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Valve train natural frequency measurement and analysis

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Cynthia A. Greene
Abstract

VALVE TRAIN NATURAL FREQUENCY MEASUREMENT AND ANALYSIS

by Cynthia A. Greene

Chairperson of the Thesis Committee: Professor Josef Torok
Department of Mechanical Engineering

Natural frequencies are very important characteristics of a valve train system. From the measured natural frequencies of a valve train the following can be determined: will the system have acceptable dynamic behavior, is one valve train system design superior to another, and is a dynamic model of the system properly setup. The three most important modes of vibration of a valve train system are the fundamental mode of vibration of the entire linkage (system natural frequency), translation of the valve spring, and torsion of the camshaft. Valve trains are nonlinear systems due to varying system geometry throughout the lift event.

It is important to have a natural frequency measurement technique that accommodates the nonlinearity of the valve train and that takes out as much subjectivity in determining the natural frequencies as possible. In this discussion, three different natural frequency measurement techniques are reviewed, which characterize one or more the three most important modes of vibration of the valve train.

A significant part of the work done for this thesis project is on the development of a step input method for measuring valve train natural frequencies. Also discussed in this thesis are the effects of system nonlinearity and two nearby frequencies of one system on the frequency analysis results. The differences between low and high natural frequency systems are also discussed.
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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>List of Tables ...............................................................</td>
</tr>
<tr>
<td>List of Figures ..............................................................</td>
</tr>
<tr>
<td>1. Introduction ........................................................................</td>
</tr>
<tr>
<td>1.1. Background .................................................................</td>
</tr>
<tr>
<td>1.2. Objectives .......................................................................</td>
</tr>
<tr>
<td>2. Natural Frequency Concepts and Measurement Methods ............</td>
</tr>
<tr>
<td>2.1. Basic Vibration Theory for Linear Systems .......................</td>
</tr>
<tr>
<td>2.1.1. Single Degree-of-Freedom Systems ...............................</td>
</tr>
<tr>
<td>2.1.2. Multiple Degree-of-Freedom Systems .........................</td>
</tr>
<tr>
<td>2.2. Signal Processing .........................................................</td>
</tr>
<tr>
<td>2.2.1. Theory .......................................................................</td>
</tr>
<tr>
<td>2.2.2. Examples ..................................................................</td>
</tr>
<tr>
<td>2.3. Valve Train Systems .....................................................</td>
</tr>
<tr>
<td>2.4. Measurement Methods ....................................................</td>
</tr>
<tr>
<td>2.4.1. Manual ......................................................................</td>
</tr>
<tr>
<td>2.4.2. Step Input ..................................................................</td>
</tr>
<tr>
<td>2.4.3. Waterfall Plot ..........................................................</td>
</tr>
<tr>
<td>3. System Modeling ...............................................................</td>
</tr>
<tr>
<td>3.1. Overhead Cam Finger Follower System Model ....................</td>
</tr>
<tr>
<td>3.2. Valve Train Dynamic Simulation Results ..........................</td>
</tr>
<tr>
<td>4. Experimental Analysis ......................................................</td>
</tr>
<tr>
<td>4.1. Step Input Method .........................................................</td>
</tr>
<tr>
<td>4.2. Test Setup and Data Analysis .........................................</td>
</tr>
<tr>
<td>4.3. Test Conditions and Results ...........................................</td>
</tr>
<tr>
<td>4.3.1. Trial One .................................................................</td>
</tr>
<tr>
<td>4.3.2. Trial Two .................................................................</td>
</tr>
<tr>
<td>4.3.3. Trial Three ...............................................................</td>
</tr>
<tr>
<td>4.3.4. Trial Four ...............................................................</td>
</tr>
<tr>
<td>4.4. Comparison of Experimental and Analytical Results ............</td>
</tr>
<tr>
<td>5. Conclusions and Recommendations .....................................</td>
</tr>
<tr>
<td>Appendix A - Sample of Valve Train Simulation Results ............</td>
</tr>
<tr>
<td>References ...........................................................................</td>
</tr>
<tr>
<td>Bibliography .........................................................................</td>
</tr>
</tbody>
</table>
LIST OF TABLES

<table>
<thead>
<tr>
<th>Number</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Types of Valve Trains..............................................3</td>
</tr>
<tr>
<td>2.</td>
<td>Variable Definitions for the System Model.......................35</td>
</tr>
<tr>
<td>3.</td>
<td>Simulated Natural Frequencies for the Left Side Cylinder Head..38</td>
</tr>
<tr>
<td>4.</td>
<td>Simulated Natural Frequencies for the Right Side Cylinder Head.39</td>
</tr>
<tr>
<td>5.</td>
<td>Valve Spring Manufacturer Specifications..........................45</td>
</tr>
<tr>
<td>6.</td>
<td>Step Input Testing Trial Four Test Conditions........................73</td>
</tr>
<tr>
<td>7.</td>
<td>Comparisons of Experimental and Analytical Results...............102</td>
</tr>
</tbody>
</table>
LIST OF FIGURES

Number Page
1. Types of Valve Trains.................................................................2
2. Example of Frequency Response Plot...........................................7
3. Undamped Single Degree-of-Freedom System.................................7
4. Damped Single Degree-of-Freedom System.....................................8
5. Example of Periodic Signal..........................................................11
6. Example of Aperiodic Signal.........................................................12
7. Example of Random Signal..........................................................12
8. Examples of Frequency Domain Amplitude and Phase Spectrums............13
9. Amplitude Spectrums of Linear and Nonlinear Systems.......................16
10. Amplitude Spectrums of a System with Two Different Frequencies.........18
11. Example of Acceleration Trace for Determining Natural Frequency.......22
12. Waterfall Plots of Systems at Different Natural Frequency Levels..........25
13. Overhead Cam Finger Follower Valve Train System Diagram...............29
14. Hydraulic Lash Adjuster Diagram................................................31
15. Overhead Cam Finger Follower Valve Train System Model....................34
16. Free Body Diagram for Valve/Spring Retainer Assembly......................41
17. Step Input Test Fixture Assembly Drawing.....................................43
18. Valve Train Test Fixture Setup for Step Input Test............................44
19. Wiring Diagram for Load Cell Used in Step Input Fixture....................47
20. Time and Frequency Responses for Step Input Test, Trial 1................51
21. Time and Frequency Responses for Step Input Test, Trial 2, Setup 1.....56
22. Time and Frequency Responses for Step Input Test, Trial 2, Setup 2.....57
23. Time and Frequency Responses for Step Input Test, Trial 2, Setup 3.....58
24. Time and Frequency Responses for Step Input Test, Trial 2, Setup 4.....59
25. Time and Frequency Responses for Step Input Test, Trial 2, Setup 5.....60
26. Time and Frequency Responses for Step Input Test, Trial 2, Setup 6.....61
27. Time and Frequency Responses for Step Input Test, Trial 3, Setup 1.....66
28. Time and Frequency Responses for Step Input Test, Trial 3, Setup 2.....67
29. Time and Frequency Responses for Step Input Test, Trial 3, Setup 3.....68
30. Time and Frequency Responses for Step Input Test, Trial 3, Setup 4.....69
31. Time and Frequency Responses for Step Input Test, Trial 3, Setup 5.....70
32. Time and Frequency Responses for Step Input Test, Trial 3, Setup 6.....71
33. Time and Frequency Responses for Step Input Test, Trial 3, Setup 7.....72
34. Time and Frequency Responses for Step Input Test, Trial 4, Setup 1.....80
35. Time and Frequency Responses for Step Input Test, Trial 4, Setup 2.....81
36. Time and Frequency Responses for Step Input Test, Trial 4, Setup 3.....82
37. Time and Frequency Responses for Step Input Test, Trial 4, Setup 4.....83
38. Time and Frequency Responses for Step Input Test, Trial 4, Setup 5...84
39. Time and Frequency Responses for Step Input Test, Trial 4, Setup 6...85
40. Time and Frequency Responses for Step Input Test, Trial 4, Setup 7...86
41. Time and Frequency Responses for Step Input Test, Trial 4, Setup 8...87
42. Time and Frequency Responses for Step Input Test, Trial 4, Setup 9...88
43. Time and Frequency Responses for Step Input Test, Trial 4, Setup 10...89
44. Time and Frequency Responses for Step Input Test, Trial 4, Setup 11...90
45. Time and Frequency Responses for Step Input Test, Trial 4, Setup 12...91
46. Time and Frequency Responses for Step Input Test, Trial 4, Setup 13...92
47. Time and Frequency Responses for Step Input Test, Trial 4, Setup 14...93
48. Time and Frequency Responses for Step Input Test, Trial 4, Setup 15...94
49. Time and Frequency Responses for Step Input Test, Trial 4, Setup 16...95
50. Time and Frequency Responses for Step Input Test, Trial 4, Setup 17...96
51. Time and Frequency Responses for Step Input Test, Trial 4, Setup 18...97
52. Time and Frequency Responses for Step Input Test, Trial 4, Setup 19...98
53. Time and Frequency Responses for Step Input Test, Trial 4, Setup 20...99
54. Time and Frequency Responses for Step Input Test, Trial 4, Setup 21...100
Chapter 1

INTRODUCTION

1.1 Background
Natural frequencies are very important characteristics of any vibrating mechanical system. The natural frequencies of a system tell how aggressively the system can be driven and allow relative comparison of two or more systems. With respect to valve train systems, natural frequencies are well known parameters. The three most important natural frequencies of a valve train system are those that correspond to the following modes of vibration of the valve train system: the fundamental mode of vibration of the entire linkage (the associated natural frequency is referred to in this discussion as the system natural frequency), torsion of the camshaft, and translation of the valve spring. Those that are familiar with valve train systems generally recognize whether or not measured or calculated natural frequencies will result in acceptable dynamic behavior for that particular type of valve train. The natural frequencies of a particular valve train system also allow relative comparison of valve trains to determine, for example, if one system is dynamically superior to the other or if a particular design change results in dynamic performance improvements. Additionally, natural frequencies calculated by a dynamic model of the valve train system can be matched to measured natural frequencies to setup the model.

There are several types of valve trains. The inherent designs of these different types of valve trains result in differing geometries (i.e., effective masses, stiffnesses, and damping factors) and, consequently, differing natural frequencies. The types of valve trains are shown in Figure 1 and described in Table 1.
Figure 1
Types of Valve Trains

![Diagram of valve trains]

- **Type I**: Overhead cam direct acting
- **Type II**: Overhead cam with finger follower
- **Type III**: Overhead cam rocker arm lash adjuster
- **Type IV**: Cam in head
- **Type V**: Cam in block
Table 1
Types of Valve Trains

<table>
<thead>
<tr>
<th>Type</th>
<th>Description</th>
<th>Typical System Natural Frequency (Hz)</th>
<th>Maximum Engine Speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>Overhead Cam Direct Acting</td>
<td>2000-3000</td>
<td>6500 + +</td>
</tr>
<tr>
<td>II</td>
<td>Overhead Cam Finger Follower</td>
<td>1200-1500</td>
<td>6500+</td>
</tr>
<tr>
<td>III</td>
<td>Overhead Cam Rocker Arm Lash Adjuster</td>
<td>900-1400</td>
<td>6000+</td>
</tr>
<tr>
<td>IV</td>
<td>Cam In Head</td>
<td>900-1400</td>
<td>6000+</td>
</tr>
<tr>
<td>V</td>
<td>Cam In Block</td>
<td>400-700</td>
<td>4000-6000</td>
</tr>
</tbody>
</table>

The forcing function of a valve train is the cam profile. The cam profile is a radial function of the cam angle, i.e. for each cam angle there is a corresponding value of cam lift. As illustrated in Table 1, different types of valve trains have differing system natural frequencies, and the allowable design parameters of a cam profile are determined based on the system natural frequency, as well as engine performance requirements (torque and power). The cam profile design parameters define how aggressive the cam profile is. An aggressive cam profile opens and closes the engine valve with a very high acceleration in order to flow as much as possible in or out of the cylinder (air/fuel mixture or exhaust gas). An aggressive cam profile allows the engine to produce high levels of torque and power, which are characteristics used to rate the performance of one engine against another. The higher the system natural frequency, the more aggressive the cam profile can be. This is because the higher the system natural frequency, the lower the amplitude of vibration of the valve train system for a given input. So a valve train system with a high system natural frequency can be forced with an aggressive cam profile while still maintaining dynamic stability (vibration amplitudes not too large to cause excessively high loads, separation between components of the valve train and/or impacts between the valve and its seat).
The development of a valve train system for acceptable dynamic behavior typically includes two main activities. First, the valve train is analytically modeled to predict dynamic behavior. Second, dynamic testing of the valve train system is conducted to verify acceptable dynamic behavior and/or allow the selection of the optimal valve train design from two or more potential configurations. For a new valve train system, once testing results are compared to the analytical results, the model may need to be adjusted to accurately represent the actual system behavior. The model can then be used to reliably predict the behavior of variations of the original valve train design.

Delphi-E performs dynamic valve train modeling with a program called Valve Train Simulation (VTS). VTS is a lumped parameter model, and it includes system nonlinearities. The outputs of VTS include the predicted eigenvalues, or natural frequencies, for a linearized system at one point in time.

Valve train dynamic testing includes several pieces of instrumentation to measure some or all of the following: valve lift and acceleration, valve spring load, rocker arm load, and/or push rod load. From the measured data, valve train natural frequencies can be determined. To date several methods for experimentally determining one or more of the valve train natural frequencies have been used.

Until a few years ago, Delphi-E used the very simple, yet imprecise, experimental method of manually calculating the valve train system natural frequency from the measured acceleration of the valve. The output of an accelerometer attached to the valve was viewed on an oscilloscope or on a print out. The number of oscillations within a particular period of time were counted, and a frequency was calculated.

Within the last few years, another method of determining valve train natural frequencies from experimental data has been used by Delphi-E. The valve acceleration data, taken at a number of different cam rpm's (typically, 600 to 1200 cam rpm) is analyzed with the
Fast Fourier Transform (FFT) method and overlaid on a plot of amplitude versus frequency, a waterfall plot. The natural frequencies are then determined by observation of the locations of the spikes which generally overlay each other for each natural frequency for all cam speeds.

The above two described methods of determining valve train natural frequencies from measured data require measurements of non-stationary valve train systems. A third method has been briefly investigated and is still under consideration. It requires measurements of a stationary valve train. The valve train system is loaded by pulling on the valve, and then the load is suddenly released, causing the valve train to vibrate. Thus a step load input is applied to the system. The resultant vibration is recorded with an accelerometer on the valve and possibly with a strain gage on the valve spring. The measured acceleration and strain gage data undergo a frequency analysis to determine the natural frequencies.

1.2 Objectives
The primary objective of this thesis project was to develop and proceduralize valve train natural frequency measurement techniques for both low and high natural frequency valve trains. The measurement systems characterized one or more of the three most important modes of vibration of the valve train system, the fundamental mode of vibration of the entire linkage, torsion of the camshaft, and translation of the valve spring. The measurement systems also had to account for system nonlinearity, as the geometry of the valve train changes throughout the lift event, thus resulting in changing effective masses, stiffnesses, and damping factors. The measurement systems were developed in an effort to take the subjectivity out of determining natural frequencies. The theoretical explanation of why certain measurement methods work well for some valve train systems, but not for others, was investigated and documented.
Chapter 2

NATURAL FREQUENCY CONCEPTS AND MEASUREMENT METHODS

2.1 Basic Vibration Theory for Linear Systems

For a single degree-of-freedom system, the natural frequency is the frequency of free vibration of the system. For a multiple degree-of-freedom system, the natural frequencies are the frequencies of the normal modes of vibration. A normal mode of vibration is one that can exist independently of the other modes of vibration of a system. In an eigenvalue problem, the normal modes are called eigenvectors for a discrete system and eigenfunctions for a continuous system, and the natural frequencies are called eigenvalues.

A system has limited response to vibrations except those at frequencies near the natural frequencies of the system. Near the natural frequencies, the amplitude of the vibration is large and the system is said to resonate. Critical frequency is defined as that at which, if the frequency is increased or decreased, the vibration amplitude will decrease, as can be see in the example of a frequency response plot in Figure 2. Without damping, the critical frequency corresponds to the natural frequency. With damping, the critical frequency, or the damped natural frequency, is lower than the natural frequency.
2.1.1 Single Degree-of Freedom Systems

A single degree-of-freedom system experiencing free vibration without damping is shown in Figure 3 and defined by the following equation of motion:

\[ m\ddot{x} + kx = 0 \]

![Figure 3: Undamped single degree-of-freedom system](image)

The circular natural frequency in radians/second and the natural frequency in hertz are defined by the following equations, respectively:

\[ \omega_n = \sqrt{\frac{k}{m}} \]

\[ f_n = \frac{\omega_n}{2\pi} = \frac{1}{2\pi} \sqrt{\frac{kg}{W}} = \frac{1}{2\pi} \sqrt{\frac{g}{\delta}} \]
The solution for the undamped, single degree-of-freedom system, experiencing free vibration is

Initial Conditions: $x_o$ and $\dot{x}_o$

$x(t) = \frac{\dot{x}_o}{\omega_n} \sin \omega_n t + x_o \cos \omega_n t$

and in terms of amplitude and phase,

$x(t) = C \sin(\omega_n t + \phi)$

where,

$$C = \sqrt{\left(\frac{\dot{x}_o}{\omega_n}\right)^2 + x_o^2}$$

$$\phi = \tan^{-1}\left(\frac{x_o \omega_n}{\dot{x}_o}\right)$$

A single degree-of-freedom system experiencing free vibration with damping is shown in Figure 4 and defined by the following equation of motion:

$$m\ddot{x} + c\dot{x} + kx = 0$$

**Figure 4: Damped single degree-of-freedom system**

The critical damping coefficient and damping ratio are defined, respectively, as follows:

$$c_c = 2\sqrt{km} = 2m\omega_n$$

$$\zeta = \frac{c}{c_c}$$
The damped natural frequency is defined by the following equation:

\[ \omega_d = \omega_n \sqrt{1 - \zeta^2} \]

The response of the system depends on the damping ratio (see Figure 2 above). The system is underdamped if the damping ratio is less than one. The system is critically damped if the damping ratio is equal to one, and the system in overdamped if the damping ratio is greater than one.

Many vibrating systems experience forced vibration. The equation of motion for an undamped, single degree-of-freedom system experiencing a sinusoidal force \( F = F_0 \sin \omega t \) is

\[ m\ddot{x} + kx = F_0 \sin \omega t \]

The solution to this system has a homogeneous (transient) part and a particular (steady state) part. The homogeneous part depends on the initial conditions, and the particular part depends on the forcing function.

Initial Conditions: \( x_0 \) and \( \dot{x}_0 \)

\[ x(t) = A \sin \omega_n t + B \cos \omega_n t + \frac{F_0}{k} \frac{1 - \omega^2}{\omega_n^2} \sin \omega t \]

where,

\[ A = \frac{\dot{x}_0}{\omega_n} + \frac{\omega F_0 / \omega_n k}{1 - \omega^2 / \omega_n^2} \]
\[ B = \dot{x}_0 \]
\[ \omega_n = \sqrt{k / m} \]

The oscillation at the natural frequency eventually dies out in actual systems with damping, so the first two terms in the solution go to zero. The equation of motion for a
damped, single degree-of-freedom system experiencing a sinusoidal force \( F = F_0 \sin \omega t \) is the following:

\[ m\ddot{x} + c \dot{x} + kx = F_0 \sin \omega t \]

The response can be written as

\[ \frac{x}{F_0 / k} = \frac{\sin(\omega t - \phi)}{\sqrt{(1 - \omega^2 / \omega_n^2)^2 + (2\zeta \omega / \omega_n)^2}} = R_d \sin(\omega t - \phi) \]

where,

\[ \phi = \tan^{-1}\left(\frac{2\zeta \omega / \omega_n}{1 - \omega^2 / \omega_n^2}\right) \]

The factor \( R_d \) is a dimensionless response factor and can be plotted as shown in Figure 2.

2.1.2 Multiple Degree-of-Freedom Systems

Systems that consist of multiple masses connected together by springs and/or dampers are considered to be multiple degree-of-freedom systems. The equations of motion, for an \( n \) degree of freedom damped system, can be represented in matrix form as follows:

\[ [m][\ddot{x}] + [c][\dot{x}] + [k][x] = \vec{F} \]

where,

\([m], [c], \) and \([k]\) are the mass, damping, and stiffness matrices, respectively, and are defined based on the particular system

\( \ddot{x}, \dot{x}, x, \) and \( \vec{F} \) are the acceleration, velocity, displacement, and force vectors, respectively, and each contains \( n \) terms

The equations of motion can be solved through simulation by converting the \( m \)-th order system to \( m \) first order equations. This is known as state space form, which can be represented in matrix form and solved, given the initial conditions:
\[ \dot{z} = [A]z + [B] \]

where,
the eigenvalues of \([A]\) represent the roots of the characteristic equation
\([B]\) is the input matrix

The natural frequencies (eigenvalues) come from solving for the roots of the A-matrix. For a multiple degree-of-freedom system it quickly becomes most efficient to solve for the natural frequencies through the use of a computer.

2.2 Signal Processing
Signal processing accomplishes the transformation between the time domain and the frequency domain. Therefore, time-based data from a transducer can be converted to the frequency domain. This allows for the identification of the natural frequencies from the system data.

2.2.1 Theory
Signal processing characterizes the signals. There are three types of signals, periodic, aperiodic and random. An example of a periodic signal is shown in Figure 5.

**Figure 5: Example of a Periodic Signal**

For a periodic signal, \(x(t + T) = x(t)\) for some \(T > 0\) at each instant in time for all time. For practical purposes, though, some signals can be assumed to be periodic for a defined window in time. This describes the case for a valve train system.

An example of an aperiodic signal is shown in Figure 6.
An aperiodic signal is not periodic, but it can be described by a mathematical function, for example $x(t) = t$, meaning that it is deterministic. An example of a random signal is shown in Figure 7.

A random signal is neither periodic nor deterministic, but it does have statistical properties. This signal is collected data. Examples include a coin toss, the stock market, or the outside temperature.

A Fourier series is a representation of a periodic function with an infinite series:

$$x(t) = \frac{A_0}{2} + \sum_{n=1}^{\infty} \left[ A_n \cos(2\pi f_n t) + B_n \sin(2\pi f_n t) \right]$$

The variable $f_n$ is a whole number multiple of the fundamental frequency of the periodic function. A Fourier series is a sum of an average value, a fundamental wave, and higher harmonics. The higher harmonics are added to correct the fundamental wave to make it more accurately represent the actual function. A Fourier analysis is the process of
finding the right combinations of the three components listed above. It is completed by determining the proper Fourier coefficients, $A_o$, $A_n$ and $B_n$. The Fourier coefficients are defined by the following equations:

\[
A_n = \frac{2}{T} \int_0^T x(t) \cos(2\pi f_n t) dt \\
B_n = \frac{2}{T} \int_0^T x(t) \sin(2\pi f_n t) dt
\]

The coefficients, $A_n$, $B_n$ or both can sometimes be zero, depending upon the function being represented. The coefficients can be combined to get an amplitude and a phase at each value of frequency.

\[
C_n = \sqrt{A_n^2 + B_n^2} \\
\phi_n = \tan^{-1}\left(\frac{B_n}{A_n}\right)
\]

\[
x(t) = \sum_{n=0}^{\infty} C_n \cos(2\pi f_n t - \phi_n)
\]

These amplitudes and phases may be plotted versus frequency to give frequency domain amplitude and phase spectrums. Figure 8 below depicts examples of these spectrums.

**Figure 8: Examples of Frequency Domain Amplitude and Phase Spectrums**
The transformation from the time domain to the frequency domain can be made with the Fourier transform:

\[ X(f) = \int_{-\infty}^{\infty} x(t)e^{-j2\pi ft} dt \]

Experimental data which is collected at a particular sampling rate can be transformed from the time domain to the frequency domain using the discrete Fourier transform (DFT):

\[ X(n\Delta f) = \Delta t \sum_{k=0}^{N-1} x(k\Delta t)e^{-j2\pi kn/N} \]

The \( X(n\Delta f) \) terms are the Fourier coefficients. The sampling rate must be greater than the highest frequency expected, if that frequency is to be perceived. The fast Fourier transform (FFT) gives the same result as the DFT, but it is much quicker.

2.2.2 Examples

In the frequency analysis of valve train system data, the following two issues are of interest. First, how do the frequency analysis results of linear and nonlinear systems differ? Second, what effect do natural frequencies that are close together have on the frequency analysis results? As will be discussed in more detail in the next section, the valve train system is a nonlinear system. The frequency domain amplitude spectrum for a linear system exhibits sharp "spikes" at each frequency of the periodic function, as long as the frequencies are not too close together. A periodic function has more than one frequency when it is the combination of more than one periodic functions with differing frequencies.

What does the frequency domain amplitude spectrum of a nonlinear system look like though? To answer this question, data is generated with sine functions with constant and
non-constant frequencies. As this discussion is focused on valve train systems, the generated data is somewhat representative of the acceleration of the engine valve (see the acceleration curve at 1000 cam rpm in Appendix A). The main exception to this is that a sine function is used to represent only the oscillations during the lift event, not the shape of the acceleration curve. The lift events shown in Appendix A are approximately 130 degrees in duration, and at 1000 cam rpm there are approximately 30 oscillations in the acceleration curve. The period of the event is calculated based on the cam speed of 1000 cam rpm:

\[
130 \text{ deg} \times \frac{\text{rev}}{360 \text{ deg}} \times \frac{\text{min}}{1000 \text{ rev}} \times \frac{60 \text{ sec}}{\text{min}} = 0.0217 \text{ sec}
\]

The average frequency is calculated as follows:

\[
\frac{30 \text{ cycles}}{0.0217 \text{ sec}} = 1384.615 \text{ Hz}
\]

This average frequency was rounded to 1385 Hz for this evaluation. Microsoft Excel was used to generate the data and perform the Fourier Analysis by FFT. The number of data points must be a power of two. Analyses with various numbers of data points were completed to give sufficient resolution to the sine function(s), and 512 data points were selected. As the period of the valve event was 0.0217 sec, the frequency resolution of the FFT was 1rev/0.0217sec, or 46.06 Hz. Three sine functions (one linear and two nonlinear) were analyzed: constant frequency of 1385 Hz, linearly ramping frequency between 1285 Hz and 1485 Hz, and linearly ramping frequency between 1185 Hz and 1585 Hz.

A comparison of the frequency domain amplitude spectrums of these three sine functions is shown in Figure 9 below. For the period evaluated, the frequency range of the FFT is
actually 0 to about 11,800 Hz, but since the functions evaluated have no frequencies as high as this, the plot below is only from 0 to 6000 Hz.

Figure 9 Frequency Domain Amplitude Spectrums of Linear and Nonlinear Systems

The FFT of the constant frequency sine function results in a relatively sharp, tall spike near 1385 Hz. The FFT of the sine function with the frequency linearly ramping from 1285 Hz up to 1485 Hz and then back down to 1285 Hz results in a much lower amplitude frequency response, which is spread out over a larger range of frequencies. This range of frequencies is even wider than the sine function frequency range of 1285 to 1485 Hz, and it is shifted toward the lower end of the frequency scale relative to 1385 Hz, even though the average of the sine function frequency range is 1385 Hz. Although not shown in Figure 9, the FFT of a sine function with the frequency linearly ramping from 1485 Hz down to 1285 Hz and then back up to 1485 Hz results in a frequency range that is shifter toward the upper end of the frequency scale relative to 1385 Hz.

Over the 0.0217 sec period, the constant 1385 Hz sine function results in almost exactly 30 cycles. The sine function with the frequency first ramping up and then down results
in approximately 27.75 cycles. The sine function with the frequency first ramping down and then up results in approximately 32.25 cycles. As the generated data is meant to be representative of the valve acceleration during a single 130 degree lift event at 1000 cam rpm, the variation from 30 cycles and a non-whole number of cycles is representative of what the actual valve acceleration event would appear like under these same conditions. So in this case a leakage error in the FFT (due to a non-whole number of cycles) is not a concern, as the periodic signal is not an individual sine wave, but all of the sine waves together representing the oscillation of the valve, from valve opening to valve closing.

The FFT of the sine function with the frequency linearly ramping from 1185 Hz up to 1585 Hz and then back down to 1185 Hz results in an even lower amplitude frequency response, which is spread out over an even larger range of frequencies. This range of frequencies is again wider than the sine function frequency range of 1185 to 1585 Hz, and it is shifted even further toward the lower end of the frequency scale.

In summary, it appears that compared to a linear system, the FFT of a nonlinear system has a lower amplitude frequency response which is spread out over a wider range of frequencies, even wider than the range of frequencies of the system. Depending on the nature of the frequency variation (increasing, decreasing, etc.), the frequency domain amplitude spectrum may be shifted along the frequency scale. In other words, the FFT of a nonlinear system is not as meaningful as that of a linear system. In comparing the frequency domain amplitude spectrums of two nonlinear systems, the greater the frequency variation the lower the amplitude frequency response is, the wider the frequency range is, and the more shifted the frequency range is.

In addition to being non-linear, valve train systems often have natural frequencies that are close to one another. For example, the second and/or third harmonics of the valve spring and the system natural frequency. Again, data is generated to evaluate this scenario.
Two sine functions, with different frequencies, were added together and the result was divided by two. The division by two was performed to keep the amplitude of the resultant function at approximately the same level as the functions evaluated and presented in Figure 9 above. The difference in the frequencies of each of the two sine functions was evaluated at three levels.

A comparison of the frequency domain amplitude spectrums of these three functions is shown in Figure 10 below. For the period evaluated, the frequency range of the FFT is actually 0 to about 11,800 Hz, but since the functions evaluated have no frequencies as high as this, the plot below is only from 0 to 6000 Hz.

Figure 10a Frequency Domain Amplitude Spectrum of a System with Two Frequencies (1000 & 1385 Hz)
Figure 10b  Frequency Domain Amplitude Spectrum of a System with Two Frequencies (1300 & 1385 Hz)

In Figure 10a, the FFT of the function results in separate, sharp, tall spikes near 1000 and 1385 Hz. In Figure 10b, the FFT of the function results in partially separate, sharp, tall
spikes near 1300 and 1385 Hz, but the spikes have begun to merge together at their bases. In Figure 10c, the FFT of the function results in a single, wider spike spanning 1335 to 1385 Hz. Recall that, based on the period of 0.0217 sec, the frequency resolution of the FFT is 46.06 Hz. So in Figure 10b the spikes are two frequency increments apart, and in Figure 10c the spikes are only one frequency increment apart.

The data generated here considers the amplitude of the vibration at the two different frequencies to be of the same magnitude. In the actual valve train system, this is not the case. The different modes of vibration are of different magnitudes, depending on the input with which the system is excited. This affects the results of the frequency analysis of two natural frequencies that are close together. In summary, the closer that two natural frequencies of a valve train system are to one another, the more difficult it is to distinguish between them, especially depending upon the frequency resolution of the FFT. Although not shown here, one can easily imagine that by combining what is shown in Figure 9 for a nonlinear system with what is shown in Figure 10 for two nearby natural frequencies, it becomes even more difficult to differentiate between two nearby natural frequencies.

2.3 Valve Train Systems

As a valve train system consists of multiple components connected together (see Section 3.1) and exhibits more than one natural frequency, it is a multiple degree-of-freedom system. The valve train experiences forced vibration. The forcing function is the cam profile. Although there is some variation from cycle to cycle and although the valve closes against its seat each revolution, the operation of a valve train can be assumed to be periodic.

As the valve train system translates and rotates through one lift event cycle, the natural frequencies of the valve train are varying. This is a result of the varying geometry of the valve train system during the lift event, resulting in varying effective masses, stiffnesses
and damping factors. This makes the valve train a nonlinear system. Modeling a nonlinear vibrating system is much more complicated than modeling a linear system.

As the valve springs of the valve train are typically non-linear, their stiffness values increase as they are compressed. This is because, as the spring compresses, some of the coils collapse on one another, changing the number of active coils. Additionally, some component effective masses, stiffnesses, and damping factors vary as the points at which the component is loaded and supported vary. Referencing the Type I valve train shown in Figure 1, the eccentricity at which the cam lobe contacts the lifter foot varies from zero to some maximum value as the cam rotates. Similar varying contact points occur as rocker arms slide against valve tips, cam lobes slide along rocker arms, rocker arms rotate about lifters, etc. The explanation of the varying stiffness of valve train components can be simplified and compared to a simply supported beam where the locations of the supports are varying. It can be easily understood that varying the locations of the supports of a beam varies the stiffness of the beam.

2.4 Natural Frequency Measurement Methods
Several natural frequency measurement methods have been proposed and evaluated to varying degrees in the past. The Manual method was used in the past and was a quick method. The Step Input method was briefly evaluated and provided some concerns because the natural frequencies were measured with the system in a stationary state, but it was actually a non-stationary system. The waterfall plot was used with success in the past, but required further development.

2.4.1 Manual Method
Using this method, the system natural frequency is determined by manually counting the number of cycles displayed in an acceleration trace over a given period of time. The acceleration trace (see Figure 11 below) comes from the accelerometer attached to the valve with the system running at a given cam rotation speed.
Typically, the oscillation count excludes opening, closing, maximum acceleration, and around the nose (maximum lift). If the abscissa coordinate represents cam rotation degrees, as is standard for dynamics testing output plots, cam degrees are converted to time by knowing the cam rotation speed at which the acceleration data was collected. Manually counting the number of oscillations to the nearest whole or half cycle results in an approximate estimate of the system natural frequency, but it is not precise enough to allow valid comparison of two systems.

**Figure 11 Example of an Acceleration Trace**

for Determining System Natural Frequency

![Graph](image)

Although this method is a quick estimate of the general system natural frequency, it can result in a significant error compared to what is considered to be a significant difference in system natural frequency. For example, given a system which actually has a system natural frequency of 700 Hz and is running at 1500 cam rpm, a one-quarter cycle error in the cycle count results in a predicted system natural frequency of 756 Hz. This result is 56 Hz higher than the actual system natural frequency, which is considered to be a significant difference when comparing two systems of this system natural frequency level.

22
Two different engineers can come up with significantly different estimates of the system natural frequency by selecting different cycles within the lift event to evaluate and/or by measurement error in the duration of selected cycle(s). Additionally, determining system natural frequency for one isolated section of the lift event is inaccurate, as the system natural frequency is changing over the lift event because the system geometry and therefore effective masses, stiffnesses, and damping factors are non-constant (nonlinear system). This is discussed in more detail in Section 2.3.

Additionally, if this method is used to determine the system natural frequency from an acceleration trace at only one speed it may give an incorrect result. This is because the forcing frequencies, which vary with speed ($\omega = \text{harmonic number} \times \text{rpm}$), show up in the response. A forcing frequency near the system natural frequency looks like the system natural frequency.

2.4.2 Step Input Method

Using this method, the natural frequencies are determined by loading the valve train system by pulling on the valve, which is held off of the seat on the base circle, and suddenly releasing the load, applying a step input load to the system. The system is allowed to freely vibrate with the valve off of the seat. The natural frequencies are determined from the acceleration data from an accelerometer attached to the valve, which is more sensitive than the ones used for typical valve train dynamic testing, and possibly from a strain gage on the valve spring. This is done by performing a frequency analysis of the accelerometer and strain gage data.

To characterize the natural frequencies as they change with changing system geometry (i.e., changing effective masses, stiffnesses, and damping factors), the cam can be set at various points in the lift event which span the geometry of the system. At each of these points, a step input is given to the system and the natural frequencies are determined.
from the valve acceleration and valve spring strain gage data as described in the above paragraph. The natural frequencies can then be reported as a range.

As mentioned above, there is some concern that determining the natural frequencies of a stationary system will not adequately represent the non-stationary system. One example of the difference between the stationary and non-stationary systems is the absence of the oil film between the cam lobe and the follower in the stationary system. This oil film is generated as the system is running. Additionally, the natural frequency associated with the camshaft torsion cannot be determined using this method, as the camshaft is not rotating during this test. Additional detail on this method is included in Chapter 4, as a significant amount of testing has been completed using this method.

2.4.3 Waterfall Plot Method

Using this method, the natural frequencies are determined by running the system over a range of cam speeds, performing a frequency analysis (FFT) on the valve acceleration data, and overlaying the results at various cam speeds on a frequency versus amplitude plot, a waterfall plot (see Figure 12 below).

Often, all three of the most important natural frequencies of the valve train (system (fundamental mode of vibration of entire linkage), cam torsion, and valve spring (base natural frequency)) are observed in the waterfall plot. The spike representing the system natural frequency appears as the largest amplitude and densest spike. This technique seems to work better for low natural frequency valve train systems, compared to the higher natural frequency systems, as will be discussed further below.

The question, though, is how to precisely and accurately determine the natural frequencies from the experimental data. As mentioned several times earlier, how can the natural frequencies be effectively characterized with single values when the system geometry (effective masses, stiffnesses and damping factors), and therefore the natural
frequencies, are varying over the lift event? Some possibilities are to automatically evaluate (via a computer program) the waterfall plot data to determine ranges of natural frequencies which characterize the system or to determine mean and standard deviation values to describe the natural frequencies.

**Figure 12a Waterfall Plot of a High System Natural Frequency Valve Train (Typically 1200 to 1500 Hz)**

![Waterfall Plot of a High System Natural Frequency Valve Train (Typically 1200 to 1500 Hz)](image1)

**Figure 12b Waterfall Plot of a Low System Natural Frequency Valve Train (Typically 400 to 700 Hz)**

![Waterfall Plot of a Low System Natural Frequency Valve Train (Typically 400 to 700 Hz)](image2)
For systems with higher system natural frequencies such as a Type II system, the peak in the waterfall plot representing the system natural frequency is found to not be so sharp, is lower in amplitude, and is wider compared to a system with a lower system natural frequency such as a Type V system (see Figure 12 above). Therefore, the systems with higher system natural frequencies would have wider ranges of natural frequencies with larger standard deviations.

Why does this variation exist between high and low system natural frequency systems? One explanation for the difference in amplitudes is that the lower system natural frequency system with higher amplitude vibrations has a higher amplitude frequency response. This is because the frequency content of the input that excites the lower natural frequency system is larger in amplitude. The amplitude of the frequency content decreases with increasing frequency. For example, at a cam speed of 600 rpm (10 Hz) the 70th harmonic of the cam profile excites a valve train with a 700 Hz system natural frequency (10 Hz x 70th harmonic). While the 140th harmonic of the cam profile excites a valve train with a 1400 Hz system natural frequency (10 Hz x 140th harmonic). The amplitude of the 70th harmonic is larger than that of the 140th harmonic.

Another explanation for the difference in the amplitude of the frequency response, as well as the spread of the frequency response, is taken from the evaluation presented in section 2.2.2 of linear versus nonlinear systems. Referencing Figure 9, it is shown that one system that is more nonlinear than another system (i.e., has greater system natural frequency variation in this case) has a frequency domain amplitude spectrum that is lower in amplitude and wider in frequency range than the less nonlinear system.

The valve train systems for which data is presented in Figure 12 have both been modeled with VTS (dynamic valve train simulation). For the valve train system in Figure 12a, the Premium V6 left side cylinder head intake system (Type II), the data presented is for the alpha level roller finger follower1. With the beta level roller finger follower, VTS
predicts that the system natural frequency ranges from 1525 Hz at valve opening to 1620 Hz at maximum lift, a range of 95 Hz$^3$. For the valve train system in Figure 12b, the XTC intake system (Type V), VTS predicts that the system natural frequency ranges from 720 Hz at valve opening to 685 Hz at maximum lift, a range of 35 Hz$^4$. This supports the statement in the above paragraph that a more nonlinear system (Figure 12a) has a frequency amplitude spectrum that is lower in amplitude and wider in frequency range than a less nonlinear system (Figure 12b).
Chapter 3

SYSTEM MODELING

3.1 Overhead Cam Finger Follower System Model

Chapter 1 discusses the most common types of Valve Train systems. The remainder of this discussion will focus on the Type II Valve Train, an Overhead Cam Finger Follower system, for discussion of both system modeling and experimental results. The experimental methods suggested for use to determine natural frequencies can be used, though, for any type of Valve Train.

Figure 13 below is a diagram of the actual Valve Train system. Referencing Figure 13, there is a cam lobe for each valve position. The cam lobes are all part of a camshaft, which is driven by the crankshaft through a drive system. The circular portion of the cam lobe is known as the base circle. When the base circle contacts the roller of the finger follower, the valve is closed. The eccentric portion of the cam lobe is known as the lift event, and the finger follower is a special type of rocker arm. During the lift event the valve is opening and closing. As shown in Figure 13, when the eccentric portion of the lobe begins to contact the roller, the finger follower begins to rotate downward in a clockwise fashion. The finger follower pivots about the hydraulic lash adjuster. Since the finger follower is in contact with the valve, when the finger follower moves downward it forces the valve open. The most eccentric point on the cam lobe is called the nose and corresponds with maximum lift of the cam. When the cam is contacting the roller at the nose, the valve is very near or at its maximum open position. Beyond the nose of the cam, the finger follower begins to rotate back toward its initial position in a counterclockwise direction. As a result of this, the valve spring is able to force the valve closed.
Figure 13 Overhead Cam Finger Follower Valve Train
System Diagram

- Cam Lobe
- Roller Finger Follower
- Oil Gallery
- Hydraulic Lash Adjuster
- Cylinder Head
- Valve Head
- Key (2)
- Spring Retainer
- Valve Stem
- Valve Spring
- Valve Guide
- Valve Seat
The valve spring and the valve are connected by the spring retainer or cap and a pair of keys, which lock the retainer and valve together under the installed load of the valve spring. The head of the valve closes against a hardened steel valve seat, whose geometry mates with the valve head to ensure a tight seal under the installed load of the valve spring. The installed load of the valve spring is sufficiently large enough to seal the valve against the valve seat during the compression and power strokes of the engine operating cycle. The valve stem is guided very closely by the valve guide to help ensure precise opening and closing movement. To ensure a close geometry relationship between the valve guide and valve seat, for proper movement and sealing of the valve, these components are finish machined relative to one another after they are installed in the cylinder head.

A more detailed diagram of the hydraulic lash adjuster (HLA) is shown below in Figure 14. The HLA automatically expands or contracts to assure no lash, or clearance, is in the system and to assure that the valve is able to seat. This is necessary to accommodate for manufacturing tolerances, wear, and thermal expansion of all of the valve train components.

The inner part of the HLA is called the plunger, and the outer part is called the body. They are two separate pieces, and the outer diameter of the plunger and the inner diameter of the body are precisely ground. The two components are matched together to give a certain very tight clearance. The reason for this will be described later. Pressurized oil is fed from the oil gallery through holes in the body and plunger to the inside of the plunger, which is called the low pressure chamber. The volume above the inside of the body bottom and below the outside of the plunger bottom is called the high pressure chamber. The plunger spring, which sits in the bottom of the body pushes the plunger upward to assure no lash in the system over the operating life of the engine. Normally the ball, which is held against a seat on the bottom of the plunger by the ball
spring, is closed. The ball spring is supported by the ball retainer, which presses onto the plunger.

Figure 14 Hydraulic Lash Adjuster Diagram
When the plunger spring pushes the plunger upward relative to the body, a lower oil pressure is created in the high pressure chamber, due to the increasing volume, as compared to the oil pressure in the low pressure chamber. Above a particular level of pressure differential the ball will be forced open against the ball spring, allowing oil to flow into the high pressure chamber until the oil pressures in both chambers are equal. With the exception of the abnormal case where there is separation between the valve train components during operation, this expansion of the HLA will occur during the base circle event. This is because during the lift event there will be valve spring and inertia loads on the plunger. These loads will be supported by the column of oil in the high pressure chamber, causing slight compression of the oil and a large increase in the oil pressure in the high pressure chamber. The high pressure in the high pressure chamber will act to keep the ball closed.

As mentioned above, there is a certain very tight clearance between the body inner diameter and the plunger outer diameter. This tight clearance is held over a certain length called the leakdown land. Together, the clearance and the leakdown land control how much oil will leak out of the high pressure chamber between the body and plunger. Leakdown will occur under three different conditions: initial installation if there is oil in the high pressure chamber, during the lift event, and over time due to valve seat recession. Before the HLA is installed in the engine, the plunger spring will push the plunger to its full extended position against the plunger retainer. At installation, the HLA is placed in its bore, the finger follower is placed on top of the HLA and valve tip, the camshaft is placed on top of the finger followers, and bearing caps are placed on top of the cam bearings and bolted down. As the valve train system is designed to nominally operate near the center of the HLA adjustment range, the HLA plunger will have to adjust downward. If there is oil in the high pressure chamber, the HLA will cause the valve to be held off of its seat, and the HLA plunger will see a load due to the valve
spring. This load will cause the HLA to begin to leakdown to the position where the valve will be able to seat. During each lift event during operation of the engine, the plunger will be loaded as described above. This loading will cause a slight amount of leakdown of the HLA. During the base circle event, the HLA can recover. Over time the valve seat will recess due to wear. Considering this type of wear only, the HLA would have to collapse slightly to allow the valve to seat. In actuality, wear on other valve train components, combined with valve seat recession, will require the HLA to expand.

The other types of Valve Trains have many similarities to the Overhead Cam Finger Follower Valve Train. This includes rotational motion of the cam being converted to linear translation of the valve through the other components of the valve train and operation of the hydraulic element, when one is present. Now that the operation of the Overhead Cam Finger Follower Valve Train has been covered, the system model is presented in Figure 15 below. Variable definitions are included in Table 2 following the figures.

As described in Section 1.1, an internally developed computer program called VTS is used for dynamic modeling of Valve Train systems. VTS is a lumped parameter model, which takes into account system nonlinearities. The program consists of Core Routines, Sub-models, Connections and Post-Processors. Each Valve Train component is represented by a sub-model, for example, the camshaft, the valve spring and the hydraulic element. A global valve train model is created by the core routine, which joins the appropriate sub-models using the connection routines. The connections are piece-wise linear as a function of whether or not there is contact at the connection. The resulting differential equations of the model are solved over a specified time interval, and the results are organized by the post-processors into graphs or reports.
Figure 15a Overhead Cam Finger Follower System Model

Front View
Figure 15b Overhead Cam Finger Follower System Model

Side View - Camshaft Only

![Diagram of overhead cam finger follower system model](image)

Table 2

<table>
<thead>
<tr>
<th>Variable</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>CK(1,2,2)</td>
<td>Effective stiffness of cam lobe and finger follower</td>
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<tr>
<td>CC(1,2,2)</td>
<td>Damping ratio associated with CK(1,2,2)</td>
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<tr>
<td>ROCMAS</td>
<td>Mass of finger follower</td>
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<td>SFRI</td>
<td>Rotational Inertia of finger follower wrt pivot of follower</td>
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<td>CK(1,2,3)</td>
<td>Stiffness, finger follower to valve tip</td>
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<tr>
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<tr>
<td>VMS</td>
<td>Mass of valve stem, keepers and cap</td>
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<tr>
<td>VSC</td>
<td>Damping ratio between valve stem and valve guide</td>
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<td>VK</td>
<td>Stiffness of valve stem</td>
</tr>
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<td>VZ</td>
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<td>VMH</td>
<td>Mass of valve head</td>
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<td>VKSEAT</td>
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<td>VZSEAT</td>
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<td>SPNGMI(i,n)</td>
<td>Mass of the individual spring nodes</td>
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<td>SPNGKI(i,3,n)</td>
<td>Coil stiffness between the $i^{th}$ and $(i-1)^{th}$ nodes; the sum of these stiffnesses in series is the overall spring stiffness</td>
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<tr>
<td>SPNGZI(i,3,n)</td>
<td>Coil damping ratio for the $i^{th}$ node</td>
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<tr>
<td>SPNGKI(i,2,n)</td>
<td>Coil clash stiffness between the $i^{th}$ and $(i-1)^{(#}$ of modeling nodes per coil when there is node contact</td>
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<td>Coil clash damping ratio for the $i^{th}$ node when there is contact between the nodes</td>
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<td>CK(1,2,4)</td>
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<tr>
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<td>Damping ratio associated with CK(1,2,4)</td>
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<td>AM2</td>
<td>Mass of HLA plunger</td>
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<td>AK12T</td>
<td>Stiffness, plunger to retainer, fully extended</td>
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<td>PSR</td>
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<td>PSZ</td>
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<td>HLA foot stiffness</td>
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<td>Viscous damping coefficient between each inertia element and ground</td>
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<td>CC(1,1,1)</td>
<td>Damping ratio associated with CK(1,1,1)</td>
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### 3.2 Valve Train Dynamic Simulation Results

The VTS program is currently designed to be run at one cam speed at a time. The program is run multiple times to evaluate the cam speed range of interest, typically around 1000 to 3500 cam rpm in increments of 500 cam rpm. Cam rpm is the rotational speed at which the camshaft is turning and is one-half the rotational speed of the engine crankshaft. As the cam speed is increased, the inertia loads are obviously increasing. While VTS can output dozens of operating parameters, two of the typical parameters that are plotted and reviewed are valve lift and valve acceleration. Natural frequencies (eigenvalues), which are linearized for one point in time, are also an output of VTS that
are typically reviewed. Along with the dynamic simulation results, kinematic behavior and/or test data may be overlaid for comparison. A set of simulation results, along with test and kinematic data, for an overhead cam finger follower valve train can be found in Appendix A.3.

As mentioned above, the natural frequencies, or eigenvalues, calculated by VTS are for a linearized system at specific points in time. This is typically done at two cam positions, maximum lift and valve opening (near base circle). These two positions are typically the extremes for system geometry variation (i.e., effective masses, stiffnesses and damping factors), thus capturing the range of the natural frequencies. The finger follower is one of the components with varying geometry. For example, the finger follower stiffness is input as a linear value at the cam-follower interface. As the contact points on the finger follower change throughout the lift event, the moment arm, and thus the finger follower effective stiffness, changes. The stiffness of the valve spring also changes throughout the lift event, with an increase in stiffness from the base circle installed position to the maximum lift position.

The system natural frequency, the base valve spring natural frequency and the lower harmonics of the spring (2nd to 4th), and the camshaft torsional natural frequency are typically selected from the resultant eigenvalues and reported. Ranges or average values may be reported. Selection of the system natural frequency from all of the natural frequencies is sometimes difficult, as it is often close to one of the valve spring harmonics. Typical expected values for the system natural frequency of a particular type of valve train are known (see Table 1 in Chapter 1). The valve spring harmonics are approximately multiples of the base valve spring natural frequency, which is often known from calculation. If it is still not apparent which natural frequency corresponds to the system natural frequency, the system natural frequency can be identified by increasing the combined cam-follower stiffness and re-running the model. Of the components that significantly affect the system natural frequency, the combined value
used for the cam-follower typically has the lowest stiffness value and, therefore, has the biggest affect on the system natural frequency. The system natural frequency is the one that has the largest sensitivity to an increase in the cam-follower stiffness. So the change in the cam-follower stiffness allows the system natural frequency to be identified.

The sample simulation results presented in Appendix A are for an intake position of the left side cylinder head of the Premium V6 application used for the experimental analysis presented in Chapter 4. The experimental evaluations presented in Chapter 4 are for exhaust positions on both the left and right side cylinder heads. There are geometry variations between both the intake and exhaust positions and the left and right side cylinder heads. To date all simulation work on the application tested has been done only for an intake position, but on both the right and left side cylinder heads. Although the geometry variations between the intake and exhaust positions cause at least slight differences in the natural frequencies, the simulation natural frequencies are presented below in Tables 3 and 4 for later comparison to the testing results. As the VTS model is correlated to the waterfall plot natural frequency results for an intake position of the left side cylinder head, the above mentioned comparison gives a general idea of whether or not the step input test method presented in Chapter 4 gives believable results. As the step input test does not provide the cam torsion natural frequency, which tends to cause a simulated increase in system natural frequency, the natural frequencies presented in Tables 3 and 4 are without cam torsion included in the model.

<table>
<thead>
<tr>
<th>Natural Frequency</th>
<th>Cam Position: -50° (near base circle)</th>
<th>Cam Position: 0° (maximum lift)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Valve Spring</td>
<td>471</td>
<td>617</td>
</tr>
<tr>
<td>Spring 2nd Harmonic</td>
<td>940</td>
<td>1206</td>
</tr>
<tr>
<td>System</td>
<td>1525</td>
<td>1620</td>
</tr>
</tbody>
</table>
Table 4 Simulated Natural Frequencies for an Intake Position of the Right Side Cylinder Head (with Valve Spring P/N 12553351EB)\textsuperscript{5}

<table>
<thead>
<tr>
<th>Natural Frequency</th>
<th>Cam Position: -50° (near base circle)</th>
<th>Cam Position: 0° (maximum lift)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Valve Spring</td>
<td>516</td>
<td>668</td>
</tr>
<tr>
<td>Spring 2\textsuperscript{nd} Harmonic</td>
<td>1029</td>
<td>1301</td>
</tr>
<tr>
<td>System</td>
<td>1647</td>
<td>1598</td>
</tr>
</tbody>
</table>
Chapter 4

EXPERIMENTAL ANALYSIS

4.1 Step Input Method

The objective here is to document the development of an experimental method to measure valve train natural frequencies. The method needs to account for valve train system nonlinearity, which is due to the changing valve train geometry throughout the lift event. The method also needs to take the subjectivity out of experimentally determining valve train natural frequencies.

The step input method for determining valve train natural frequencies has both advantages and disadvantages. One of the advantages is that the natural frequencies can be determined before all of the hardware is available and/or setup is complete for a running dynamics test. Another advantage is that the step input method allows the natural frequencies at specific points within the lift event to be measured. Those points can be selected to span the range of natural frequencies. One disadvantage is that the step input method measures natural frequencies of a stationary system. The effect of the actual non-stationary system on the natural frequencies is not completely known. Another disadvantage is that the measurements are completed with the hydraulic lash adjuster converted to a mechanical unit, which makes it stiffer and, therefore, not as representative of the actual system. Of the three measurement methods described earlier in this discussion, the least amount of work done has been on the step input method. For this reason and for the advantages listed above, the step input method has been selected as the focus for the experimental analysis section of this thesis project.
In the step input method, the valve of an assembled valve train is pulled on and released, allowing the system to vibrate freely. As the HLA is converted to a mechanical unit and shimmed to create a few thousandths clearance between the valve and seat (which is required to allow free vibration), there is initially a load at the valve tip which is very near the valve spring pre-load. There are contact loads between the other components in the system as well. When the valve is pulled, with a device that will be described later, the tension load that is applied directly subtracts from the load at the valve tip. If the tension load exceeds the pre-load of the valve spring, separation is created between the components. See Figure 16 below. When the tension load is released, the load at the valve tip will increase to the level of the original load, initially with some oscillation of the load as the components vibrate. Thus a step input load is applied to the valve train system in this experimental method.

Figure 16 Free Body Diagram for Valve/Spring Retainer Assembly

F = Valve Tip Contact Load
F_s = Valve Spring Pre-load
F_{so} = Initial Valve Spring Pre-load
T = Tension Load Applied to Valve

**Case 1:** \( T = 0 \)
\[ F_s = F_{so} \]
\[ F = F_s \]

**Case 2:** \( 0 < T < F_{so} \)
\[ F_s = F_{so} \]
\[ F + T = F_s \]

**Case 3:** \( T \geq F_{so} \)
\[ F_s > F_{so} \]
\[ T = F_s \]
\[ F = 0 \]
The resultant vibration is recorded with an accelerometer on the valve, and from this acceleration data the natural frequencies are determined. Additionally, a strain gaged valve spring may be used to determine the natural frequencies of the spring only. Prior to this investigation, a fixture was designed and fabricated for this type of testing. The fixture is designed to attach to the valve of nearly any type of valve train.

4.2 Test Setup and Data Analysis

The test fixture (Delphi-E part number RP184189) is shown on the following page in Figure 17. The adhesive accelerometer cage (Delphi-E part number RP184433) screws into a threaded hole in the valve where an accelerometer is normally installed for valve train dynamics testing. An adhesive accelerometer is attached to the cage using beeswax. Attachment of the accelerometer with epoxy is also evaluated here. The accelerometer used for this testing is Kistler model number 8720A500. The features of this accelerometer are high shock resistance, low impedance signals, and small size (4.9 g). The acceleration range of this accelerometer is ±500 g's. The frequency response within ±5% is 500 to 9000 Hz. Per the calibration certificate on the accelerometer case, the sensitivity of the accelerometer used is 9.75 mV/g. Other information on this accelerometer model can be found in Kistler Piezo-Instrumentation catalog.

The accelerometer cage is attached to the wire coupler (Delphi-E part number RP184186) through a cable. It is this cable that is cut to release the valve. Aircraft cable is used for this testing. Each end of the cable is double over and securely clamped to attach these two components together. The wire coupler threads into a load cell. The load cell used for this testing is Sensotec model number 34/1894-01, serial number 452615. Is has a capacity of 250 pounds in tension and a calibration factor of 2.0535 mV/V. It will be explained later how the conditioner box used with this load cell can be setup to directly read out in pounds.
The load cell is attached to the turnbuckle coupler (Delphi-E part number RP184434), which attaches through a threaded rod to the turnbuckle (Delphi-E part number RP184187). At the other end of the turnbuckle is an eyebolt through which the fixture can be anchored (to the test stand, for example) so that a load can be applied to the valve.

Figure 17 Step Input Test Fixture Drawing
A schematic of the valve train test fixture, including the step input fixture, is shown below in Figure 18.

**Figure 18 Valve Train Test Fixture Setup for Step Input Test**

In preparation for completing this test, the HLA is converted to a mechanical lifter by removing the plunger spring, ball retainer, ball and ball spring. These components are replaced by a solid shim, and the plunger is reinstalled in the body. The shim size is selected to result in the valve being held off of its seat by 0.003” to 0.005”. The selection
of the proper shim size is done by trial and error, and the clearance between the valve and seat is measured with feeler gages. The HLA is converted to a mechanical lifter, otherwise it would leak down over time, allowing the valve to seat. The HLA is shimmed to hold the valve off of the seat by 0.003” to 0.005” to allow the valve train system to vibrate freely because once the valve seats, the valve train becomes a different system. The valve spring manufacturer specifications are listed below in Table 5. The stiffness of the valve spring is actually non-linear.

<table>
<thead>
<tr>
<th>Part Number</th>
<th>12553351AA</th>
<th>12553351EB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring Load @ Base Circle (35.00mm) (N/lb)</td>
<td>220±11/49.5±2.5</td>
<td>222.4±11/50.0±2.5</td>
</tr>
<tr>
<td>Spring Load @ Max Lift (24.50 mm) (N/lb)</td>
<td>570±28/128.1±6.3</td>
<td>605±28/136.0±6.3</td>
</tr>
<tr>
<td>Average Measured Spring Rate (N/mm)</td>
<td>31</td>
<td>not measured</td>
</tr>
<tr>
<td>Spring Load @ Base Circle w/ XXX add’l compression (N/lb)</td>
<td>222.4±11/50.0±2.5 at 0.003”</td>
<td>not known</td>
</tr>
<tr>
<td>Spring Load @ Max Lift w/ XXX add’l compression (N/lb)</td>
<td>572.4±28/128.7±6.3 at 0.003”</td>
<td>not known</td>
</tr>
<tr>
<td>Natural Frequency at Base Circle (Hz)</td>
<td>447</td>
<td>525</td>
</tr>
<tr>
<td>Natural Frequency at Max Lift (Hz)</td>
<td>685</td>
<td>651</td>
</tr>
</tbody>
</table>

The next step is to reinstall the roller finger follower, the cam (if it was removed; in this case it was just tipped out of the way), and the cam bearing caps. The bearing cap bolts must be tightened down. The valve cover does not need to be installed, as the system will not be running so there will be no splashing oil. The cylinder head is rigidly mounted to a motored test stand. It is important that the cylinder head be rigidly mounted to assure valid test results. Unless otherwise noted for this testing, the part number of the HLA used is 17123524A. The serial number of the roller finger follower used is F193. The part numbers of the valve springs used are 12553351AA (color code yellow) and 12553351EB.
Next, the accelerometer cage is screwed into the valve, and the accelerometer is securely attached to the cage with beeswax. One end of a piece of cable is formed into a loop and clamped. This loop is fed onto the removable dowel pin of the cage, and the dowel is reinstalled. The other end of the cable is fed through a hole in the wire coupler, formed into a loop and clamped. The eye bolt, which is screwed into the far end of the turnbuckle is then securely fastened to the test stand. To do so, a piece of cable is fed through the eye bolt and then through the bottom of the frame of the test stand cover. The test stand cover is clamped to the test stand, and the ends of the cable are clamped together.

As mentioned above, the instrumentation for this test setup includes an accelerometer, a load cell, and possibly a strain gage for the valve spring. Additionally, a proximeter, which may already be installed in the cylinder head to collect valve lift data for dynamics testing, may used to monitor the motion of the valve during the test. The load cell is powered by a strain gage conditioner box, which also provides a display for the load cell output. The conditioner box used for this testing is Daytronic model number 3370 (serial number VT-AA-4 for this testing).

Before attaching the step input test fixture to the test stand, the conditioner box is setup to read directly out in pounds. This is useful, as the output of the load cell is used to determine how much load to apply to the valve. At initial setup and at the beginning of each test, the conditioner box must be allowed the warm up for fifteen minutes. A generic wiring diagram for the load cell is shown below in Figure 19.
In actuality, the power supply and display used for this test are combined in one unit. Labeled in Figure 19 are the wire colors that are used for this particular setup. A special connector that is already in existence is used to make the above described connection between the load cell and the conditioner box. Per the load cell calibration certificate, a 59 kΩ shunt resistor is used. Also per the calibration certificate, the shunt cal factor is 1.4801 mV/V, the calibration factor is 2.0535 mV/V, and the capacity is 250 pounds. The factor to dial in for the span to allow the display to read out pounds is calculated from the following equation:

\[
\frac{\text{Shunt Cal Factor}}{\text{Calibration Factor}} \times \text{Capacity} = \frac{1.4801 \text{mV/V}}{2.0535 \text{mV/V}} \times 250 \text{LB} = 180.19 \text{LB}
\]

The following steps are followed to setup the conditioner box:

1. Connect the red, black, green and white wires between the load cell and conditioner box.
2. Set the proper decimal place using the switches on the back of the conditioner box.
3. Zero the read out using the balance screws.
4. Install the shunt resistor between the black and green wires. A male-to-female banana connector, with the shunt resistor installed in it, is used to do this.

5. Dial in 180.19 using the balance screws.

6. Disconnect the shunt resistor, reconnect the black and green wires, and dial in zero using the balance screws.

7. Repeat steps 5 and 6 until both settings are met (0 and 180.19).

8. Hang a known weight from the load cell to verify the read out.

The accelerometer is connected through a cable to a Kistler model number 5004 dual mode amplifier (serial number VT-G-6 for this testing). The sensitivity factor of the accelerometer, which is listed in the equipment description above, is dialed into the amplifier. The amplifier output is set to 200 g’s/V. The output of the charge amplifier, as well as the output of the proximeter and/or strain gage conditioner box, are fed into a TEAC PCM Data Recorder, model number RD-180T, so that the data can be recorded and analyzed later. The outputs of the accelerometer, strain gage, and proximeter are also displayed on an oscilloscope for monitoring purposes. The data analysis is conducted with a Hewlett-Packard (HP) 3566A Dynamic Signal Analyzer. The time-based voltage data is converted to engineering units using the proper calibration factor for the accelerometer data. The valve spring strain gage calibration is not completed. So the data from the strain gage is left in voltage units, which is acceptable as only relative comparisons are required. A frequency analysis of the time-based data is completed so that the natural frequencies can be identified.

To actually conduct a test, the fixture is setup as shown in Figure 18. The turnbuckle is turned to apply tension to the system until the load cell display reads the desired value. The TEAC is turned on to record, and a verbal test description is recorded. The cable is then cut, and the TEAC is stopped. This is repeated for the desired number of test runs and conditions.
4.3 Test Conditions and Results

4.3.1 Trial One

For Trial One of data collection, the test fixture was setup on the right side Premium V6 cylinder head at exhaust position number six with valve spring part number 1255351AA. The cam was set at the base circle position, and two starting loads were tested. The first load was 50 pounds (less than the valve spring installed load as determined by checking for clearance between the components), and the second load was 60 pounds, which is greater than the valve spring installed load, resulting in clearance between the valve train components. Two runs were completed at 50 pounds and ten runs were completed at 60 pounds.

A frequency analysis of only the accelerometer data for ID# 4 (60 pound starting load) has been completed, and the time and frequency responses are shown in Figure 20 on the following page, where Figure 20a is the entire event, and Figure 20b is the ring down portion of the same event. The period of the event evaluated is 125 msec, resulting in an 8 Hz resolution of the FFT. The frequencies that correspond with the peaks, or natural frequencies, are noted on Figure 20b.

The first natural frequency, 480 Hz, corresponds with the natural frequency of the valve spring at the base circle condition. By calculation the base spring natural frequency is estimated to be 447 Hz, and one spring of this design level has been measured separately to have a base natural frequency of 470 Hz. With the exception of one frequency, all the rest of the frequencies, 928 Hz, 1400 Hz, 1848 Hz, 2344 Hz, and 2832 Hz, are approximately whole number multiples of the valve spring base natural frequency. This seems to indicate that these frequencies are the other modes of vibration of the spring, or the harmonics of the valve spring. The only frequency that is not a multiple of the valve spring base natural frequency is at 1600 Hz, and its amplitude is smaller than most of the other peaks. From previous dynamic testing of this system it has been determined, from
“eyeballing” the center of the peak on a waterfall plot, that the average system natural frequency is 1500 Hz.

At this point, it is hypothesized that applying a load to the valve which creates clearance between the valve train components imparts enough energy to the valve spring so that mainly the natural frequencies of the valve spring are being observed in the frequency analysis. Further testing at other conditions is conducted and reported in the following sections to further determine what the results of this test indicate.
Figure 20a Time and Frequency Responses for the Step Input Test, Trial One, ID#4, Base Circle, Start Load 60 lb.
Figure 20b  Time and Frequency Responses for the Step Input Test, Trial One, Ring Down, ID#4, Base Circle, Start Load 60 lb.
4.3.2 Trial Two

For Trial Two of data collection, the test fixture was setup on the left side Premium V6 cylinder head at exhaust position number six with valve spring part number 12553351AA, as the right side cylinder head had been removed from the test stand. The testing was completed at the base circle position and at the maximum lift position. The roller finger follower, the valve and the valve spring were positioned at the maximum lift condition by placing gage blocks equal to the maximum cam lift (5.90 mm) between the cam base circle and the roller of the roller finger follower. Gage blocks were used between the cam and roller instead of rotating the cam to the maximum lift position to avoid having to lock the cam in place. The cam would need to be locked in place to prevent movement during the test.

Three starting load conditions were tested at both the base circle and maximum lift conditions, 40, 50, and 60 lb., for a total of six test conditions. At the base circle condition, the 40 and 50 lb. starting loads were not large enough to overcome the installed valve spring load and create clearance between the valve train components, but the 60 lb. starting load was large enough to do so. At the maximum lift condition, none of these starting loads were enough to overcome the installed load of the valve spring (nominally 128.1 lb.). A total of six runs each were completed at the base circle and maximum lift positions (two runs at each starting load condition in case there was a problem with the data from one test run).

Frequency analysis of the accelerometer data has been completed for each of the test conditions described above. The period of the events evaluated is 62.5 msec, resulting in a 16 Hz resolution of the FFT's. The actual TEAC tape ID#'s are 13, 16, 17, 20, 21, and 23. The time and frequency responses are shown on the following pages in Figures 21 to 26. The frequency values that correspond with the peaks, or natural frequencies, are noted on each figure.
At the base circle condition, the responses at starting loads of 40 and 50 lb. are very similar (see Figures 21 and 22). The largest spike in each case is at 1344 Hz, and this is believed to be the system natural frequency at the base circle position. It is expected that the spike corresponding to the system natural frequency would be the largest, as the step input load applied to the valve excites the system natural frequency the most. The exception to this, as shown in Trial One, is the case where the tension load applied to the valve creates clearance between the valve train components. The first spike (at 496 Hz) is believed to correspond to the base natural frequency of the valve spring at the base circle condition. By calculation the base spring natural frequency is estimated to be 447 Hz, and one spring of this design level has been measured separately to have a base natural frequency of 470 Hz. The valve spring being installed in the valve train should not affect the base natural frequency of the valve spring, as this frequency is so much lower than the system natural frequency. With the exception of the spike at 1344 Hz, the other spikes are believed to correspond to the harmonics of the valve spring as they are approximate multiples of 496 Hz (960, 1824, 2288, and 2784 Hz). A spike is expected at around 1488 Hz for the third harmonic of the valve spring, but is not clearly seen, which may have to do with this frequency being so close to the suspected system natural frequency. At this point it is not well understood why the spike that is believed to correspond to the second harmonic of the valve spring (960 Hz) is larger in amplitude than the spike that is believed to correspond to the valve spring base natural frequency (496 Hz).

At the base circle position with a starting load of 60 lb. (see Figure 23), the responses are very similar to what is described in the previous section on Trial One (see also Figure 20). Again the frequency response data shows spikes at frequencies that are believed to correspond to the base natural frequency of the valve spring and the harmonics of the valve spring, 496, 944, 1440, 1904, 2384, and 2832 Hz, with no obvious spike corresponding to the system natural frequency. The slight differences between the
results shown in Figures 20a and b and Figure 23 could be attributed to several factors, including testing on a different cylinder head, testing on the left head instead of the right, a different valve spring (same part number though), and a different roller finger follower (the mechanical HLA is the same part). It should be noted that in comparing Figure 23 with Figures 21 and 22 there is a difference in the y-axis scales of both the time and frequency response plots. In particular, note that the y-axis scale of Figures 21 and 22 is 0 to 400 mg’s, and the y-axis scale of the Figure 23 is 0 to 6.4 g’s. Looking more closely at Figure 23, the spike just to the left of 1440 Hz has roughly the same amplitude as the 1344 Hz spike in Figures 21 and 22. This seems to support what has been initially hypothesized in the above section on Trial One, that by creating separation between the valve train components and then releasing the valve to vibrate imparts a significant amount of energy to the valve spring, causing it to vibrate at an amplitude which overshadows the other modes of vibration of the valve train. This is further investigated in later testing.

At all starting loads (40, 50 and 60 lb.) at the maximum lift position (see Figures 24 to 26), there are spikes in the frequency response plots, but there is also a lot of noise at many other frequencies as well. Note that at the maximum lift position the pre-load on the valve spring is nominally 570.0 N (128.1 lb.). Based on the valve spring pre-load and the appearance of the frequency response plots, it is hypothesized that not enough energy is imparted to the system to sufficiently excite it. Further testing is completed to confirm this.
Figure 21 Time and Frequency Responses for the Step Input Test, Trial Two, ID#13, Base Circle, Start Load 40 lb.
Figure 22 Time and Frequency Responses for the Step Input Test, Trial Two, ID#16, Base Circle, Start Load 50 lb.
Figure 23 Time and Frequency Responses for the Step Input Test, Trial Two, ID#17, Base Circle, Start Load 60 lb.
Figure 24  Time and Frequency Responses for the Step Input Test, Trial Two, ID#20, Max. Lift, Start Load 40 lb.
Figure 25  Time and Frequency Responses for the Step Input Test, Trial Two, ID#21, Max. Lift, Start Load 50 lb.
Figure 26  Time and Frequency Responses for the Step Input Test, Trial Two, ID#23, Max. Lift, Start Load 60 lb.
4.3.3 Trial Three

For Trial Three of data collection, the test fixture was again setup on the left side Premium V6 cylinder head at exhaust position number six with valve spring part number 12553351AA. The testing was completed at the base circle position and at the maximum lift position (see Trial Two for an explanation of how the system was positioned at maximum lift). Three starting load conditions were tested at the maximum lift condition, 50, 110, and 120 lb. Three starting load conditions were also tested at the base circle condition, 40, 50, and 60 lb. The following observations were made at each position and load condition:

**Base Circle:** 40 lb. - RFF is not loose at all under finger pressure; 50 lb. - RFF is not loose, but can be shifted around; 60 lb. - RFF is loose and can be moved side to side and up and down

**Maximum Lift:** 0 to 60 lb. - RFF and gage blocks not loose at all under finger pressure; 60 to 90 lb. - gage blocks can twist, but RFF not loose; 90 to 110 lb. - gage blocks twist more easily, but RFF not loose; 110 to 130 lb. - gage blocks twist very easily, but RFF not loose

The above observations confirmed that as the tension load on the valve is increased, the other contact loads within the valve train decrease. Again, in most cases two runs were completed at each position and starting load condition in case there was a problem with the data from one test run. Additionally, testing was repeated at each position and starting load condition, with the exception of base circle with a starting load of 60 lb., with the accelerometer attached by five minute epoxy, instead of beeswax. Although the beeswax is supposed to be good up to frequencies of 12,000 Hz, the testing with the epoxied accelerometer was completed to verify that the beeswax mounting method was not the cause of the noise seen in the frequency response plots at the maximum lift condition as shown and discussed in the Trial Two results.
Frequency analysis of the accelerometer data has been completed for some of the test conditions described above. The period of the events evaluated is 62.5 msec, resulting in a 16 Hz resolution of the FFT's. The actual TEAC tape ID #’s analyzed are 29, 31, 33, 34, 39, 40, and 43. The time and frequency responses are shown on the following pages in Figures 27 to 33. The frequency values that correspond with the peaks, or natural frequencies, are noted on each figure.

Figure 27 shows the time and frequency responses at the maximum lift position with a starting load of 50 lb. These conditions are the same as for the results shown in Figure 25, and the testing and analysis are repeated here to verify that the results repeated. Figures 25 and 27 display very similar results with spikes observed at some frequencies, but also with noise at many other frequencies.

Figure 28 shows the time and frequency responses at the maximum lift position with a starting load of 110 lb. The largest spike in this frequency response plot is at 288 Hz. This is an unexpected result, as none of the natural frequencies are expected to be this low. Further testing is conducted to try to explain this result. The next spike is at 656 Hz, which is believed to correspond to the base natural frequency of the valve spring at the maximum lift condition (estimated by calculation to be 685 Hz; not measured separately). The natural frequencies of the spring increase as the spring is compressed due to coils collapsing on each other, which reduces the number of active coils. The other frequencies displayed in this plot are believed to be harmonics of the spring, based on the spikes occurring approximately every 656 Hz. It is expected that the system natural frequency would also be seen in this plot, but it is not. The only possible explanation for this is that the 110 lb. starting load resulted in an input that only excited the spring. Further testing is required to better understand this.

Figures 29 to 32 show results for the same test condition, maximum lift with a starting load of 120 lb., with the exception that in Figures 29 and 30 the accelerometer is held on
with beeswax and in Figures 31 and 32 it is held on with epoxy. The frequency responses show similar results in Figures 29, 31, and 32. Similar to Figure 28, there is a spike near 300 Hz, but in these cases the amplitude of this spike is not nearly as large as in Figure 28. Again more testing is required to understand the cause of this spike. In each of these three plots, there is a spike at 672 Hz, which is believed to correspond to the base natural frequency of the valve spring at the maximum lift condition (estimated by calculation to be 685 Hz). Next, in each case there is a large mound spanning from about 1300 Hz to 1500 Hz. In each of the three plots, the mound has the largest amplitude frequencies.

It is hypothesized that a mound is observed instead of distinct spikes because the second harmonic of the spring (expected to be about 1340 Hz) and the system natural frequency are very close together. This is evaluated in further testing by attempting to move the second harmonic of the spring away from the system natural frequency by shimming the spring to increase its compression (i.e., increase its natural frequencies). The spikes at 1950 to 2100 Hz and near 2600 Hz are believed to be the third and fourth harmonics of the valve spring. The natural frequencies of the spring are evaluated in further testing where a frequency analysis of data from a valve spring strain gage is completed.

The cause for other spikes near 2200 Hz and 2860 Hz is not currently known. In Figure 30, both the time and frequency response plots display very different results from those seen in Figures 29, 31, and 32. These results would seem to support the practice of collecting at least two sets of data at each test condition in case there is a problem with the data, which is suspected to be the case here.

Figure 33 shows the results at base circle with a starting load of 50 lb. and the accelerometer held on with epoxy. The purpose of testing at these conditions is to confirm that the results with the accelerometer held on with beeswax or epoxy are the same at the base circle condition. Comparing the frequency response results in Figures
22 and 33, spikes are seen at the same or nearly the same frequencies, confirming that there is apparently no difference between these two attachment methods.
Figure 27 Time and Frequency Responses for the Step Input Test, Trial Three, ID#29, Max. Lift, Start Load 50 lb.
Figure 28 Time and Frequency Responses for the Step Input Test, Trial Three, ID#31, Max. Lift, Start Load 110 lb.
Figure 29 Time and Frequency Responses for the Step Input Test,
Trial Three, ID#33, Max. Lift, Start Load 120 lb.

ID 33 Max Lift Start Load 120 lbs

Inst Time

Chan 1

200

60

/div

Res

q

-280

0

Lin

Sec

62.5m

X: 8.48389mSec
Y: 5.39911g

Inst Freq

Chan 1

600m

75m

/div

Mog

Peak

9

0

X: 1.408kHz
Y: 410.287mng
Figure 30  Time and Frequency Responses for the Step Input Test, Trial Three, ID#34, Max. Lift, Start Load 120 lb.
Figure 31 Time and Frequency Responses for the Step Input Test,
Trial Three, ID#39, Max. Lift, Start Load 120 lb., Epoxy
Figure 32  Time and Frequency Responses for the Step Input Test,  
Trial Three, ID#40, Max. Lift, Start Load 120 lb., Epoxy
Figure 33  Time and Frequency Responses for the Step Input Test, Trial Three, ID#43, Base Circle, Start Load 50 lb., Epoxy

ID 43 Base Circle Load 50 lbs Epoxy

Inst. Time

Chan 1

Inst. Freq

Chan 1

X:2.31934mSec  Y:94.3131g

X:1.344kHz  Y:318.474mg
4.3.4 Trial Four

For Trial Four of data collection, the test fixture was setup on the right side Premium V6 cylinder head at exhaust position number six with valve spring part number 12553351EB (a different spring from Trial One to Three testing). For Trial Four, the valve spring was strain gaged. This allowed a closer look at the natural frequencies of the valve spring in an attempt to better differentiate them from the system natural frequency. The testing was completed at the base circle position and at the maximum lift position (see Trial Two for an explanation of how the system was positioned at maximum lift). As many different test conditions were evaluated in Trial Four, Table 6 below is used to summarize the different conditions.

Table 6 Trial Four Test Conditions

<table>
<thead>
<tr>
<th>Lift Condition***</th>
<th>Start Load (lb.)</th>
<th>Valve Spring Shim Height (mm)**</th>
<th>RFF Clearance Evaluation*</th>
<th>TEAC Tape ID #’s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Circle</td>
<td>40</td>
<td>None</td>
<td>shifts, not loose</td>
<td>45, 46</td>
</tr>
<tr>
<td>Base Circle</td>
<td>45</td>
<td>None</td>
<td>shifts, not loose</td>
<td>51, 52</td>
</tr>
<tr>
<td>Base Circle</td>
<td>50</td>
<td>None</td>
<td>loose</td>
<td>47, 48</td>
</tr>
<tr>
<td>Base Circle</td>
<td>60</td>
<td>None</td>
<td>very loose</td>
<td>49, 50</td>
</tr>
<tr>
<td>Max Lift</td>
<td>110</td>
<td>None</td>
<td>shifts, not loose</td>
<td>55, 56</td>
</tr>
<tr>
<td>Max Lift</td>
<td>120</td>
<td>None</td>
<td>shifts, not loose</td>
<td>57, 58</td>
</tr>
<tr>
<td>Max Lift</td>
<td>130</td>
<td>None</td>
<td>shifts, not loose</td>
<td>59, 60</td>
</tr>
<tr>
<td>Base Circle</td>
<td>50</td>
<td>1.55 mm</td>
<td>shifts, not loose</td>
<td>67, 68</td>
</tr>
<tr>
<td>Base Circle</td>
<td>55</td>
<td>1.55 mm</td>
<td>shifts, not loose</td>
<td>69, 70</td>
</tr>
<tr>
<td>Base Circle</td>
<td>60</td>
<td>1.55 mm</td>
<td>loose</td>
<td>71, 72</td>
</tr>
<tr>
<td>Base Circle</td>
<td>70</td>
<td>1.55 mm</td>
<td>very loose</td>
<td>73, 74</td>
</tr>
<tr>
<td>Max Lift</td>
<td>120</td>
<td>1.55 mm</td>
<td>shifts, not loose</td>
<td>61, 62</td>
</tr>
<tr>
<td>Max Lift</td>
<td>130</td>
<td>1.55 mm</td>
<td>shifts, not loose</td>
<td>63, 64</td>
</tr>
<tr>
<td>Max Lift</td>
<td>140</td>
<td>1.55 mm</td>
<td>shifts, not loose</td>
<td>65, 66</td>
</tr>
<tr>
<td>Base Circle</td>
<td>14 (light spring)</td>
<td>None</td>
<td>shifts, not loose</td>
<td>75</td>
</tr>
<tr>
<td>Base Circle</td>
<td>16 (light spring)</td>
<td>None</td>
<td>shifts, not loose</td>
<td>76, 77</td>
</tr>
<tr>
<td>Base Circle</td>
<td>17 (light spring)</td>
<td>None</td>
<td>loose</td>
<td>78</td>
</tr>
<tr>
<td>Max Lift</td>
<td>19 (light spring)</td>
<td>None</td>
<td>shifts, not loose</td>
<td>79, 80</td>
</tr>
<tr>
<td>Max Lift</td>
<td>20 (light spring)</td>
<td>None</td>
<td>loose</td>
<td>81, 82</td>
</tr>
<tr>
<td>Base Circle</td>
<td>110</td>
<td>10 mm</td>
<td>shifts, not loose</td>
<td>83, 84</td>
</tr>
<tr>
<td>Lift Condition***</td>
<td>Start Load (lb.)</td>
<td>Valve Spring Shim Height (mm)**</td>
<td>RFF Clearance Evaluation*</td>
<td>TEAC Tape ID #’s</td>
</tr>
<tr>
<td>------------------</td>
<td>------------------</td>
<td>---------------------------------</td>
<td>---------------------------</td>
<td>------------------</td>
</tr>
<tr>
<td>Base Circle</td>
<td>120</td>
<td>10 mm</td>
<td>shifts, not loose</td>
<td>85, 86</td>
</tr>
<tr>
<td>Base Circle</td>
<td>130</td>
<td>10 mm</td>
<td>very loose</td>
<td>87, 88</td>
</tr>
<tr>
<td>Base Circle - Spring Only</td>
<td>19 (light spring)</td>
<td>None</td>
<td>N/A</td>
<td>89-91</td>
</tr>
<tr>
<td>Max Lift - Spring Only</td>
<td>23 (light spring)</td>
<td>None</td>
<td>N/A</td>
<td>92-94</td>
</tr>
<tr>
<td>Base Circle - Spring Only</td>
<td>50 (approx.)</td>
<td>None</td>
<td>N/A</td>
<td>95, 96</td>
</tr>
<tr>
<td>Max Lift - Spring Only</td>
<td>136 (approx.)</td>
<td>None</td>
<td>N/A</td>
<td>97, 98</td>
</tr>
</tbody>
</table>

*shift = can be shifted around, but no apparent clearance
loose = clearance between RFF, cam, valve and HLA
**Spring load increase due to shim
1.55 mm x 31 N/mm = 10.8 lb. (start loads increased by 10 lb. to result in approx.
same step input load as without shim)
10 mm shim equal to maximum valve lift
***In spring only test, springs were tested outside of cylinder head by compressing them in an arbor press.

Frequency analysis of the accelerometer and strain gage data has been completed for all of the test conditions described above. The period of the events evaluated is 62.5 msec, resulting in a 16 Hz resolution of the FFT’s. The results are presented via time and frequency response plots on the following pages in Figures 34 to 54, with the actual TEAC tape ID #’s noted on each plot. The frequency values that correspond with the peaks, or natural frequencies, are noted on each figure.

To identify the natural frequencies at base circle, the results at the following test conditions are presented and compared: 1) base circle cam position with the valve spring at base circle height, 2) base circle cam position with the valve spring shimmed to max lift height (to differentiate between the valve spring natural frequencies and the system natural frequency by shifting the spring frequencies and maintaining the system natural
frequency), and 3) valve spring only outside of cylinder head at base circle height. Figures 34 to 43 show the results for these test conditions.

Figures 34 to 38 show the results for the base circle cam position with the valve spring at base circle height and starting loads of 40, 45, 50, and 60 lb. The starting loads of 40 and 45 lb are not enough to create clearance between the components. With the EB design level spring used, the starting loads of 50 and 60 lb are enough load to create clearance between the components. Although at 50 lb, there is very little clearance (not detectable by eye). Whereas at 60 lb there are several millimeters of clearance between the cam and roller.

In the frequency response plots of the accelerometer data in Figures 34 to 38, peaks at the following frequencies are observed in most every instance: 512-528 Hz, 992-1008 Hz, 1392 Hz, and 1632-1648 Hz. Additionally, in Figure 38 (60 lb start load), peaks are seen at 1504 and 2032 Hz. In each case, the peak at 992-1008 Hz or at 1392 Hz is the largest in amplitude. In the frequency response plots of the strain gage data in Figures 34 to 38, peaks are seen at 512-528 Hz, 992-1008 Hz, 1392 Hz, and 1648 Hz. In some cases there are also peaks at other intermediate frequencies. In each case the peak at 512-528 Hz is the largest in amplitude.

Figures 39 to 42 show the results for the base circle cam position with the valve spring at max lift height and starting loads of 110, 120, and 130 lb. The starting loads of 110 and 120 lb are not enough to create clearance between the components. The starting load of 130 lb is enough load to create clearance between the components.

As seen in the frequency response plots of the accelerometer data in Figures 39 and 40 (starting loads of 110 and 120 lb), there are some peaks, but also a lot of noise. This is similar to some of the results presented in the section on Trial Two. This leads to the conclusion that, for some reason, at the base circle cam position with the valve spring at
max lift height starting loads of 110 and 120 lb are not enough to excite the system. In
Figures 41 and 42 (starting load of 130 lb), though, prominent peaks are seen at 608 Hz,
1168 Hz, and 1600-1616 Hz. Recall that the 130 lb starting load creates clearance
between the components, which, as discussed in previous sections, seems to really excite
the valve spring. In Figures 41 and 42, shorter peaks are also seen at other frequencies,
including around 1400 Hz. The amplitude of the peaks seen near 1400 Hz are
comparable to the amplitude of peaks at 1392 Hz in Figures 34 to 38. In the frequency
response plots of the strain gage data in Figures 39 to 42, peaks are seen at 608 Hz, 1168-
1184 Hz, and 1616-1632 Hz. In some cases there are also peaks at other intermediate
frequencies. In each case the peak at 608 Hz is the largest in amplitude.

Figure 43 shows the results for the valve spring outside of the cylinder head at the base
circle height. For this testing, the strain gaged valve spring is compressed by an arbor
press to the installed base circle height. The objective of this testing is to isolate the
behavior of the valve spring from the rest of the valve train. The part of the arbor press
contacting the top of the valve spring is struck with a mallet to cause the valve spring to
vibrate, and the strain gage output is recorded. In the frequency response plots of the
strain gage data in Figure 43, peaks are seen at 512 Hz, 992 Hz, 1408 Hz, and 1648 Hz.
The 512 Hz peak has the largest amplitude.

From the results of this testing, presented in Figures 34 to 42, the following conclusions
are made. The base natural frequency and second harmonic of the valve spring at the
base circle height are 512-528 Hz and 992-1008 Hz, respectively (calculated spring base
natural frequency is 525 Hz). The third harmonic of the valve spring may be 1632-1648
Hz, but it is very difficult to make this determination experimentally. In Figure 38,
where the starting load creates clearance between the components which seems to really
excite the valve spring, there are peaks at other frequencies which may be higher
harmonics of the valve spring (1504 and 2032 Hz). The system natural frequency is 1392
Hz.
All of these frequencies show up in the data from both transducers (accelerometer and strain gage). This is because at each transducer we are picking up what is happening at the other transducer. This is expected as all of the components of the valve train are in contact with one another.

In the frequency response plot of the accelerometer data, the second harmonic of the valve spring is larger in amplitude than the spring base natural frequency because the system is exciting the second harmonic more than the base, which may have something to do with the second harmonic of the spring being so close to the system natural frequency. The accelerometer detects this, while the strain gage shows the amplitude of the spring base natural frequency to be the largest. This may be because the accelerometer can measure absolute motion, and the strain gage measure relative motion. Additionally, the accelerometer is accurate within 5% over a frequency range of 500 to 9000 Hz, and the strain gage likely has a much narrower range, which is unknown as strain gages are not actually intended for this type of measurement. Another possible reason for this result is due to the location of the strain gage on the valve spring. If the strain gage is near a node point it may not measure the actual amplitude of the vibrations.

Finally, in some cases, even where there is not clearance between the components due to the starting load, the peak at 992-1008 Hz is larger than the peak at 1392 Hz, and in other cases it is just the opposite. This can be explained by the fact that these two frequencies are the primary natural frequencies of the system, and slight variations in the initial conditions, including the exact starting load and the length of the cable which is cut, cause one or the other to be dominant.

With both of these observations regarding the second harmonic of the valve spring, it would be very useful to have the dynamic system model calculate the eigenvectors. With this information, it would be better understood how the energy input to the system is distributed by the eigenvectors.
To identify the natural frequencies at maximum lift, the results at the following test conditions are presented and compared: 1) maximum lift cam position with the valve spring at maximum lift height and 2) valve spring only outside of cylinder head at maximum lift height. Figures 44 to 48 show the results for these test conditions.

Figures 44 to 47 show the results for the maximum lift cam position with the valve spring at maximum lift height and starting loads of 110, 120, and 130 lb. None of these starting loads are enough to create clearance between the components, although 130 lb is close to creating this condition. No testing is done at this cam position with clearance between the components as there is not a way to keep the gage blocks from slipping out from between the cam and roller once clearance is created and the contact loads go to zero.

In the frequency response plots of the accelerometer data in Figures 44 to 47, peaks at the following frequencies are observed in most every instance: 272-304 Hz, 640 Hz, 1216 Hz, and 1584-1600 Hz. In each case, the peak at 288-304 Hz or at 1216 Hz is the largest in amplitude, and the peak at 1584-1600 Hz is the second or third largest in amplitude. In the frequency response plots of the strain gage data in Figures 44 to 47, peaks are seen at 640 Hz, 1216 Hz, and 1584 Hz. In each case the peak at 640 Hz or at 1216 Hz is the largest in amplitude.

Figure 48 shows the results for the valve spring outside of the cylinder head at the maximum lift height. This testing is completed as described above for the base circle condition. In the frequency response plots of the strain gage data in Figure 48, peaks are seen at 624 Hz and 1200 Hz. Smaller peaks are seen at other frequencies as well including 1824 Hz. The 624 Hz peak has the largest amplitude.

From the results of this testing, presented in Figures 44 to 48, the following conclusions are made. The base natural frequency and second harmonic of the valve spring at the
maximum lift height are 640 Hz and 1216 Hz, respectively (calculated spring base natural frequency is 651 Hz). The third harmonic of the valve spring may be 1824 Hz (see Figure 48), but again it is very difficult to make this determination experimentally. The system natural frequency is 1584-1600 Hz. The explanations above of the results at base circle apply here as well. There are several different occurrences in the max lift data though compared to the base circle data. First, the frequency response plots of the accelerometer data (Figures 44 to 47) show peaks at 272-304 Hz with significant amplitude. These are possibly sub-harmonics of the nonlinear valve spring, which are being excited similar to what is believed to be the second harmonic of the valve spring. Again the accelerometer is able to pick up this mode of vibration, while the strain gage is not. Second, in some cases in the frequency response plots of the accelerometer data, there is a “mound” between the peaks at 1216 Hz and 1584-1600 Hz (see Figures 44 and 47). One explanation for this may be the effect that the nearby peaks at 1216 Hz and 1584-1600 Hz have on the frequency analysis (FFT). As shown in the examples in section 2.2.2 (see Figure 10b), the FFT of nearby frequencies can result in the bases of the peaks starting to blend together. This is possibly why a mound is seen between the two nearby peaks.

Figures 49 to 54 show the results at base circle and max lift with the valve spring shimmed 1.55 mm. The intent here is to try to shift the spring natural frequencies, while keeping the system natural frequency at the same level. All of the starting loads are increased by 10 lb to account for the additional valve spring load in order to keep the step input load at the same level. The results show that shimming the valve spring by 1.55 mm is not enough to significantly change the spring natural frequencies. So the accelerometer and strain gage frequency response plots here look similar to those presented in Figures 34 and 42 and 44 to 47.
Figure 34 Time and Frequency Responses for the Step Input Test, Trial Four, ID#45, Base Circle, Start Load 40 lb.
Figure 35 Time and Frequency Responses for the Step Input Test, Trial Four, ID#51, Base Circle, Start Load 45 lb.
Figure 36 Time and Frequency Responses for the Step Input Test, Trial Four, ID#52, Base Circle, Start Load 45 lb.
Figure 37  Time and Frequency Responses for the Step Input Test, Trial Four, ID#47, Base Circle, Start Load 50 lb.
Figure 38  Time and Frequency Responses for the Step Input Test, Trial Four, ID#49, Base Circle, Start Load 60 lb.

ID 49 Base Circle Start Load 60 lb.

- Chan 1
  - Inst Time
  - Real
  - Accel
  - X: 4.02832 m
  - Y: 90.5726 g

- Chan 2
  - Inst Freq
  - Mag
  - rms
  - Spring
  - X: 992 Hz
  - Y: 13.2773 mV
Figure 39 Time and Frequency Responses for the Step Input Test, Trial Four, ID#84, Base Circle, Valve Spring Shimmed 10 mm, Start Load 110 lb.
Figure 40 Time and Frequency Responses for the Step Input Test, Trial Four, ID#86, Base Circle, Valve Spring Shimmed 10 mm, Start Load 120 lb.
Figure 41 Time and Frequency Responses for the Step Input Test, Trial Four, ID#87, Base Circle, Valve Spring Shimmed 10 mm, Start Load 130 lb.

ID 87 BC, Valve Spring Shimmed 10mm, Start Load 130 lb.

4/16/97 6:39PM
Hanning

4/16/97 6:39PM
Hanning

4/16/97 6:39PM
Hanning

4/16/97 6:39PM
Hanning

87
Figure 42 Time and Frequency Responses for the Step Input Test,
Trial Four, ID#88, Base Circle, Valve Spring Shimmed 10 mm, Start Load 130 lb.
Figure 43  Time and Frequency Responses for the Impact Test of Spring Only, Trial Four, ID#95, Base Circle, Valve Spring at 35 mm Height

ID 95 BC, EB Spring Only, 35mm hl.

Ch 1 : Ch 1:
Start=0.0005Sec 1=62.500mSec Lines=1024
Inst Time  No Ovlp  Channel 1

4/16/97 9:16PM
Hanning

Ch 1 : Ch 1:
Start=0Hz  Span=6.4kHz Lines=401
Inst Freq  No Ovlp  Channel 1

4/16/97 9:16PM
Hanning

89
Figure 44 Time and Frequency Responses for the Step Input Test, Trial Four, ID#55, Max. Lift, Start Load 110 lb.
Figure 45  Time and Frequency Responses for the Step Input Test, Trial Four, ID#56, Max. Lift, Start Load 110 lb.

[Graph showing time and frequency responses for the step input test, trial four, ID#56, max. lift, start load 110 lb.]
Figure 46  Time and Frequency Responses for the Step Input Test,  
Trial Four, ID#57, Max. Lift, Start Load 120 lb.
Figure 47  Time and Frequency Responses for the Step Input Test, Trial Four, ID#59, Max. Lift, Start Load 130 lb.
Figure 48 Time and Frequency Responses for the Impact Test of Spring Only, Trial Four, ID#97, Max. Lift, Valve Spring at 24.5 mm Height

ID 97 Max Lift, EB Spring Only, 24.5mm ht.

Ch 1 : Ch 1 :
Start=0.000Sec T=62.500mSec Lines=1024 Inst Time
4/16/97 9:30PM
4/16/97 9:30PM
Hanning
Hanning

200m
50m
/div
Real

-200m
0

Lin
Sec

X:21.7896mSec
Y:-108.561mV
Spring

Ch 1 : Ch 1 :
Start=0Hz Span=6,400Hz Lines=401 Inst Freq
4/16/97 9:30PM
4/16/97 9:30PM
Hanning
Hanning

60m
7.5m
/div
Mag
rms

V

0

Lin
Hz

X:62.9Hz
Y:53.07mV
Spring

94
Figure 49  Time and Frequency Responses for the Step Input Test, Trial Four, ID#67, Base Circle, Valve Spring Shimmed, Start Load 50 lb.

ID 67 Base Circle, Shimmed Valve Spring, Start Load 50 lb.
Figure 50  Time and Frequency Responses for the Step Input Test,
Trial Four, ID#71, Base Circle, Valve Spring Shimmed, Start Load 60 lb.

Insl Time
Chan 1

Insl Freq
Chan 1

Insl Freq
Chan 2

96
Figure 51  Time and Frequency Responses for the Step Input Test, Trial Four, ID#73, Base Circle, Valve Spring Shimmed, Start Load 70 lb.
Figure 52  Time and Frequency Responses for the Step Input Test,  
Trial Four, ID#61, Max. Lift, Valve Spring Shimmed, Start Load 120 lb.  

ID 61 Max Lift, Shimmed Valve Spring, Start Load 120 lb.
Figure 53 Time and Frequency Responses for the Step Input Test, Trial Four, ID#63, Max. Lift, Valve Spring Shimmed, Start Load 130 lb.
Figure 54  Time and Frequency Responses for the Step Input Test, Trial Four, ID#65, Max. Lift, Valve Spring Shimmed, Start Load 140 lb.
4.4 Comparison of Experimental and Analytical Results

The valve train natural frequencies presented in this discussion are from four sources: spring base natural frequency calculated by the spring manufacturer\textsuperscript{6}, spring and system natural frequencies from a dynamic valve train simulation model\textsuperscript{3,5}, spring and system natural frequencies measured with the step input method, and the system natural frequency measured with the waterfall plot method\textsuperscript{1}. As the dynamic simulation is not the focus of this thesis project, the dynamic simulation results (VTS) presented are ones that were readily available from other studies that were completed on the Premium V6 valve train. The simulation results are of an intake positions of the right and left side cylinder heads without cam torsion turned on in the model. The step input experimental results, though, are from exhaust positions of the right and left side cylinder heads. The waterfall plot experimental results are from an intake position of the left side cylinder head. The differences in valve train geometry between the intake and exhaust positions are likely to affect the system natural frequency, but not the valve spring natural frequencies. Additionally, the step input measurements are made with a mechanical HLA, which is stiffer and probably affects the results. Also note that the VTS base circle results are not actually at the base circle, but at -50 cam degrees, which is just beyond valve opening. The valve must be off of the valve seat in order that the valve train be able to vibrate freely. The system geometries at base circle and just beyond valve opening are very similar. The valve train natural frequency comparison is presented below in Tables 7a and 7b.
### Table 7a Comparison of Experimental and Analytical Results for Left Side Cylinder Head (with Valve Spring P/N 12553351AA)

<table>
<thead>
<tr>
<th>Natural Frequency</th>
<th>Spring Base</th>
<th>Spring 2nd Harmonic</th>
<th>System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step Input - Base Circle, Exhaust (Hz)</td>
<td>496</td>
<td>960</td>
<td>1344</td>
</tr>
<tr>
<td>Step Input - Max Lift, Exhaust (Hz)</td>
<td>656-672</td>
<td>**</td>
<td>**</td>
</tr>
<tr>
<td>VTS - Base Circle, Intake (Hz)</td>
<td>471</td>
<td>940</td>
<td>1525</td>
</tr>
<tr>
<td>VTS - Max Lift, Intake (Hz)</td>
<td>617</td>
<td>1206</td>
<td>1620</td>
</tr>
<tr>
<td>Manufac. Spec. - Base Circle (Hz)</td>
<td>447</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Manufac. Spec. - Max Lift (Hz)</td>
<td>685</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Waterfall Plot, Intake (Hz)</td>
<td>N/A</td>
<td>N/A</td>
<td>1350-1650</td>
</tr>
</tbody>
</table>

*Could not be determined from test data

**Could not differentiate between spring 2nd harmonic and system

### Table 7b Comparison of Experimental and Analytical Results for Right Side Cylinder Head (with Valve Spring P/N 12553351EB)

<table>
<thead>
<tr>
<th>Natural Frequency</th>
<th>Spring Base</th>
<th>Spring 2nd Harmonic</th>
<th>System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step Input - Base Circle, Exhaust (Hz)</td>
<td>512-528</td>
<td>992-1008</td>
<td>1392</td>
</tr>
<tr>
<td>Step Input - Max Lift, Exhaust (Hz)</td>
<td>640</td>
<td>1216</td>
<td>1584-1600</td>
</tr>
<tr>
<td>VTS - Base Circle, Intake (Hz)</td>
<td>516</td>
<td>1029</td>
<td>1647</td>
</tr>
<tr>
<td>VTS - Max Lift, Intake (Hz)</td>
<td>668</td>
<td>1301</td>
<td>1598</td>
</tr>
<tr>
<td>Manufac. Spec. - Base Circle (Hz)</td>
<td>525</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Manufac. Spec. - Max Lift (Hz)</td>
<td>651</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

*Could not be determined from test data

Comparing the results for the left side cylinder head, at the base circle position there is good agreement on the spring base natural frequency between the step input method and VTS, but the manufacturer's specification is quite a bit lower (one separately measured spring of the AA design level was measured to be at 470 Hz). At maximum lift, the step input method and the manufacturer's specification agree well, but the results from VTS are quite a bit lower. Some of the variation may be due to VTS not predicting further
collapse of the spring coils. For the second harmonic of the valve spring, there is good agreement at base circle between the step input method and VTS, but at maximum lift this natural frequency could not be differentiated from the system natural frequency with the step input method. For the system natural frequency at base circle, there is good agreement between the step input method and the waterfall plot method. The result from VTS is higher though. At maximum lift, VTS and the waterfall plot method agree well. While, again, the system natural frequency at maximum lift could not be differentiated from the second harmonic of the valve spring with the step input method data. At maximum lift, the spring second harmonic and the system natural frequency could not be differentiated because of their close proximity. So although there is no engineering need to know the second harmonic of the valve spring, it is sometimes necessary to be able to it in order to differentiate it from the system natural frequency.

Comparing the results for the right side cylinder head, at the base circle and maximum lift positions there is good agreement on the spring base natural frequency between the step input method, VTS, and the manufacturer’s specification. For the second harmonic of the valve spring, there is good agreement at base circle between the step input method and VTS, but at maximum lift this natural frequency is much lower with the step input method than with VTS. For the system natural frequency at base circle, the step input method gives a much lower value than VTS. At maximum lift, VTS and the step input method agree well.

There is not consistently good agreement between the various analytical and experimental methods. The reasons for this requires further investigation. In most cases, though, for the spring base natural frequency and second harmonic there is good agreement between the analytical methods and the step input method. The analytical results can be used in conjunction with the step input results to differentiate between the spring natural frequencies and the system natural frequency.
Chapter 5

CONCLUSIONS AND RECOMMENDATIONS

Being able to measure the natural frequencies of a valve train system is very important in characterizing the system as this information allows prediction of acceptable dynamic behavior, comparison of two systems, and verification of a dynamic model. The step input method for measuring natural frequencies of a valve train system has been developed as a part of this thesis project. The test setup and method are documented in this discussion. The step input method characterizes the natural frequencies of two of the most important modes of vibration, the fundamental mode of vibration of the entire linkage (system natural frequency) and translation of the valve spring. Analytical modeling or the waterfall plot method, both of which are completed on a non-stationary system, are ways for potentially determining the natural frequency of the third important mode of vibration of the valve train system, camshaft torsion.

The manual method for determining system natural frequency with a cycle count from a valve acceleration trace is a quick estimate of the system natural frequency. It can result in a significant error compared to what is considered to be a significant difference in system natural frequency. This method can be used rigorously to minimize potential errors. This method does not account for varying system natural frequency over the lift event due to system nonlinearity.

Experimental and analytical comparisons of valve trains with low and high system natural frequencies are presented in this discussion. The peak representing the system natural frequency in a waterfall plot for the lower natural frequency system is higher in amplitude and narrower in range than for the higher natural frequency system, making it
easier to determine the system natural frequency for the lower natural frequency valve train. The reason for the difference in amplitude is partially attributed to the fact that a lower natural frequency system with higher amplitude vibrations has a higher amplitude frequency response. This is because the frequency content of the input that excites the lower natural frequency system is larger in amplitude. Secondly, it is shown that the lower natural frequency valve train has less total system natural frequency variation (i.e., is less nonlinear). Furthermore, it is shown that the frequency domain amplitude spectrum from the FFT of a more nonlinear system is lower in amplitude, wider in range (even wider than the actual natural frequency variation), and more shifted along the frequency scale than the less nonlinear system. Therefore, the waterfall plot method works better for valve trains with lower system natural frequencies.

Valve train systems are nonlinear due to varying system geometry throughout the lift event, which results in varying effective masses, stiffnesses and damping factors of the valve train. The step input method is able to account for system nonlinearity, as measurements are taken of a stationary system at one point in the lift event, which is similar to how VTS determines natural frequencies. Measurements can be taken at two or more points in the lift event to characterize the ranges of the natural frequencies. It is important to characterize the ranges of the natural frequencies, especially when comparing two or more systems to determine which is dynamically superior as the ranges of natural frequencies may overlap. Additionally, in most cases the step input method results in a sharp peak at each natural frequency. This makes it easier to determine the natural frequency, thereby taking out some of the measurement subjectivity.

Analytical and experimental comparisons of a system with two nearby natural frequencies are presented in this discussion. With generated data it is shown that the closer two frequencies of a system are, the more the FFT of the data blends together the peaks of the frequency domain amplitude spectrum. Experimental data also showed that,
in the frequency domain, nearby peaks could blend together. This makes it difficult to differentiate between natural frequencies. In particular, in the step input data that has been collected for this thesis project, the second harmonic of the valve spring and the system natural frequency are often close to one another and sometimes hard to differentiate between. Simulation results and/or separate measurements of the spring can sometimes be used to differentiate between the valve spring second harmonic and the system natural frequency.

Step input testing results show that, if the tension load applied to the valve is large enough to overcome the pre-load of the valve spring, this imparts a large amount of energy to the valve spring. This causes the amplitude of vibration of the valve spring to be larger, sometimes much larger, than the amplitude of vibration of the valve. So in the frequency response plots the largest amplitude peaks are those corresponding to the valve spring natural frequencies. Conversely, if the tension load applied to the valve is not large enough, the valve train vibration is very small as the natural frequencies are not excited. This results in the frequency response plots having some peaks, but also a lot of noise. Additionally, testing has been completed with two attachment methods for the accelerometer to the valve, beeswax and epoxy. Comparison testing with these two attachment methods shows no significant difference.

In the step input testing, system and spring natural frequencies show up in the data from both transducers (accelerometer and strain gage). At each transducer we are picking up what is happening at the other transducer because all of the components of the valve train are in contact with one another.

Above, one effect of two nearby natural frequencies is discussed. Another effect is that, in the frequency response plots of the accelerometer data, the peak corresponding to the second harmonic of the valve spring is larger in amplitude than the peak corresponding to the spring base natural frequency. This may be due to the second harmonic of the
valve spring being so close to the system natural frequency and thus being excited the most. The accelerometer detects this, while the strain gage on the spring does not. This may be because the accelerometer measures absolute motion, and the strain gage measures relative motion. Another reason for these results may be the location of the strain gage on the spring. In some cases, even where the starting load does not create clearance between the valve train components, the amplitude of the peak corresponding to the second harmonic of the valve spring exceeds the amplitude of the peak corresponding to the system natural frequency. This is believed to be due to variations in initial conditions. Additionally, in some of the step input testing, peaks in the frequency response plots are seen at frequencies that are approximately one-half the base spring natural frequency. These peaks may correspond to sub-harmonics of the nonlinear valve spring.

In comparing the various sources of experimental and analytical results, there is not consistently good agreement between the sources. Some of this may be due to the fact that the analytical models from which results were obtained for this discussion are not completely representative of the experimental systems. Additional investigation of the correlation between the experimental and analytical results is warranted.

It is also suggested that an interesting and useful continuation of the study of valve train natural frequency measurement and analysis is modal analysis. From the analytical side this includes modifying VTS to calculate the eigenvectors (or mode shapes). From the experimental side this includes instrumenting one or more valve trains with a series of accelerometers to determine all of the modes of vibration of the valve train. The analytical and experimental results could then be correlated. From this study a more in depth understanding of valve train system vibrations could be obtained.
Appendix A
Dynamic Simulation Results for an Overhead Cam Finger Follower Valve Train
PV6 Dynamics, Left, AA Spring, Intake 5l, 1999BB1x1

2600 CRPM

2700 CRPM

2800 CRPM

2900 CRPM

Valve Lift - mm

Rotation Angle - Deg

-90 -60 -30 0 30 60 90

-90 -60 -30 0 30 60 90

-90 -60 -30 0 30 60 90

-90 -60 -30 0 30 60 90

Test 95314201

VTS Simulation

Kin 1999BB1x1
PV6 Dynamics, Left, AA Spring, Intake 5I, 1999BB1x1

3400 CRPM

3500 CRPM

Valve Lift - mm

Rotation Angle - Deg

-90 -60 0 30 60 90

-90 -60 0 30 60 90

Test 95314201
VTS Simulation
Kin 1999BB1x1

GM DELPHI MKT
PV6 Dynamics, Left, AA Spring, Intake 5I, 1999BB1x1
REFERENCES


BIBLIOGRAPHY


