Design and Analysis of a Lightweight Crankshaft for a Racing Motorcycle Engine

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Contents

Introduction

World Superbike championship

- Petronas designed engine when rules allowed 900cc I3 to compete with 750cc I4
- Rule changes meant that 900cc I3 must race 1000cc I4

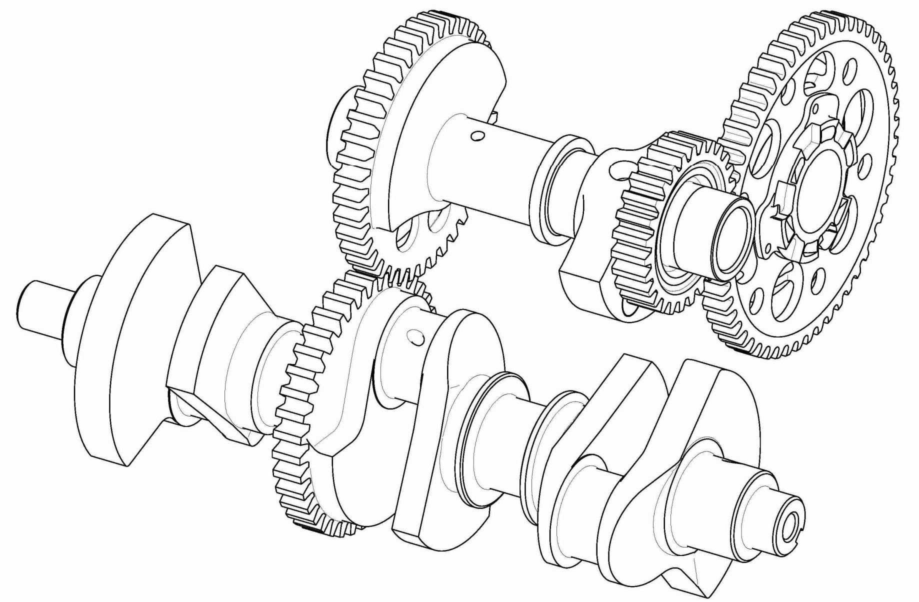
Main engine development target was to maximise power

- Baseline engine had rev limit of 14000 rpm but changes to valve train enabled 16000 rpm engine speed
- Piston and rod were also redesigned to enable operation at higher speed
- This presentation covers design and analysis of crankshaft

Crankshaft design objectives

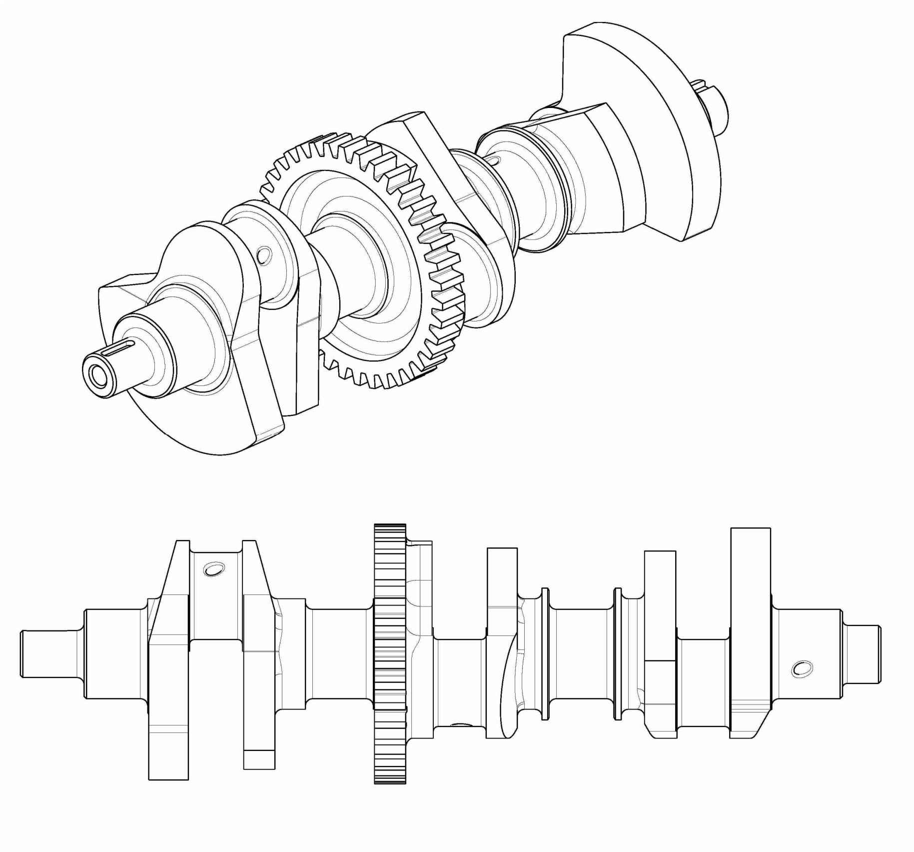
Main objectives

- 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

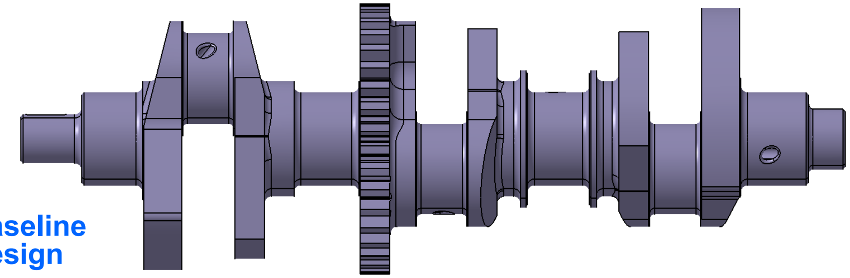
- Fully machined crank
- Integral drive gear
- Double vacuum remelted steel 31CrMoV9
- Gas nitrided to 800Hv to depth of 0.3 mm
- Polished bearing journal surfaces
- Full circumferential grooves in main bearings
- Big end bearings supplied from main bearings via drillings



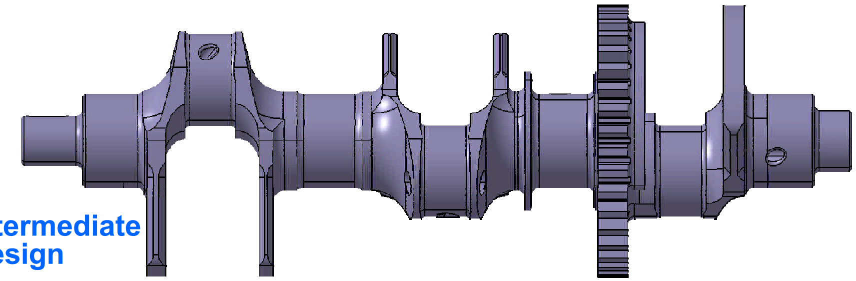
Crankshaft Design Iterations

- 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 rod were lightened during the project

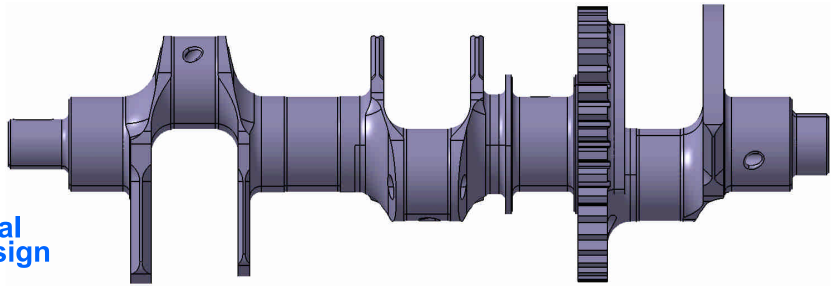
Component	Baseline	Final
Piston assembly mass (kg)	0.292	0.249
Connecting rod mass (kg)	0.278	0.245



Baseline Design



Intermediate Design

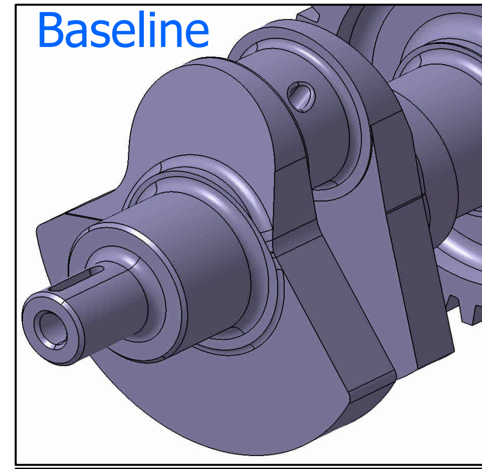


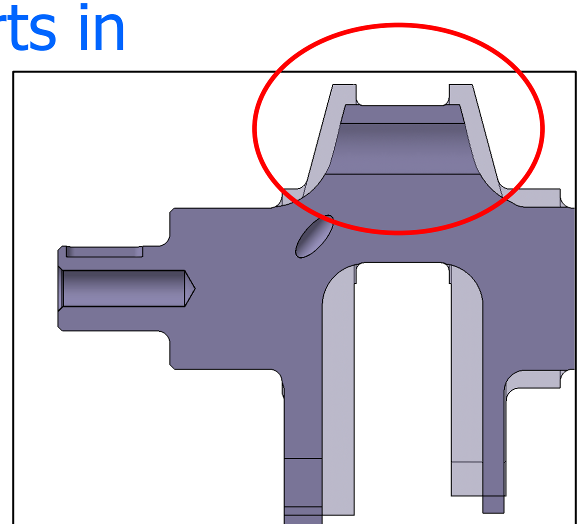
Final Design

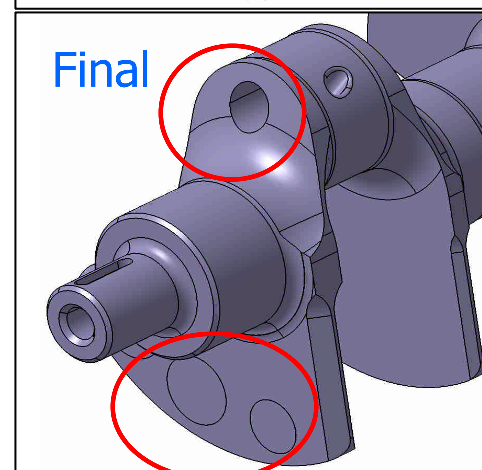
Reducing Mass and Rotating Inertia

- 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

Design	Baseline	Intermediate	Final
Mass (kg)	6.136	4.375	4.282
Inertia (kgmm^2)	5070	3255	3265

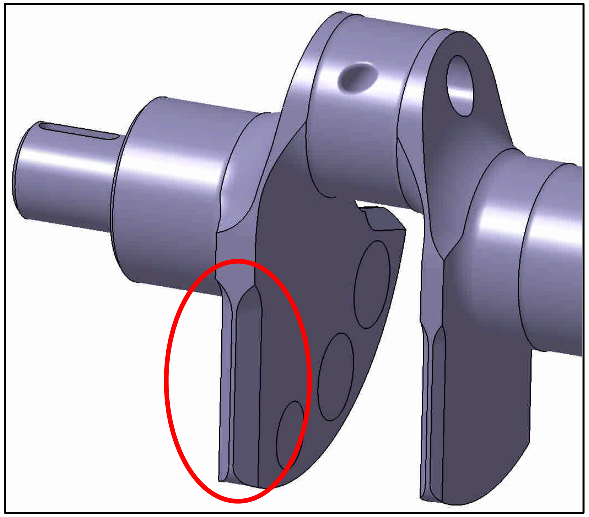


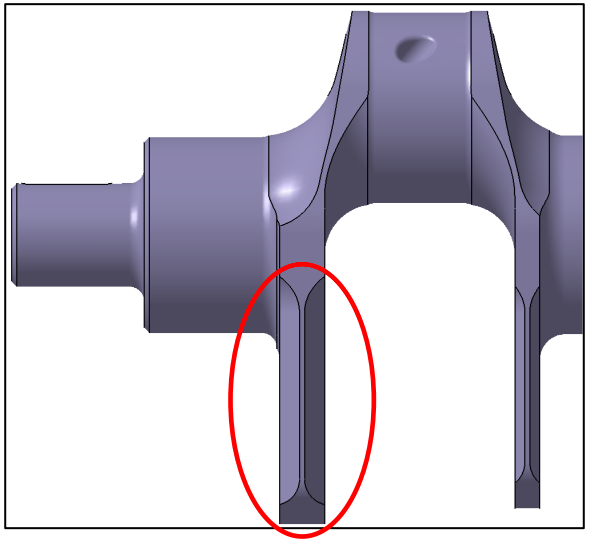


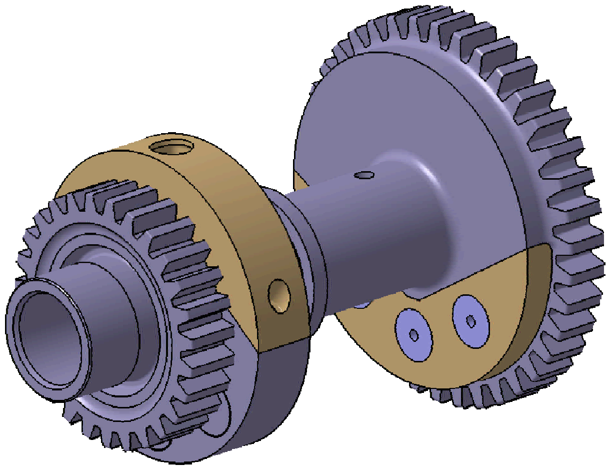


Minimising Friction Losses

- Windage loss reduction
  - Thinner webs with chamfered edges
  - Shrouded balance shaft
- Bearing friction loss reduction
  - Journal diameters were not changed due to cost and lead time implication

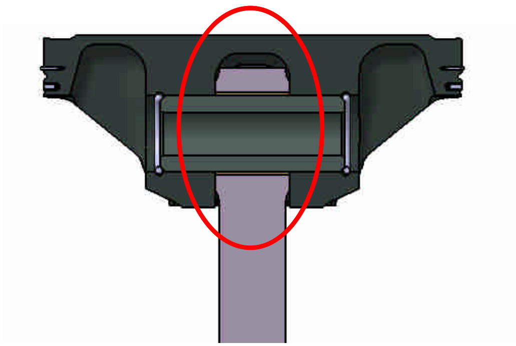


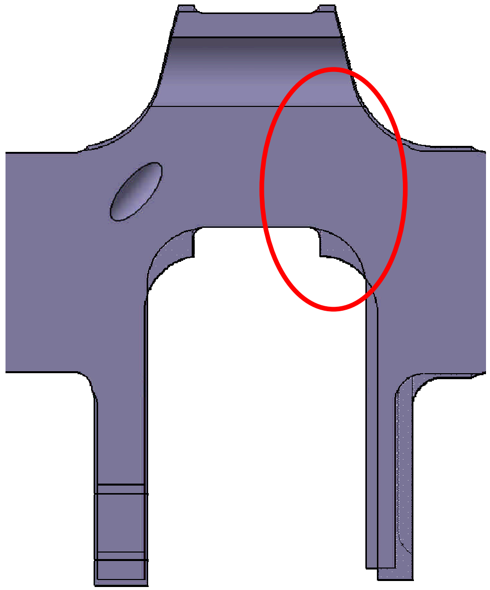


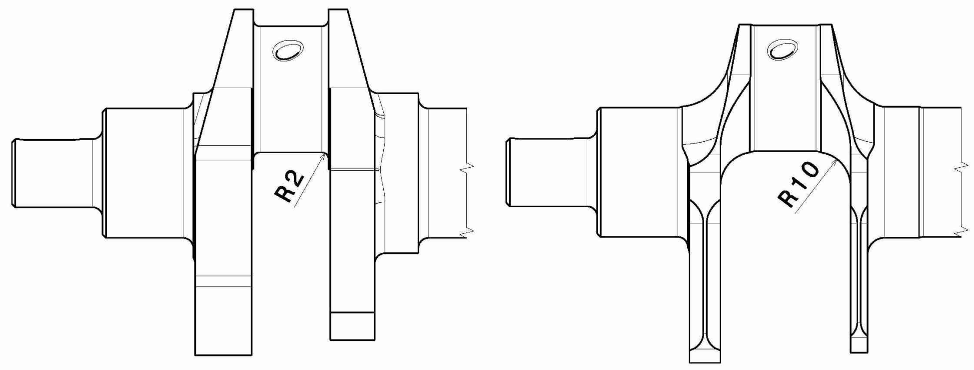


Increasing Strength

- Final design had 'piston guided rod'
- This eliminated the need for thrust faces on big end journals
- Thus permitting use of a large fillet radius in the critical area of crankshaft overlap region

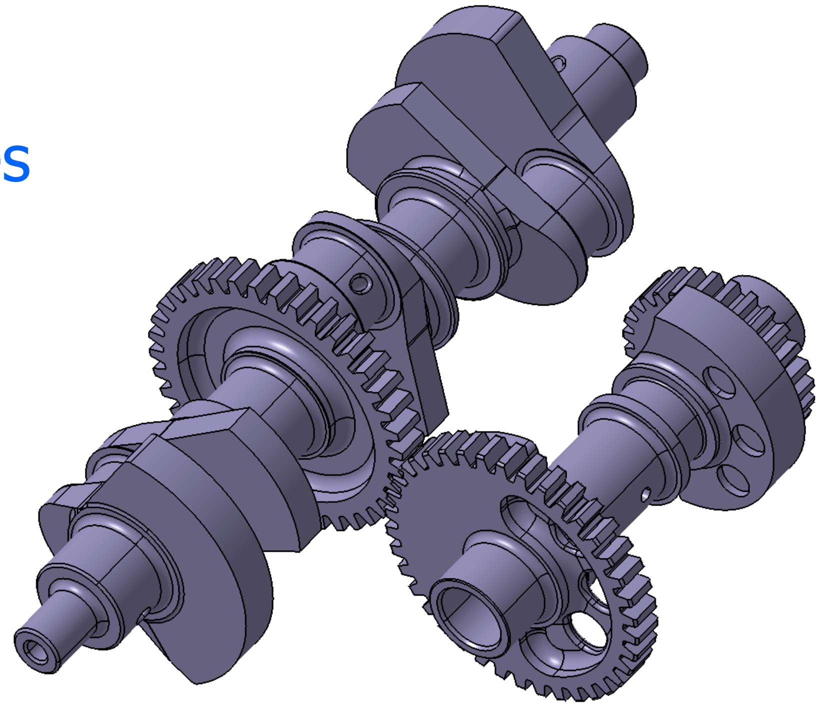




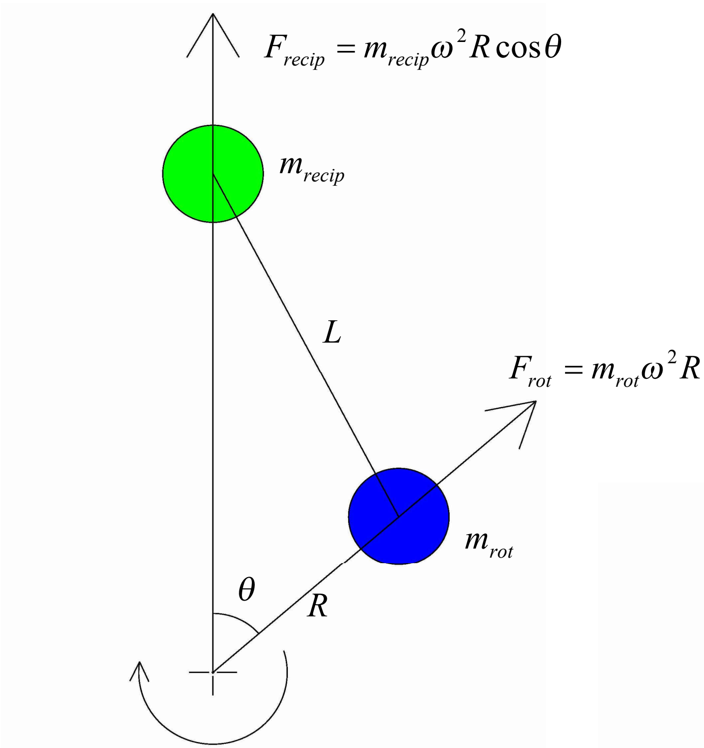


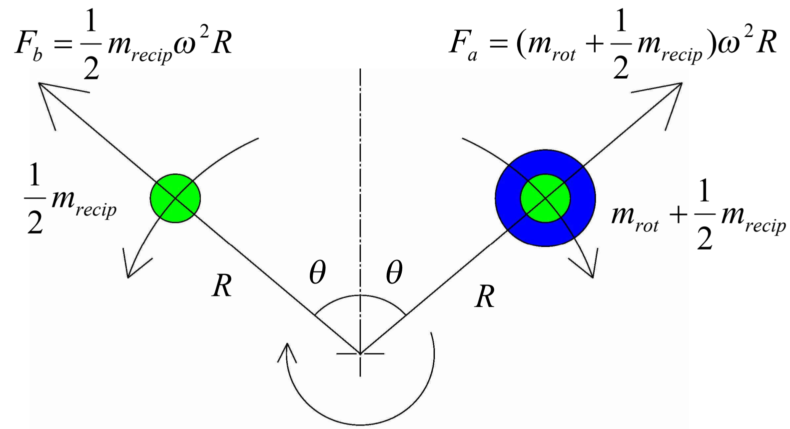
In-Line 3 Cylinder Engine Balance

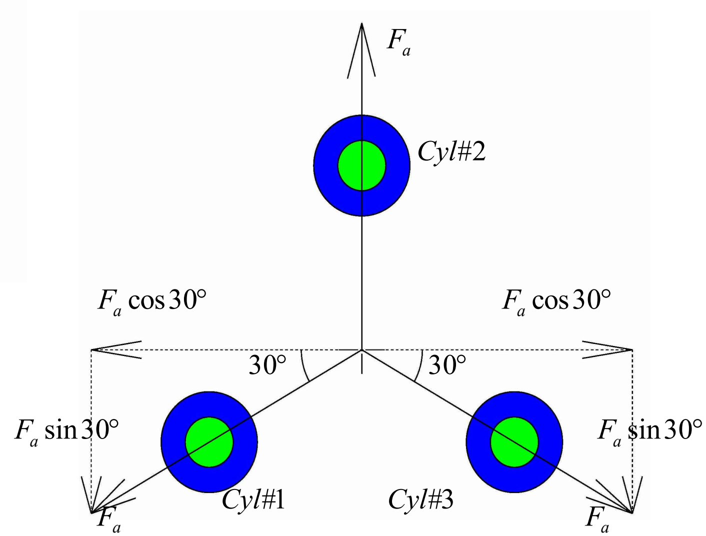
- In-line 3 cylinder engine has
  - Balanced primary and secondary reciprocating forces
  - Unbalanced primary and secondary reciprocating moments
- Baseline FP1 engine had crank counterweights and balancer shaft arranged to give complete balance of primary moment
- But is complete moment balance necessary ?

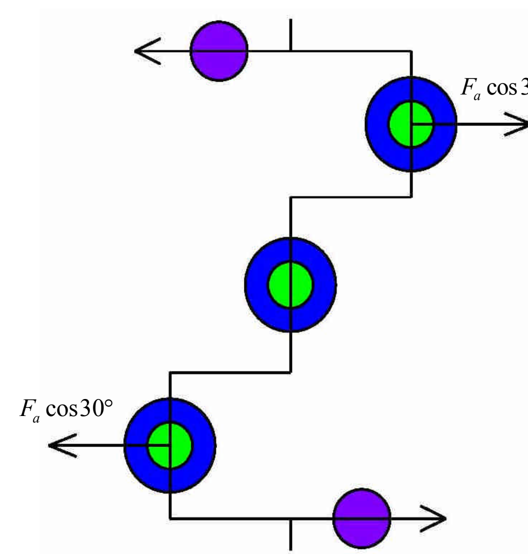


In-Line 3 Cylinder Engine Balance





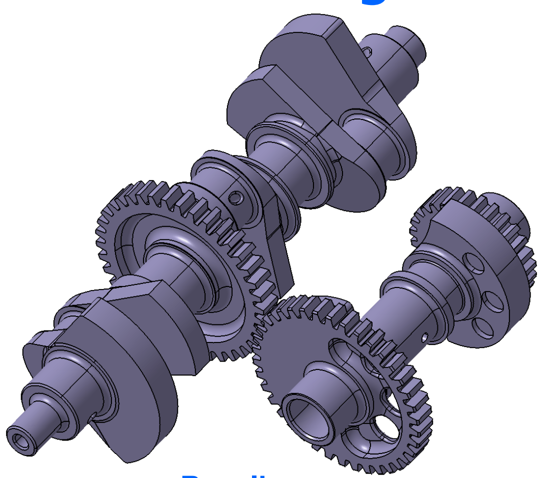




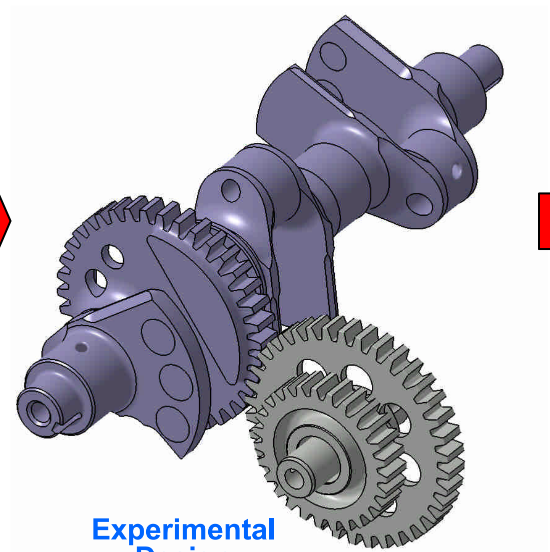
Crank/Balancer Design Iterations

- Removing the counterweights on the balance shaft was tried
- Level of vibration was acceptable to riders but a frame failure occurred
- A compromise was adopted for the final design

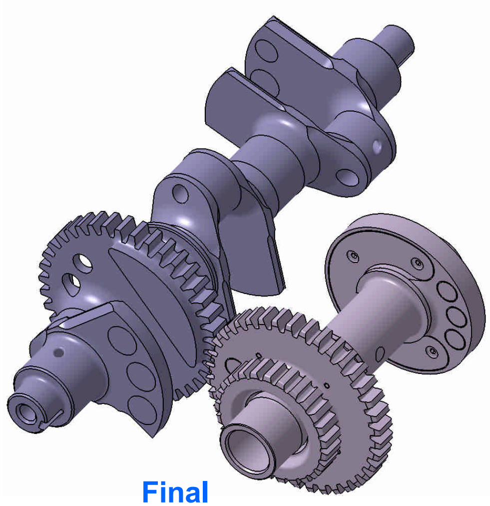
Component	Baseline Design	Final Design
F_a force balance factor	50%	30%
Primary forces	100%	100%
Primary moment, M_a	100%	90%
Primary moment, M_b	100%	60%



Baseline Design



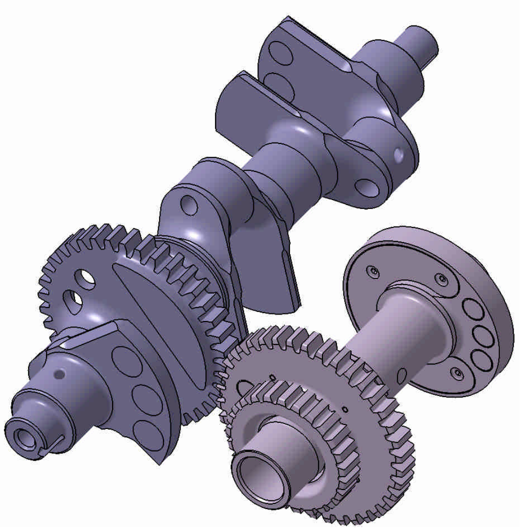
Experimental Design



Final Design

Residual out-of-balance moments

- Numerical values of residual out-of-balance moments are shown
  - For engine with no counterweights
  - For baseline design
  - For final design

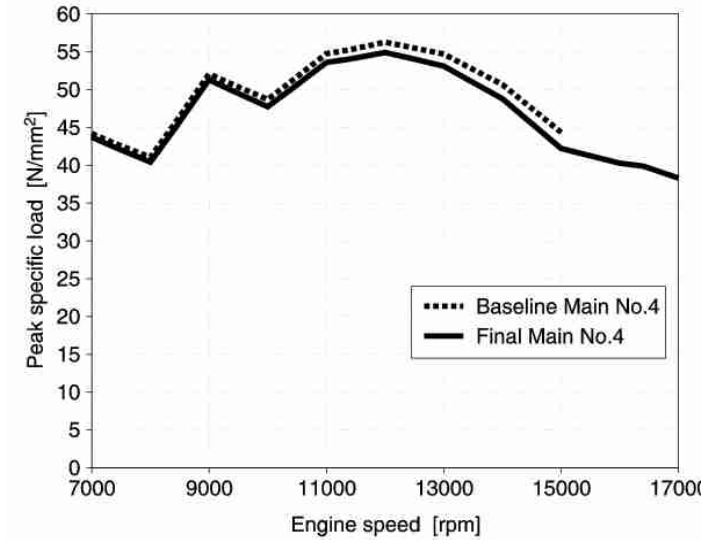


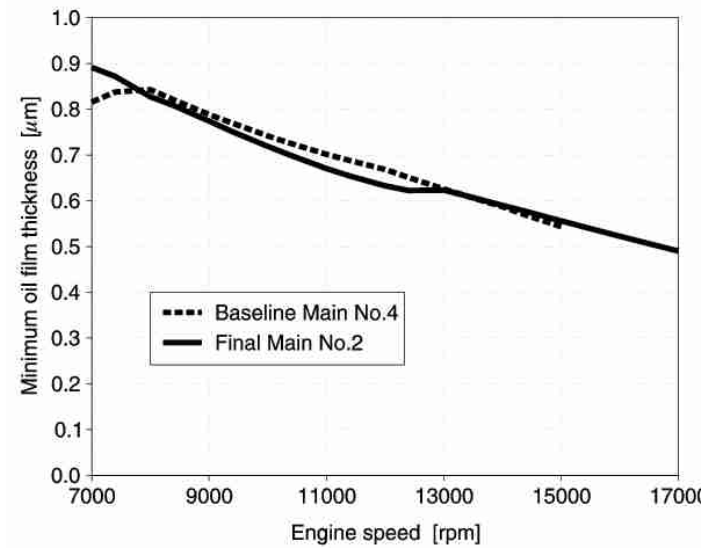
Parameter	No c/w	Baseline	Final
Primary shaking moment – pitch at 14000 rpm (Nm)	9158	0	1031
Primary shaking moment – pitch at 16000 rpm (Nm)	-	-	1345
Secondary shaking moment – pitch at 14000 rpm (Nm)	803	803	683
Secondary shaking moment – pitch at 16000 rpm (Nm)	-	-	892
Primary shaking moment – yaw at 14000 rpm (Nm)	5816	0	215
Primary shaking moment – yaw at 16000 rpm (Nm)	-	-	281

Main bearing analysis

- 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

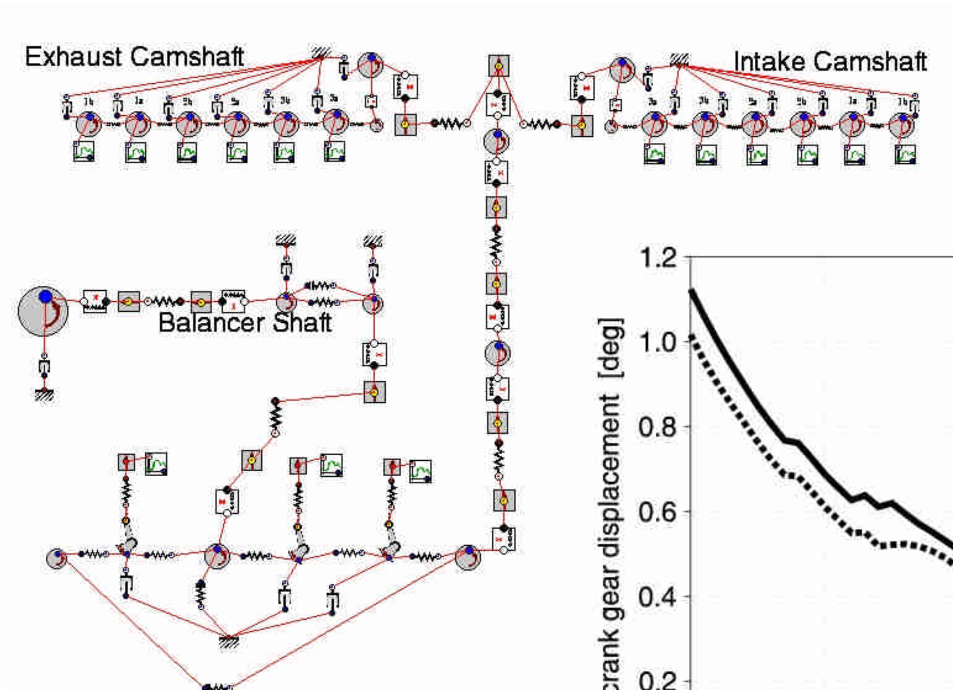
Parameter	Baseline	Final
Maximum peak specific main bearing load (N/mm^2)	56.3 @ Main No.4 12000 rpm	54.9 @ Main No.4 12000 rpm
Minimum oil film thickness (μm)	0.59 @ Main No.4 14000 rpm	0.53 @ Main No.2 16000 rpm
Maximum predicted oil temperature (°C)	159.3 @ 14000 rpm	160.2 @ 14000 rpm 165.5 @ 16000 rpm
Total power loss at all main bearings (kW)	3.022 @ 14000 rpm	3.144 @ 14000 rpm 4.001 @ 16000 rpm

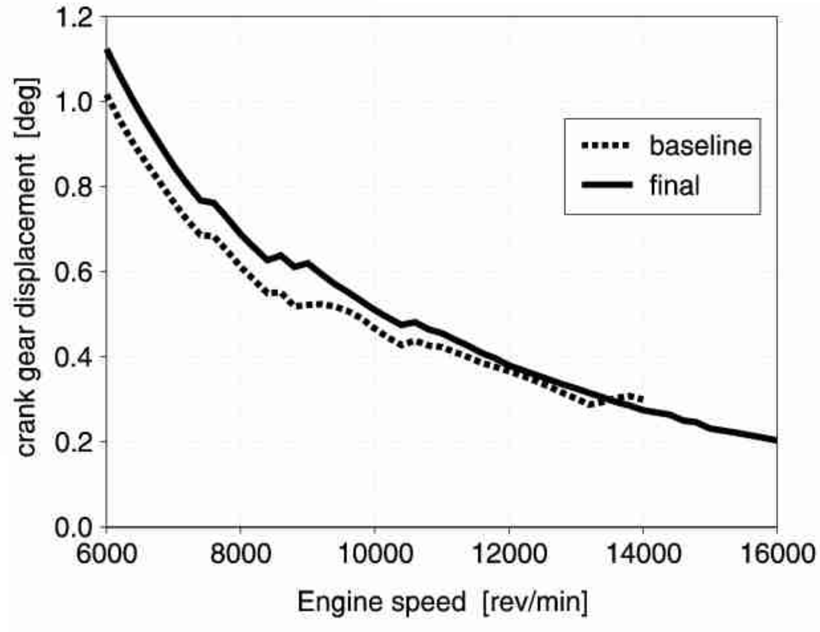




Torsional vibration analysis

- VALDYN linear frequency domain analysis
- Reduction in inertia results in more crank motion at low speed

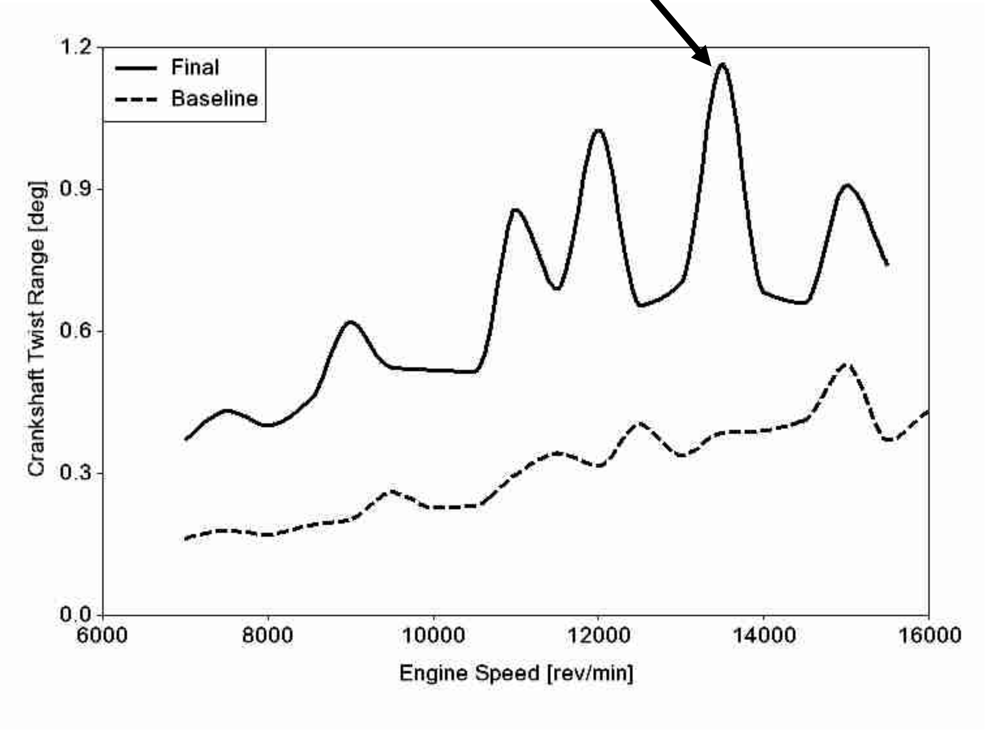




Torsional vibration analysis

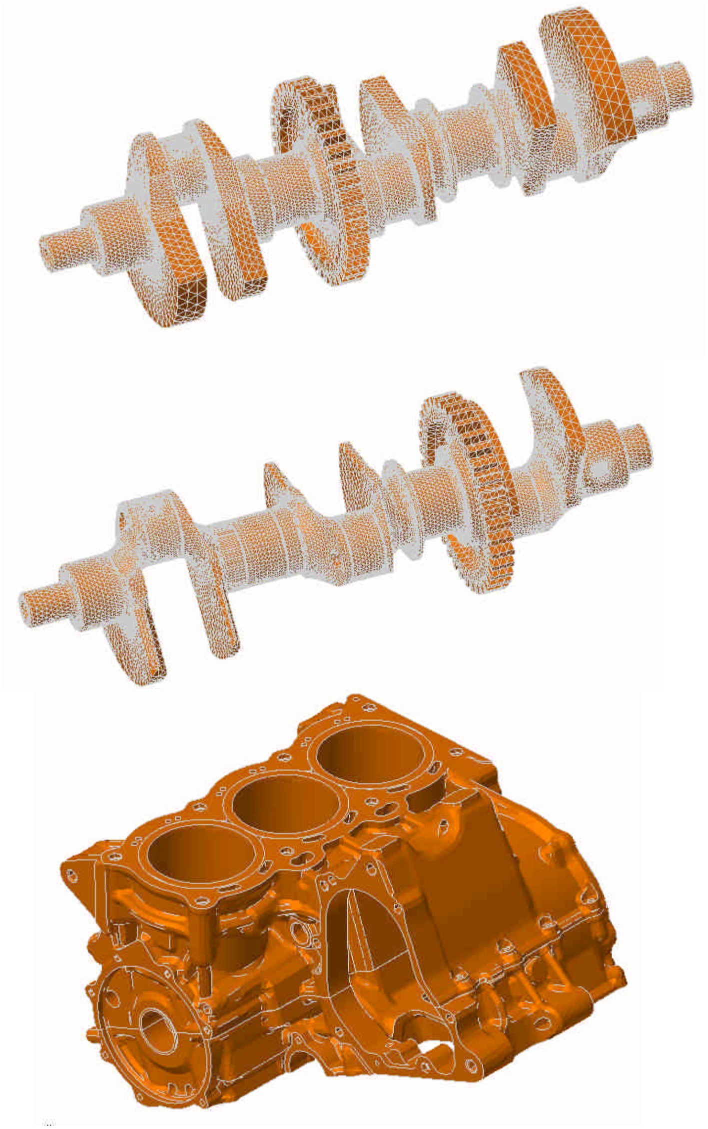
- ENGDYN 3D crankshaft dynamics analysis shows significant increase in crankshaft twist for final design
- Baseline crank natural frequency of 1317 Hz
- Final crank natural frequency of 971 Hz

4.5 order peak



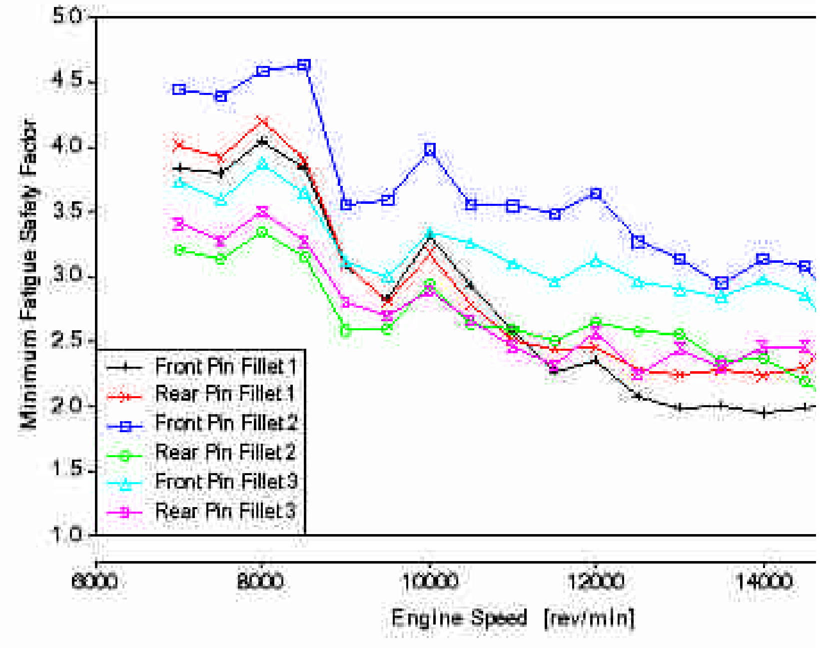
Stress analysis

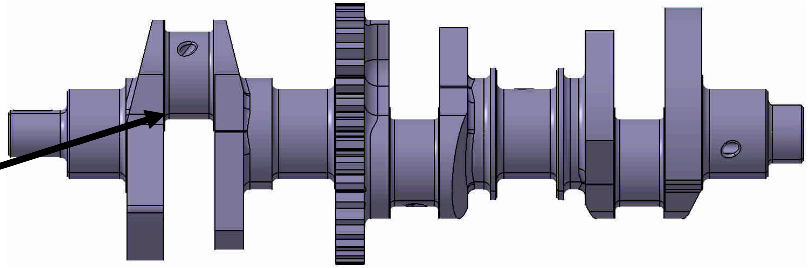
- Finite element analysis was performed on the baseline and final crankshafts
- ENGDYN used to
  - Calculate boundary conditions
  - Combine FE models
  - Solve equations of motion
  - Calculate combined stresses at 5 degree intervals for each engine speed
  - Calculate Goodman safety factors at fillets and oil holes



Stress analysis

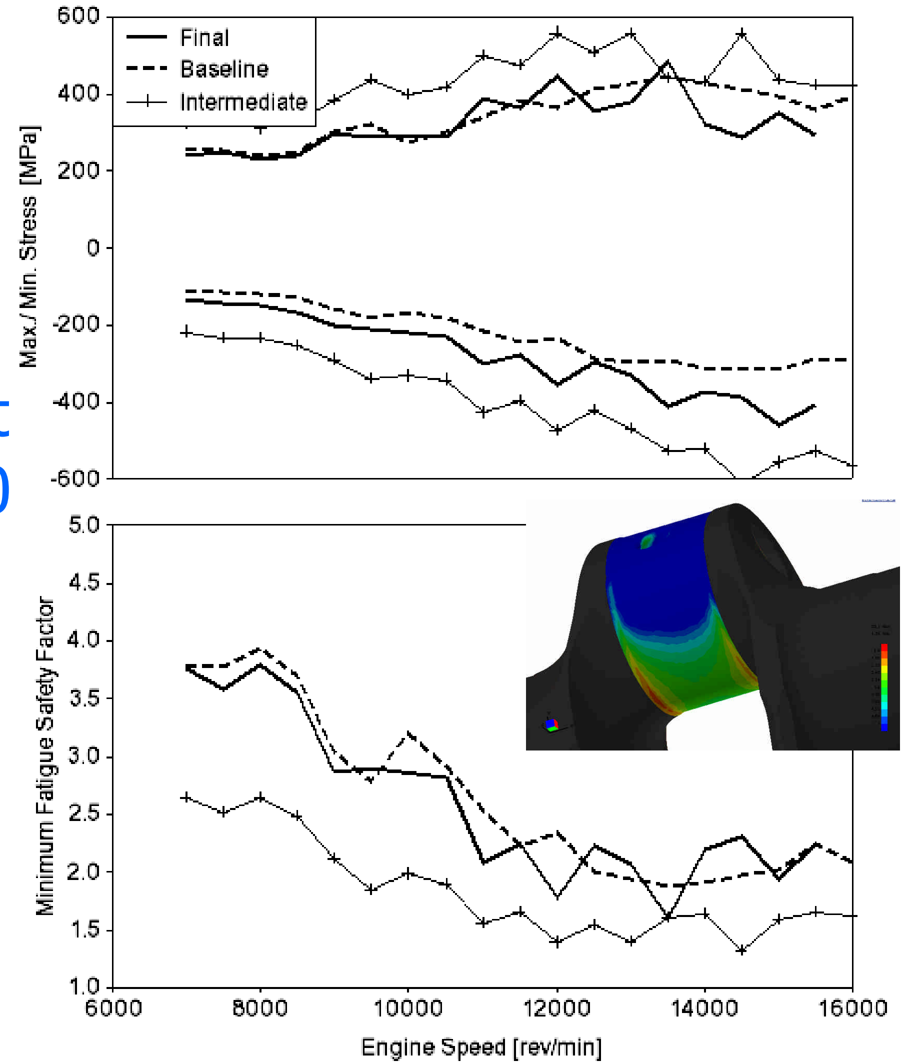
- UTS 1050 N/mm^2
- Yield strength 900 N/mm^2
- Fatigue strength estimated accounting for influence of nitriding and size effect
  - At pin fillets 745 N/mm^2
  - At main fillets 747 N/mm^2
  - At pin oil holes 745 N/mm^2
- Baseline results indicate that lowest safety factor occurred at crank pin fillet on web No.1





Stress analysis

- Results compared for pin fillet at web No. 1
- Lower safety factor for intermediate design
- Lowest value for final design at 4.5 order resonance at ~13500 rpm
- Intermediate crank design did fail at pin fillet on Web No.1



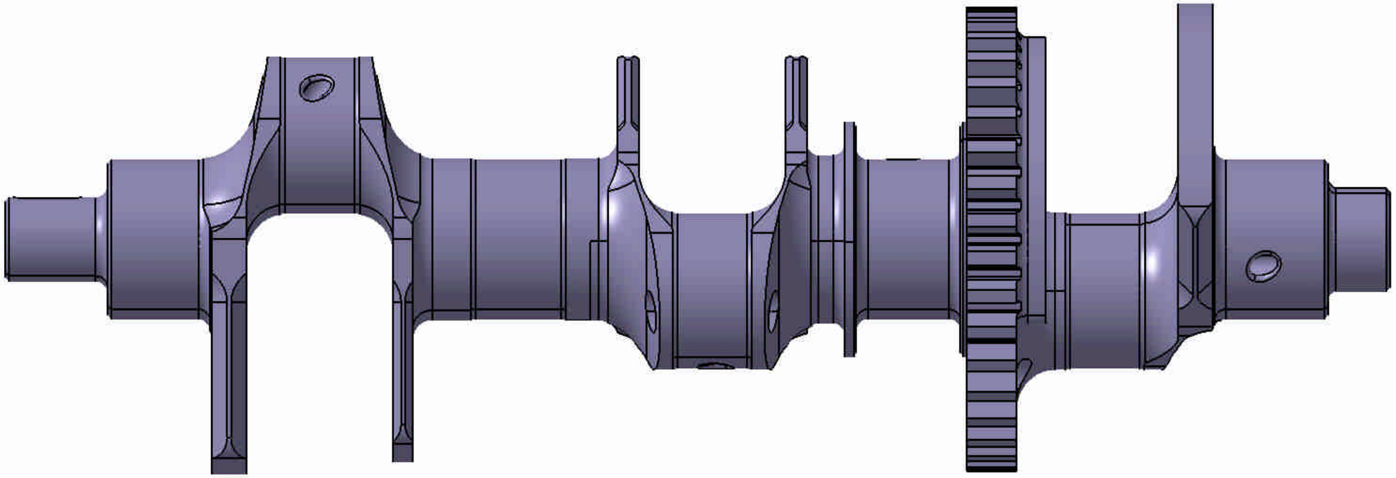
Practical experience

- Track testing was performed with engines having various degrees of unbalance
  - Riders preferred low inertia of final design
  - Riders were prepared to tolerate increased vibration
- Crankshaft was very durable
  - No failures of baseline or final design on test or during racing
  - Crankshaft was usually replaced after 4 million cycles
    - some baseline cranks experienced 7 million cycles
    - some final cranks experienced 6 million cycles
  - No significant wear of main bearings

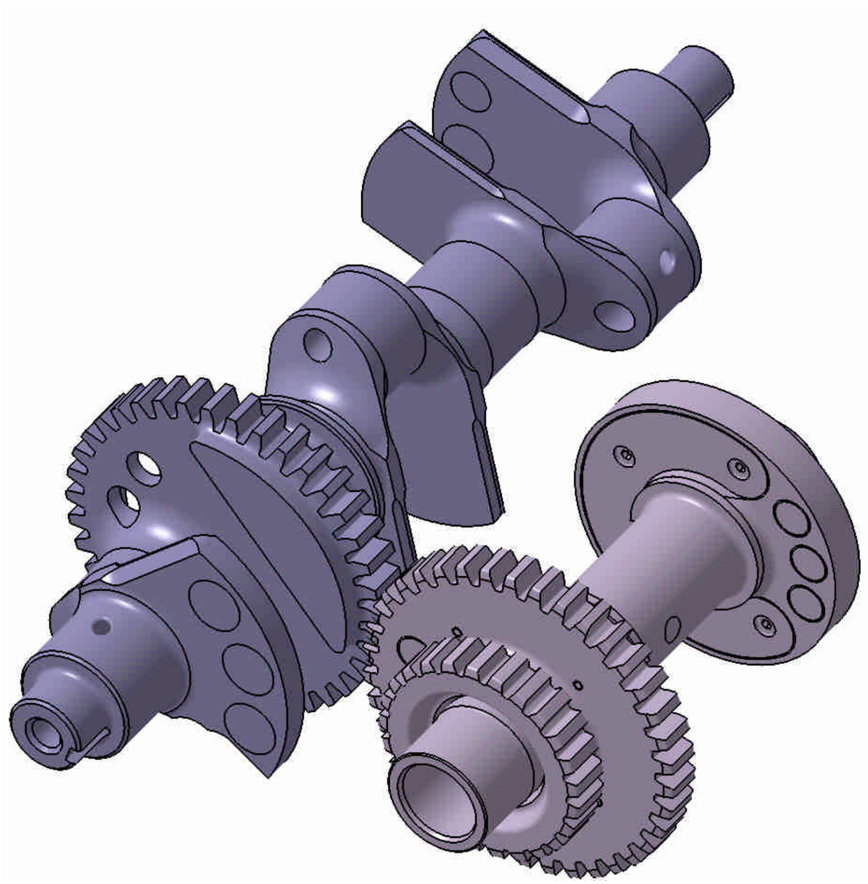
Conclusions

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
- had partially balanced primary reciprocating moment
- was guided by analysis using Ricardo Software



Thank you for your attention



Any questions ?