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Software&CAE

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Racing to 16,000 rpm

When rule changes allowed some teams in the World Superbike series to use larger engines, PETRONAS needed to uprate its 900 cc FP1 unit to stay competitive. This ambitious challenge was tackled using Ricardo's CAE analysis tools, allowing the in-line triple ultimately to rev safely to beyond 16,000 rpm. Jesse Crosse reviews the work first revealed at this year's SAE Congress

PETRONAS had been competing in the World Superbike series since 2003, but a set of changes to the regulations for the 2003 season enabled some competitors to use 1000cc in-line 4 cylinder engines. PETRONAS needed to think fast if its smaller 900 cc unit was to remain competitive, and approached Ricardo. This exciting project, aimed at turning a compact racing motorcycle engine into a giant killer, was presented in the form of three papers at this year's SAE Congress.

Rather than go down the expensive route of homologating a new engine, the PETRONAS plan was to improve both the power output and the width of the power band by increasing engine speed. Although the regulations did not allow an increase in capacity of an existing engine, there was no limit on maximum operating speed: this, it was decided, would prove to be an elegant solution.

An increase of 2000 rev/min to 16,000 rev/min would deliver the required power but the valvetrain was identified as being a limiting factor. The crankshaft design would also need consideration, as would the gas flow both above and below the pistons. By far the most challenging of these three would be the valvetrain design. Regulations forbid the modification of major castings, so it would not be possible to change the design from direct-acting to the finger pad follower favoured by most F1 engine designers, a solution which can potentially reduce effective mass. Neither could the team introduce the pneumatic springs also prevalent in F1 engines. The only course of action left open by

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Design & analysis of components of a high speed motorcycle engine

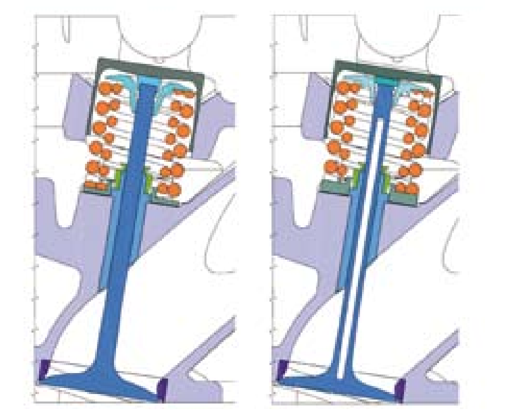
Due to the requirement for extreme confidentiality it is rare that Ricardo is able to publicise details of its motorsport work. At the 2007 SAE World Congress however, three papers were presented which documented in detail some of the CAE work which was carried out by Ricardo in support of the up-rating of the PETRONAS 900cc FP1 engine in order to enable it to compete with larger displacement competitors. Given the level of interest in this area of CAE application we are devoting this issue of Software & CAE to a feature describing the content of these highly interesting technical papers.

the regulations was to reduce the mass of the moving parts to the minimum required to deliver sufficient durability. Ricardo used its VALDYN software to analyse the valvetrain dynamics and the design of the cam profile.

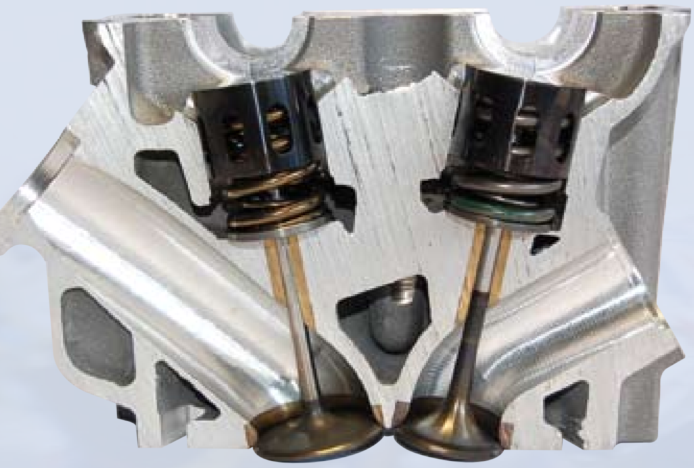
Inlet valvetrain: the biggest challenge

The software includes a kinematics solver used to design cam profiles and calculate pseudostatic forces, oil film thicknesses and other factors. It also includes a dynamics solver to determine dynamic valve motion, dynamic forces and spring surge vibration. By far the biggest challenge lay with the inlet valvetrain due to the larger valve and more aggressive, higher-lift cam profile. A compromise was needed between valve lift and period for higher speed operation, and simulation using WAVE played a big part in testing a series of cam profiles before proceeding to engine testing.
In the final design, the peak lift was reduced by 1 mm and the peak period increased slightly. Ramps are required on the ends of the cam profile to control the speed at which the valves open and close. The kinematic ramp velocity remained unchanged despite the increase in engine speed and an increase in actual ramp velocity. Peak kinematic valve accelerations were decreased by between 13 per cent and 16 per cent while dynamic peak acceleration values increased by between 12 per cent and 15 per cent due to the increased engine speed.

Baseline and final intake valve designs



The durability of the contact point between cam and tappet is an important area. The team assessed this in terms of contact stress, lubricant film thickness and the proximity of the contact line between cam and tappet, to the edge of the tappet. Analysis showed that the highest stress occurred at low engine speeds, but the final design incorporated a larger nose radius on the cam lobe to reduce this. This did mean, however, that contact stress at high speed was increased slightly compared to baseline due to higher spring force and lower valve train mass.



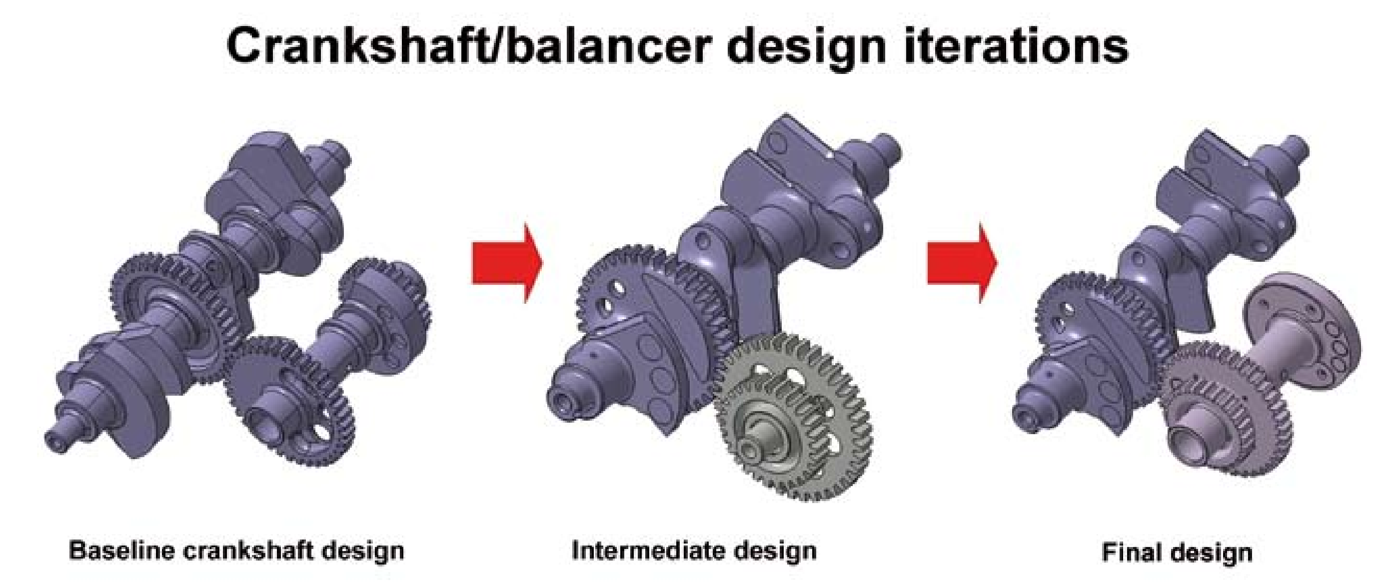
Section through cylinder head showing intake and exhaust valves

Critical lubricant film thickness

The cam/tappet stress over the nose was also calculated using dynamic forces and the lubricant film thickness plotted against the crank angle. The analysis also demonstrated that film thickness is high on the flanks and low over the nose and on the transition between the two, the film thickness passes through a low region for just under 10 degrees. On the baseline cam (with high lift) the tappet edge clearance was very low at just 0.3 mm. But as the lift reduced, this increased to a higher value than necessary. The team considered reducing the tappet diameter but it was decided that the potential benefits of doing so would be small because the tappet size was limited by the need to pass over the valve springs.
The Ricardo VALDYN product has from the outset been focused on the needs of valvetrain dynamic analysis - something the team was able to usefully exploit during this project.
There are many ways in which a valve train can be represented using VALDYN but the following breakdown of mass and stiffness was found to acceptable results even for the very high speed valve train. Firstly, a cam node was suspended on a stiffness element representing the camshaft bending stiffness and camshaft bearing support stiffness. The tappet top stiffness was then modelled as a function of eccentricity of the cam/tappet contact and the valve stem was generally modelled as a single stiffness. The valve, spring retainer, shim and collets were modelled as a single, lumped mass. Each valve spring was modelled as a series of lumped masses (eight per coil) connected by stiffnesses using a special macro element that accounts for coil clash effects as the spring closes, as well as loss of contact between spring ends and mating parts.
The level of damping due to interference between valve springs is dependent on the fit between the springs (which changes as the springs are compressed) and so is very difficult to model explicitly. For this project the approach taken was to make two analysis runs to assess each design; one with high damping (assumed 20 per cent of critical damping) and one with very low damping typical of independent springs with no interference (0.5 per cent of critical damping). The sensitivity of all aspects of system dynamics to damping was therefore considered at every stage.
The finished model was used to calculate the dynamic response of the valvetrain in the high speed range and the results used mainly to plot critical parameters against engine speed. A good example was valve seating velocity. At low speeds seating velocity was controlled by the closing ramp on the cam but as engine speed increased, control deteriorated, resulting in a sharp transition to very high seating velocity at around 14,800 rev/min.

Robustness for in-race over-revving

However, engine and rig testing revealed that the titanium intake valves being used would fail suddenly at the valve stem just below the retainer should the seating velocity exceed 4 m/s. This information was used to set the engine speed limiter for each valvetrain build.
In the final build, the valve seating velocity did not exceed 2 m/s below 16,500 rev/min and there was no sharp transition to a very high seating velocity below 17,000 rev/min. As a

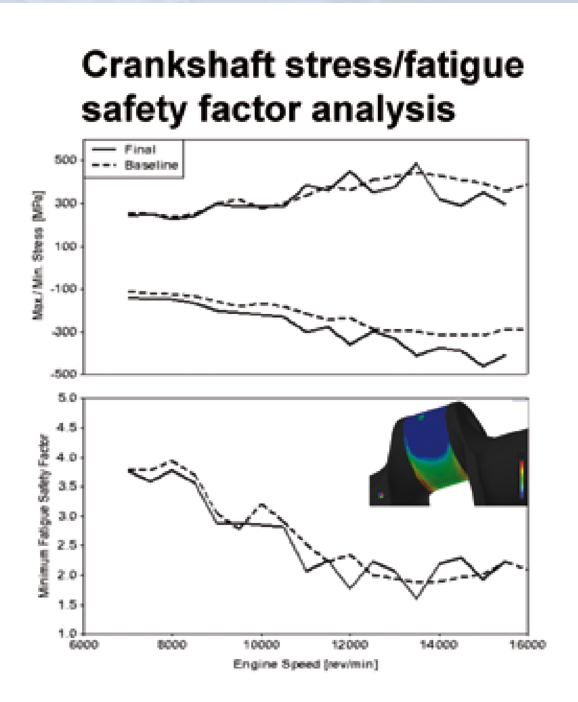


result, robustness in the face of those inevitable over-rev events in racing was greatly improved. But interestingly, valve seating velocity was not especially dependent on the assumed damping effect due to interference between the inner and outer springs.

Dynamic analysis also revealed useful data on the phenomenon of valve bounce and several other issues relating to high speed valvetrains. On the baseline engine the valve would hit the seat before the top of the ramp was reached. Seating velocity was so high that at 15,000 rev/min initial contact between the valve and the seat would be followed by a bounce of 0.44 mm. The team considered a bounce in excess of 0.1 mm unacceptable and were able to reduce bounce to just over 0.05 mm at 16,000 rev/min in the final design.

Valve jump, or loss of contact between the cam and tappet, happens when inertia force exceeds spring force. Again, the baseline design exhibited a significant loss of contact between cam and tappet at only 14,500 rev/min but in the final design separation did not exceed 0.2 mm at speeds below 17,000 rev/min.

Spring surge is another well known phenomenon associated with the high-speed coil-sprung valvetrain: it arises where the spring is excited by the harmonic content of the cam profile. When this occurs, vibration may continue following valve closure, possibly to the extent of affecting motion during the next valve event. It also causes spring seat hammering as the spring literally bounces on its seat, generating large impact forces. This problem manifests itself with broken spring end coils .



The software was also used to calculate dynamic stress in the springs. Although the spring fatigue strength was not known, the effect of valvetrain dynamics on spring stress was quantified, something that proved invaluable as a basis for comparison between designs.

In addition to simulation of individual details of the valvetrain, an analysis model of the whole engine was created. This enabled the team to investigate the effects of crankshaft dynamics and timing drive dynamics on valve motions, dynamic loads on timing gears and dynamic torques at gear fasteners.

Crankshaft design

The team also needed to address the crankshaft design if the engine was to rev safely to 16,000 rev/min; accordingly, a number of objectives were determined. Mass would be reduced, mainly to lower the overall mass of the bike. Rotating inertia would also be reduced in order to improve engine response and acceleration. Friction-related power loss through the bearings would be addressed and it would be important to minimise crankshaft windage loss.

Adequate crankshaft strength and bearing durability would clearly need to be retained at the higher engine speed and the engine should be well balanced. That said, Ricardo engineers first questioned whether the degree of balancing on the baseline engine was strictly necessary given that the counterweight arrangements were compromising the need to reduce mass and inertia. Several iterations later, the crankshaft and balancer shaft counterweight arrangements were substantially reviewed.

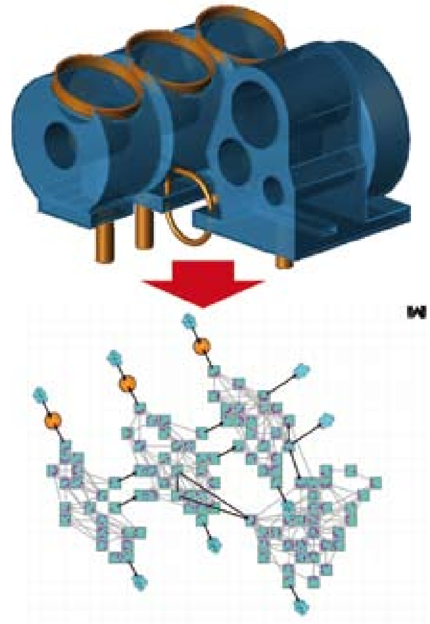
Performance of the main bearings was analysed using Ricardo's ENGDYN software, a program whose wide-ranging capacity for analysis of bearings, crankshaft strength and dynamics, and crankcase loading makes it eminently suitable for the task. Although ENGDYN offers a hierarchy of crankshaft and bearing analysis methods, comparative analysis of bearings on this project was achieved using the most simple method. Using these techniques, peak main bearing loads were reduced by 2.5 per cent, although minimum oil film thickness was

10 per cent lower at 16,000 rev/min than for the baseline engine at 14,000 rev/min. There was a small increase in maximum predicted oil temperature and small increase in predicted power loss at all main bearings at 14,000 rev/min.

Torsional vibration analysis

VALDYN was also used in the design of the crankshaft to analyse torsional vibration: this was done by developing a model to perform a linear frequency domain analysis. The model was used to calculate the rotational displacement of the crank gear relative to a node rotating at constant speed. Crankshaft twist amplitude relative to engine speed was predicted using ENGDYN software. The final design had a lower natural frequency than the baseline engine and so the twist range of the final design was increased at higher engine speeds. This also resulted in a corresponding increase in peak vibration torques.

Both the baseline and final versions of the crankshaft underwent finite element (FE) stress analysis to predict stresses under full load. These were subsequently used to calculate safety factors through the engine cycle across a sweep of speeds. FE models were also created of the cylinder block assembly, including the block itself, bedplate and a stiffness representation of the cylinder head. ENGDYN was used to calculate boundary conditions, combine the FE models and solve the equations of motion within the system. Neither the baseline crank nor the final design failed during testing – an intermediate design did, but only after the failure had been predicted by simulation.



WAVEBuild was used to construct the detailed WAVE crankcase flow model

Final design – liked by riders

The final design proved to have exceptional durability at the rated speed of 16,000 rev/min and could survive over-speeding to as much as 17,000 rev/min. The crank was not fully balanced but the trade-off was the low inertia (much preferred by riders) of the final design, whose success was due mainly to the extensive use of ENGDYN.

In order to fully realise the benefits of the modified valvetrain and crankshaft, it was also important to minimise pumping losses and aeration in the crankcase. Ricardo's WAVE software was used to model the crankcase gas flow during the design phase of a dry sump version of the FP1 racing engine. Significant losses can be incurred by gas exchange between crankcase bays, through external breathers, between cylinder and crankcase volumes and in the form of heat transfer to crankcase walls. Interaction of components with crankcase gas and oil also causes losses and high velocity gas flows also cause oil aeration. Following extensive analysis, the team was able to show a potential reduction of 4.7 kW in crankcase pumping losses using a dry sump system. Key factors affecting crankcase pumping mean effective pressure (CPMEP) include breather size and discharge coefficient, engine displacement, crankcase compression ratio and scavenge flow rate.

Clearly, Ricardo software packages such as VALDYN, VALKIN, ENGDYN and WAVE played a key role in the transformation of the PETRONAS FP1 racing engine, meeting all performance targets and avoiding the high cost of homologating and all-new, larger capacity engine.

For further details of the work described in this article, see:

1) *SAE Paper 2007-01-0264: Design and Development of the Valve Train for a Racing Motorcycle Engine Phil Carden and Ken Pendlebury, Ricardo UK, Naji Zuhdi, PETRONAS Malaysia, and Andrew J G Whitehead, Del West USA*

2) *SAE Paper 2007-01-0265: Design and Analysis of a Lightweight Crankshaft for a Racing Motorcycle Engine*
Naji Zuhdi, PETRONAS Malaysia, and Carden and David Bell, Ricardo UK

3) *SAE Paper 2007-01-0266: Crankcase Flow Modeling for a Racing Motorcycle Engine*
Thomas Deighan, Ricardo UK, and Naji Zuhdi, PETRONAS Malaysia ■

SABR 1.1 released

Ricardo Software is pleased to announce the release of version 1.1 of its gear and transmission design package SABR (for a description of function and applications of this product see Software&CAE Q1, 2006). This new release builds on the capabilities of the package and includes a number of significant enhancements to both the main SABR package and the GEAR module.

With version 1.1 of SABR the Koyo bearing has now been integrated into the program's database allowing users to select from over 3000 bearings. A wide range of improvements have also been made to the user interface, for example allowing models to be manipulated and viewed more easily and also improving the presentation and analysis of results. The many functional improvements include enabling the simulation of designs of increased complexity and load cases, and also allowing the solution of sub-sets of shafts and load cases.

For further information contact the support team at: rs_support@ricardo.com.