World SuperBike simulation and optimisation of gas dynamics

Ken Pendlebury – Ricardo UK

Contents

Introduction

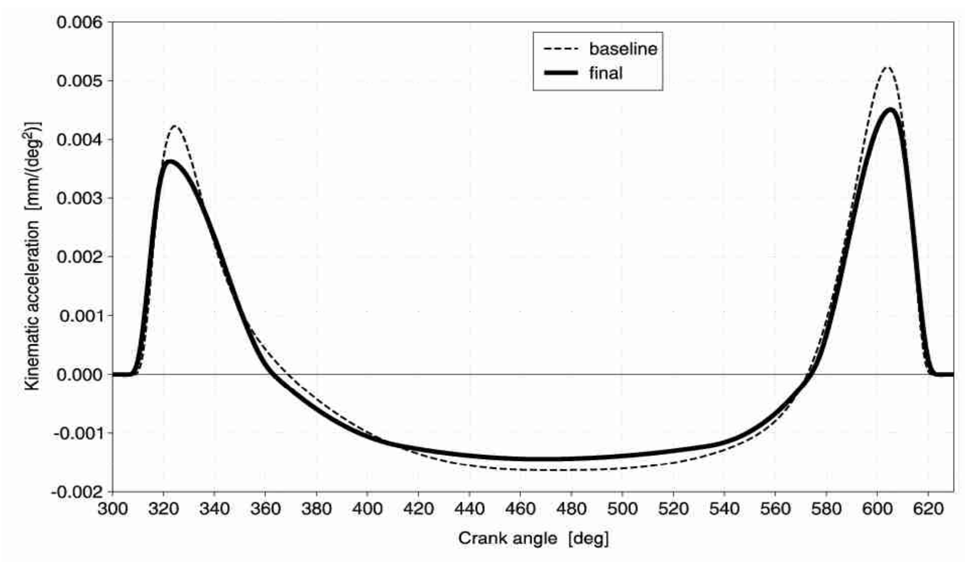
- World Superbike championship
  - Initial engine was designed when rules allowed 900cc I3 to compete with 750cc I4
  - Rule changes meant that 900cc I3 must race 1000cc I4 since a new engine could not be homologated
- Main engine development target was to maximise power whilst maintain good driveability
  - Baseline engine had engine speed limit of 14000 rpm
    - Limited by valve train dynamics
  - Target of 16000 rpm was identified to achieve target performance
  - Extensive use of analytical techniques to minimise testing
- Areas of focus for this presentation
  - Valvetrain
  - Crankshaft
  - Crankcase

Valvetrain design objectives

- Maintain vavletrain control and airflow at new rated engine speed of 16000rpm
- Allow accidental over-speed to 17000 rpm without piston-valve contact or instantaneous failure of any valvetrain component
- Minimise valvetrain friction within constraints of the homologated design
- By use of rig techniques determine limits and mode of valvetrain failures

Initial design analysis focused on cam profile design using kinematic analysis

- Kinematics module of Ricardo VALDYN used
- Focused mainly on intake valve train
- Changes made to enable high speed operation
  - Peak intake lift reduced by 1 mm
  - Intake period increased by 2.8 deg

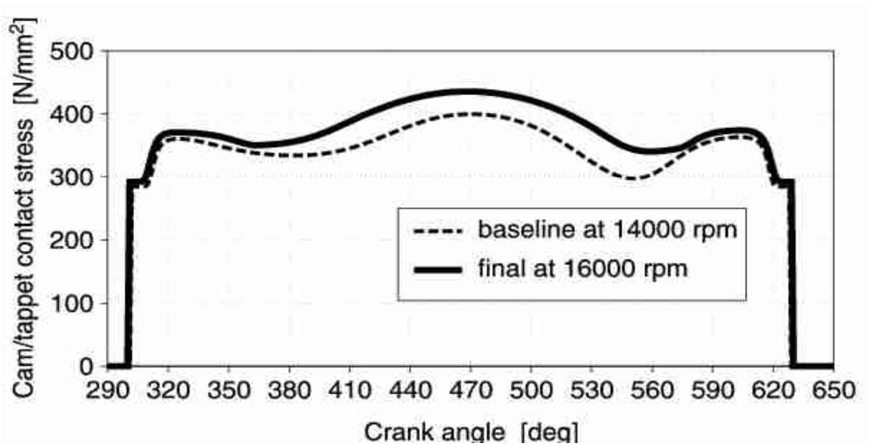


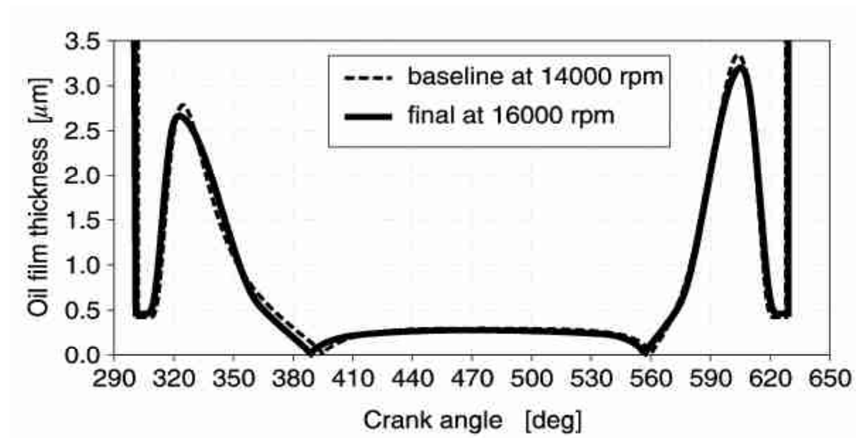
Parameter	Baseline	Final
Peak kinematic valve lift L (mm)	12.0	11.0
Inner seat diameter D (mm)	35.0	35.0
L/D	0.343	0.314
Lift area integral	0.555	0.557
Period – top of ramp (deg)	307.2	310.0
Ramp height (mm)	0.20	0.20
Ramp velocity (m/s)	0.432 @ 14000 rpm	0.500 @ 16000 rpm
Valve acceleration on opening flank (m/s^2)	29818 @ 14000 rpm	33404 @ 16000 rpm
Valve acceleration on cam nose (m/s^2)	11530 @ 14000 rpm	13305 @ 16000 rpm
Valve acceleration on closing flank (m/s^2)	36962 @ 14000 rpm	41554 @ 16000 rpm
Opening side acceleration ratio	2.51	2.51
Closing side acceleration ratio	3.21	3.12

This then moved to the cam/tappet interface (kinematic analysis)

- Kinematics module of Ricardo VALDYN used
- High speed contact stress increased
- Low speed contact stress reduced
- Film thickness at nose reduced slightly
- Film thickness at transition improved
- Tappet edge clearance increased

Parameter	Baseline	Final
Peak cam tappet contact stress at idle (N/mm^2)	831 @ 3500 rpm	764 @ 3500 rpm
Peak cam tappet contact stress at rated speed (N/mm^2)	400 @ 14000 rpm	436 @ 16000 rpm
Lubricant film thickness at peak cam lift (μm)	0.295	0.278
Deschler and Wittman number at peak lift	0.207	0.272
Maximum number of consecutive crank degrees at which oil film thickness is less than 0.1 μm at rated speed	8.26	7.86
Minimum tappet edge clearance (mm)	0.30	1.90





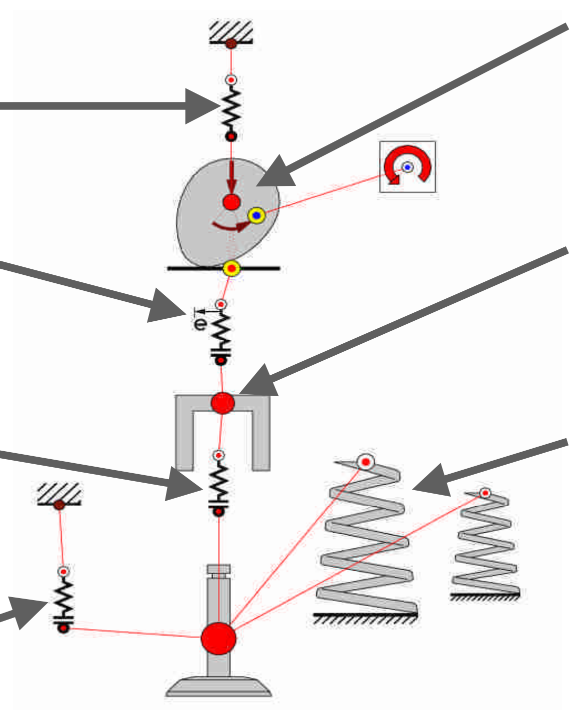
Following this a full valvetrain dynamics model was built using VALDYN

Camshaft bending stiffness and support stiffness

Cam/tappet stiffness (dependent on eccentricity)

Stiffness of valve stem between tip and centre of mass

Combined stiffness to represent valve seat contact and valve head bending



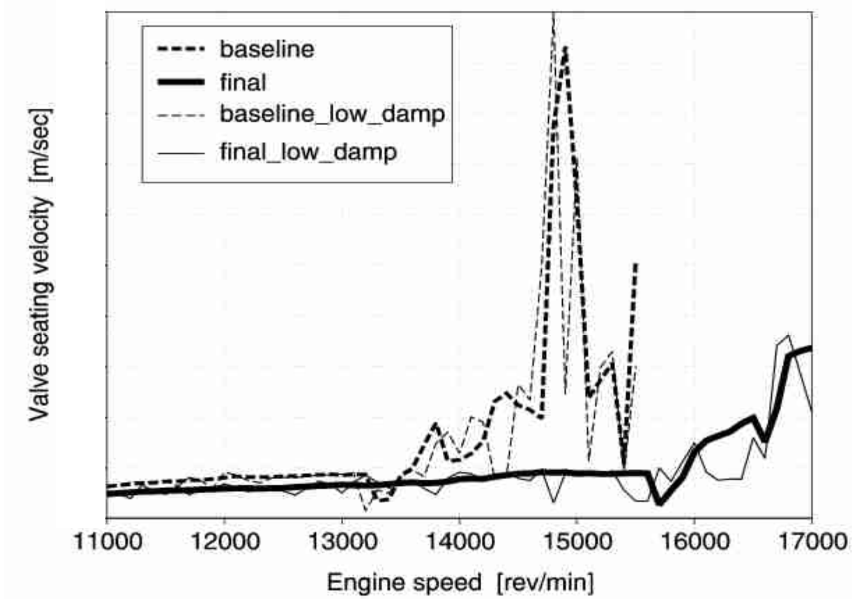
Node representing effective local mass of camshaft

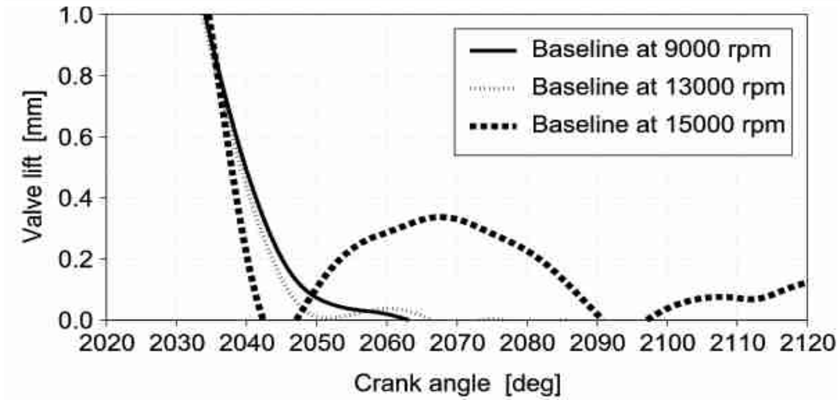
Node representing tappet mass

Valve spring models
- 8 masses per coil
- Connected by stiffness
- Coil clashing model
- Spring interference damping uncertainty so modelled with 2 levels for each run

Design Analysis – Valve seating

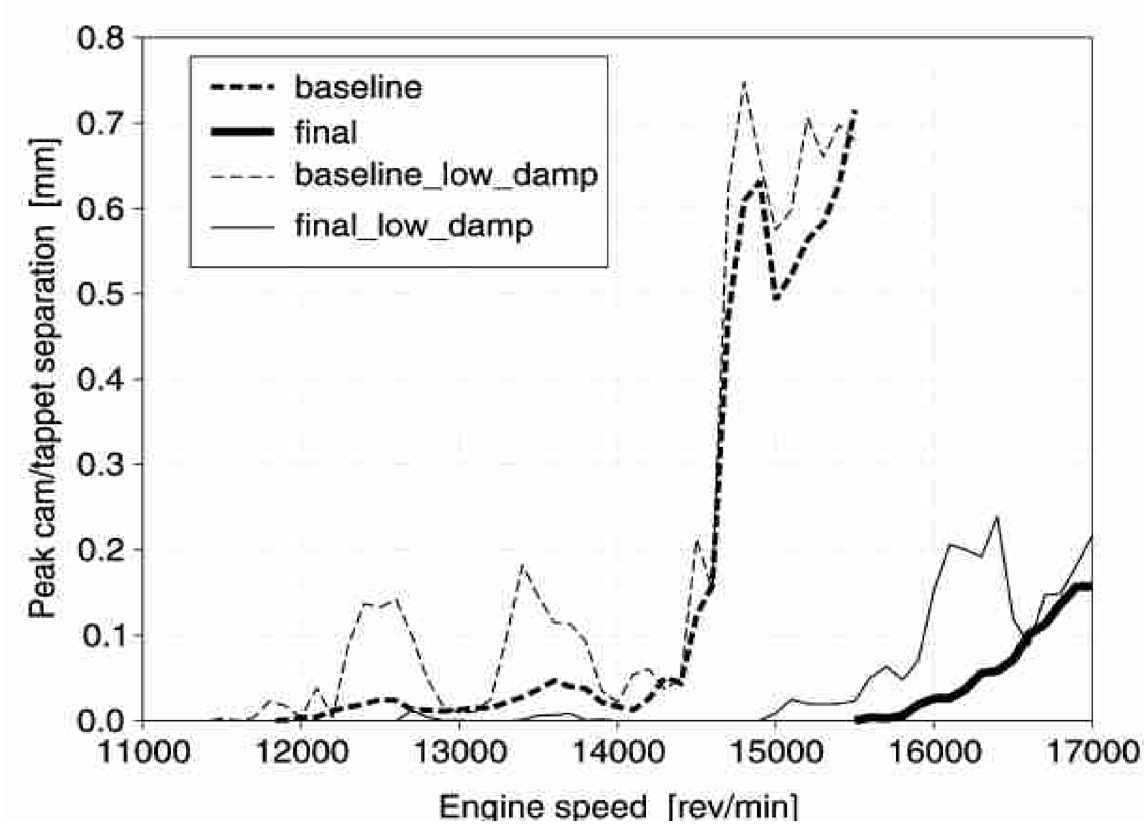
- Baseline design
  - Loss of control from ~14000 rpm
  - Sharp transition to high velocity at ~14800 rpm
  - Large valve bounce evident at 15000 rpm
  - Failures of valve stem observed
- Final design
  - Loss of control from ~16000 rpm
  - Below 4 m/s even at 17000 rpm
  - No failures
- Results not dependent on spring damping assumption





Design Analysis – Valve jump

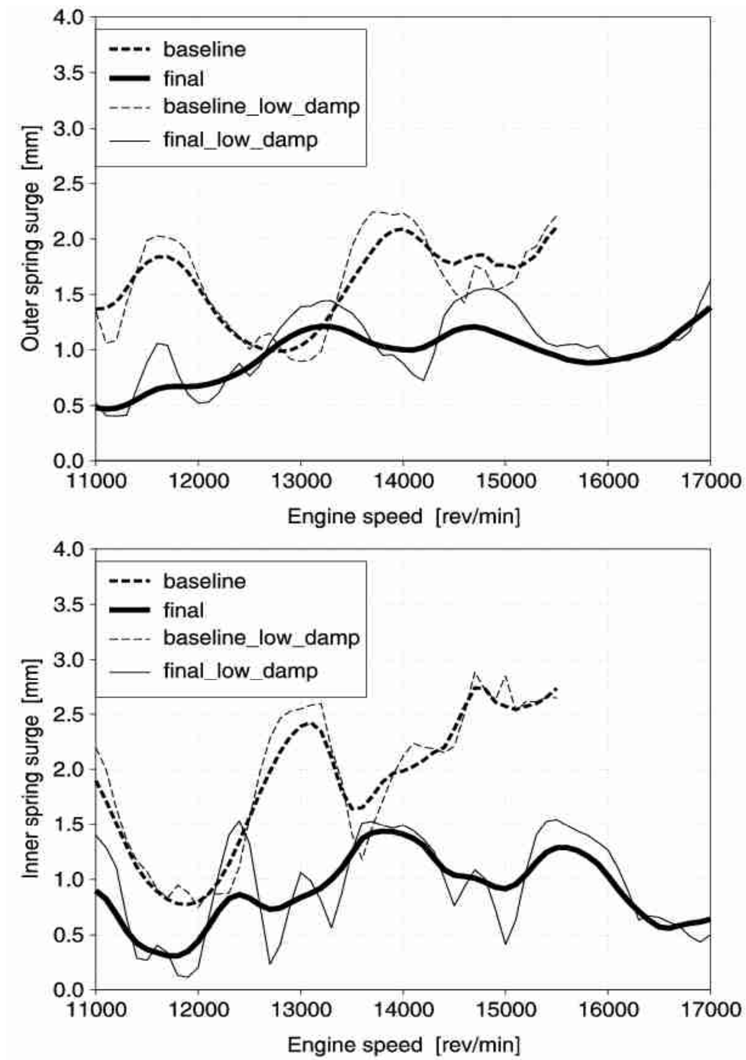
- Baseline design
  - Sudden transition at ~14600 rpm
- Final design
  - Progressive increase in separation from ~16000 rpm with high damping
  - Less than 0.2 mm peak separation at 17000 rpm
- Results sensitive to spring interference damping assumption



Note:

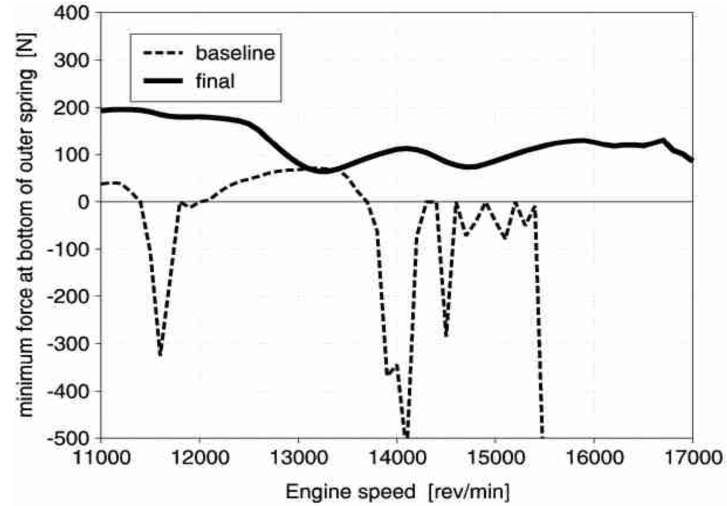
Design Analysis – Spring surge (1)

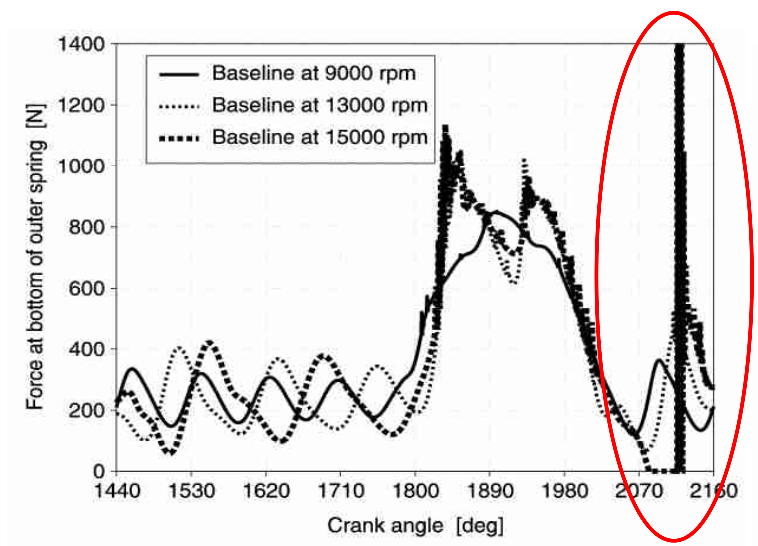
- Baseline design
  - High surge amplitude on both springs
  - +/- 1mm normal target for passenger car engines
- Final design
  - Significant reduction in surge across speed range
- Results moderately sensitive to spring interference damping assumption

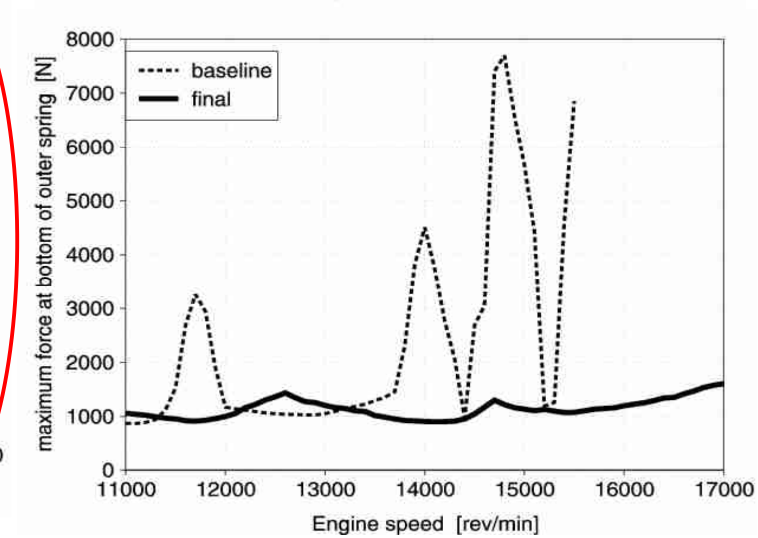


Design Analysis – Spring surge (2)

- On baseline design surge led to loss of contact between spring and seat at high speed just after valve closing
  - High force when contact re-established
    - Spring seat hammering
  - Some failures of spring end tangs resulted

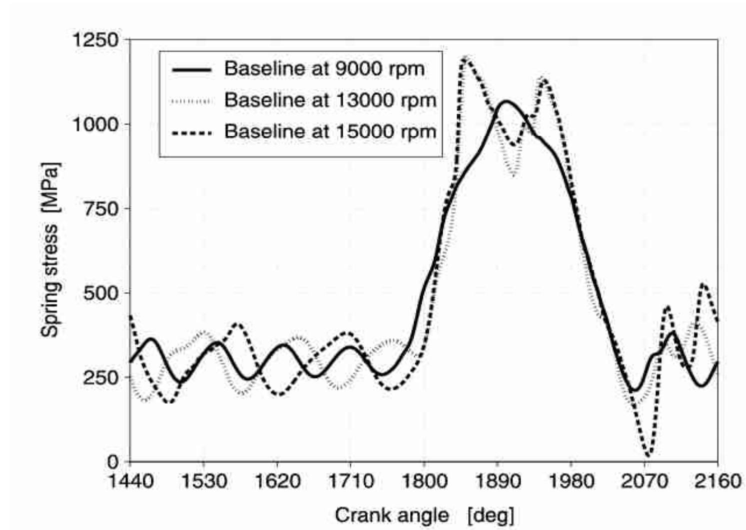


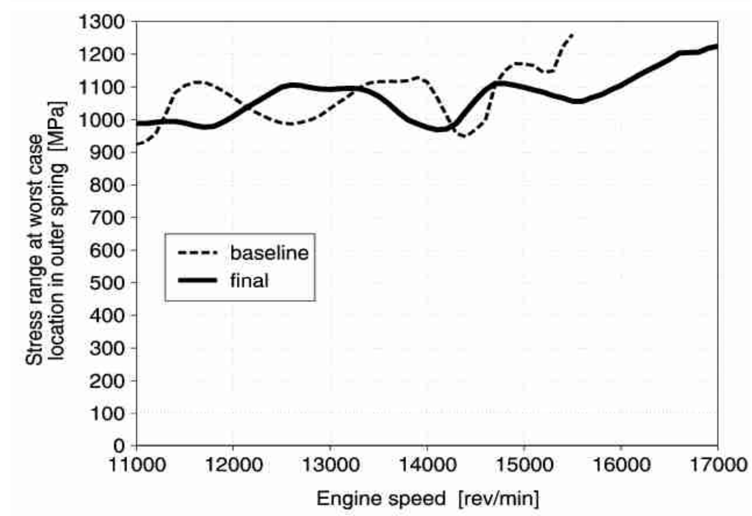




Design Analysis – Spring stress

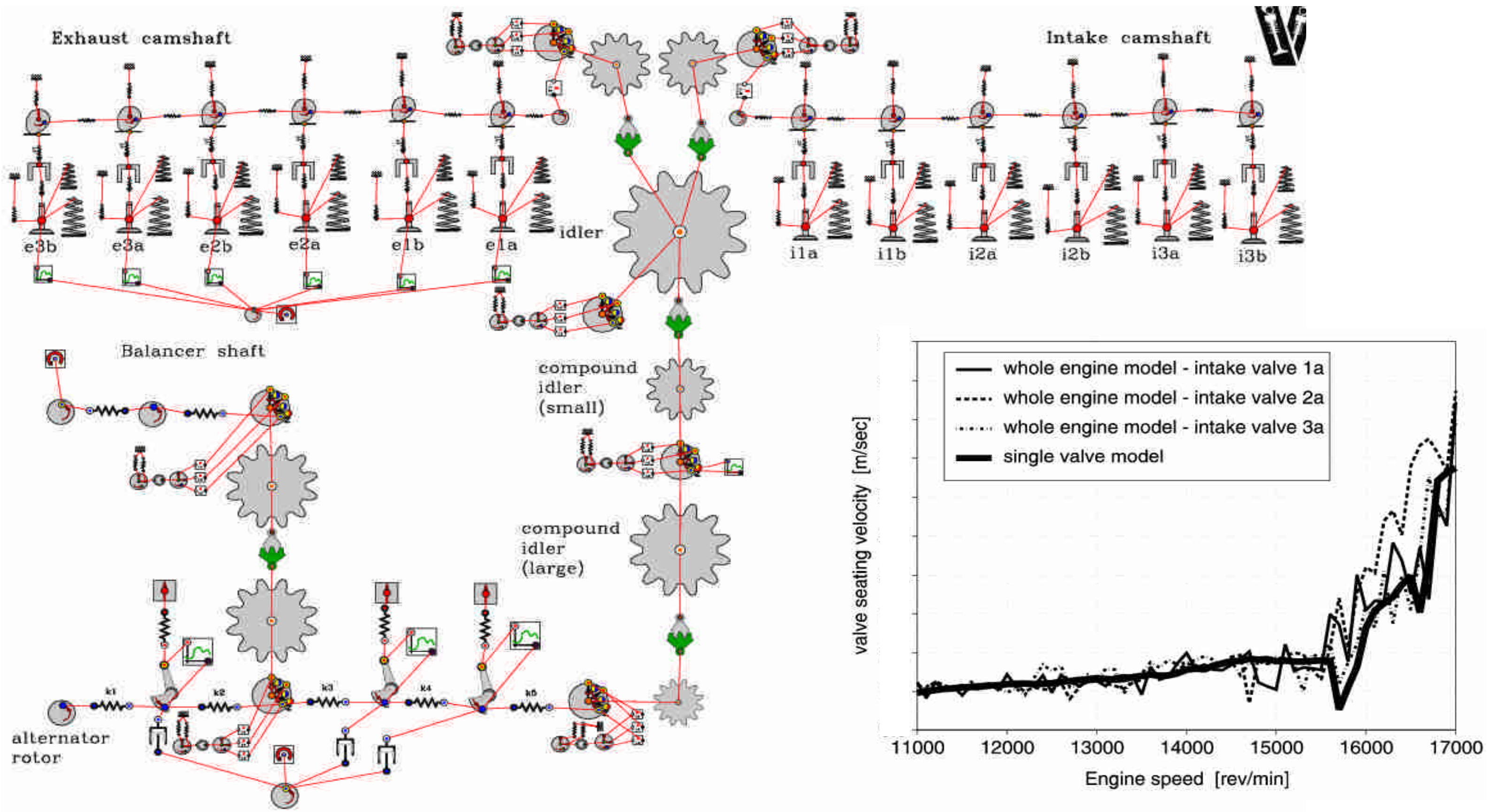
- Baseline design
  - Stress at worst case location in spring increases as valve train loses control at high speed
- Final design
  - Pseudo-static spring stress levels were increased but the spring strength was also improved
  - Dynamic stresses were controlled to similar level as baseline design





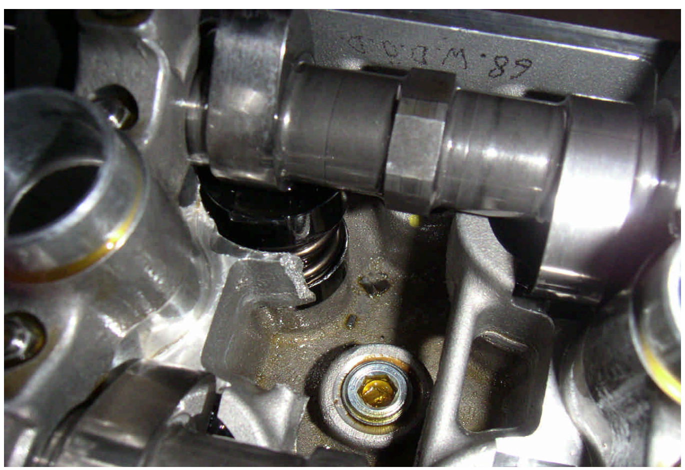
Design Analysis – Whole engine model

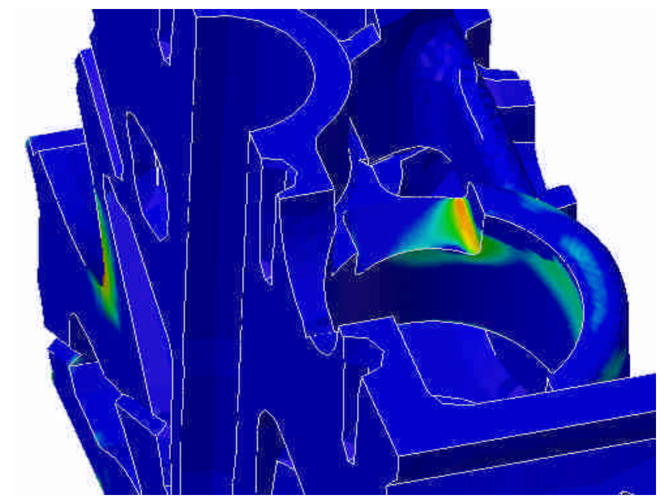
- VALDYN model was extended and used to calculate
  - Effect of timing drive on complete valve train motion
  - Dynamic loads at gears and fasteners for subsequent analysis



Example Valvetrain failure mode, Tappet Bore

- Several failures of cylinder head structure at tappet bore
- Cracks in cylinder head at machined slot for cam clearance
- VALDYN analysis used to calculate moment on tappet
- Reaction forces calculated and applied to local FE model
- FEARCE used to calculate safety factors
  - Low safety factors confirmed and alternative designs addressed
- Small change in fillet radius gave desired improvement



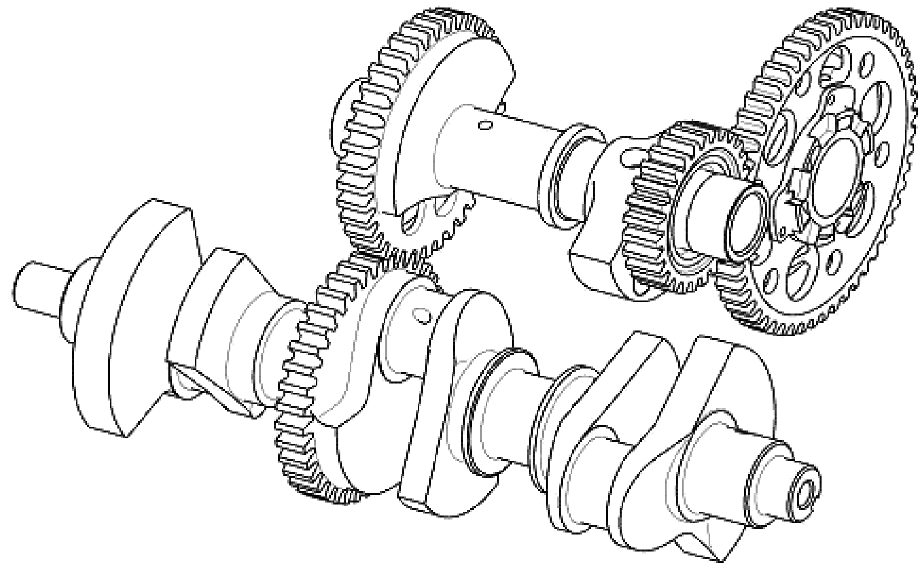


Valvetrain analysis conclusions

- The final intake valve train
  - Had effective mass reduced by 15.3g (18%)
  - Had exceptional durability with rev limiter set to 16000 rpm
  - Was able to survive over-speed events at up to 17000 rpm without failure
- Success was achieved by
  - Making extensive use of dynamic simulation
  - Combined with minimal rig testing
- The contribution of world class component suppliers to the success of the project was invaluable

Crankshaft design objectives

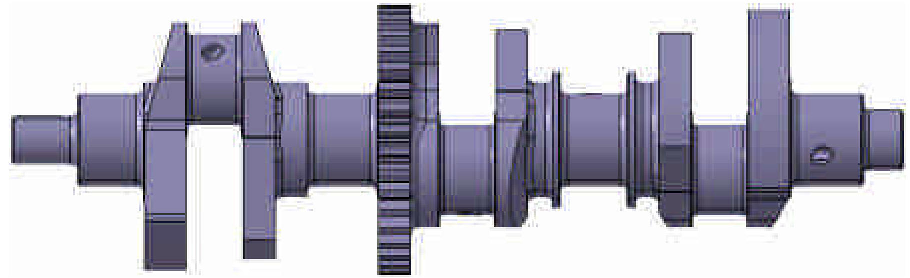
- Main objectives
  - Reduce crankshaft mass
  - Reduce rotating inertia
  - Reduce friction
  - Reduce windage
  - Maintain adequate crankshaft strength
  - Maintain adequate bearing durability
  - Maintain acceptable engine balance



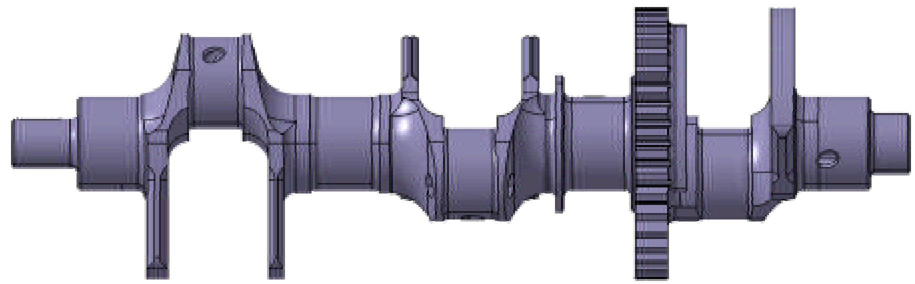
- Crankshaft design overview
  - Fully machined crank
  - Integral drive gear
  - Double vacuum re-melted steel 31CrMoV9
  - Gas-nitrided to 800Hv to depth of 0.3 mm
  - Polished bearing journal surfaces
  - Full circumferential grooves in main bearings
  - Big end bearings supplied from main bearings via drillings

Summary of crankshaft design iterations

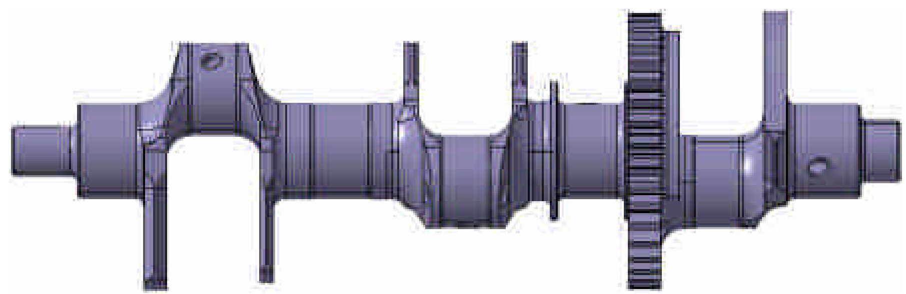
- Pictures show the design evolution of the crankshaft
- The drive gear was moved from web 3 to web 5 to avoid transmitting power through the balancer shaft
- Piston and connecting rod were also lightened during the project
- Final design was not balanced with a corresponding increase in vibration (not discussed here)



Baseline design



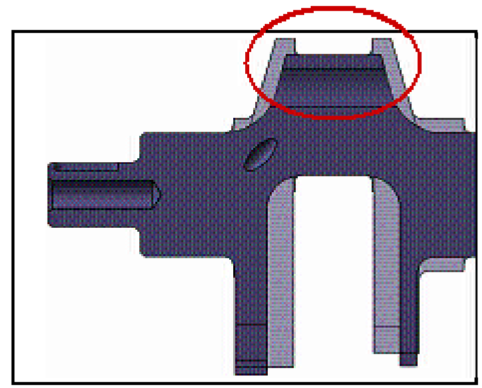
Intermediate design

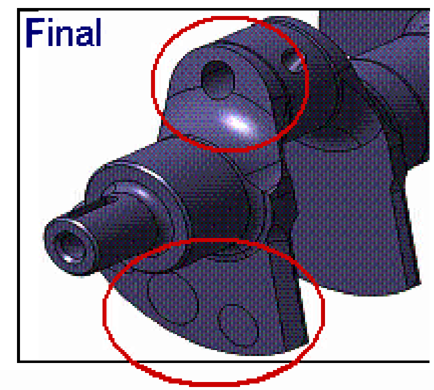


Final design

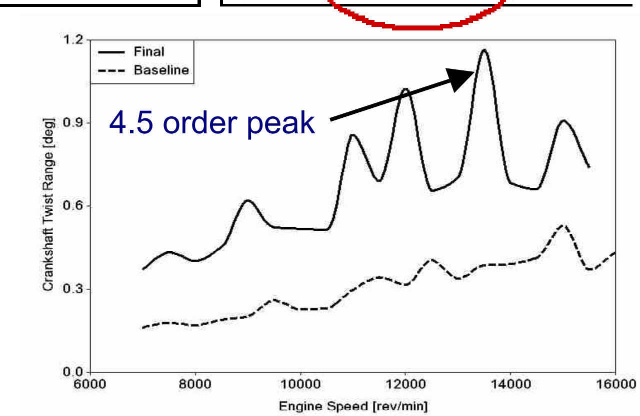
Initial focus was reducing mass and rotating inertia

- ❑ Smaller counterweights used for final design as engine was no longer fully balanced (see later section)
- ❑ Reduce the mass of ‘upper’ portion of the crankshaft
- ❑ Drill through the crank pin
- ❑ Use heavy metal inserts in counterweights
- ❑ 30% mass reduction
- ❑ 35% inertia reduction



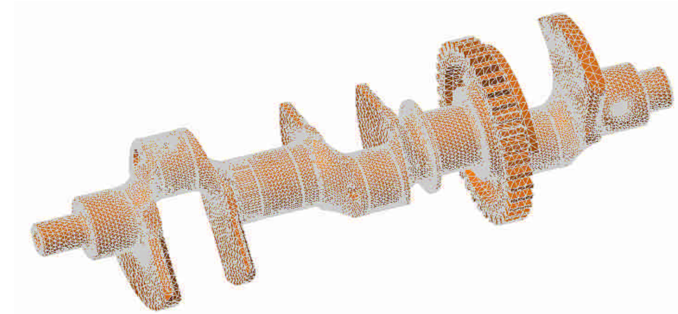


- ❑ ENGDYN 3D crankshaft dynamics analysis shows significant increase in crankshaft twist for final design
  - – Baseline crank natural frequency of 1317 Hz
  - – Final crank natural frequency of 971 Hz

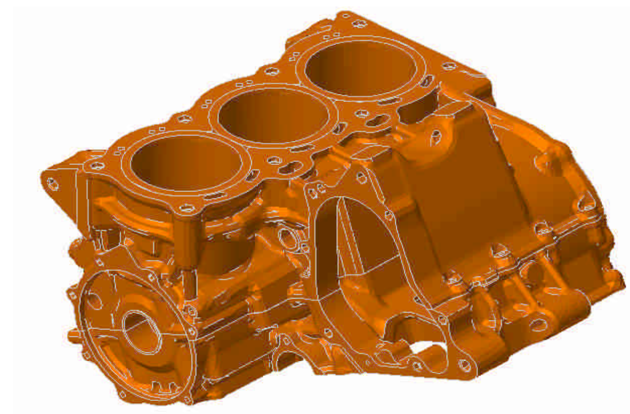


Stress analysis

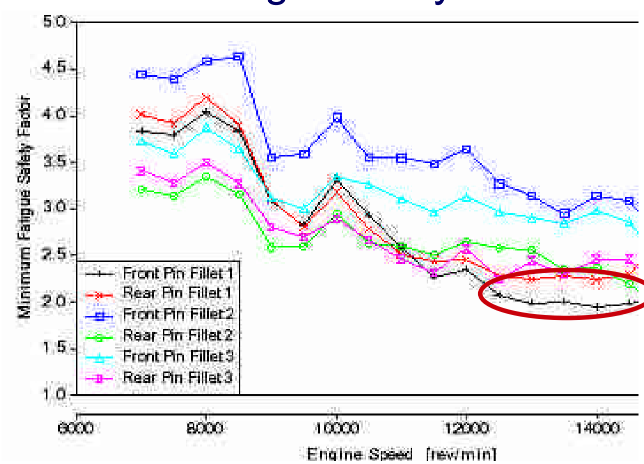
- Finite element analysis was performed on the baseline and final crankshafts

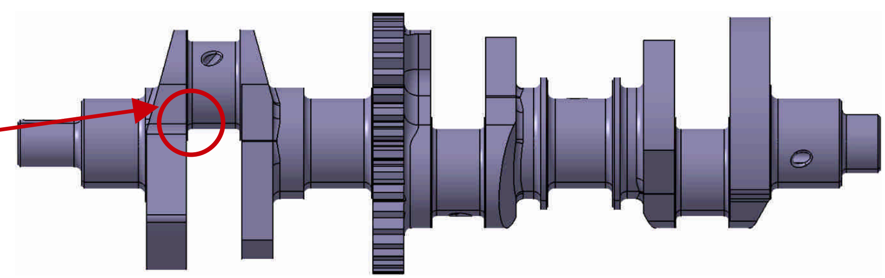


- ENGDYN used to
  - Calculate boundary conditions
  - Combine FE models
  - Solve equations of motion
  - Calculate combined stresses at 5 degree intervals for each engine speed
  - Calculate Goodman safety factors at fillets and oil holes



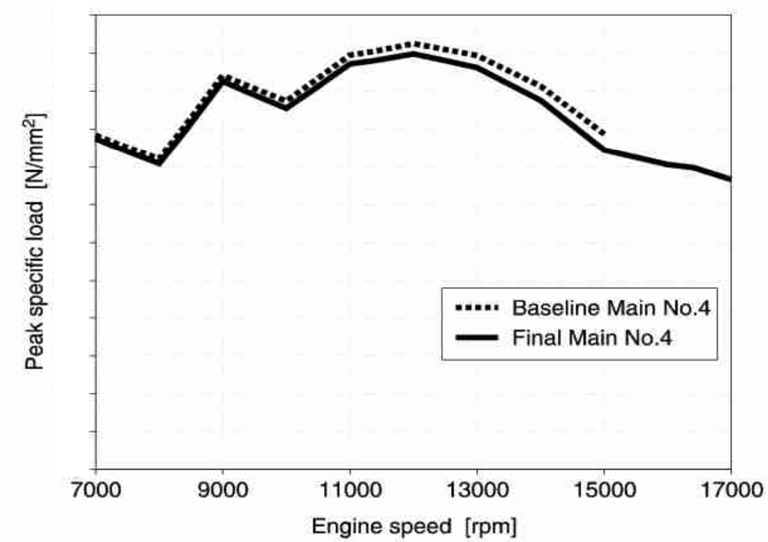
- Baseline results indicate that lowest safety factor occurred at crank pin fillet on web No.1
  - Radius significantly increased by use of piston guided rod

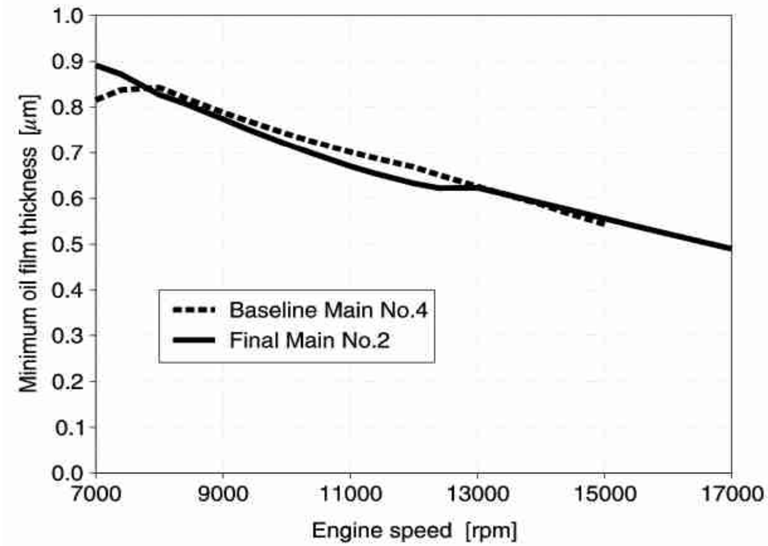




In parallel, analysis of the main bearings was carried out

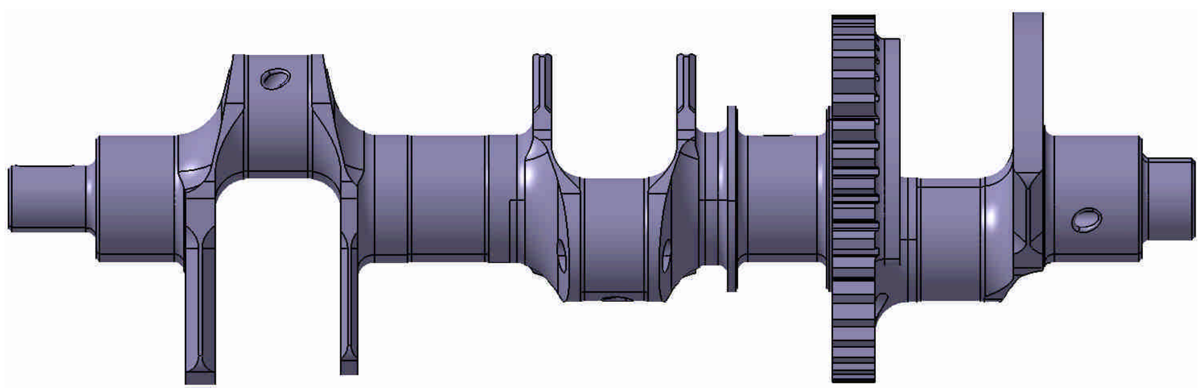
- ENGDYN bearing analysis shows
  - Reduced peak specific load at worst case speed (peak torque)
  - Slight reduction in minimum oil film thickness at high speed
  - Slight increase in hydrodynamic power loss at 14000 rpm





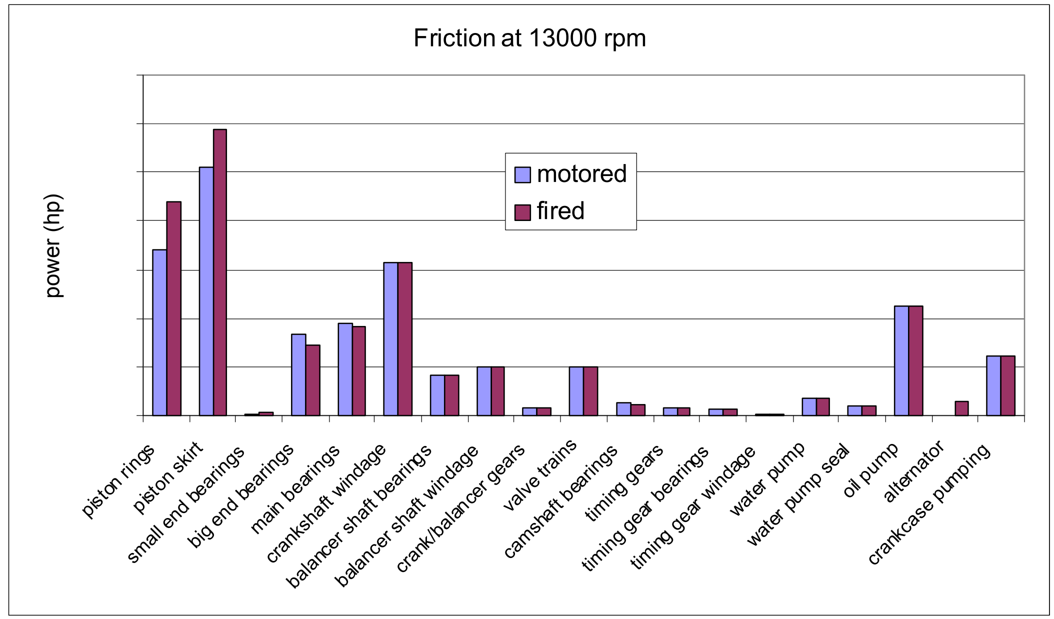
Crankshaft analysis conclusions

- Use of advanced analysis was able to significantly reduce the mass and inertia of the crankshaft whilst still maintaining acceptable levels of balance, torsional vibration and durability
- The final crankshaft design
  - Had exceptional durability even when rev limiter was set to 16000 rpm despite considerable increase in twist due to torsional vibration
    - 30% mass reduction
    - 35% inertia reduction
  - Had partially balanced primary reciprocating moment
- Riders preferred low inertia of final design and were prepared to tolerate increased vibration



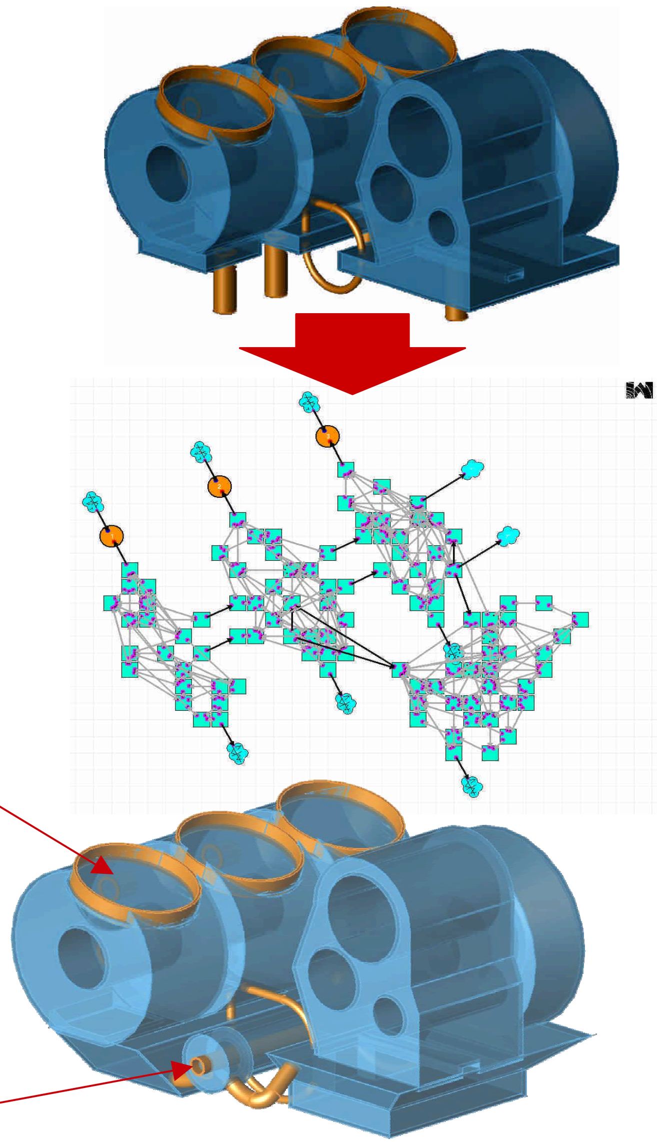
Crankcase issues and approach

- Pumping of crankcase gas incurs a power loss
  - Gas exchange between bays
  - Gas exchange through external breathers
  - Other minor losses
    - Gas exchange between cylinder and crankcase volumes
    - Heat transfer to crankcase walls
  - Losses can be significant
- Windage loss
  - Interaction of engine components with crankcase fluid
- Conversion from a wet to a dry sump system began in 2005
  - Targeted benefits
    - Reduction in CPMEP
    - Increased scope for revised mass distribution
  - Analysis required to
    - Increase knowledge and understanding
    - Limit amount of testing
- Prediction of pumping losses can be obtained from relatively simple 1D flow analysis



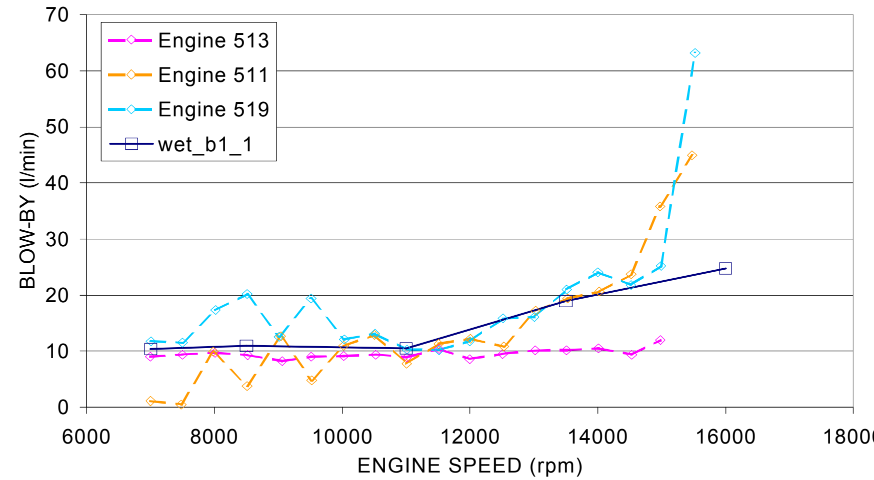
Analytical approach using WAVE

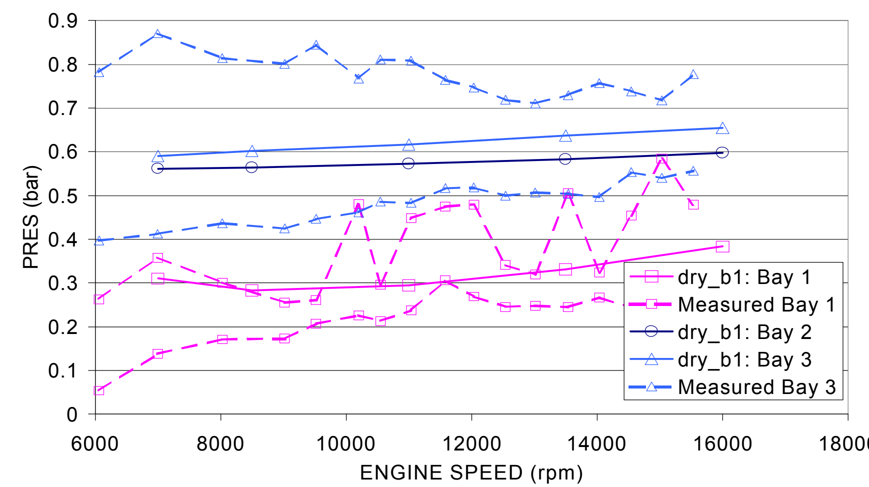
- 1D time-dependant fluid dynamics
- Complex geometries constructed using WAVEBuild3D
- Automatically meshed into 1D network components
- Program input data
  - Engine internal component volumes
  - Engine configuration, bore, stroke, rod length, firing order
  - Cylinder pressure
  - Scavenge flow rate
  - Wall temperatures
- Cylinder bases attached to variable under-piston volume
  - Connected to cylinder pressure via a duct and orifice representing blow-by path
- Scavenge points
  - Connected to gear pump with imposed constant velocity
- External breather
  - Connected to ambient conditions



Model validation

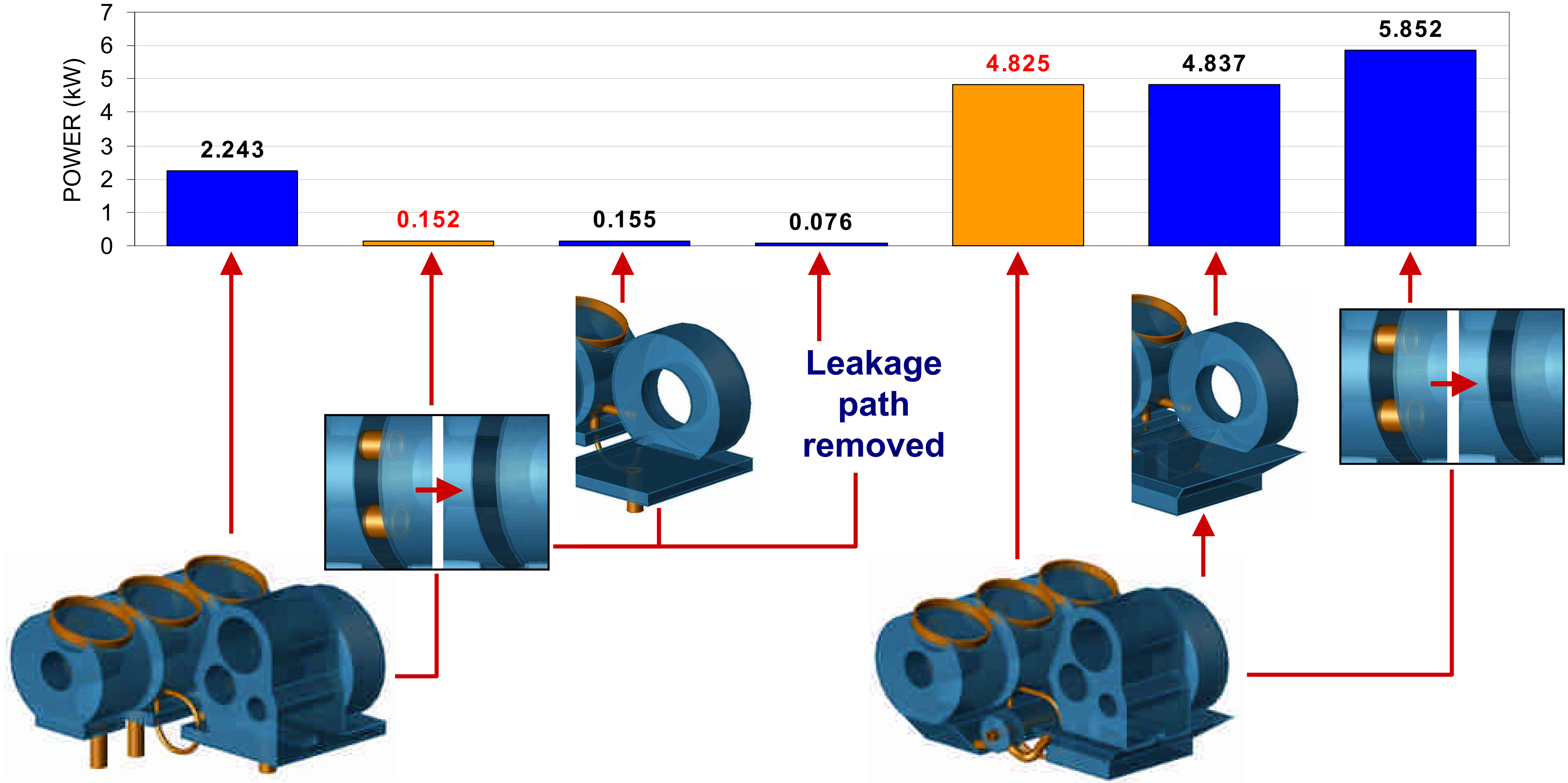
- Blow-by flow
  - Blow-by affected by driving pressure and behavior of piston and rings
    - Measurements on wet sump engines showed considerable variation
  - Blow-by orifice geometry
    - Adjusted to achieve a reasonable fit to data
    - Copied to dry sump model
- Mean crankcase pressure
  - Variation in measured data on dry sump engine
  - Scavenge velocities and leakage orifice dimensions adjusted to achieve a good fit



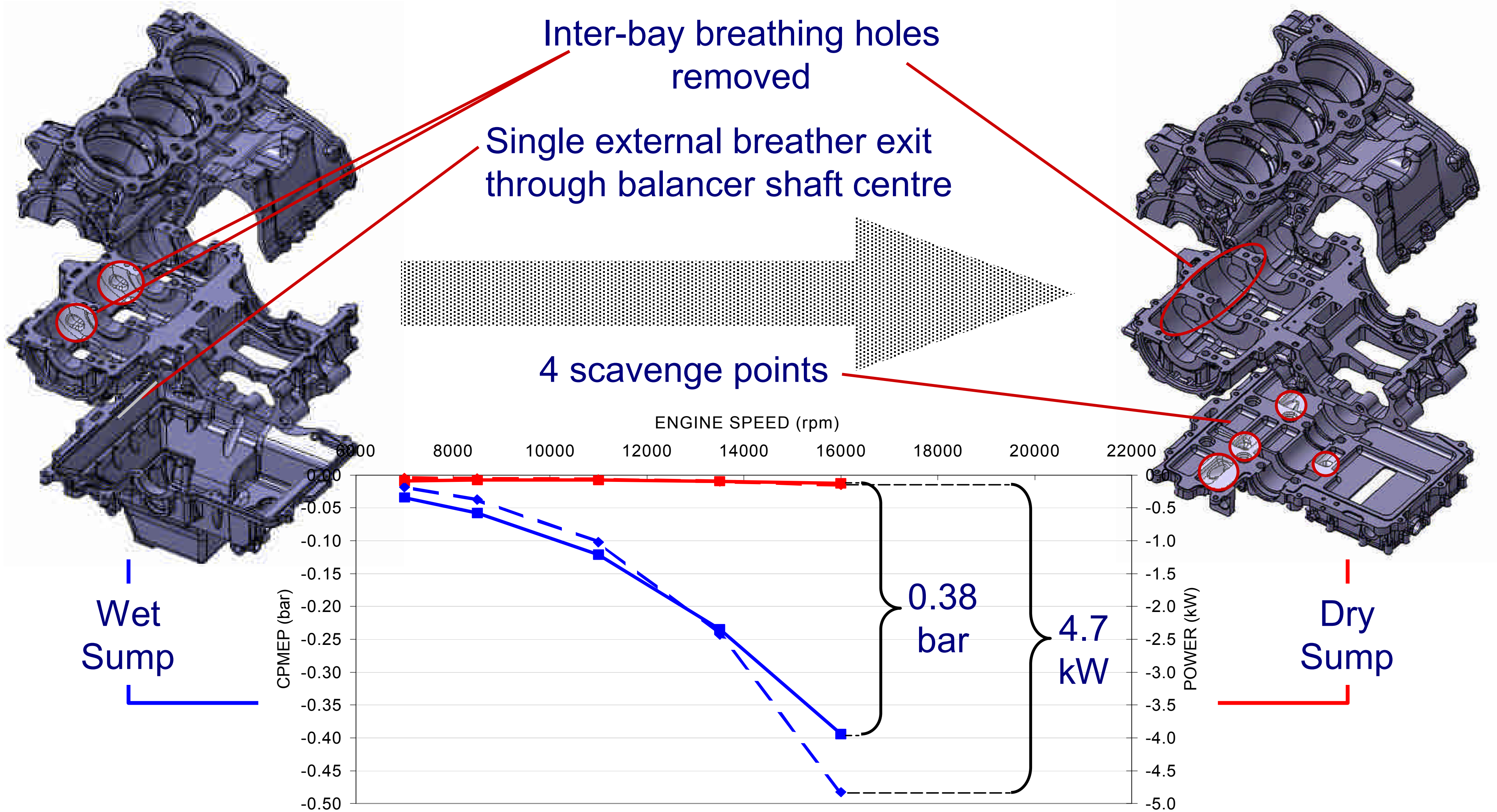


FP1 Engine – Results (1)

Power loss at 16000rpm



FP1 Engine – Results (2)



Crankcase analysis conclusions

- Analysis showed a potential 4.7kW reduction in crankcase pumping loss with a dry sump system
- ~ 4-5kW total benefit realised in practice
- Parametric studies showed dominant parameters effecting CPMEP
  - Breather size & discharge coefficient
  - Engine displacement
  - Crankcase compression ratio
  - Scavenge flow rate

Conclusions

- Maximum engine speed was increased from 14000rpm to 16000rpm which along with the improved parasitic losses released an additional 30ps over the two years of the development programme
- Valvetrain life has been improved considerably from around 600,000 cycles to >1,000,000 cycles
- Valvetrain reliability has been improved with over-speed capability to 17000rpm and no valvetrain failures recorded in race conditions in 2006
- The final crankshaft design
  - Had exceptional durability even when rev limiter was set to 16000 rpm despite considerable increase in twist due to torsional vibration
    - 30% mass reduction
    - 35% inertia reduction
  - Had partially balanced primary reciprocating moment
- Riders preferred low inertia of final design and were prepared to tolerate increased vibration