the response of a crankshaft with no TV damper (see next slide)



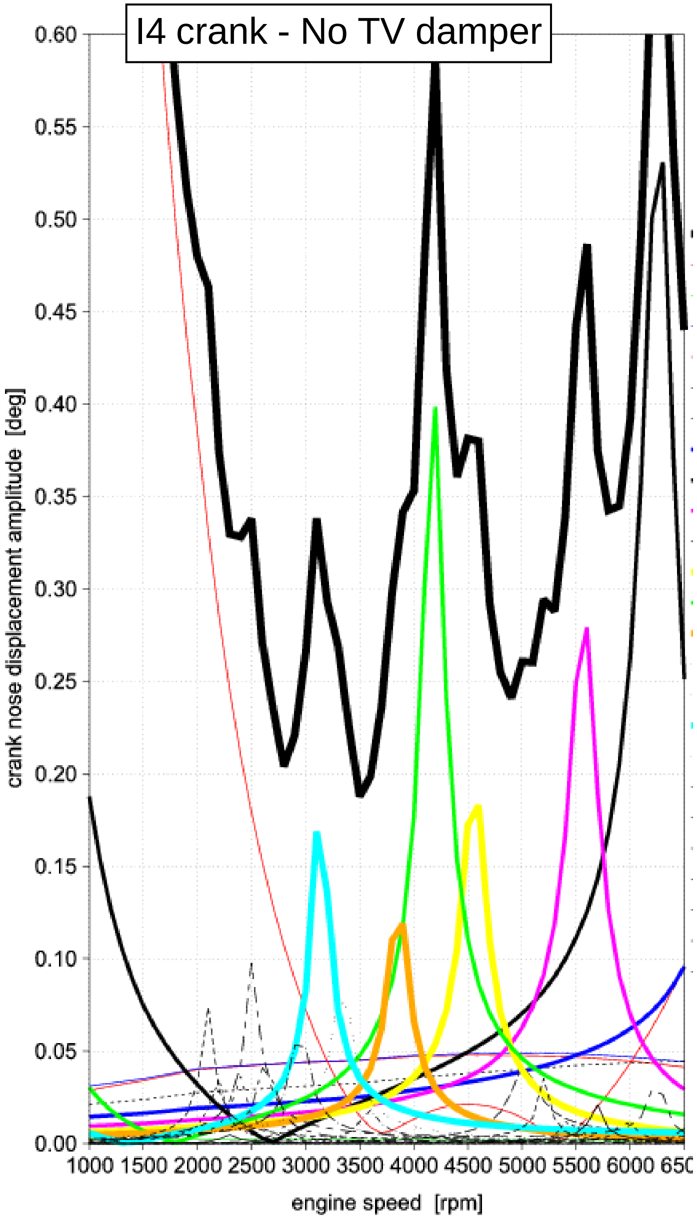
Crankshaft analysis (16) – Concept level TV analysis results

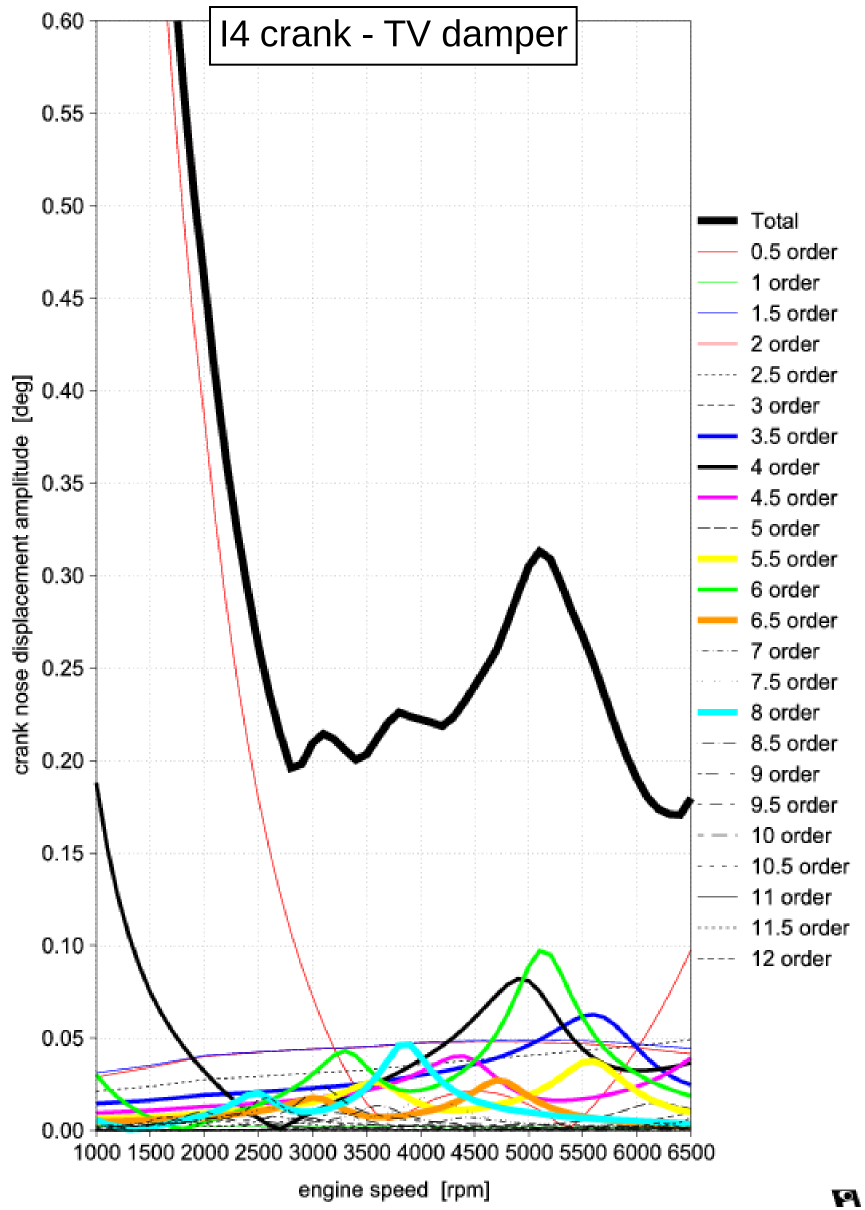
- **With no TV damper the resonant peaks are sharp, well-defined and numerous**
- **In the case shown the crank train natural frequency is ~410 Hz**
  - or 410 x 2π = 2576 rad/s
  - or 2576 x (2π/60) = 24600 rpm
- **At crankshaft speed of 24600/2 = 12300 rpm there would be an enormous resonance as the cranktrain would be excited by the engine firing frequency**
  - 2nd order for I4 engine
  - 4 firing events in 2 crank revolutions
  - 2 firing events per crank revolution
- **The higher harmonics of the gas and inertia torque applied at each crank pin excite the crank train natural frequency at speeds within the operating range of the engine**
  - 4th order resonance at 24600/4 = 6150 rpm
  - 6th order resonance at 24600/6 = 4100 rpm



Crankshaft analysis (17) – Effect of TV damper

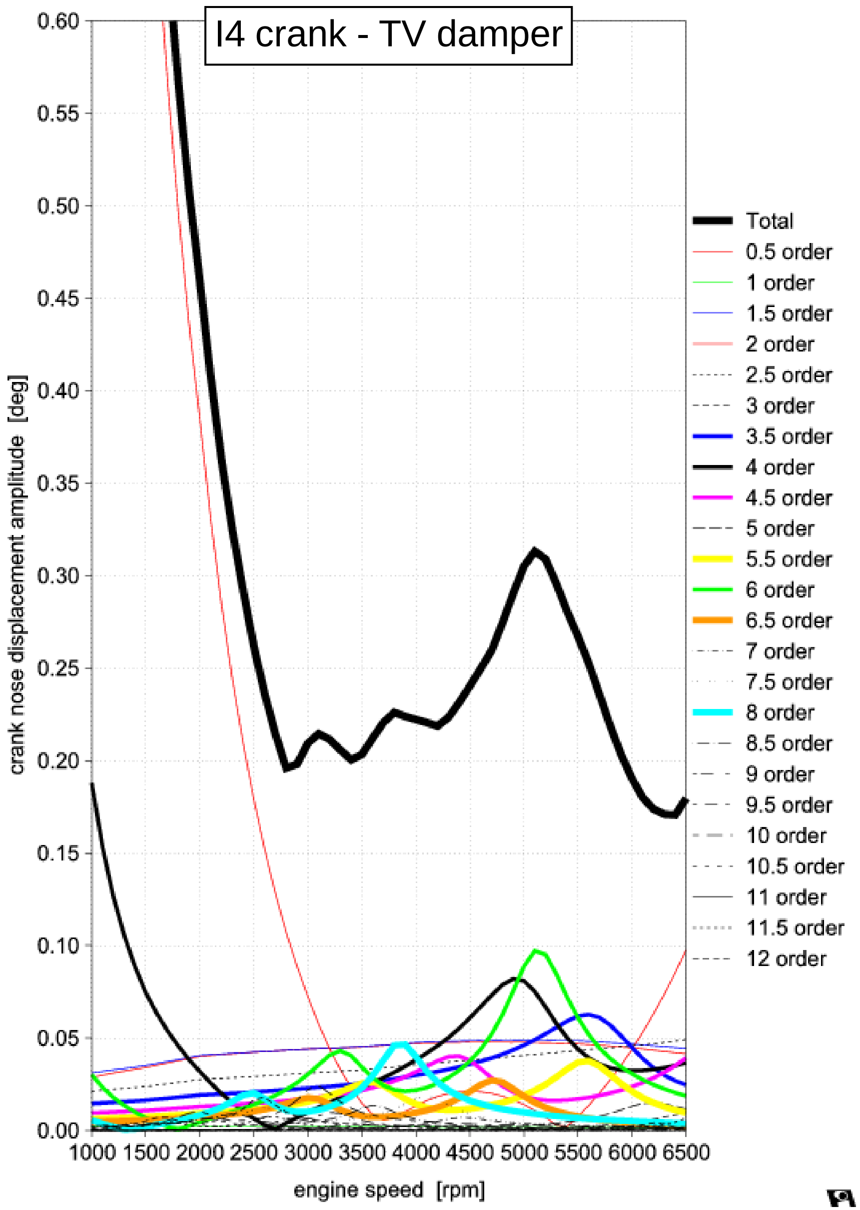
- **Crankshaft is excited by combination of gas forces and inertia forces**
- **Crank train natural frequency is given by**
- **TV damper acts to split each resonant peak into 2 smaller ones**





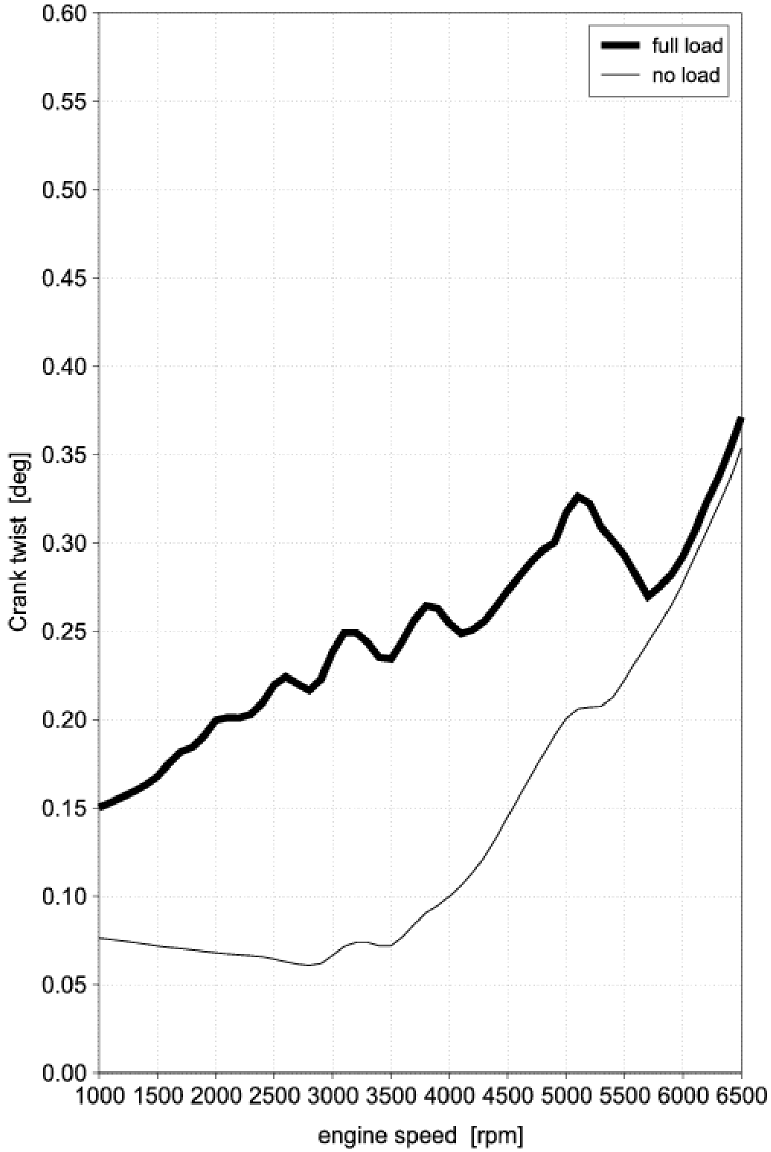
Crankshaft analysis (18) – Effect of TV damper

- **The inertia of the damper ring determines the amount by which the two peaks are separated**
- **The rubber stiffness sets the tuning ratio which determines the relative height of the two peaks**



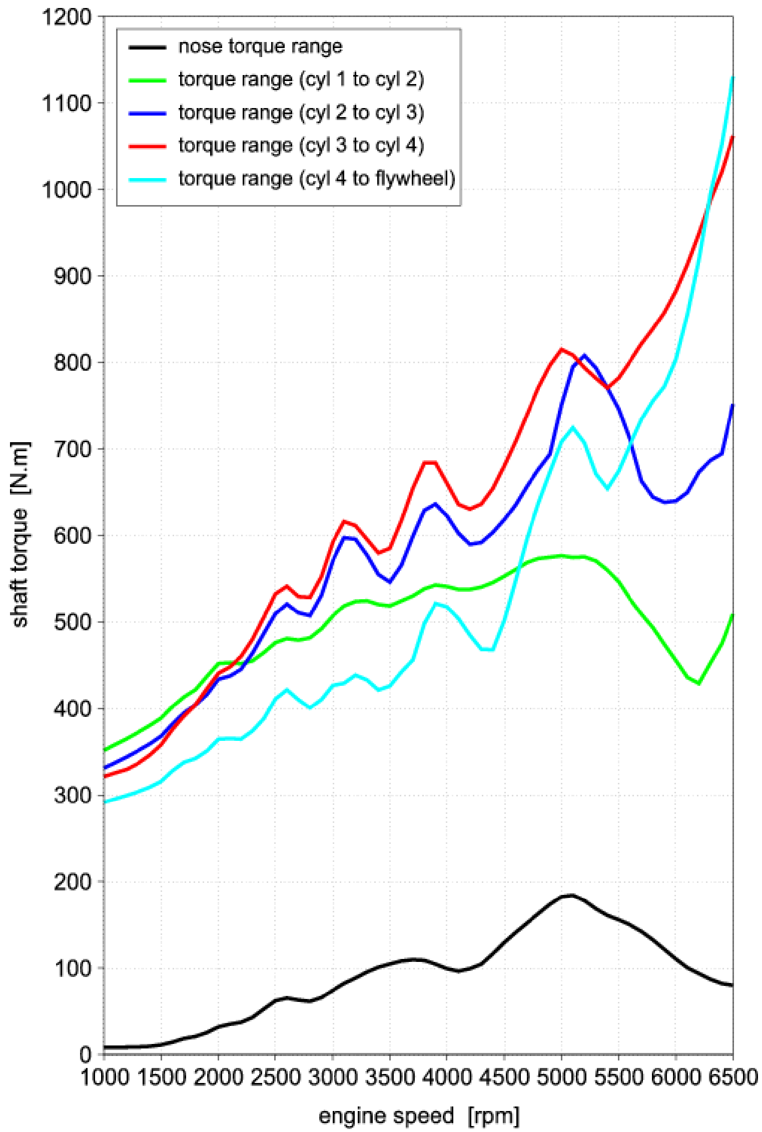
Crankshaft analysis (19) - Concept level TV analysis results

- **Ricardo normally present a graph of crank twist (1/2 peak to peak) against engine speed at full load**
- **This gives an indication of the variation of torsional stress in the crankshaft**



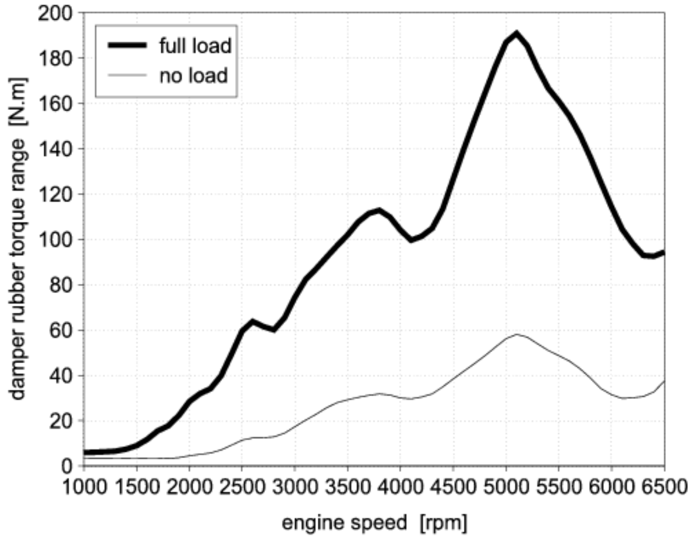
Crankshaft analysis (20) - Concept level TV analysis results

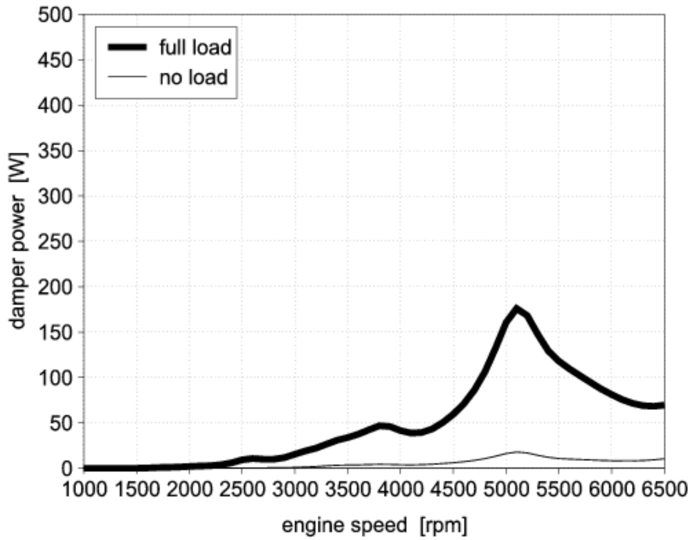
- **Ricardo normally present a graph of crank TV torque in each crankshaft element against engine speed at full load**
- **These data can be useful to show the worst case locations for TV torque**
- **For engines with a large, high inertia flywheel the TV torque is generally highest at locations close to the flywheel at high engine speed**
- **Ricardo use this data to feed into classical stress analysis**



Crankshaft analysis (21) - Concept level TV analysis results

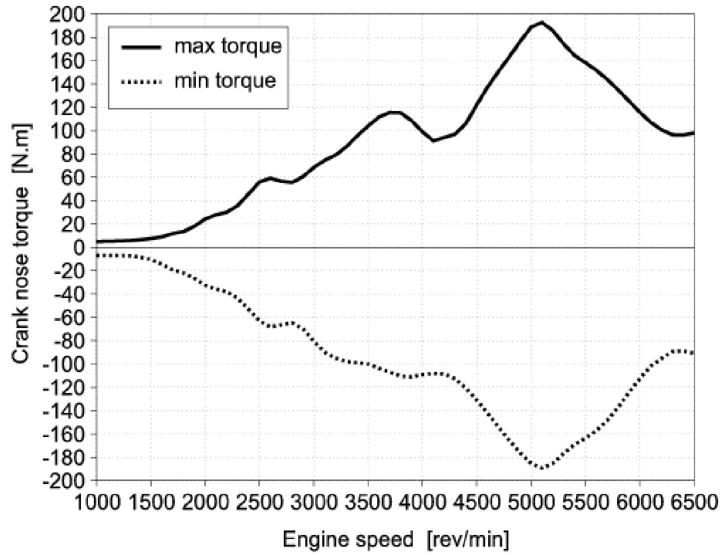
- **Ricardo also use VALDYN model to calculate**
  - torque in TV damper rubber element
  - power dissipated in TV damper rubber element
- **These data are further processed to calculate**
  - Shear stress in TV damper rubber
  - Heat generated per unit volume of TV damper rubber
- **These values are used in conjunction with well-developed limits to assess the durability of the TV damper**

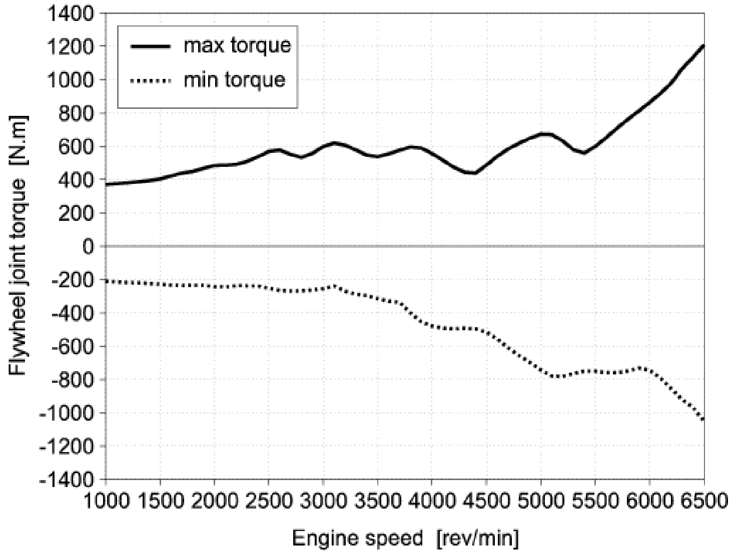




Crankshaft analysis (22) - Concept level TV analysis results

- **Ricardo also use VALDYN model to calculate**
  - TV torque at crank nose bolted joint
  - TV torque at flywheel bolted joint
- **These data are used during calculation of joint cover factor**





Crankshaft analysis (23) – Crank nose bolted joint cover factor

- **Ricardo use the procedure shown to calculate the cover factor at the crank nose joint**
- **Look at each joint if multiple joints are present**
- **Take TV torque from VALDYN (correlated model if possible)**
- **Use friction coefficient of 0.2**
  - 0.3 is possible using diamond washer and this can be a big advantage
- **Highly rated engines subject to lots of development often have cover factors between 1.3 and 1.8**
- **Lower values result in fretting or worse**

Effective outer radius of joint	$R_1 := \mathbf{18} \cdot mm$		from drg
Effective inner radius of joint	$r_1 := \mathbf{10} \cdot mm$		from drg
Coefficient of friction	$\mu := \mathbf{0.2}$		Typical value
Bolt tightening torque (min)	$T_{bolt} := \mathbf{400} \cdot N \cdot m$		From
Nominal bolt diameter	$D := \mathbf{16} \cdot mm$		from drg
Thread pitch	$p := \mathbf{1.5} \cdot mm$		From drg
Bolt tensile stress area	$csa := \frac{\pi}{4} \cdot (D - 0.938p)^2$	$csa = 167.255\,mm^2$	
Bolt stress under proofing load	$P := \mathbf{940} \cdot \frac{N}{mm^2}$		Grade 10.9 see BS EN20898-1
Peak applied torque at TV peak	$TTV := \mathbf{226} \cdot N \cdot m + \mathbf{19} \cdot N \cdot m$	$TTV = 245\,N \cdot m$	Maximum positive torque in nose from VALDYN at 7100 rpm + typical FEAD torque
Bolt load	$F := \frac{5 \cdot T_{bolt}}{D}$	$F = 1.25 \times 10^5\,N$	
Percentage of proof stress	$percent := \frac{F \cdot 100}{P \cdot csa}$	$percent = 79.507$	70-75 percent recommended
Contact stress under bolt head	$\sigma_1 := \frac{F}{\left[\frac{\pi}{4} \cdot \left[(\mathbf{24} \cdot mm)^2 - (\mathbf{17} \cdot mm)^2\right]\right]}$	$\sigma_1 = 554.547 \frac{N}{mm^2}$	Recommended maximum is 500 N/mm^2 for steel
Joint torque capacity (pulley to crank)	$T_{joint1} := \mu \cdot F \cdot \frac{2}{3} \cdot \left(\frac{R_1^3 - r_1^3}{R_1^2 - r_1^2}\right)$	$T_{joint1} = 359.524\,N \cdot m$	
Cover factor (pulley to crank)	$cfTV := \frac{T_{joint1}}{TTV}$	$cfTV = 1.467$	1.8 minimum 2.0 preferred

Crankshaft analysis (24) – Flywheel bolted joint cover factor

- **Ricardo use the procedure shown to calculate the cover factor at the flywheel joint**
- **Take TV torque from VALDYN (correlated model if possible)**
- **Use a friction coefficient of 0.2**
- **Ricardo recommend minimum cover factor of**
  - 1.8 at TV peak
  - 2 at max power
  - 1.7 at overspeed

Maximum Torque	$Tmax := \mathbf{1216} \cdot Nm$	5000 rpm brake torque 265 Nm + max TV torque 951 Nm = 1216 Nm
Flywheel face inner diameter	$d := \mathbf{47} \cdot mm$	
Flywheel face outer diameter	$D := \mathbf{84.4} \cdot mm$	
Coefficient of friction at face (Typically 0.2)	$\mu := \mathbf{0.2}$	
Number of bolts	$n := \mathbf{6}$	
Nominal bolt diameter	$Dbolt := \mathbf{10} \cdot mm$	
Bolt thread pitch	$p := \mathbf{1.0} \cdot mm$	
Tightening method (For reference)	*yield controlled*	
Bolt strength grade (For reference)	12.9	
Fraction of bolt stress (See note below)	$k := \mathbf{0.75}$	
Bolt stress (See note below)	$\sigma b := \mathbf{1100} \cdot MPa$	

Note

Snug torque + angle of turn	*Use 0.85 of the 0.2% proof sress*
Yield controlled tightening (SPS)	*Use 0.75 of the 0.2 % proof stress*
Torque wrench tighten	*Use 0.70 of the proof load stress*

Strength grade	*10.9*	*11.9*	*12.9*
Proof stress, Rp02(MPa)	*940*	*1020*	*1100*
Stress under proofing load (MPa)	*830*	*900*	*970*

Bolt Force

Tensile stress area of bolt $tsa := \left(\frac{\pi}{4}\right) \cdot (Dbolt - 0.9382p)^2$ $tsa = 64.494\,mm^2$

Pre load for one bolt $Fbolt := k \cdot \sigma b \cdot tsa$ $Fbolt = 53.207\,kN$

Pre load due to all the bolts $Fbolts := n \cdot Fbolt$ $Fbolts = 319.245\,kN$

Effective Diameter

Effective friction diameter $Deff := \frac{2}{3} \cdot \left(\frac{D^3 - d^3}{D^2 - d^2}\right)$ $Deff = 67.474\,mm$

Torque Capacity

Torque capacity $Tcap := \frac{Deff}{2} \cdot Fbolts \cdot \mu$ $Tcap = 2.154 \times 10^3\,Nm$

Nominal Cover Factor

Nominal cover factor $CF := \frac{Tcap}{Tmax}$ $CF = 1.771$

Crankshaft analysis (25) – Classical stress analysis assumptions

- **Ricardo use a classical stress analysis process during the concept design phase**
  - This is performed using ENGDYN
- **Stresses are calculated at critical locations using statically determinate loading**
  - This accounts for bending and torsion due to gas forces and inertia forces
  - The effect of torsional vibration is included but the effects of bending vibration and load sharing between main bearings are not included
- **Stresses and safety factors are calculated at the following locations**
  - Crank pin journal fillets based on crank web overlap area
  - Main journal fillets based on overlap area
  - Crank pin journal oil hole breakouts

Crankshaft analysis (26) – Classical stress analysis input data

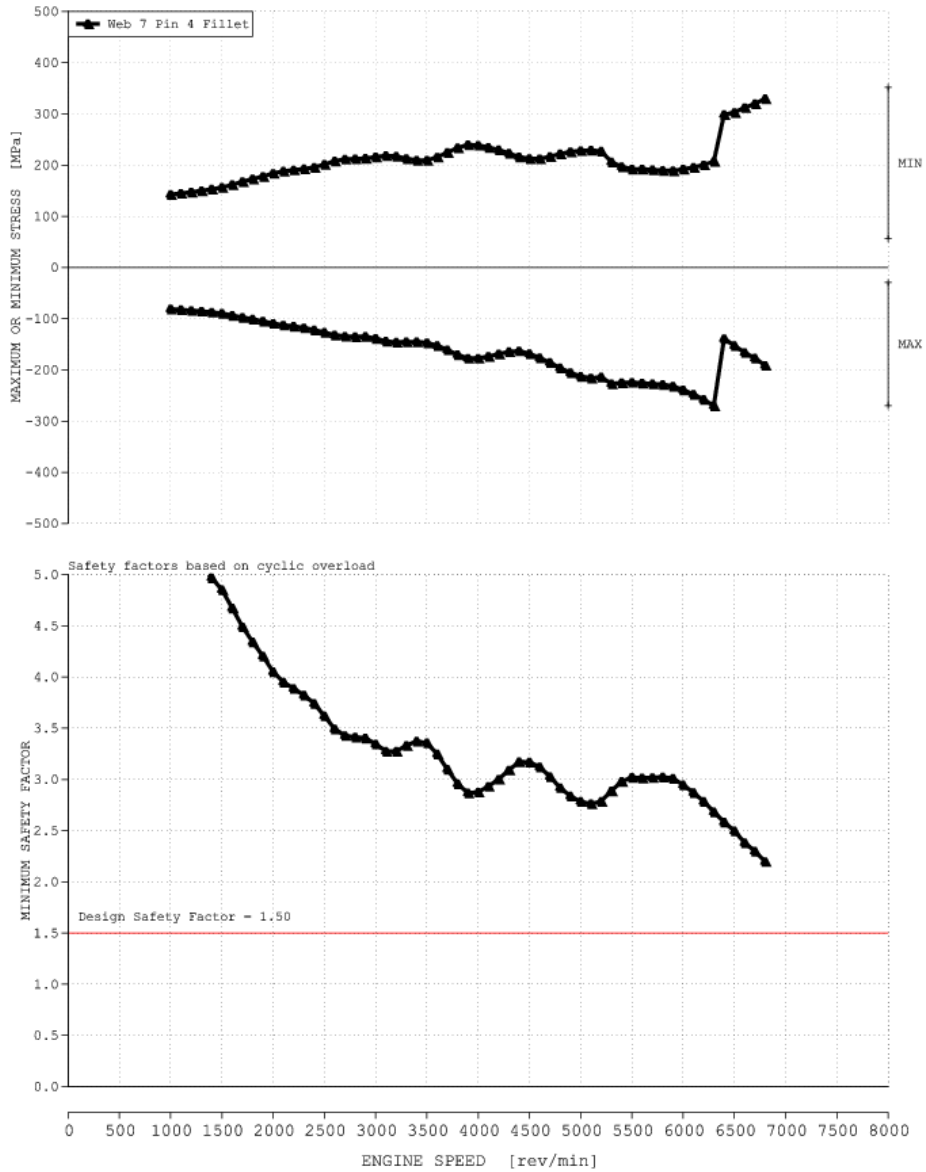
Parameter	Value	Source
Web reduced thickness (mm)	21.3	Full width 22.3 – land width
Web minimum width (mm)	71.4	From drawing
Web maximum width (mm)	80.0	From drawing
Web offset (mm)	0	from crank drg
Crank pin fillet radius (mm)	1.6	From crank drg
Crank pin fillet undercut (mm)	0.52	From crank drg
Crank pin fillet web undercut (mm)	0	From crank drg
Main journal fillet radius (mm)	1.6	From crank drg
Main journal fillet undercut (mm)	0.5	From crank drg
Main journal web undercut (mm)	0	From crank drg
Pin journal oil hole diameter (mm)	5.4	From crank drg
Pin journal oil hole height (mm)	16.9706	From crank drg
Pin journal oil hole X (mm)	48	From incoil.mcd
Pin journal oil hole Y (mm)	73.68	From incoil.mcd
Pin journal oil hole Z (mm)	30.021	From incoil.mcd
Main journal oil hole diameter (mm)	5.4	From crank drg
Main journal oil hole height (mm)	9.406	From crank drg
Pin journal land diameter (mm)	60.0	From crank drg
Pin journal land thickness (mm)	0.5	From crank drg
Main journal land diameter (mm)	70.0	From crank drg
Main journal land thickness (mm)	0.5	From crank drg
Base material UTS (N/mm^2)	750	C38+N2BY steel
UTS in fillets (N/mm^2)	920	Ricardo experience
Base tensile yield strength (N/mm^2)	450	C38+N2BY steel
Base compressive yield strength (N/mm^2)	450	C38+N2BY steel
Base infinite life fatigue strength (N/mm^2)	375	0.5 x base UTS
Base hydrostatic fatigue strength (N/mm^2)	543.75	2.5 x 217.5
Base torsional fatigue strength (N/mm^2)	217.5	0.58 x 375
Fillet UTS (N/mm^2)	920	Ricardo experience
Fillet tensile and compressive yield (N/mm^2)	750	Ricardo experience
Pin fillet fatigue strength (N/mm^2)	619	375 x 0.868 x 1.9 (rolled)
Pin oil hole fatigue strength (N/mm^2)	326	375 x 0.868 x 1.0 (induction hardened)
Main fillet fatigue strength (N/mm^2)	611	375 x 0.858 x 1.9 (rolled)
Main oil hole fatigue strength (N/mm^2)	326	375 x 0.858 x 1.0 (induction hardened)
Torques	valdyn.sdf	VALDYN files
Mean torque	User	Brake torque from WAVE

- **Tables show additional input data required to perform classical stress analysis**
  - Geometric data for webs, fillets and oil holes
  - Materials strength data
  - Brake torque data

Speed (rpm)	Brake torque (Nm)
1000	175
1500	200
2000	245
2500	255
3000	265
3500	265
4000	265
4500	265
5000	265
5500	255
6000	220
6500	190

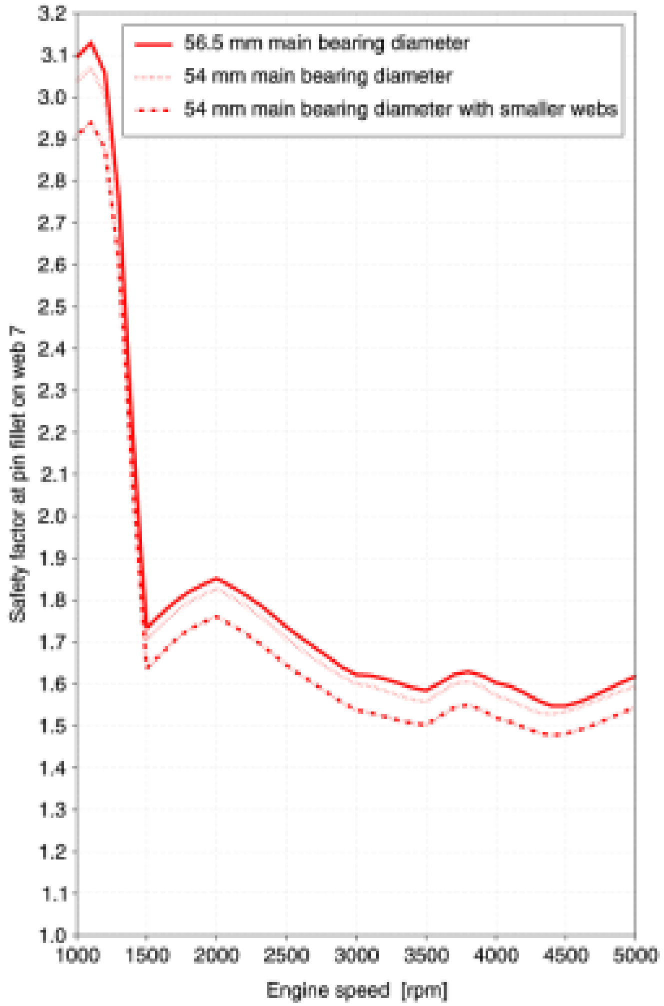
Crankshaft analysis (27) – Classical stress analysis results

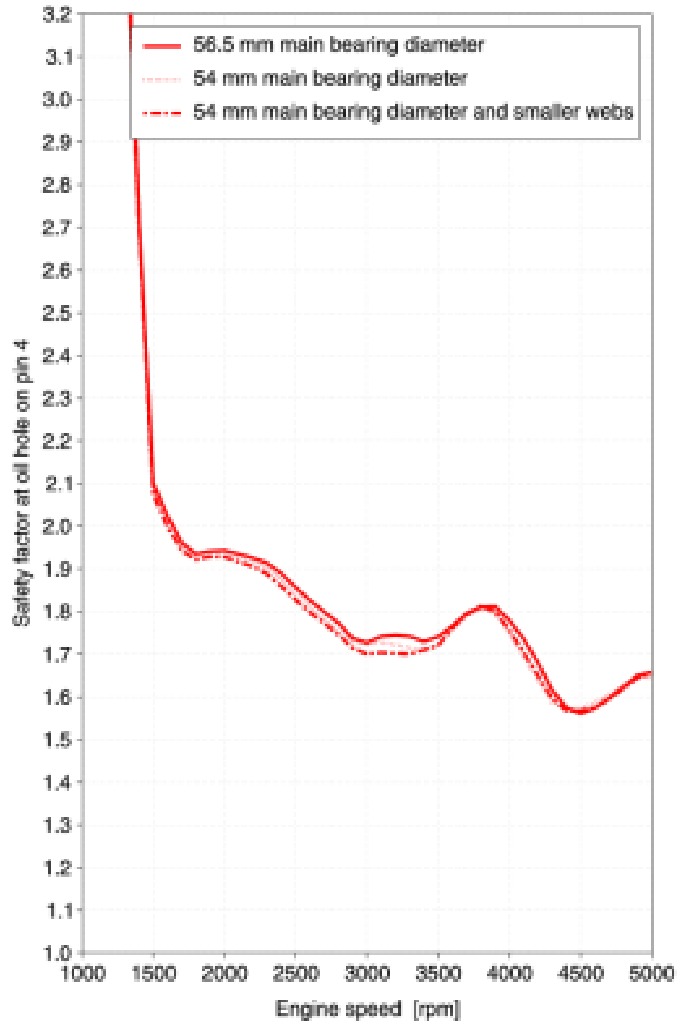
- **Graph shows typical output from ENGDYN**
  - Maximum and minimum stress plotted against speed
  - Goodman fatigue safety factor plotted against speed
- **Ricardo design to achieve minimum safety factor of 1.5 at all load/speed conditions**
- **Although stress at fillets is affected by bending stress and torsional stress**
  - Effect of bending usually dominates at fillets
  - Effect of torsion is more important at oil hole breakouts



Crankshaft analysis (28) – Classical stress analysis results

- **The classical stress analysis method is quick to set up and run and so is very useful for parametric studies to look at the effect of**
  - alternative materials
  - alternative fatigue strength improvement techniques
  - geometric changes
  - increased cylinder pressure
  - etc





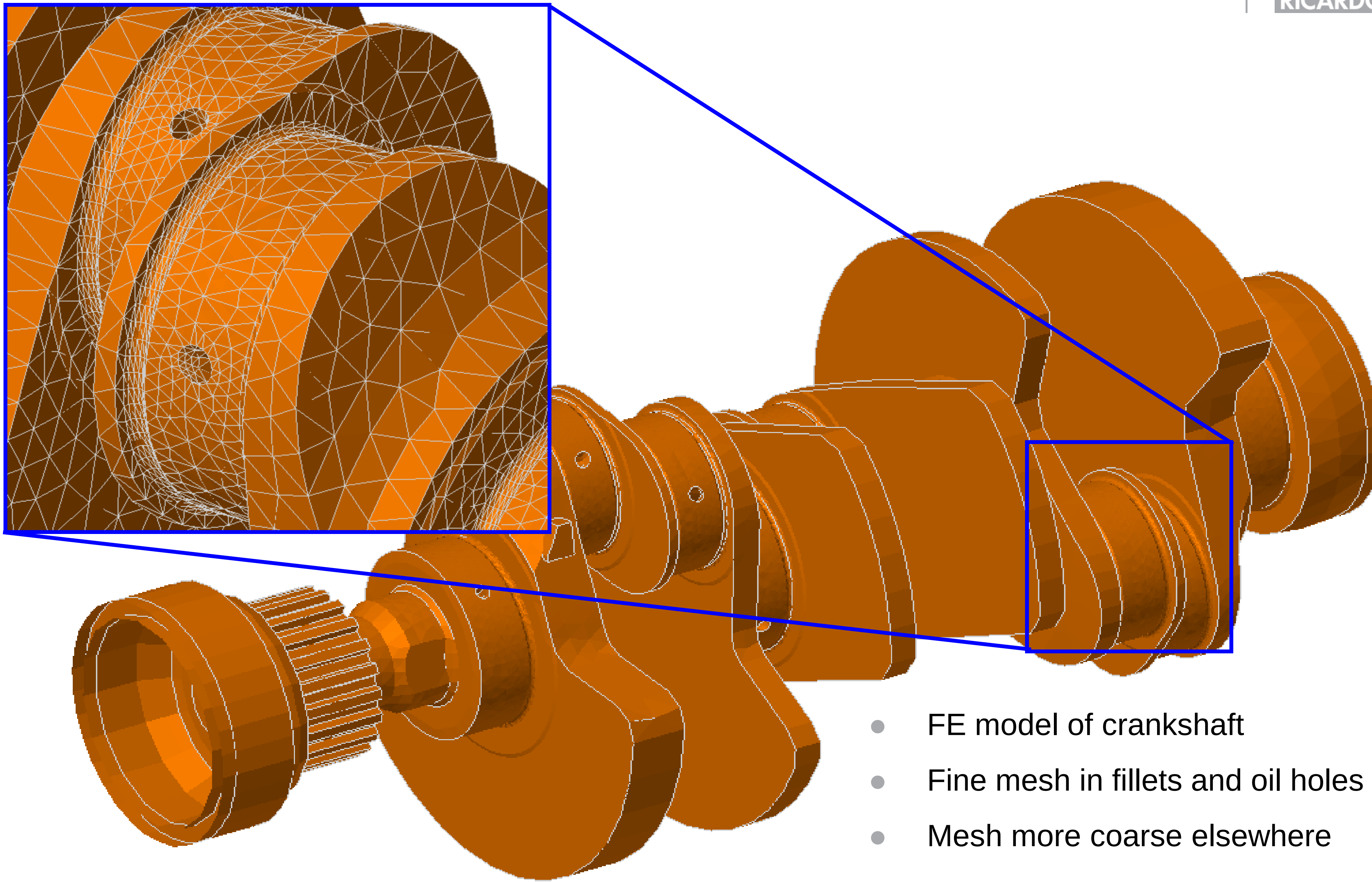
Crankshaft analysis (29) – Classical stress analysis of crank nose

- **It is important not to forget to calculate stresses in the crank nose**
  - This is particularly important as OEMs add belt driven starter alternators to existing engines
- **Ricardo have developed a classical stress analysis method considering**
  - Torsional vibration torque
  - FEAD drive torque and timing drive torque
  - Bending loads due to FEAD belts and timing drive
  - Axial loads due to crank nose bolt
- **This calculates Goodman fatigue safety factor at fillets and can be used to assess the effects of different dimensions, materials and fatigue lift improvement treatments**

Crankshaft analysis (30) – FE stress analysis

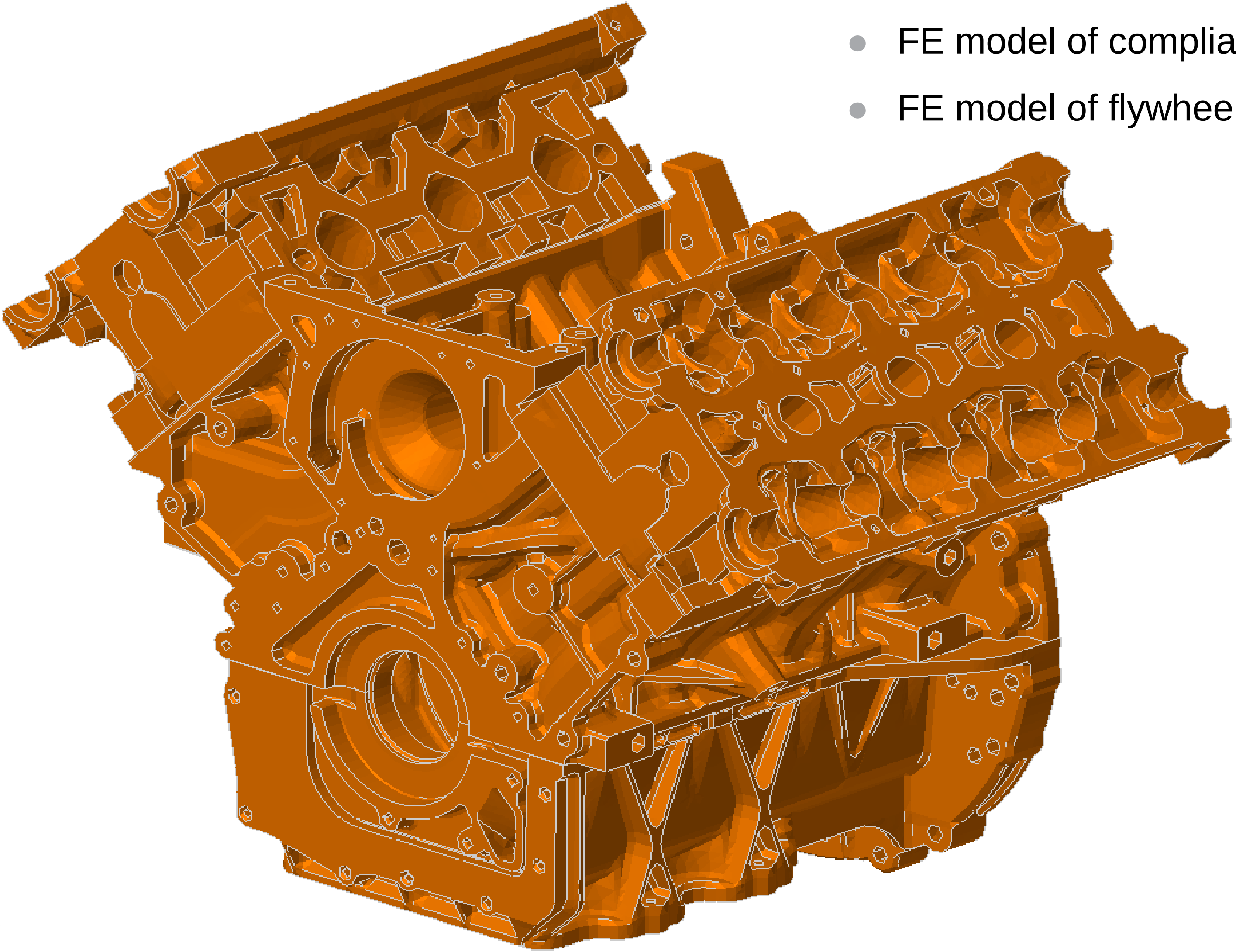
- **Ricardo usually perform FE stress analysis after a concept crankshaft design has been defined**
- **Ricardo use ENGDYN to apply loads and post-process results**
- **Ricardo prefer to perform fully dynamic analysis and this requires FE models of crankshaft and cylinder block**

Crankshaft analysis (31) – Crank models



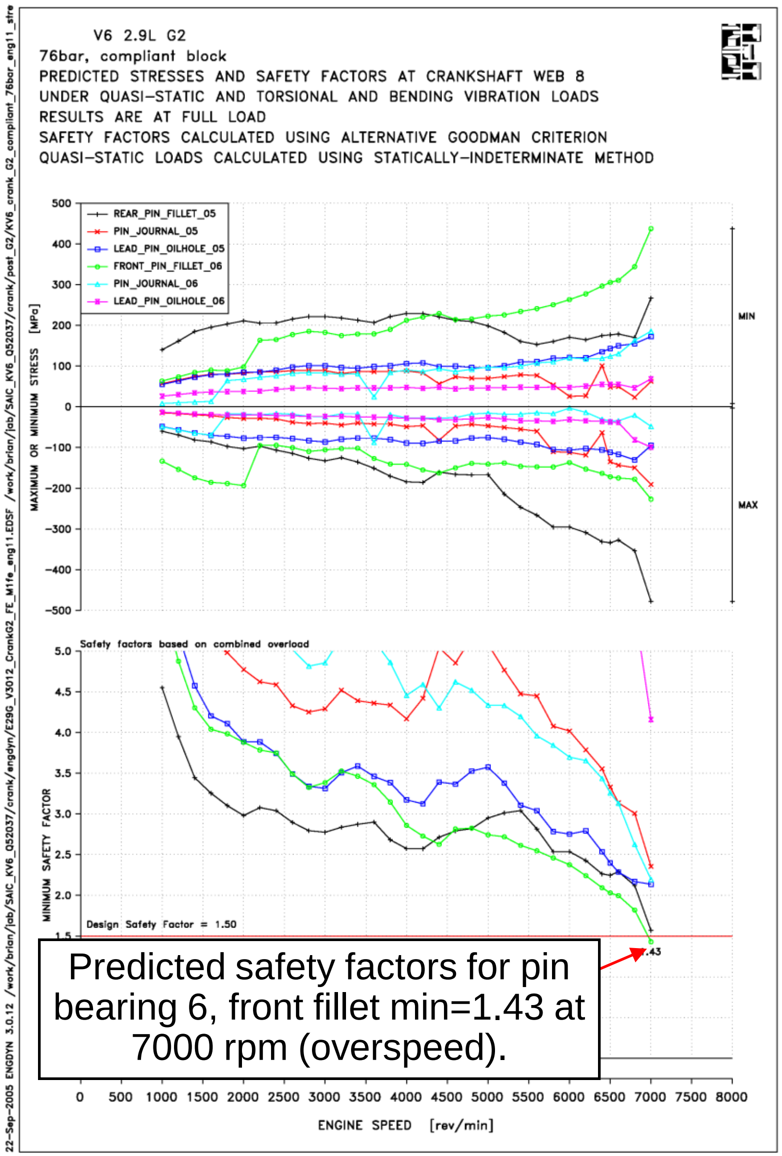
- FE model of crankshaft
- Fine mesh in fillets and oil holes
- Mesh more coarse elsewhere

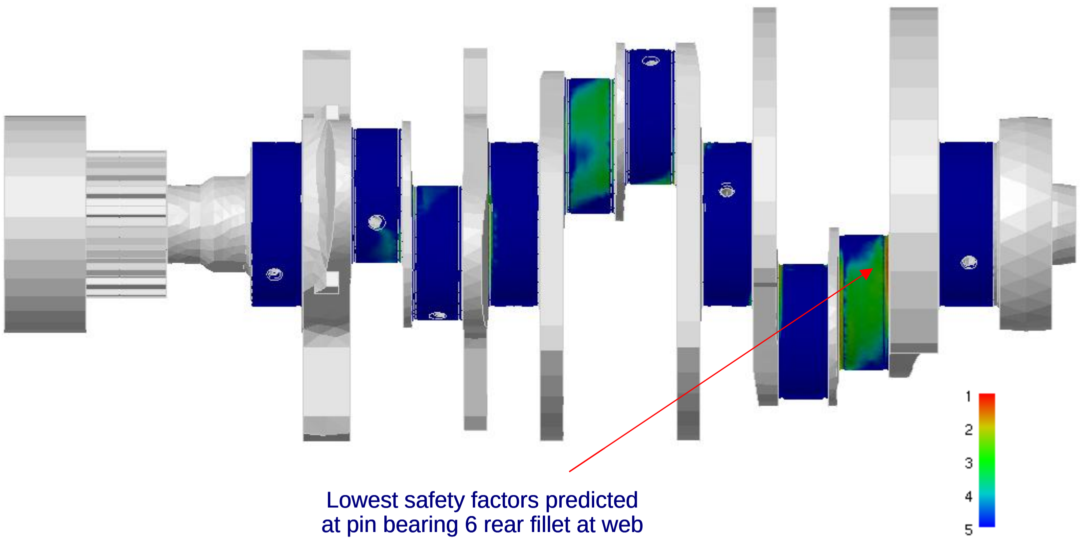
Crankshaft analysis (32) - Cylinder Block Model



- FE model of compliant block assembly
- FE model of flywheel also required

Crankshaft analysis (33) – FE stress analysis results

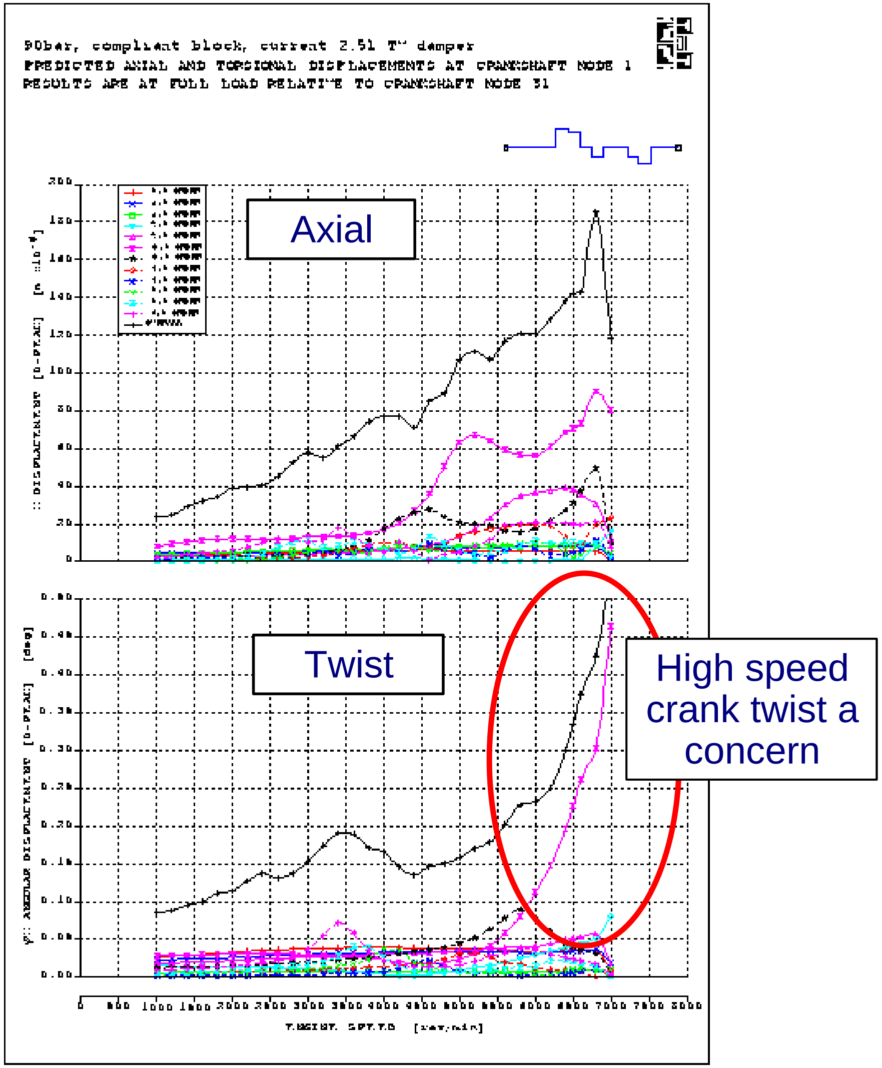


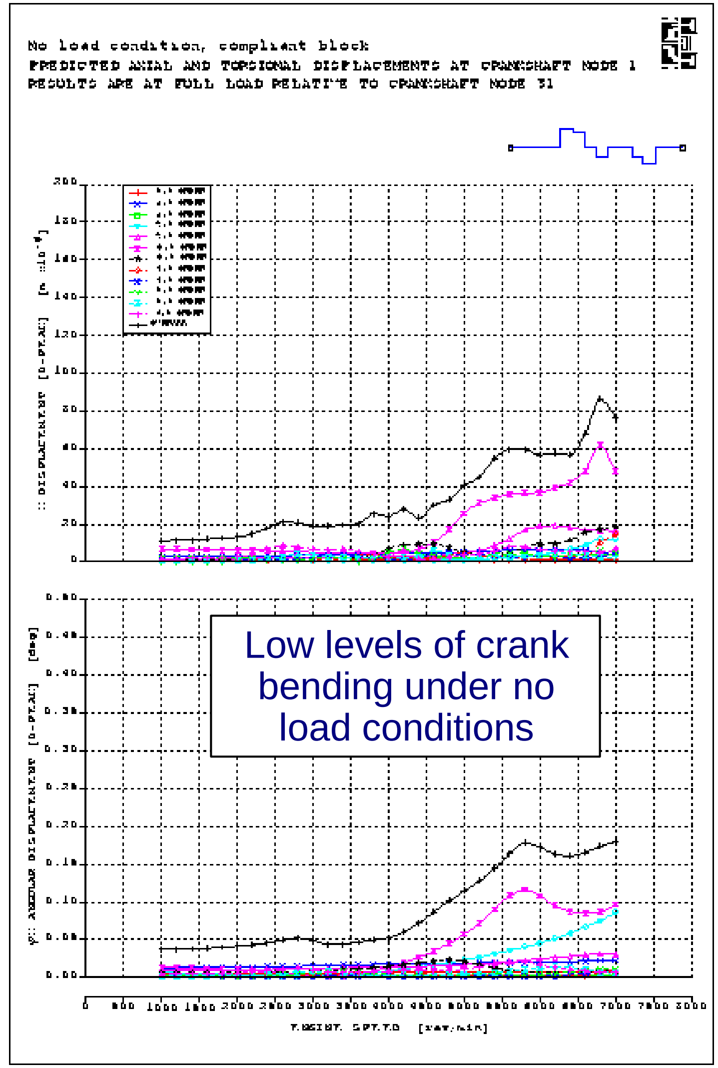


Lowest safety factors predicted at pin bearing 6 rear fillet at web 9. Min safety factor 1.41

Crankshaft analysis (34) – FE stress analysis results

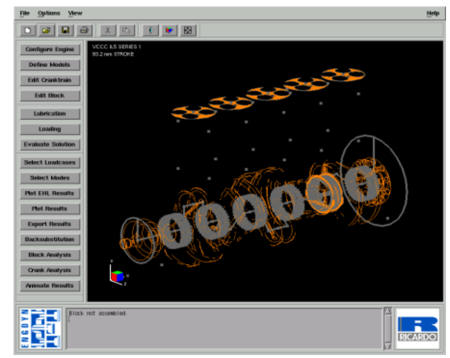
- **ENGDYN with FE models also used to investigate 3D dynamics of cranktrain**

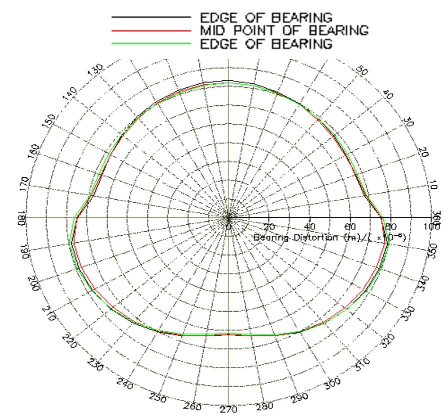




Crankshaft analysis (41) – ENGDYN bearing friction

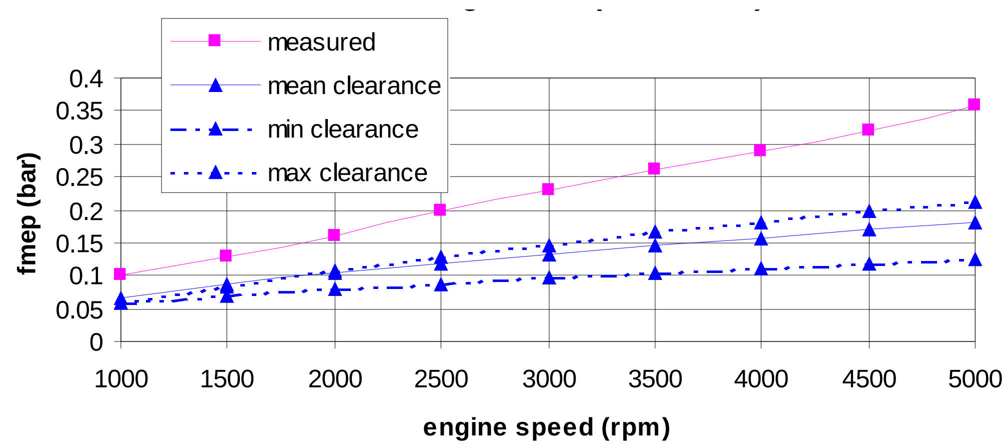
- **Journal bearings in engines normally operate in the hydrodynamic lubrication regime and so losses are dominated by oil shearing effects**
- **Surface contact (and so boundary or mixed lubrication) is possible**
  - during start-up period when oil may have drained away
  - possibly during running-in period
  - if bearing is overloaded
- **Prediction of friction losses in engine bearings is reasonably mature**
  - Ricardo use ENGDYN with mobility method to calculate power loss at main and big end bearings as part of routine design work
- **But advanced finite volume hydrodynamic and elastohydrodynamic analysis methods are now available**
  - Are they useful for friction prediction ?

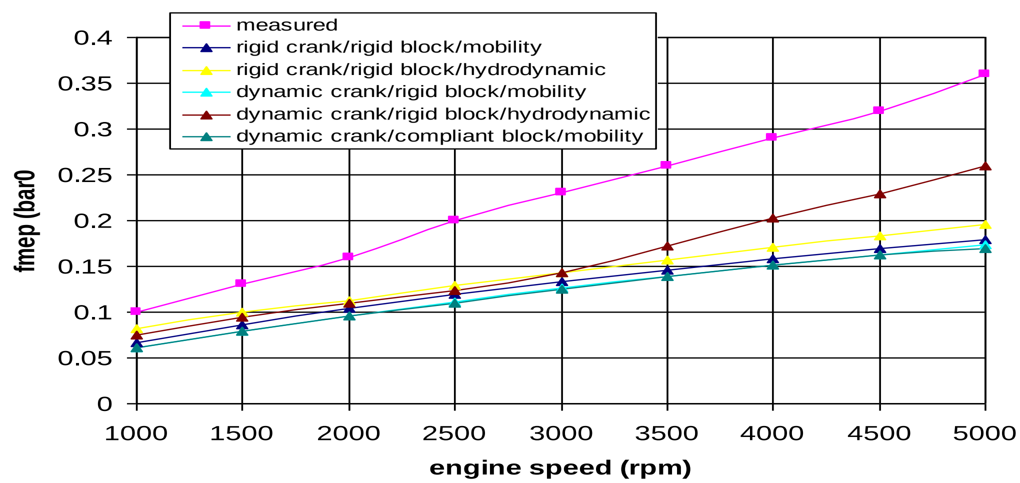




Crankshaft analysis (42) – ENGDYN bearing friction

- **On many occasions Ricardo have compared results of main bearing analysis with test data from the final stage of a motored teardown test in which the crankshaft is motored with no pistons/rods/drives**
- **The following pattern often emerges**
  – Losses due to oil shear do not account for the total measured
  – The gap between predicted and measured data is significant at low speed and grows larger with increasing speed
  – Variations in bearing clearance cannot account for the difference
  – Use of more advanced analysis methods cannot fully account for the difference

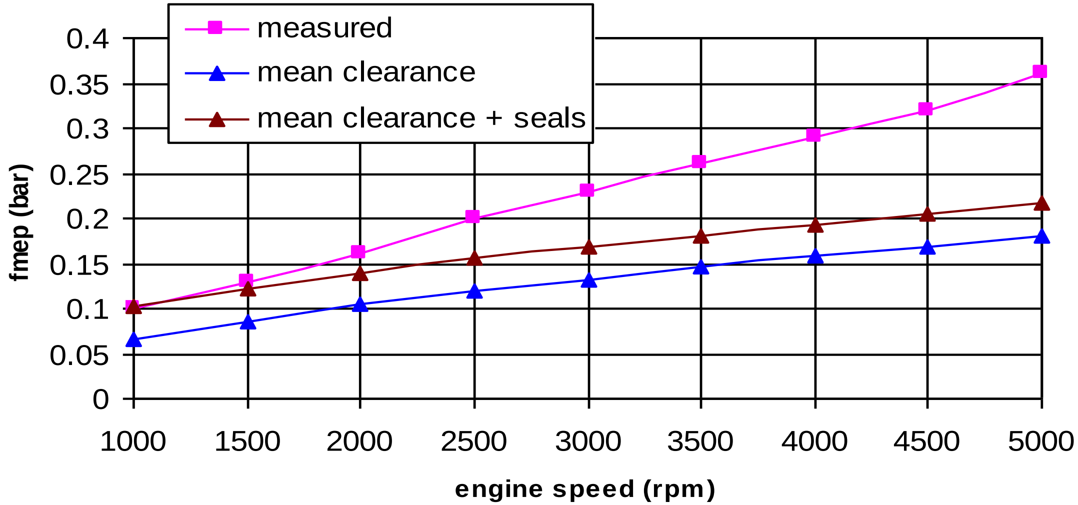


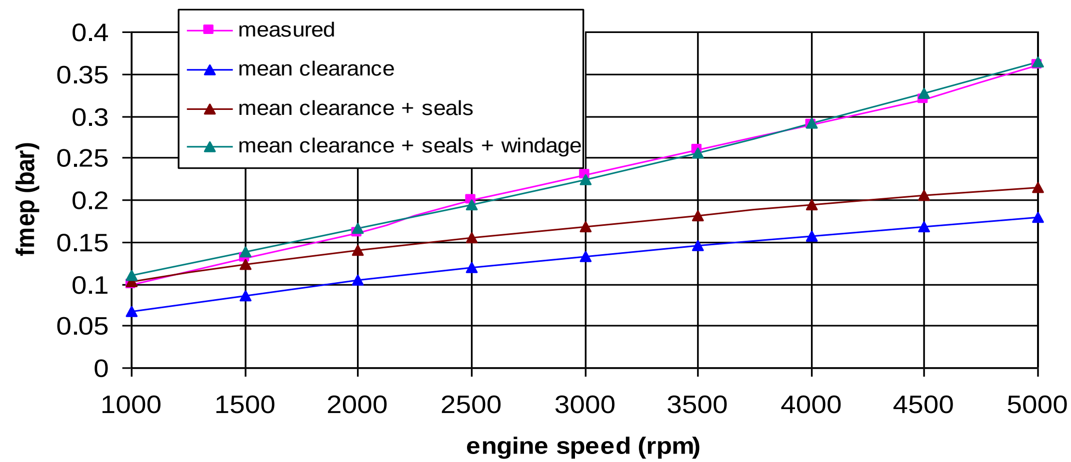


Crankshaft analysis (43) – ENGDYN bearing friction

RICARDO

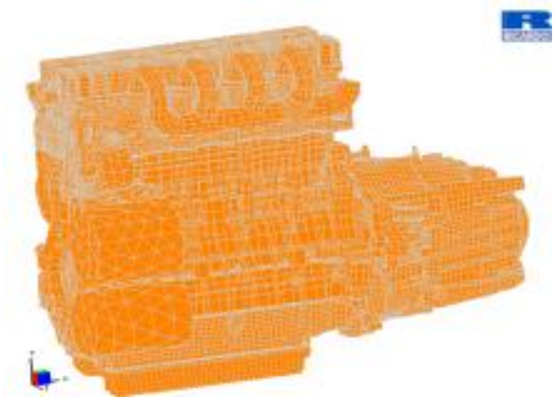
- **Equations for power loss at front and rear oil seals were obtained from seal manufacturers and incorporated with results using simple ENGDYN model**
  - This gives match to measured data at low engine speed
- **Text book equations were used to estimate windage in the crank case using dimensions of webs and counterweights**
  - This analysis indicated that windage could account for remaining loss if crank case gas was assumed to contain 2-3% oil
- **This approach has been validated on several engines**



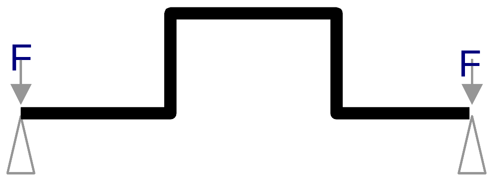


Crankshaft analysis (44) – ENGDYN bearing friction

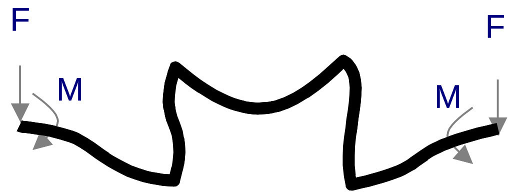
- **Ricardo performed a study to investigate the influence of bearing analysis method on predictions of main bearing performance on an in-line 5 cylinder diesel engine**
  - published at IMechE Tribology Conference 2006
  - summarised in this presentation



- **Statically determinate solution method**
  - rigid crank train element
  - lumped crankshaft masses



- **The dynamic solution method**
  - mass and stiffness matrices define crankshaft
  - matrices reduced within ENGDYN and include crankshaft gyroscopics



Crankshaft analysis (45) – Wear - assuming circular bearing shape

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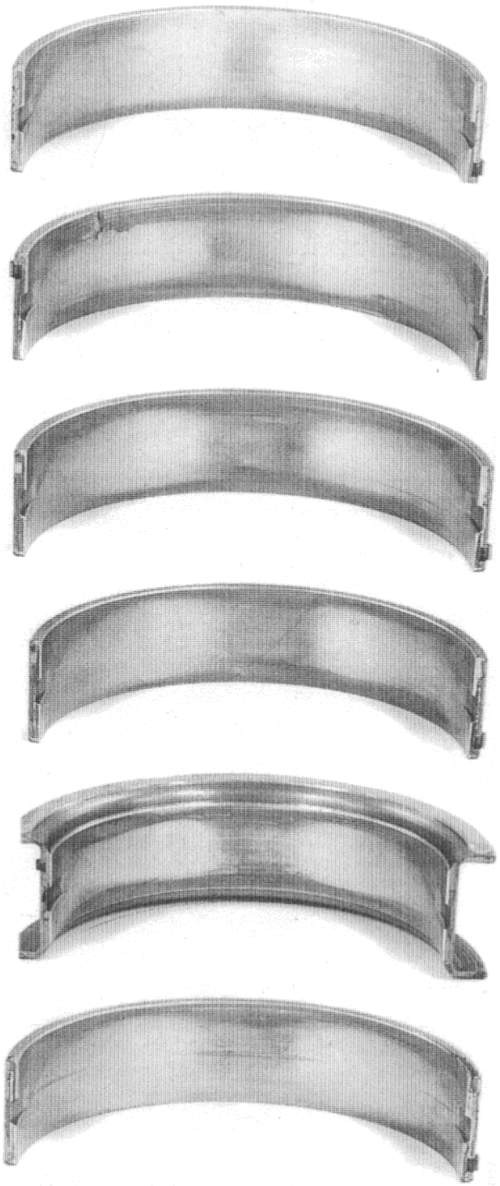

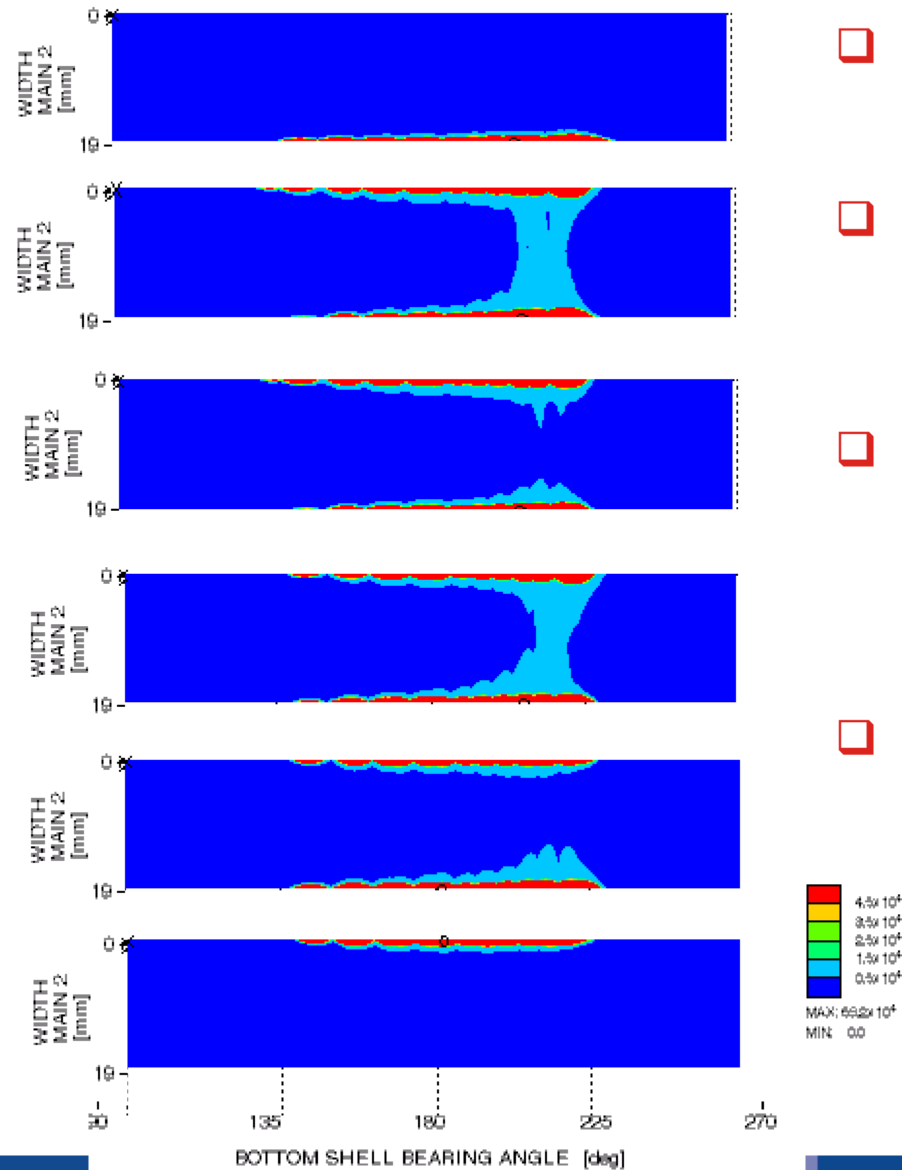

- **Photograph shows worn bearings**
  - durability engine
- **Plots show bearing wear load in W/m²**
  - Single steady state solution
- **EHD analysis predicts high wear loads at bearing edges**
  - Consistent with test observations
- **Regions of wear across the bearing width also predicted**
  - Does not match well with test observations

Crankshaft analysis (46) – wear – assuming distorted shape

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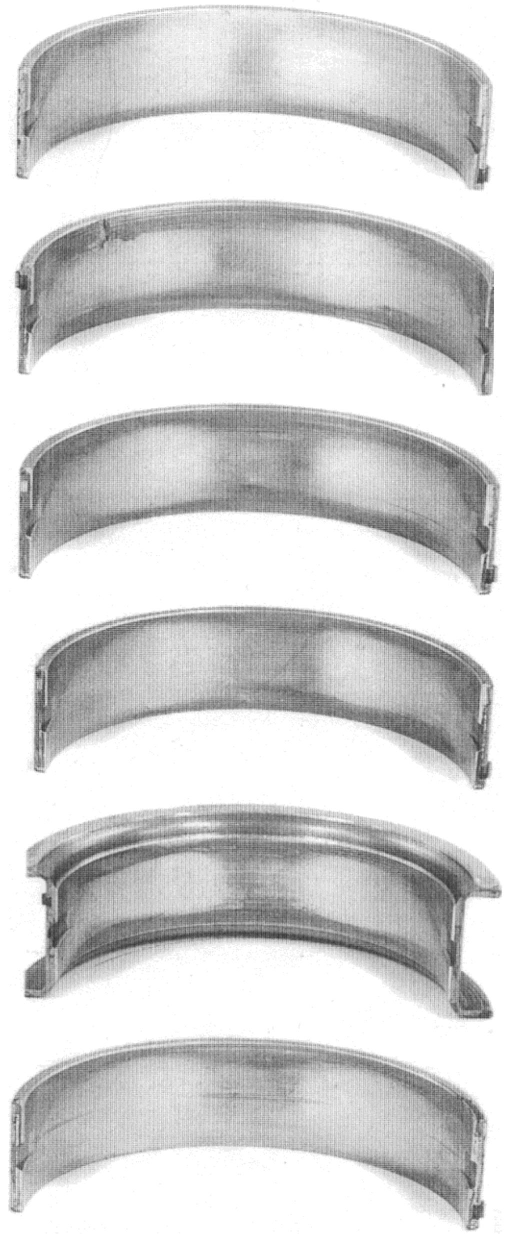
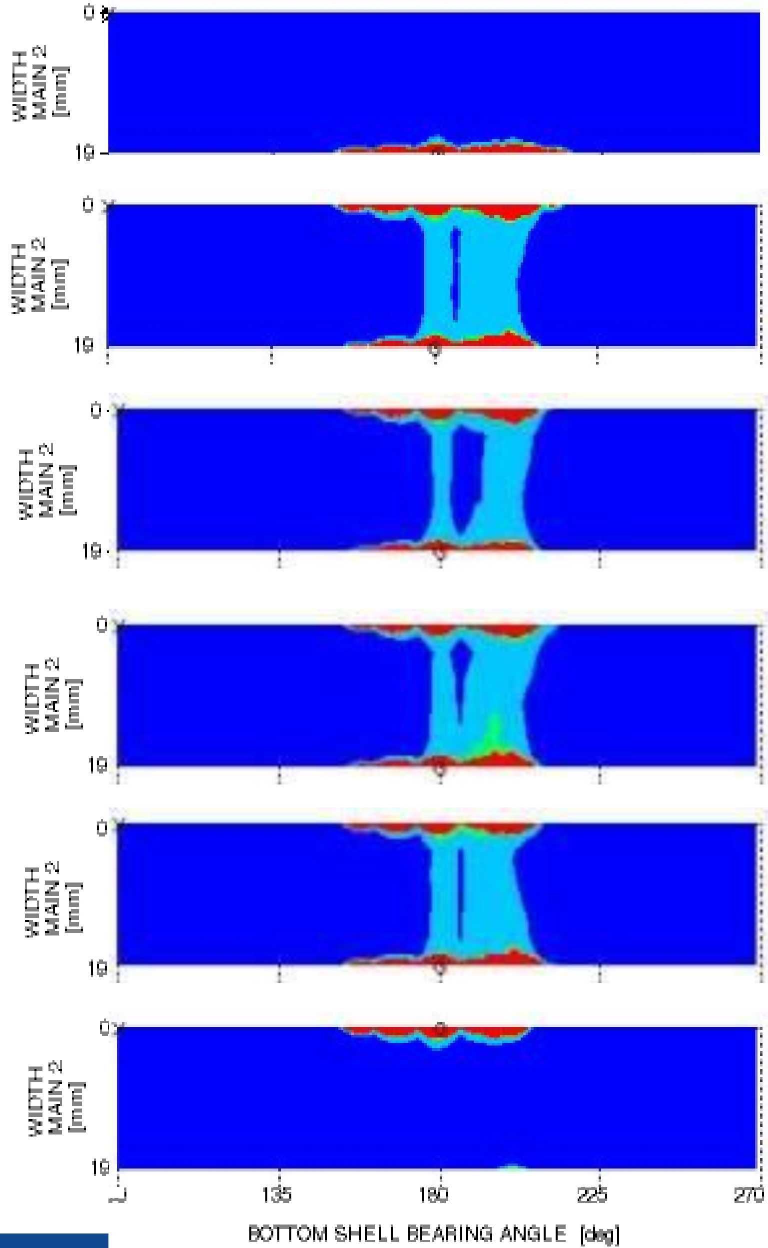
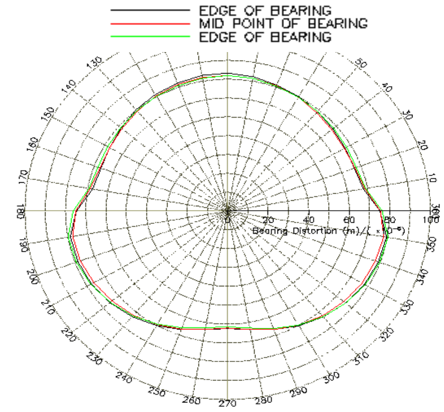
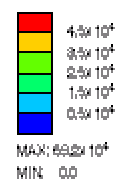




- EHD analysis repeated with thermal and assembly distortions

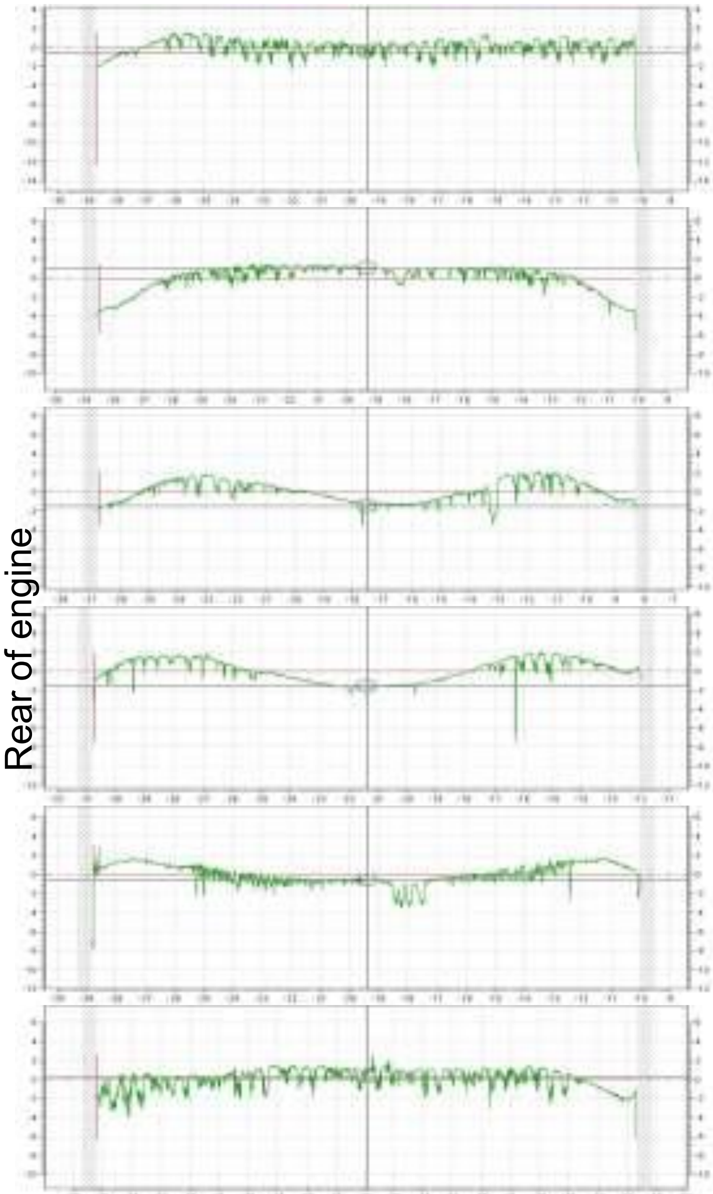


- Position of predicted central bearing wear loading now matches the observed wear patterns

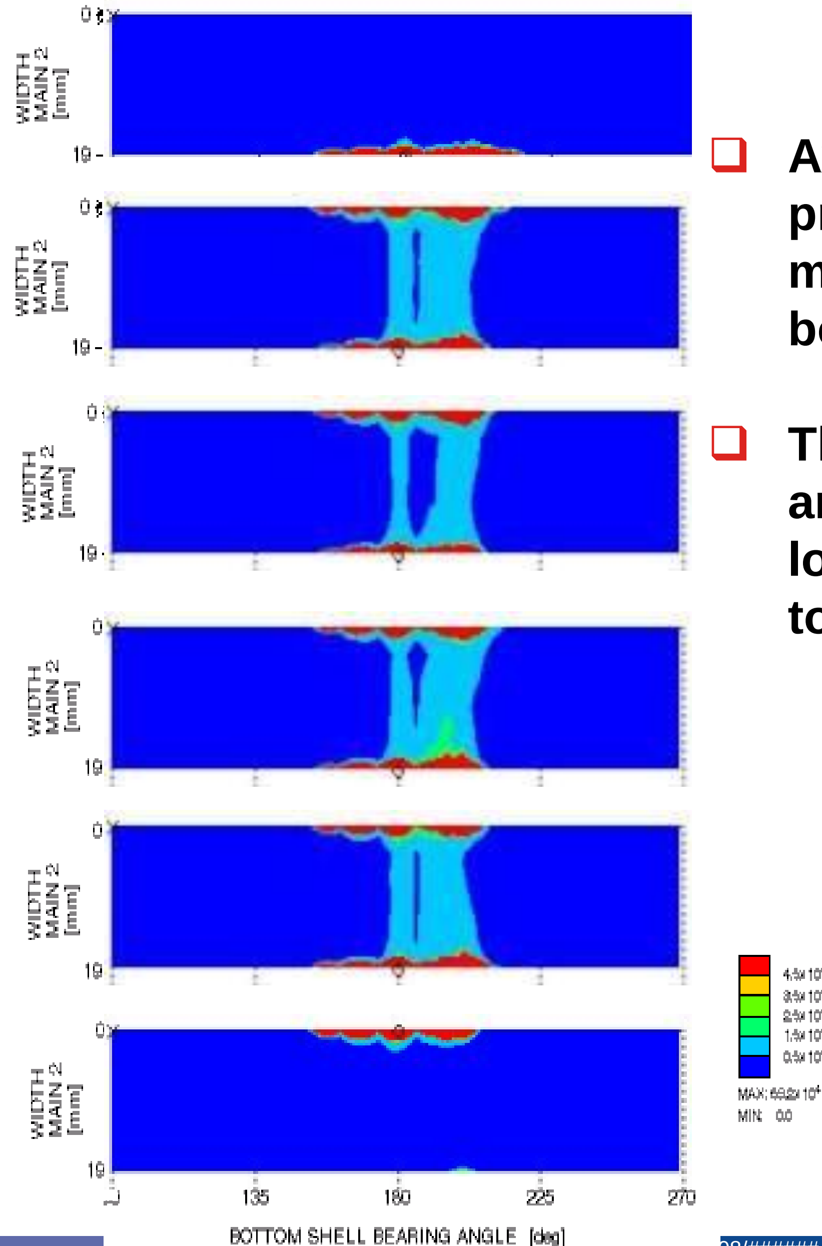


Crankshaft analysis (47) – axial profile to match worn bearings

Rear of engine

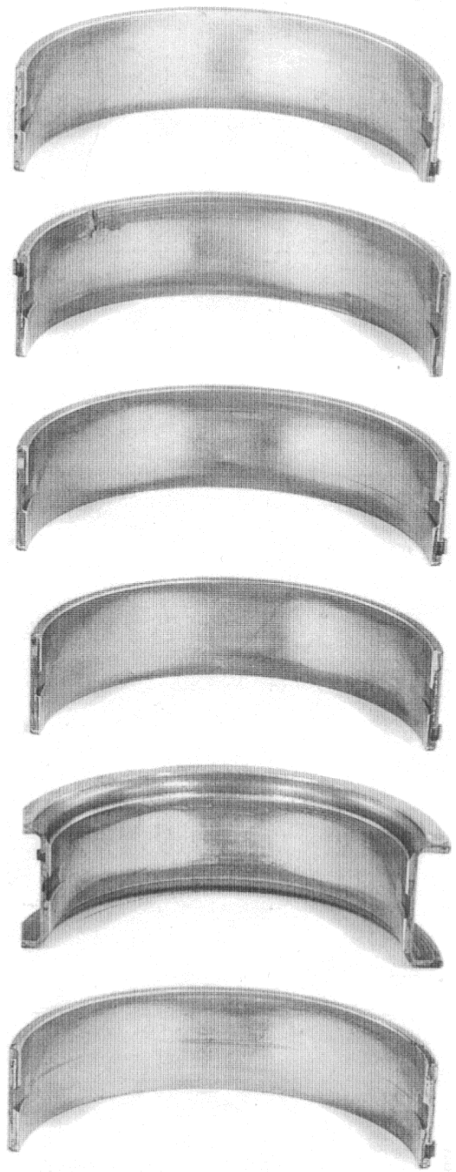


Front of engine



- Axial bearing profiles were measured at the bottom position
- The predicted areas of high wear load compare well to measurements

Crankshaft analysis (48) – assuming distortions and worn profiles



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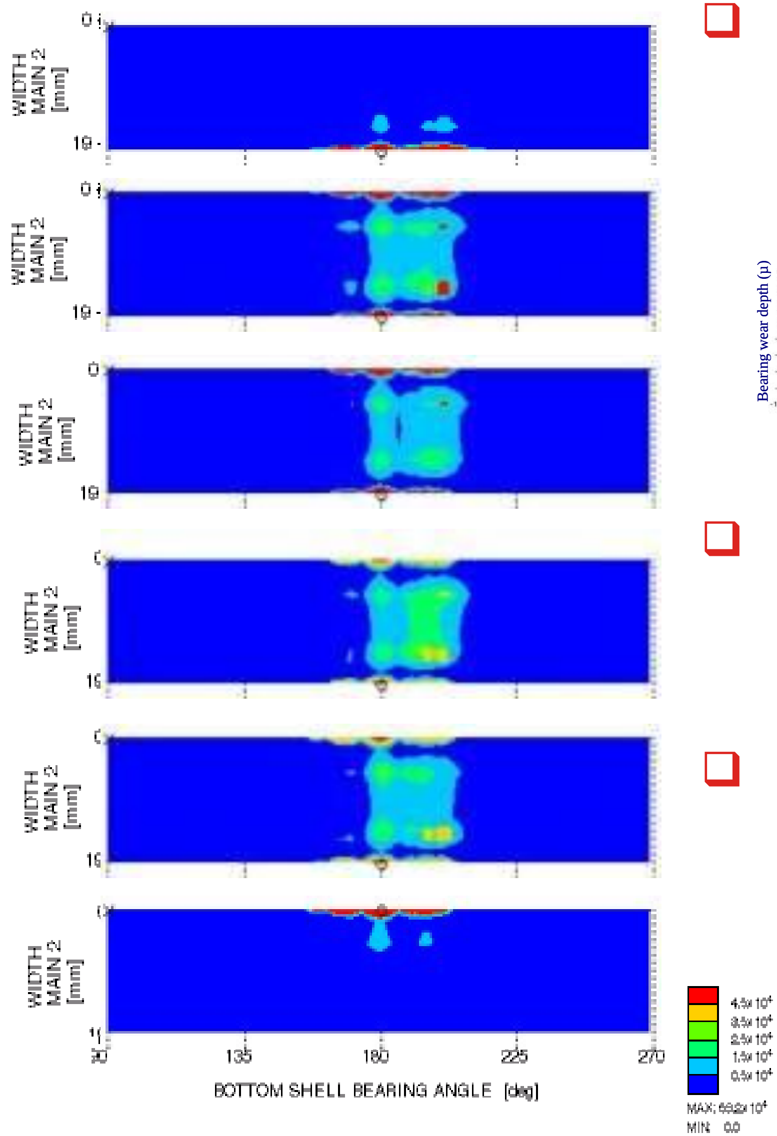
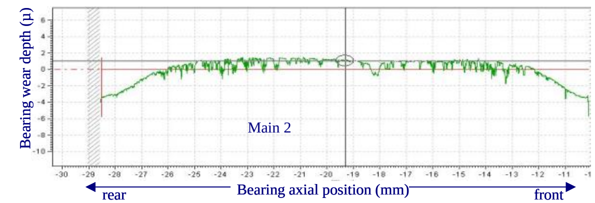


- ❑ **EHD analysis repeated**
  - – thermal and assembly distortions
  - – a simple axial profile



- ❑ **Predicted wear loading now more evenly spread across the bearings width**

- ❑ **But we need a method to iteratively predict a worn bearing profile from the wear loads**

Crankshaft analysis (49) – ENGDYN bearing friction - conclusions

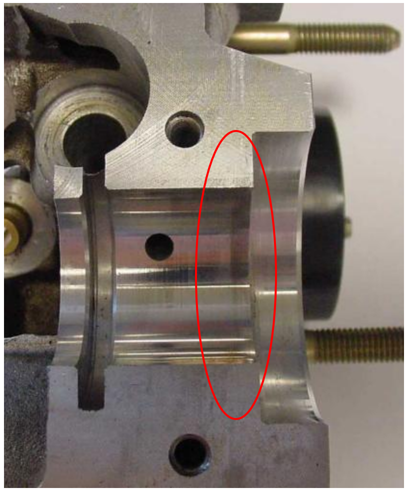
- **The use of EHD with distorted bearing shape and worn axial profile**
  - gives the best match to measured wear data but
  - the run time is very long

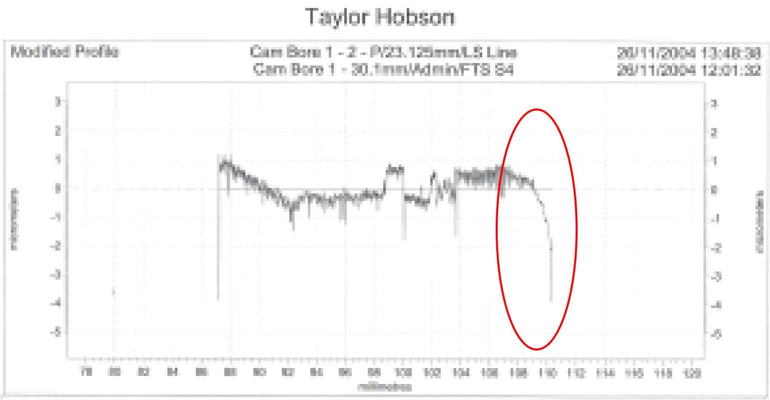
Crank model	Block model	Bearing solution	Hydrodynamic power loss (W)	Boundary power loss (W)	Total power loss (W)
rigid	rigid	mobility	1607	-	1607
dynamic	rigid	mobility	1435	-	1435
dynamic	dynamic	mobility	1448	-	1448
rigid	rigid	HD	1856	-	1856
dynamic	dynamic	EHD	1761	1061	2822
dynamic	dynamic	EHD with distortion	1779	3557	5336
dynamic	dynamic	EHD with distortion and profile	1886	637	2523

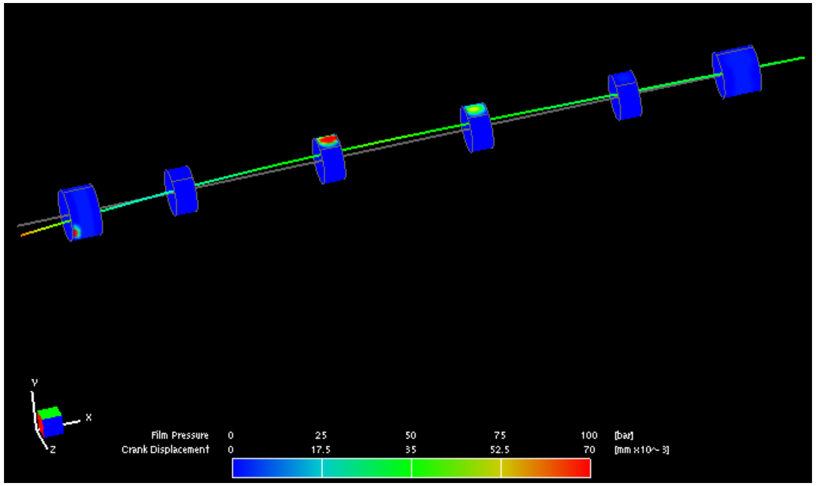
- **For a well proportioned bearing the predicted power loss due to oil shear is similar to that obtained by rigid assumptions and HD method with much shorter run time**
- **The final results shown here indicate that boundary contact friction loss was significant for this engine even when distorted bearing shape and worn axial profile shape were accounted for**

Crankshaft analysis (50) – Effect of model level on friction prediction

- **Ricardo have investigated the sensitivity of ENGDYN predictions of friction at camshaft bearings to modelling assumptions**
- **Front camshaft bearings often exhibit edge wear due to loads from timing drive**
  - This wear often occurs during run-in period and then stabilises
- **When a flexible model of the camshaft was used with dynamic loading assumptions, then edge contact was predicted at the front bearing and some other bearings**

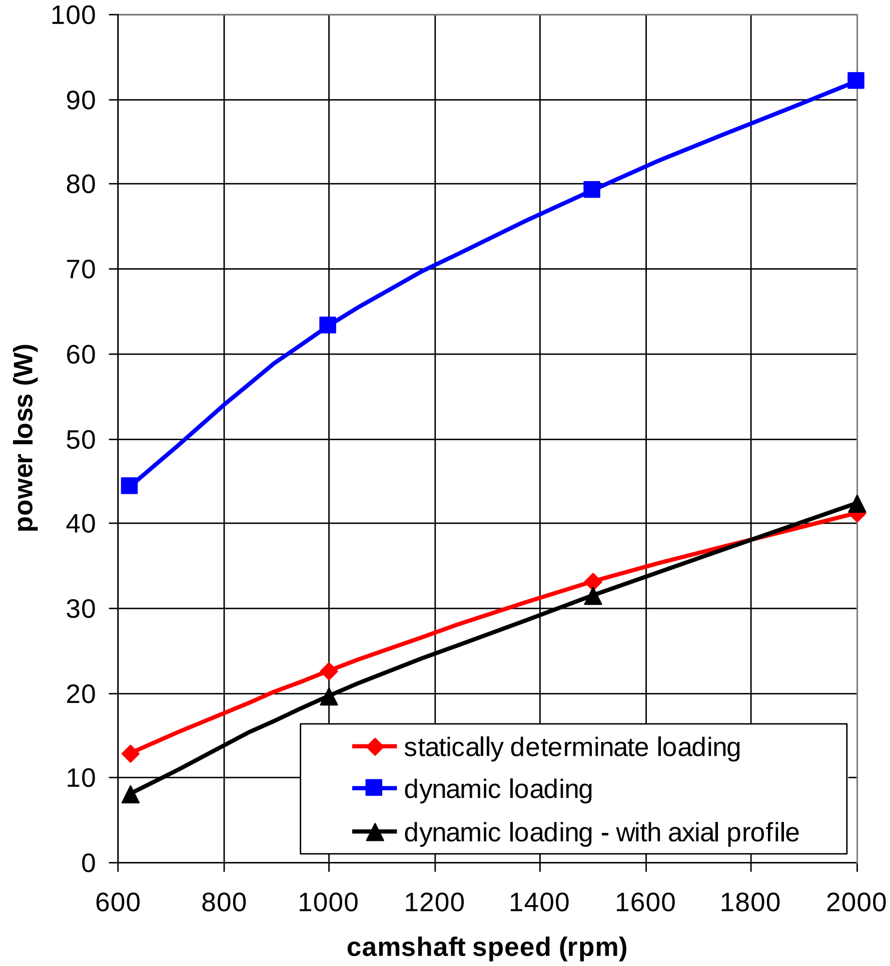






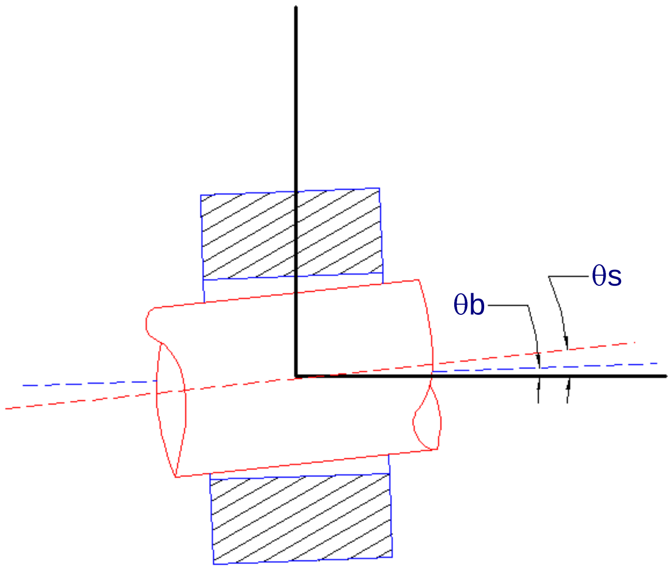
Crankshaft analysis (51) - effect of model level on friction prediction

- **The graph shows friction due to all camshaft bearings with**
  - A statically determinate loading
  - B dynamic loading
  - C dynamic loading and worn axial profile
- **The predicted friction level for Cases A and C was very similar**
  - So simple method is adequate for normal friction prediction work
- **The predicted friction for case B was much higher**
  - This increased friction level was due to boundary/mixed lubrication regime at the edge contacts
  - This occurs during run-in until the worn profile shape is obtained
  - But how to predict whether or not eventual wear level will be acceptable or not ?



Crankshaft analysis (52) – Relative shaft tilt

- **Ricardo are developing a new guideline based on relative tilt at main bearings**
  - This can be calculated using stiffness quality FE models of crankshaft and structure
  - Might indicate whether or not main bearings will suffer from edge loading
    - Without the need to perform EHD analysis



θs = Crank journal tilt

θb = Bearing housing tilt

(θs - θb) = Bearing angular misalignment (relative bearing tilt)

Crankshaft analysis (53) – Bearings - conclusions

- **Basic bearing analysis methods such as the mobility method in ENGDYN are generally acceptable for making predictions of friction at**
  - normally proportioned engine journal bearings such as
    - main bearings
    - big end bearings
    - camshaft bearings
    - balancer shaft bearings
  - under normal operating conditions
  - if seal losses and windage losses are considered separately
- **For rolling element bearings use ENGDYN to calculate loads and then calculate friction loss separately using friction coefficients**
- **Advanced journal bearing analysis methods**
  - can be used to investigate friction and wear under run-in conditions
  - cannot currently predict whether wear processes related to edge-loading are self-limiting or not