4.Experimental Studies of Valve Train Dynamics

4.1 Objective of Experiments

In the previous two chapters, manufacturing errors that can occur while grinding an automotive camshaft were investigated. These errors were a result of changing the link lengths of a rocker type grinder by a significant amount. These changes in link length resulted in a variance in the kinematic properties of the camshaft being ground. If a camshaft with changed kinematic properties is installed in an engine, changes in the dynamic behavior of the engine's valve train may result. As a result, some of the error profiles described in the previous chapter were used to fabricate several cams. These cams were then tested in an actual valve train. These tests were conducted to determine any significant changes in dynamics between the theoretical cam and cams containing profile errors. With this in mind, this chapter will describe the equipment and methods used to experimentally investigate the effects of cam manufacturing errors on valve train performance.

4.2 Experimental Apparatus

The purpose of the test apparatus is to measure valve position with respect to cam angle for a number of engine speeds in a typical small block engine. A schematic of this set-up is shown in Figure 4.1 and a photograph of the actual equipment is shown in Figure 4.2. This equipment was constructed using an actual engine block fitted with race type cylinder heads and a valve train assembly for a single valve. Since testing the valve train

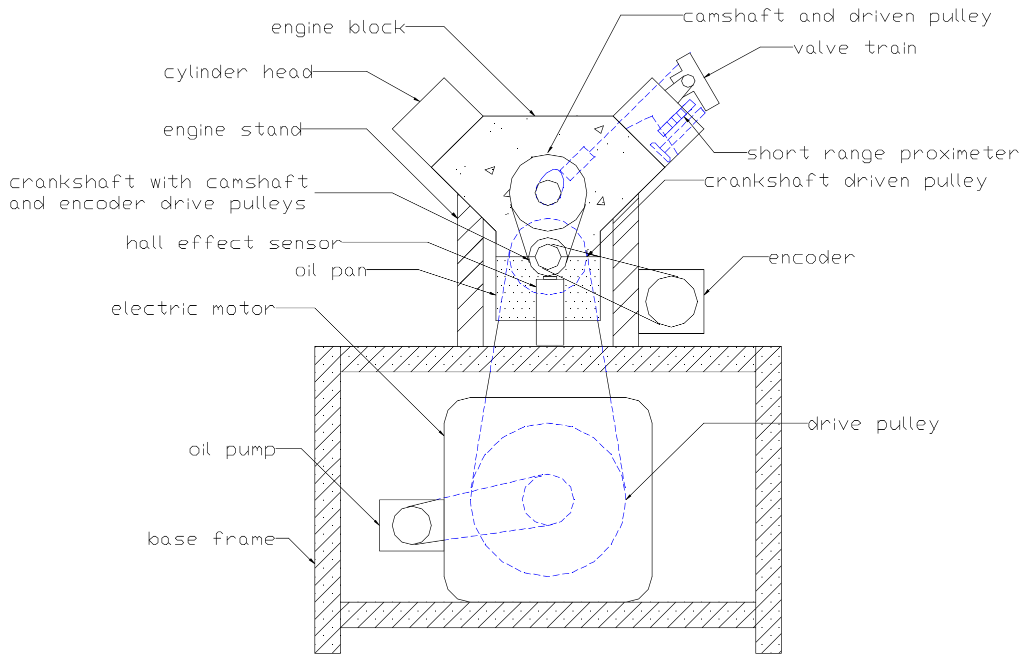

Figure 4.1: Schematic of the test rig used to experimentally investigate valve train dynamics

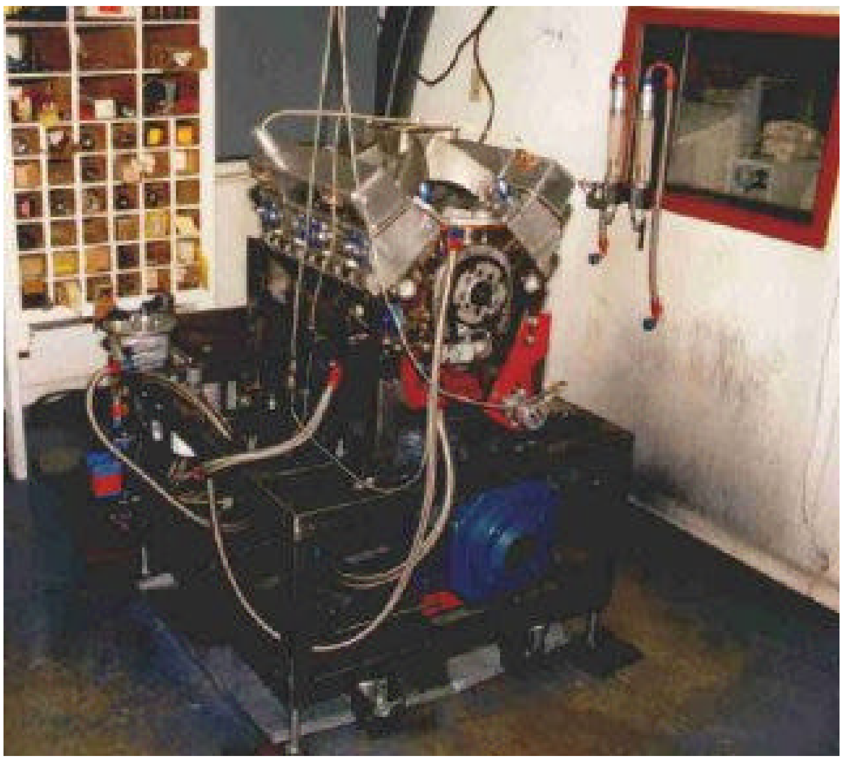

Figure 4.2: Photograph of actual test equipment

does not require a fully functioning engine, several modifications were made. The crankshaft was replaced with a straight ‘dummy’ crankshaft (i.e. no crank arms) due to the fact that piston motion is not required or is of interest in this analysis. However, a crankshaft was still required to drive the valve train. The cylinder heads were also drilled and tapped to accept the proximeter used to measure valve position. In addition, all oil galleys that were not used were plugged to maintain oil pressure in the block.

As shown in Figure 4.1 and Figure 4.2, the assembled test block is mounted to a rigid frame. This frame houses a direct current (DC) motor, which drives the dummy crankshaft. This crankshaft powers the valve train over a specific engine speed range. The typical engine speed range used during this investigation was 7,000 to 10,000 rpm. Since cam rotation is reduced by a factor of two with respect to the crankshaft, the corresponding cam speed was 3,500-5,000 rpm. As the valve train is actuated, the valve lift, crank angle and engine speed are measured by a proximeter, encoder and hall effect sensor respectively. These measurements are captured via an analog-to-digital board in a personal computer.

The experimental data in these studies were obtained by a short range proximeter. Since a short range sensor has an approximate range of 1 mm (0.040 inches), the lift data were truncated as shown in Figure 1.4. As a result, the entire valve motion was not recorded. Only the motion in the proximity of the valve/valve seat contact was recorded. This does not, however, limit the validity of the test results. The dynamics of interest in this system occurred when the valve closed and struck the seat. This occurred well within the sensing range of the proximeter used in this testing.

4.3 Modular Camshaft Construction

In order to investigate changes in valve train dynamics due to cam profile changes, it is only necessary to look at a single valve. Therefore, having an entire cam shaft ground was not necessary. Although, a cam grinder was not available for use in this investigation, a numerical control electro-discharge machine (EDM) and conventional machine tools were. With this in mind, it was decided to design a modular camshaft with a single cam lobe and the following features:

- Individual plate cams can be rigidly attached to the shaft yet easily replaced so that the shaft is reusable.
- Consistent alignment of the front and rear journals is achieved each time the camshaft is taken apart and reassembled.

The modular cam developed during this investigation contains these features.

The various components that make up the modular cam used in this investigation are shown in Figure 4.3 and Figure 4.4. These figures present the exploded and the assembled views of the modular cam respectively. The shaft shown in these figures was manufactured on a cylindrical grinder. After grinding, a flat was cut on the portion of the shaft were the cam plate and rear journal are mounted. This results in a 'D' shaped shaft. The plate cam and rear journal are also constructed with 'D' shape holes to lightly press fit onto the shaft. In order to ensure consistency in setup, the cam also registers against a step on the shaft. The journal is then pressed onto the shaft behind the cam. The journal and cam lobe are then held in place by a flat washer, lock washer and nut.

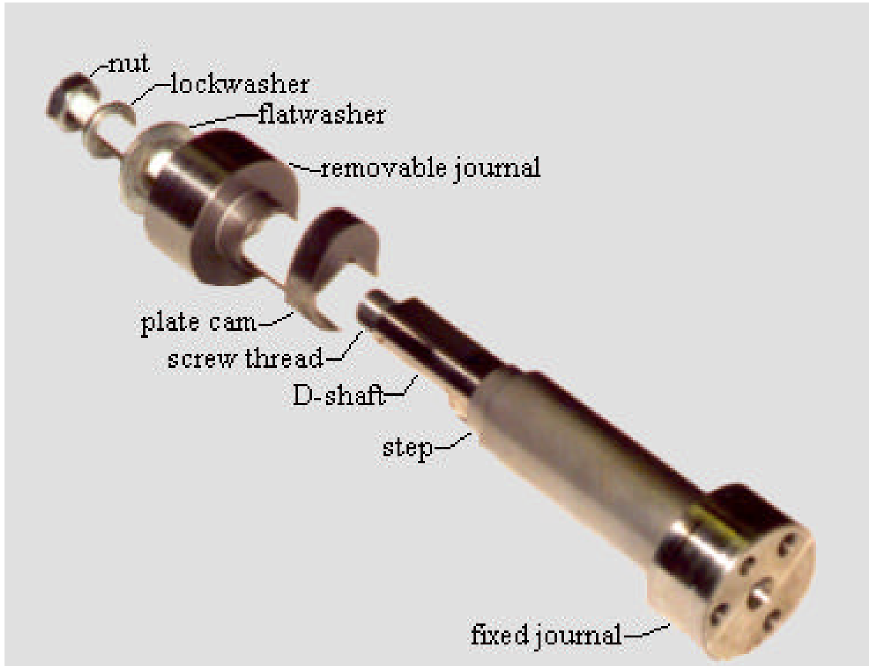

Figure 4.3: Exploded view of modular cam components

Figure 4.4: Assembled view of modular cam components

The cam plate shown in the previous figures was manufactured from F-7 tool steel which was hardened to Rockwell 60C. This is the same hardness specification as a typical racing cam however, the material specification is different. It was decided to harden the cam before machining so that the cam surface would not be distorted due to post process hardening. Since the material is this hard, conventional machine tool cutters cannot be used to fabricate the lobes. Therefore, the cam profiles were produced on an EDM. The EDM also facilitates the cutting of the 'D' shape holes in the cam blank and the rear journal which are difficult to produce on conventional machine tools.

It should be noted that the modular camshaft only uses the first two bearings in the small block engine. In order to maintain oil pressure in the block, the other cam bearings were plugged using a nylon shaft that was machined to press fit into the remaining three cam bearings.

For this study it was planned to run modular cam tests for both roller follower and flat follower systems. The initial test run for the roller follower resulted in insignificant wear on the cam lobe. Thus this setup was deemed acceptable for further testing. The flat follower cam, however, did not fair as well. As can be seen in Figure 4.5, the surface of the cam was marred from running the cam in the test block. It is believed that this cam may have failed from one or more of the following causes:

- Incompatible material combination for the cam/follower pair
- Improper taper on the cam lobe to match the taper on the cam follower.
- Too coarse of a surface finish from the EDM process on the lobe face.

Since this investigation was more focused on determining changes in system dynamics due to manufacturing errors, the roller cam was used to complete the investigation. Since cam/follower compatibility was not of primary concern, additional investigation into the causes of the wear of the flat follower cam were not conducted.

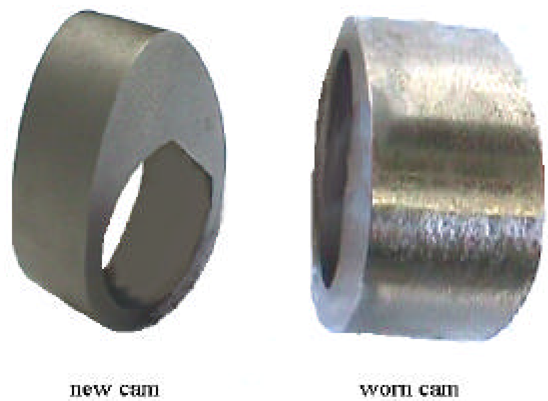

Figure 4.5: Comparison of flat follower lobes manufactured by edm to show wear due to improper cam/follower match

4.4 Baseline Experiments Concerning Valve Train Behavior

As a starting point to investigate the effects of errors on the dynamics of valve trains, some baseline testing was required. The baseline data was gathered to investigate the following:

- Behavior of valve trains of different cylinders in the same engine.
- Repeatability of successive tests for the same valve train.

The first test was to determine the operational differences between two cylinders using similar valve train set-ups. A full valve train setup was installed on the test engine (i.e. all 8 cylinders using both the intake and the exhaust valves) and valve position was monitored for the first and eighth cylinder. Figure 4.6 shows the difference between the valve bounce amplitudes for the intake valves in these cylinders. This figure shows that there is a major difference in valve train dynamics within the same engine.

Since the purpose of this investigation is to determine whether cam differences can cause this type of behavior, the cam lift curves were measured. The absolute difference of the lift curves and the nominal lift curve itself are shown in Figure 4.7. As can be seen in these figures, the largest errors are less than 0.0254 millimeters (0.001 inch) yet the valve bounce amplitudes differ by more than 0.254 millimeters (0.010 inches). This cannot necessarily be attributed to the cam profile, since it may be a function of the differences in stiffness and damping in the system for the different engine positions. What is important to note is that variation may occur in the valve train from cylinder to cylinder. It is shown later in this chapter, that changes of the cam profile of several thousandths of an inch do not cause significant changes in valve train dynamics. Therefore, performance differences for the two cylinders may be attributable to different stiffness and damping characteristics of the cylinder area.

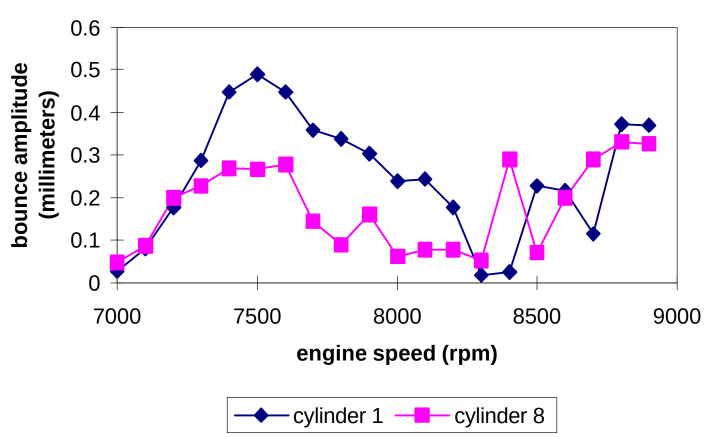

Figure 4.6: Valve bounce comparison for two cylinders during the same test using a full valve train

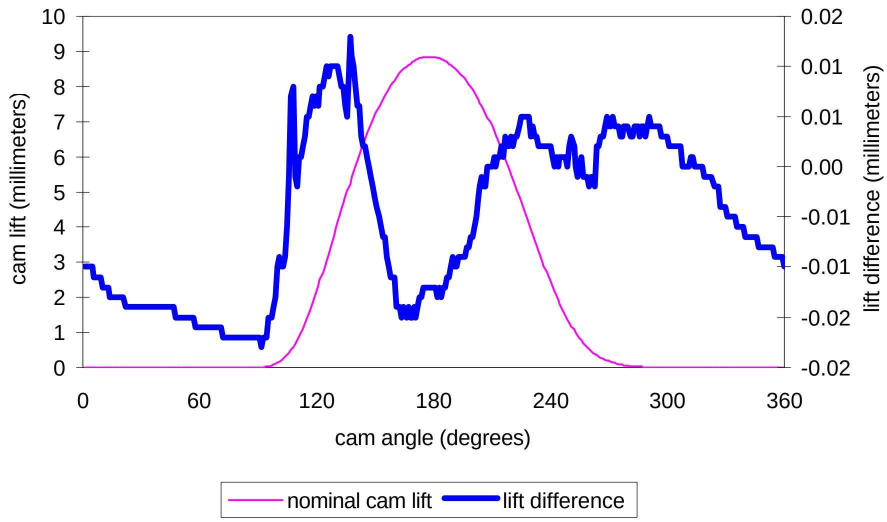

Figure 4.7: Difference in cam lift for two cylinders on same camshaft

The next baseline experiment that was performed was to investigate the repeatability of the behavior of the valve train for a single cylinder. Several limit speed tests were using the same valve train components with the exception of the valve spring. The valve spring must be changed any time limit speed is reached in an engine. It should be noted that the springs must be changed after limit speed testing since a permanent set is present in the spring after limit speed is obtained.

The baseline tests were conducted for two different spring types. These springs are the Comp Cams 938 and the K-Motion 1600 type springs. Both of these springs are used in NASCAR unrestricted engines, which have an operational range of approximately 7,000-10,000 rpm. Some of the physical properties of the springs are shown in Table 4.1. The results of these tests are shown in Figures 4.8 and 4.9. These figures show that valve bounce amplitude changes from test to test for a given set-up at a given engine speed. It also shows that valve bounce amplitude up to 0.254 millimeters (0.010 inch) occurs at typical operating speeds, therefore, valve bounce amplitudes up to this point can be considered acceptable.

For all of the baseline tests, the maximum bounce amplitude was determined for each engine speed and a bounce envelope was developed. The envelopes are shown in yellow and orange on Figures 4.8 and 4.9 for the Comp Cams 938 and K-Motion 1600 springs respectively. These envelopes will be used to compare the performance other tests performed in this work to the baseline data. It should also be noted that each of the baseline tests repeat within a band of +/- 100 rpm. As a result of this, if the experimental test results fall within this band, the test will be considered to be within tolerance of the baseline testing.

Parameter	**Comp Cams 938**	**K-Motion 1600**
Material of construction	steel	steel
Number of nested springs	2	2
Number of active coils of outer	3.75	4.125
Free length of outer: millimeters (inches)	61.7(2.430)	62.4(2.455)
Wire diameter of outer: millimeters (inches)	5.26(0.207)	5.26(0.206)
Outside coil diameter of outer: millimeters (inches)	39.3(1.547)	39.7(1.562)
Number of active coils of inner	5.5	5.75
Free length of inner: millimeters (inches)	55.1(2.170)	61.0(2.400)
Wire diameter of inner: millimeters (inches)	3.61 (0.142)	3.71(0.146)
Outside coil diameter of inner: millimeters (inches)	25.9(1.018)	27.3(1.074)

Table 4.1: Nominal parameters of the two spring types used in this investigation

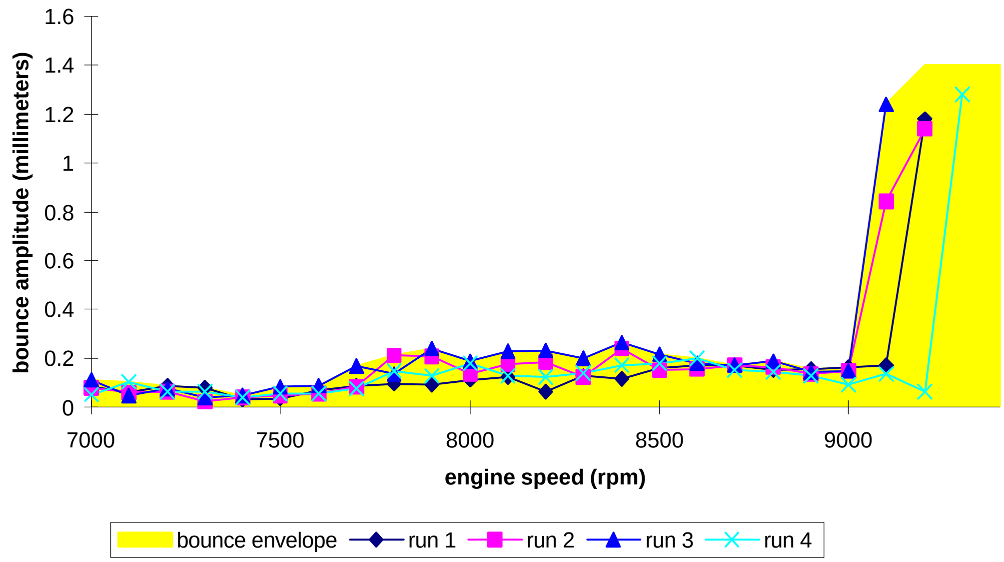

Figure 4.8: Baseline test data for comp cams 938 type springs

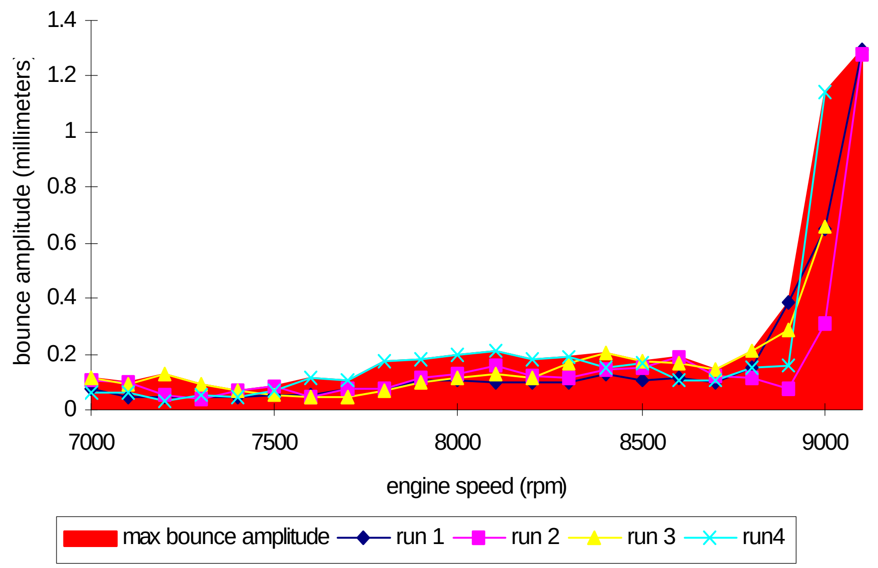

Figure 4.9: Baseline test data for k motion 1600 type springs

4.5 Comparison of Test Results for an Actual Automotive Camshaft and the Modular Cam

In order to use the modular cam to predict dynamic behavior for a typical automotive camshaft, a baseline test was run to see if the results from both cams correlate. In the previous section, multiple tests that constitute the bounce envelope of the system for the modular cam were presented. In order to compare the modular cam and an actual automotive camshaft, tests were performed using a camshaft with the same lift profile as the one on the modular cam and the same valve train components. This testing was conducted using both the Comp Cams 938 and K-Motion 1600 springs. The test results are presented in Figure 4.10 and Figure 4.11. For the Comp Cams 938 test presented in Figure 4.10, the limit speed does not "go out" as strongly as the other tests. However, the test was stopped since it was audibly obvious that it was impractical to continue to run the engine at the higher engine speeds. A raspy sound was emitted from the engine that indicated dynamic instability in the valve train even though the valve bounce was not as large as other tests. The bounce amplitudes for the entire test were well in line with the bounce envelope. Even the values that were outside of the envelope were close, and therefore acceptable. The results of the K-Motion 1600 test were even more 'in line' than the Comp Cams 938 test. The bounce amplitude only slightly exceeded the bounce envelope in four places and the limit speed is well contained in it. The test results indicate a good correlation between the modular cam and the actual automotive camshaft. Therefore, the modular cam provides an accurate basis for dynamic testing of valve trains.

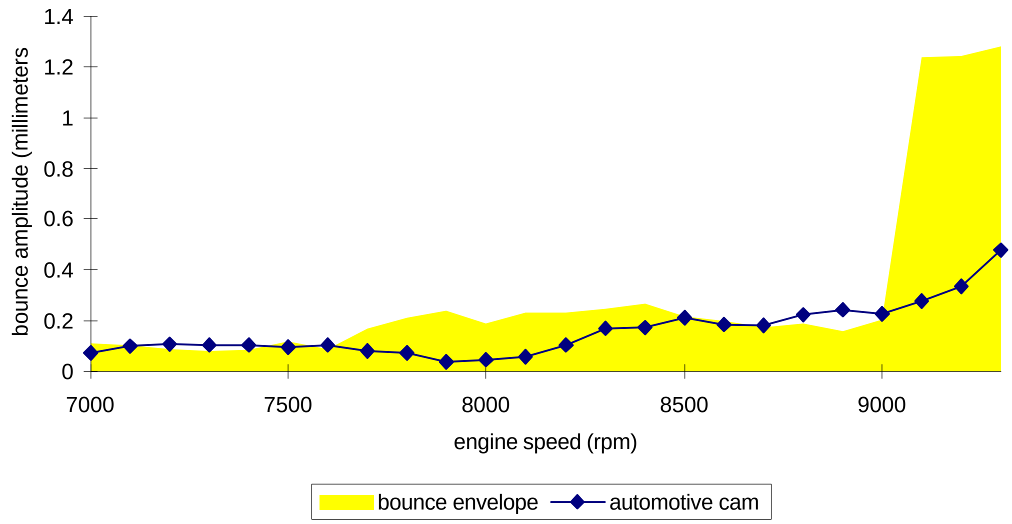

Figure 4.10: Comparison of actual camshaft and modular cam results for comp cams 938 spring type

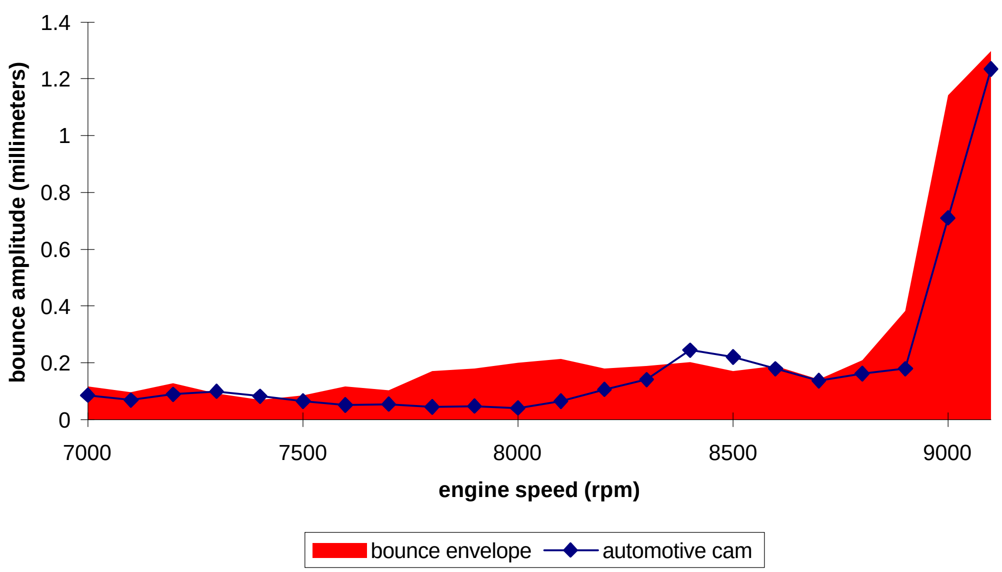

Figure 4.11: Comparison of actual camshaft and modular cam results for k-motion 1600 spring type

4.6 Experimental Determination of the Effects of Grinding Wheel Size Error on Valve Train Dynamics

In the previous chapters, error cam profiles were developed by changing the lengths of various links on the rocker type cam grinder. The changes in link length went from small to extremely large. With this in mind, errors over a few thousandths of an inch on the vertical grinding wheel position and rocker length are beyond what is considered the normal measurement accuracy of a typical machine shop. The grinding wheel, however, can change on the order of inches and may be used if the grinding velocities are appropriately adjusted to match the wheel diameter. Since the grinding wheel diameter appears to be the most likely candidate to change and limited samples were available for use in this investigation, it was decided to construct cams that included this error instead of the vertical grinding wheel position and rocker link length errors. These cams were then mounted on the modular camshaft and tested in the test rig. Results for the Comp Cams 938 and K-Motion springs are presented in Figures 4.12 and 4.13 respectively.

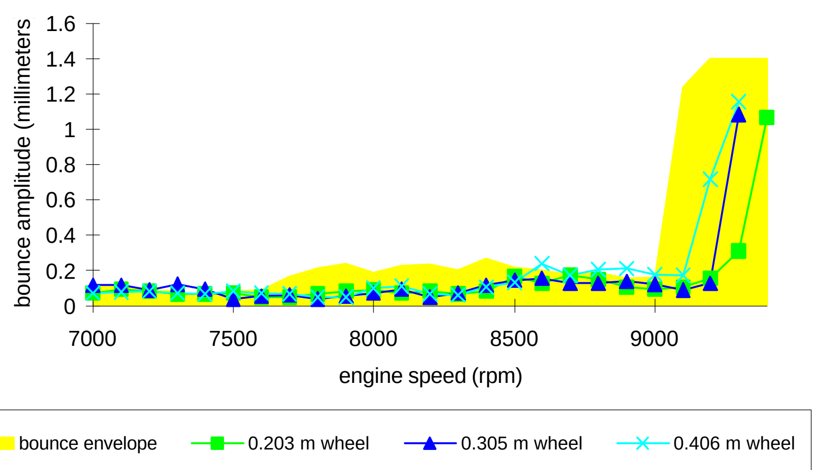

Figure 4.12: Error Cam Test Data for the Comp Cams 938 Type Spring

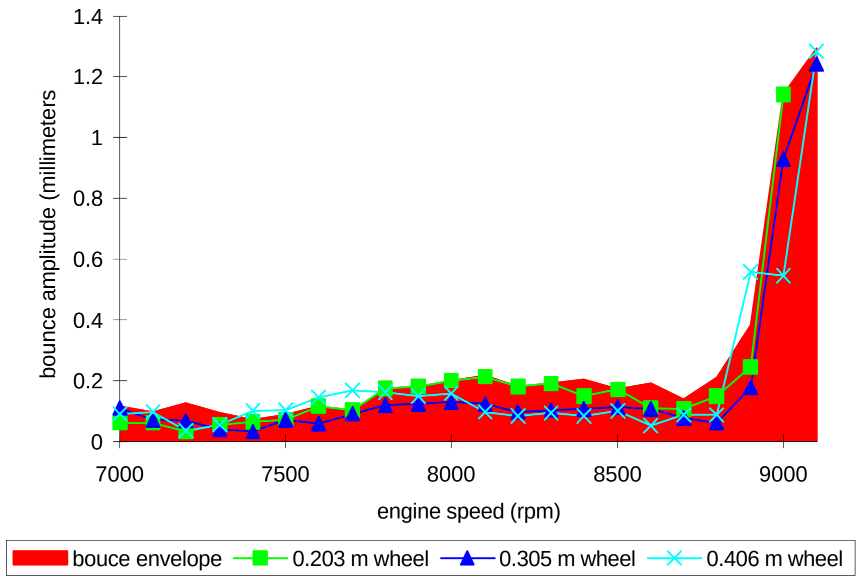

Figure 4.13: Error cam test data for the k motion 1600 type spring

As shown in Figure 4.12 and Figure 4.13, the valve bounce amplitude did not vary greatly from the performance envelope for each of the tests. Even in the few cases where the performance envelope was exceeded the amplitude did not vary greatly. This leads us to the following conclusions:

- Grinding wheel size changes do not have a great impact on the dynamics of the given valve train.
- Tolerances for cam profiles can be on the order of 0.025 millimeters (thousandths of an inch) instead of on the order of 0.0025 millimeters (ten thousands of an inch) when valve train dynamics are concerned.

Again, this analysis assumes that the grinding wheel velocities are compensated for when the wheel size is reduced. If these velocities are not compensated for, the cam surface may become unusable.