Use of full engine and detailed valvetrain models to improve valve control in a high-speed racing engine.

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Abstract
Valve control is critical to the performance of high speed racing engines. Even slight errors in valve timing can lead to significant power loss and deviations from the design valve motion may affect engine life.

Traditionally, valve-train designs have been optimised using simple rigid body models. At high loads, however, component flexibility leads to discrepancies between predicted and measured results. These can arise either from within each single valvetrain or from torsional effects in the full engine system.

This paper describes how full engine system and detailed single valvetrain models have been used in the process of developing a high-speed racing engine. Results from the models have shown how valvetrain dynamics is affected by component flexibility and have given solutions to durability problems.

Introduction
This paper follows an investigation during the development of a high-speed V10 racing engine. Special measurement techniques had shown that the inlet valves were closing significantly earlier than intended. Furthermore, the results had shown that there was also a measurable difference between the left and right hand banks.

Experience from running cycle simulation models had shown that the shape and timing of valve lift curves was critical to engine power. It was therefore important to ensure that the valve motion was as close as possible to design. ADAMS/Engine models were used to investigate the probable causes of this problem and solutions were proposed and tested.

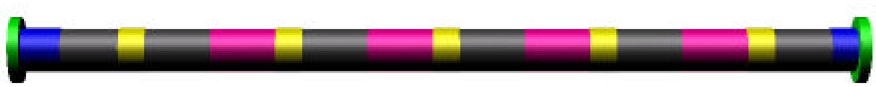
A full engine torsional model was used to investigate possible torsional effects through the whole system. Changes were made to the drive layout and corresponding improvements to valve control were predicted. The full torsional model also predicted the bank-to-bank differences measured on test.

A detailed single valvetrain model was used to investigate the effect of valve flexibility on the valve lift curve. The high accelerations in the valve lift curve had significantly affected the motion at the valve base. Significant valve bend was also evident in the results that may have been a contributing factor to valve seat wear seen on test.

Full System Model Description

The full engine model contained enough detail to fully predict any torsional effects that would alter the timing and shape of the valve lift curves. To this end the model contained the following refinements.

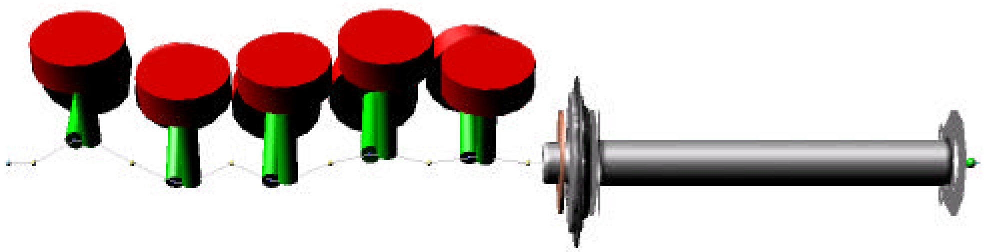
Flexible Camshafts



The camshafts were divided into many hollow cylindrical sections, connected with revolute joints and torsion springs. To keep run times low, bending effects were ignored. A representative level of torsional damping was applied to each camshaft bearing to simulate hydrodynamic losses.

The geometry and stiffness of each section was parameterised so that different diameters, engine configurations, and firing orders could be analysed with the same template. This level of parameterisation allows the same camshaft template to be used for both inlet and exhaust shafts.

Flexible Crankshaft and Drive Arrangement

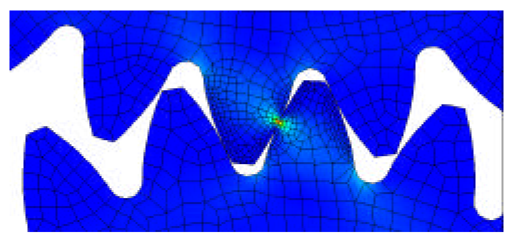


Both the crankshaft and dyno driveshaft were modelled as flex bodies. Previous analyses had shown that crankshaft torsion is sensitive to model detail within the drive arrangement. This model was created before the age of A/Engine Cranktrain, so the valvetrain interface had to be modified to create cranktrain subsystems. Faster run times may now be possible using the FEV beam crank.

Representative damping levels were applied to the main bearing, big end and piston-liner constraints. An FE analysis of the crankcase assembly was used to generate a realistic main bearing stiffness. A flexible block was not modelled as this would have slowed run times down significantly and was not relevant to the analysis.

The pistons and conrods were rigid bodies, but defined parametrically so as to investigate different configurations and vee angles

Gear Drive



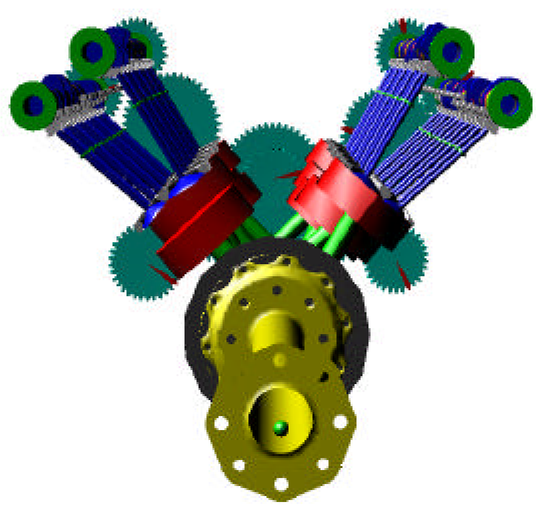
All Gears were modelled with tooth flexibility and backlash. A non-linear FE analysis was used to calculate a mean tooth stiffness value. This was found from measuring the deformation at contact points along the full sweep of the gear tooth.

The vibration isolation/compound gears were modelled with a torsional stiffness from mechanical test data. The damping produced by these gears was modelled with stick-slip friction derived again from test data.

Single Valvetrains

Initially, a 3D spline of cam torque vs angle and speed was created from running a simple single valvetrain model. This produced the correct cam stab torque for different speeds, but not for different accelerations. The resulting camshaft torsional vibrations were then much higher than those created using a comparable model with single valvetrains. A simple single valvetrain model was therefore created that had the minimum model detail possible so as to reduce run times.

General Running



The model was built up to the full engine by testing out the cranktrain, valvetrain, and gear drive assemblies separately. Getting the full assembly to solve required several refinements.

A number of perpendicular joints were added to constrain the motion of all the flexible shafts during static equilibrium. These were then de-activated after a small period of dynamic analysis. This allowed oscillations in the gear drive to settle slightly before any torsional vibration occurred in any of the other shafts.

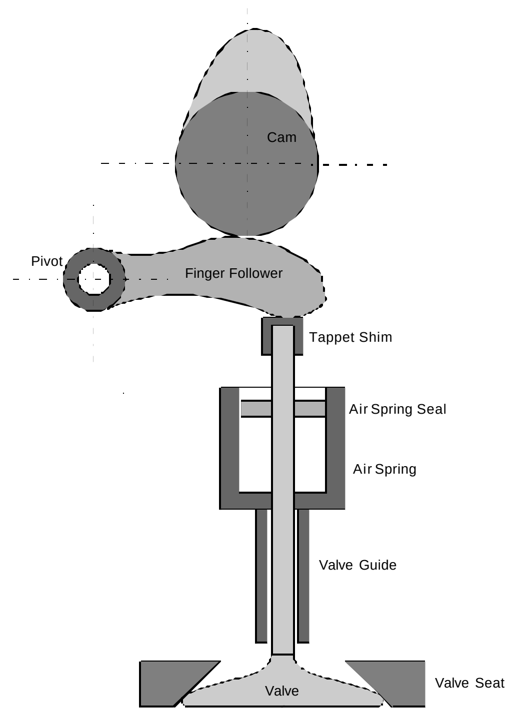
All cams were constrained so that all the valves were on their seats at the start of the analysis. Imposed motions then moved the cams into their required position within the first camshaft revolution (thanks to Christoph for this one). This was required as ADAMS was having difficulty finding static equilibrium when some of the valves were at maximum lift (>15mm worth of movement).

The damping ratio for the flexible shafts was ramped from a high level at the start to the required level for the analysis over several revolutions. The time period for each shaft was slightly different so that they did not all overlay simultaneously. This produced an efficient way to get the system to settle out within the smallest number of revolutions (10 revs approx).

The model still has some detail missing. Most notable are the time varying torques required to drive the side auxiliaries and more detailed hydrodynamic bearing representations. It is, however, still very powerful in its current state where comparative analyses can give direction to future engine configurations.

Detailed Single Valvetrain Model Description

A simplistic diagram of a finger follower valvetrain is shown to the right.

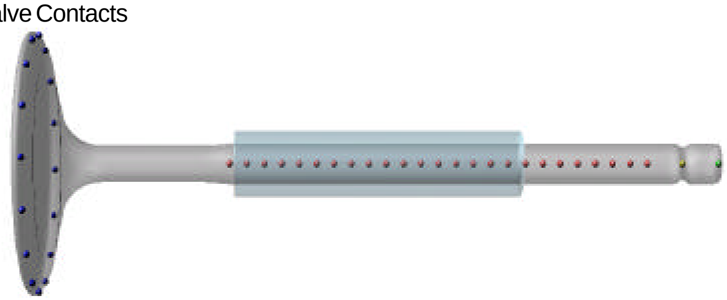


A flexible valve was added to a rigid body valvetrain model to predict deflections due to high cam accelerations. It was desirable to not only look at axial but also sideways or lateral valve vibrations. This required a three dimensional representation of the contacts between the valve and the rest of the assembly.

A flexible finger follower will be required for a more accurate representation of the flexibilities within the system. In this instance, however, the follower was very short (not indicated in the diagram) so it was assumed that its high bending stiffness would have less of an effect on the results.

The following is a summary of the detail within the model:

Valve Contacts



To simulate the contact between the valve and guide a line of interface parts were created along the valve stem centreline (see red dots above). Individual contact forces were created between the parts and the cylinder head. These forces were switched on using step functions as the parts entered the guide. This, however, did not produce a satisfactory variation in valve lateral stiffness with lift. The contacts were refined so that their magnitude when close to the guide (but not in the guide) were scaled to represent the effect of a contact force right at the edge of the guide. This gave a much better variation of lateral stiffness with lift (see verification plots below).

The valve to seat contact was also created with multiple discrete forces. This time many sphere-to-flat contact forces were used (see blue dots above). The follower to valve contact was initially created with three equi-spaced sphere-to-flat contacts, but has now been replaced by a solid-to-solid contact.

An Air Spring Lateral Stiffness Model Derived From Non-Linear FEA

A finite element model of the air spring piston, seal and cylinder was created. Several non-linear FE analyses, using measured running temperatures and material properties, were used to generate stiffness curves for the ADAMS model. The radial damping produced by the air spring seal is a low approximation based on material properties of PTFE.

Air Spring Pressure Model

The air spring is a variable volume chamber that produces a non-linear reaction force with valve lift. Its pressure is derived from simple flow equations and computational fluid dynamics models that characterise the air inlet and outlet openings. A separate ADAMS model was used to generate an air spring reaction force spline that was used in the single valvetrain model. This model correlates well to measured rig results (see figure 1).

Model Results

Full Engine Model

The full torsional model has been compared to dyno test results. The natural frequencies and vibration magnitudes compare fairly well, but a full in-depth correlation has yet to be completed. This is primarily due to a lack of running time on the dyno. The model is, however, still very powerful for comparative analyses.

Full Model Results – At Max Rpm

Figure 5 shows an example of the torsional vibration seen between the front of the crankshaft and the rear of the camshafts at 18000 rpm. There are significant 2.5E (engine order) vibrations present in the baseline results that were also evident on test. In an effort to reduce these vibrations a new drive layout was conceived that would make it much harder for the cam stab torques to excite the camshaft torsionals in this way. The improvement in torsional vibration from the new drive layout is also shown in figure 5.

Camshaft torsional vibration will affect the resulting valve motion. On observation of the acceleration curves, it was evident that the valve motion was far closer to design when the engine is run with the new drive layout.

Figure 6 shows the torsional vibration between the front of the camshafts of each bank at 18000 rpm. It is evident that there is a permanent offset between the banks. On test, a very similar offset had been observed in the timing of the inlet valves. Further analysis showed that this was a result of an a-symmetry in the gear drive. The engine was re-built and the camshafts timed with the same offset. The test results then showed negligible bank-to-bank differences.

Single Valvetrain Model

The two test assemblies in figure 2 were used to verify the static stiffness of the valve, guide contact and the air spring seal. Figures 3 and 4 show a good correlation of the ADAMS model with test. Two extremes of valve lift are shown demonstrating the effects of the valve guide and air spring seal.

Single Valvetrain Model – At Max RPM

Figures 7 to 11 show results from running the flexible valve model at 9000 cam rpm (18000 crank). Valve stem vibration is evident in both the acceleration and compression traces (see figures 7 to 9). As the valve closes, the high deceleration causes it to compress to the extent that it hits the seat with a higher closing velocity than the cam design should allow (figure 7 and 9). Furthermore, the valve seems to make more than one separate contact with the seat. This may be a contributing factor to valve seat wear seen on rig tests.

Figure 9 shows that the valve base is tilting and bending, increasing the compression at some points up to 150 microns. The valve shim is moving laterally due to the offset contact load and frictional forces between the follower and valve. The vibrations generated at the shim travel down the stem and cause the valve base to move laterally. Solutions to the issues highlighted above concentrated on the following:

A new valve design was tried to reduce stem compression and head bend. Figure 8 shows that valve stem compression can be reduced significantly by switching to the new design.

Several different cam profiles were tried to take account of the dynamic compression of the valve stem. Figure 10 shows that it is possible to get the valve base to follow the design lift curve by some adjustments to the timing of the closing acceleration peaks. A close approximation to the rigid 211_06 valve lift can be created with the flexible assembly and the 226_01 cam.

Changes to the air spring and seal were also tried to improve lateral deflections of both the shim and the valve base (see figure 11). Note that these do not have much of an effect on the axial vibrations.

Conclusions

Full torsional and detailed single valvetrain models have been used to improve valve control and durability on a high-speed racing engine.

The full torsional model has been used to improve camshaft torsional vibration by the use of a different drive arrangement. This improved the resulting valve lift curves as a consequence. The model also predicts the same difference in the valve closing points between each bank as found on test. It was shown that this was primarily due to an asymmetry in the drive arrangement.

The flexible valve single valvetrain model predicts a similar early inlet valve closing point as measured on test. This is a combined effect of the dynamic compression of the valve stem and the bending of the valve head. Several solutions, including the use of a new valve design and changes to the cam profile, were proposed and tested. These all significantly improved valve control on test.

Appendix 1 – Figures:

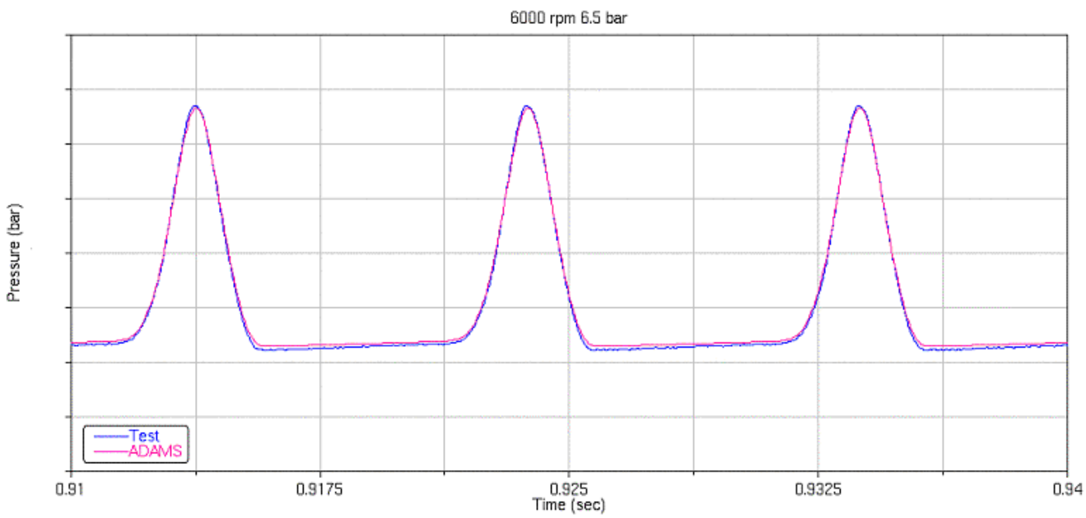


Figure 1. Comparison of air spring model with test

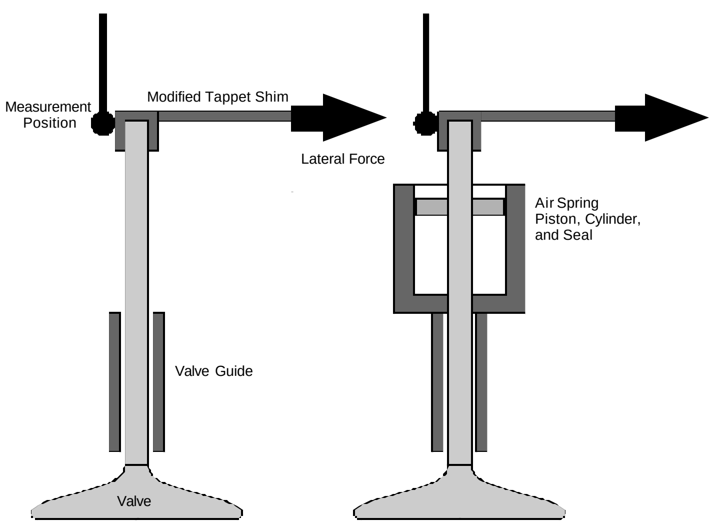


Figure 2. Test configurations for verification of flexible valve model

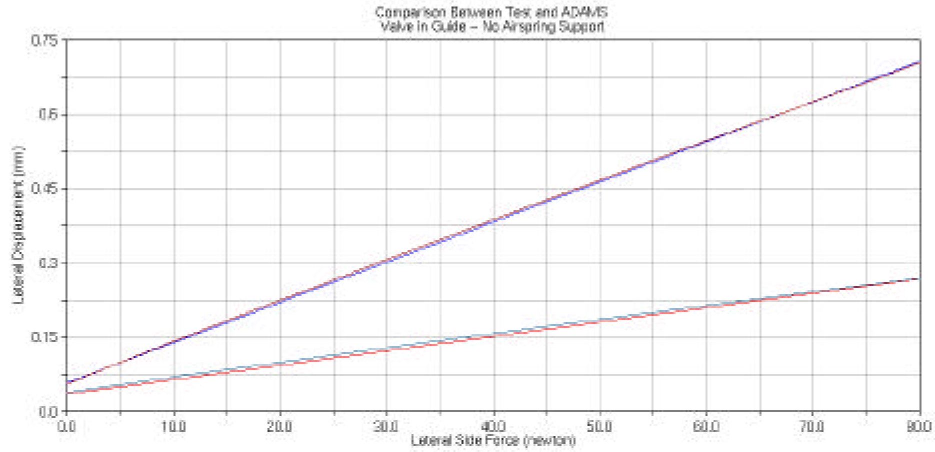


Figure 3. Comparison of valve lateral stiffness with test

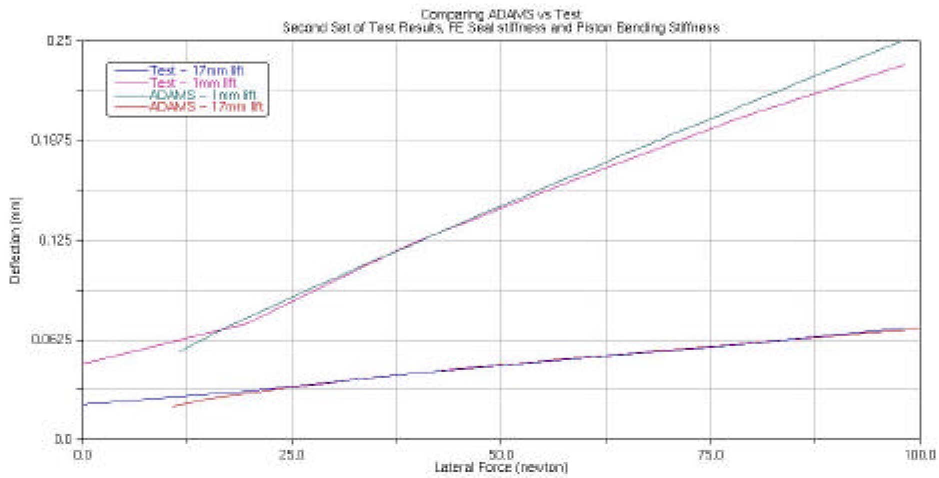


Figure 4. Comparison of valve and air spring lateral stiffness with test

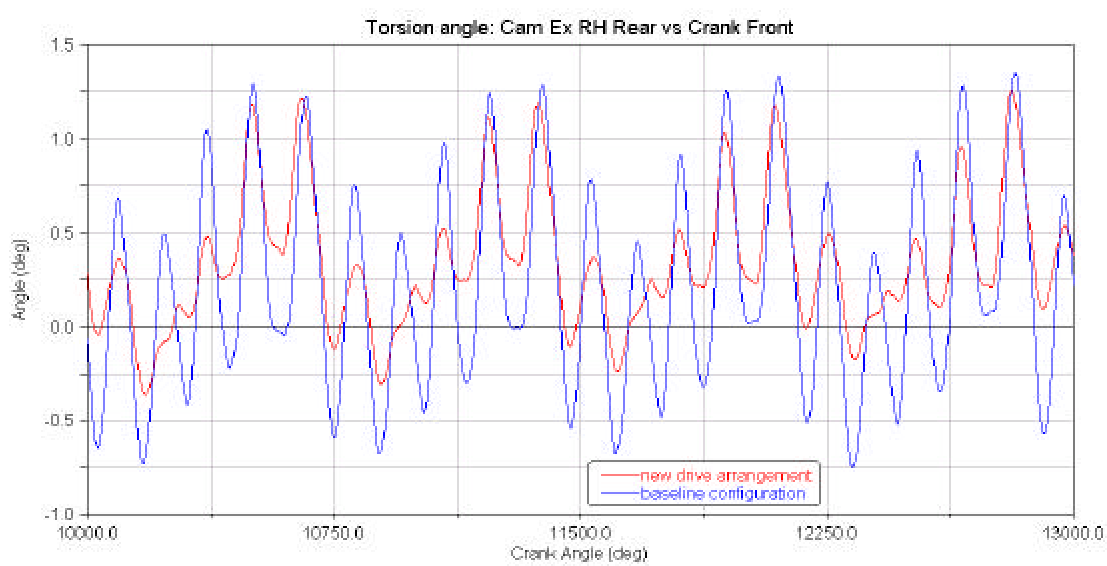


Figure 5. Rear Cam torsion relative to crank front – Cam Ex RH

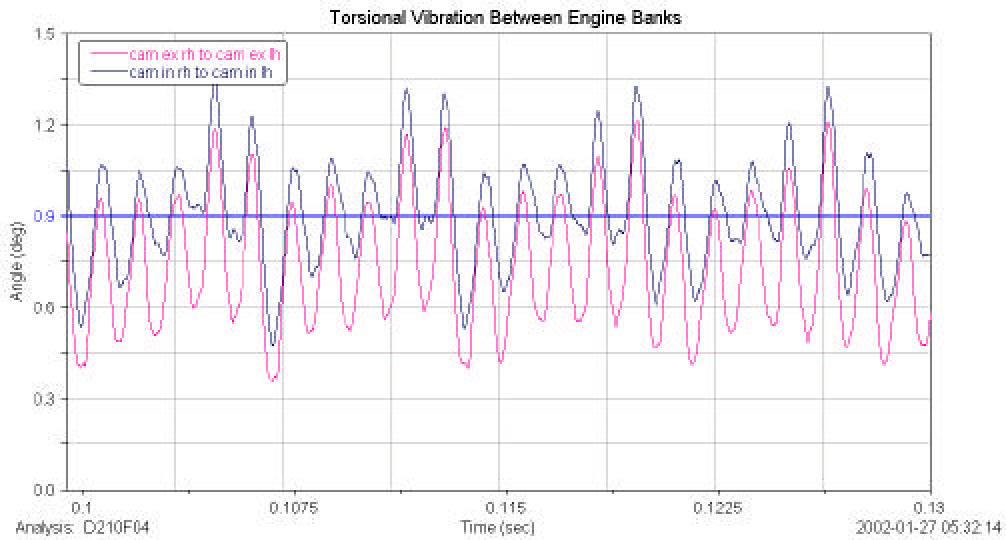


Figure 6. Bank to bank torsional vibration

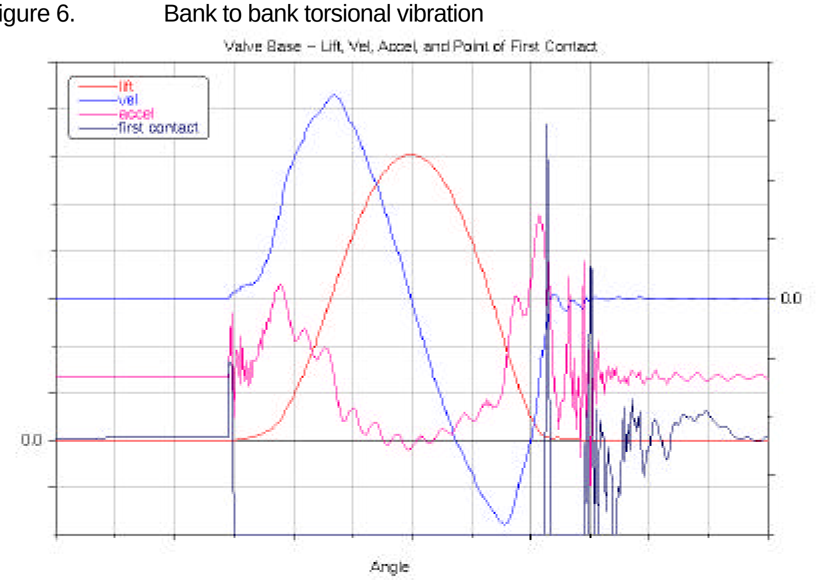


Figure 7. Lift, velocity, acceleration and seat contact force at 9000 cam rpm

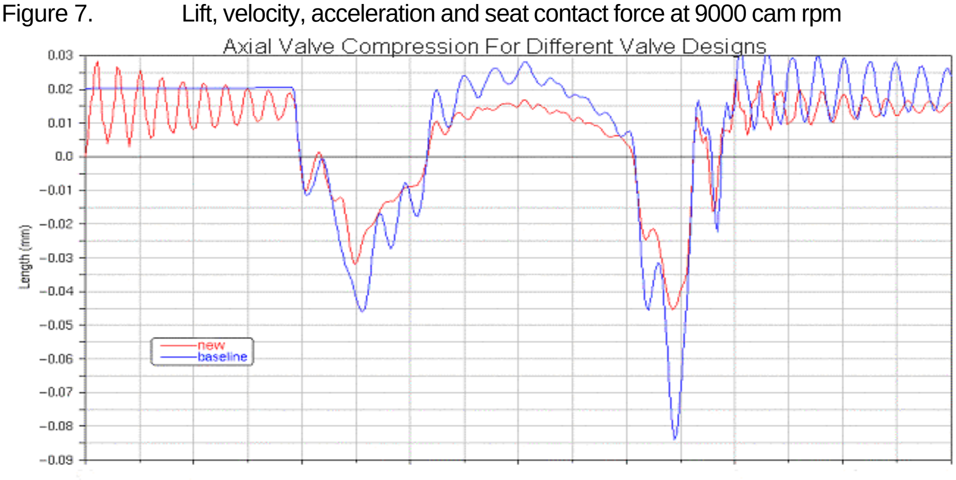


Figure 8. Axial valve deformation – showing improvement in new valve design

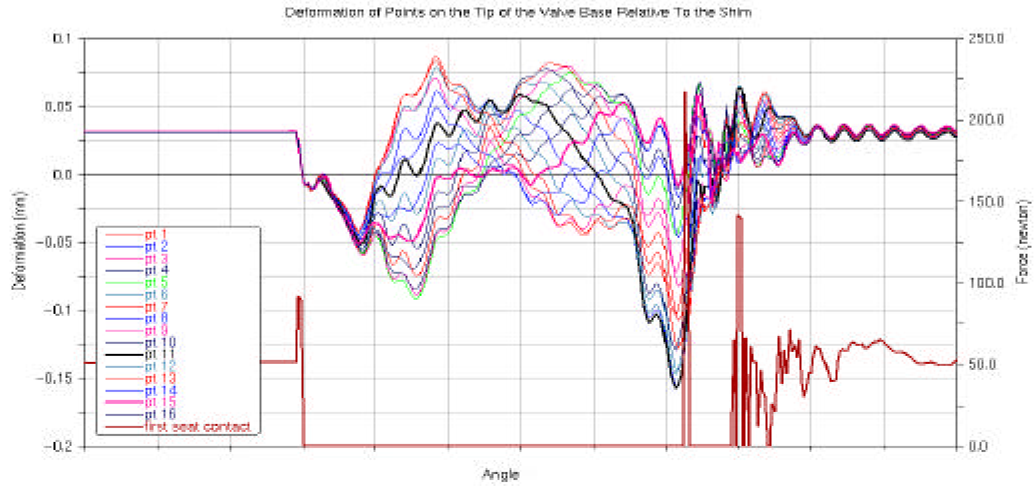


Figure 9. Showing axial d eformation at edges of valve base

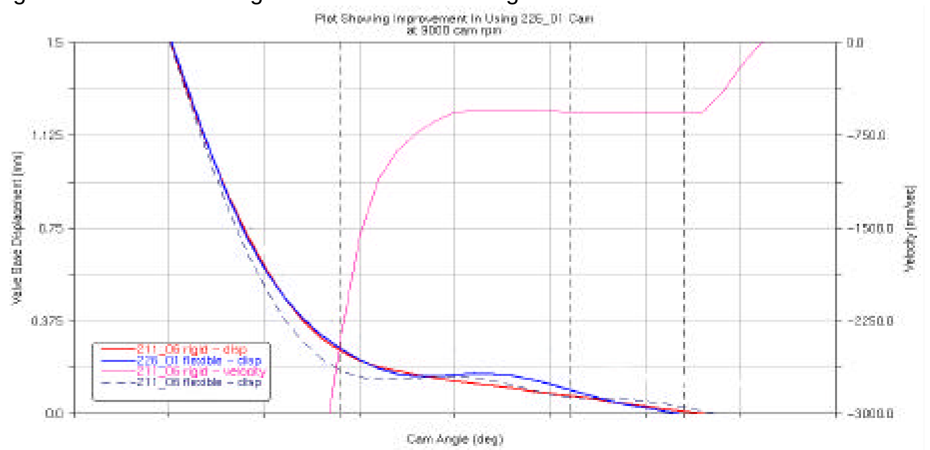


Figure 10. Showing how a cam profile change can improve valve control

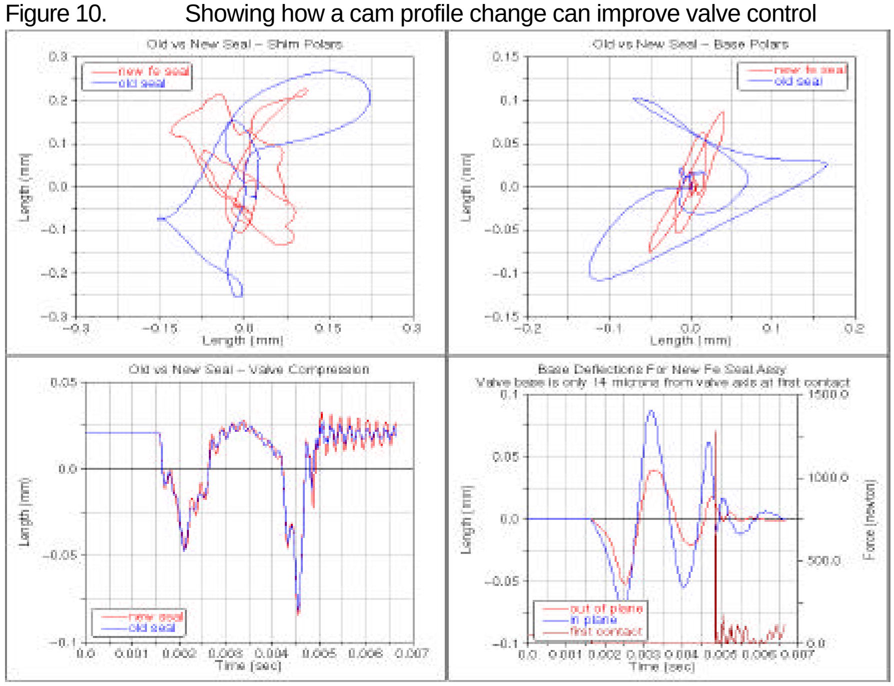


Figure 11 Shows the effect of a new seal arrangement