Final Report: Cam Dynamics DOE – Variable Force Dampener

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Abstract

The camshaft and valve train system are an integral part of the modern internal combustion engine. Sub optimal behavior of this system can greatly reduce the efficiency and integrity of an engine. Two of the most prevalent failure modes for the valve train system are valve float and valve bounce: both relate to the inability of valves to properly follow the paths defined by the camshaft and can lead to fatigue, and eventually failure, of critical components. Numerous spring, dampener, and shim combinations were evaluated in prior experiments to investigate the causes and modes of these failures as well as how to address them. It had been hypothesized that the application of a variable force dampener (VFD) to the valve spring may help mitigate these known failure modes. The VFD was successful in offsetting certain failure modes because it could apply greater amounts of friction to the outer coils of the spring, relative to other dampeners that have been utilized in prior experiments, thereby increasing the spring's effective stiffness.

Introduction

When designing an IC engine, it is necessary to have a complete understanding of the operation of its subsystems. The valve train in particular is a uniquely complicated subsystem, whose operation is dictated by numerous parameters and requires investigation through experimentation. At higher operational speeds the valve train becomes susceptible to two unique forms of failure: spring surge and valve float. Spring surge characterizes the onset of longitudinal compression waves in the valve springs while valve float denotes the loss of contact between the valve and cam head surface.[1] Both phenomena are detrimental to engine behavior. Their onset and severity is highly dependent on the operational parameters of the valve train; spring stiffness, engine speed, and spring form among them.[1] In past experiments, the effects of adding a cup dampener to a spring was analyzed and seen to greatly alter the valve train susceptibility to certain modes failure. By further exploring this trend using a variable force dampener (VFD), it is possible to determine the exact effect of damping on valve train dynamics. Monitoring the dynamics of the system using force and displacement sensors while varying VFD dampening force, the onset and behavior of resulting failure phenomena can be identified and characterized. This characterization will allow engineers to properly identify the optimal parameters for efficient engine operation.

Theory

Valve Train Operation:

In a typical valve train, a rotating camshaft controls the opening and closing of numerous intake and outtake valves arranged along its length. Engine designs vary greatly in how the camshaft motion is applied to the valves, but in this experiment, a single over-head camshaft, diagram shown in Figure 1, was used. In this setup, the rotating cam head depresses a rocker arm, which pivots around a lifter point to depress the valve. It is clear that the rocker arm acts like lever amplifying the force applied by the cam head to the valve. The profile of the cam head is shaped such that it will produce a smooth displacement profile for the valve, during normal operation. The spring attached to the valve is used to oppose the action of the cam head and to maintain contact between the rocker arm and cam head. The restoring force applied by the spring is determined by its stiffness, the slope of a force vs. compression curve. Springs are initially compressed and pre-loaded upon placement into the valve train. Shims can be placed under the springs to increase the initial compression and consequently preload force. For the Ford spring, a cup damper can be installed that damps the coils that are displaced into the cup by providing friction to the outside of the spring.

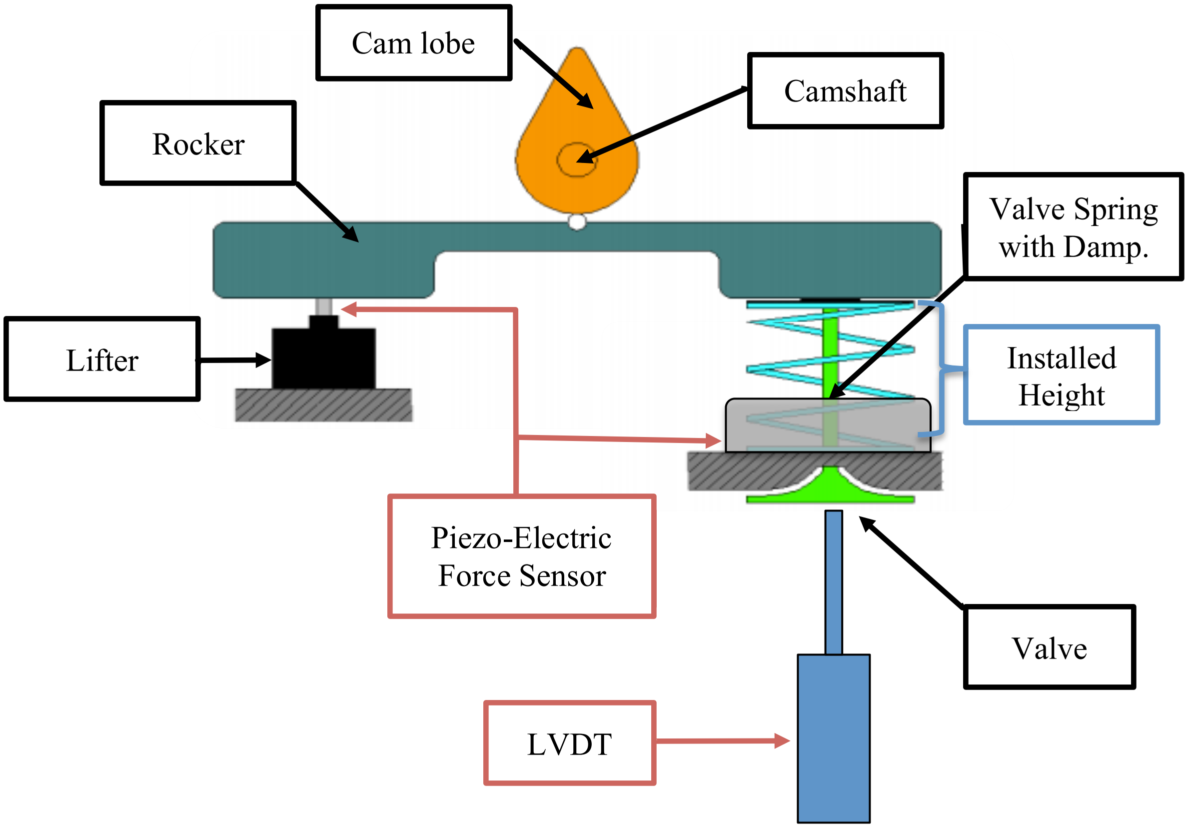


Figure 1: Drawing of valve train with components labeled. Adapted from Sherry[2]

Valve Float Failure:

During each rotation of the camshaft, the rocker arm is accelerated downwards by the lift region of the cam head. This acceleration is opposed by the valve spring which attempts to maintain contact between the cam head and rocker arm. If the camshaft is rotating at a sufficiently high speed this acceleration can become large enough that the spring force is unable to match it and a separation of the cam head and rocker arm surfaces occur. This separation is known as valve float and is detrimental to engine operation for two reasons. First it results in the valve being out of position during the intake-exhaust cycle. Secondly it results in a collision between the cam and rocker arm as the spring force regains control of the valve and reseats the rocker arm against the cam surface. Repeated collisions can produce significant wear on the valve components as well as decrease the efficiency of the engine.

Dampeners:
The term dampener refers to a wide variety of devices that resist vibrational motions within a spring. A common method employed by dampeners to reduce vibration is the application of small frictional forces to the spring during its motion. Two common forms of these so called friction dampeners are flat wire dampeners and cup dampeners. Flat wire dampeners are placed within the center of springs and provide constant frictional force along its length. Cup dampeners on the other hand enclose one end of the spring and provide frictional force to outer edge of the coils that rest inside it. Images of installed cup and wire dampers are shown in Figures 2A-B.

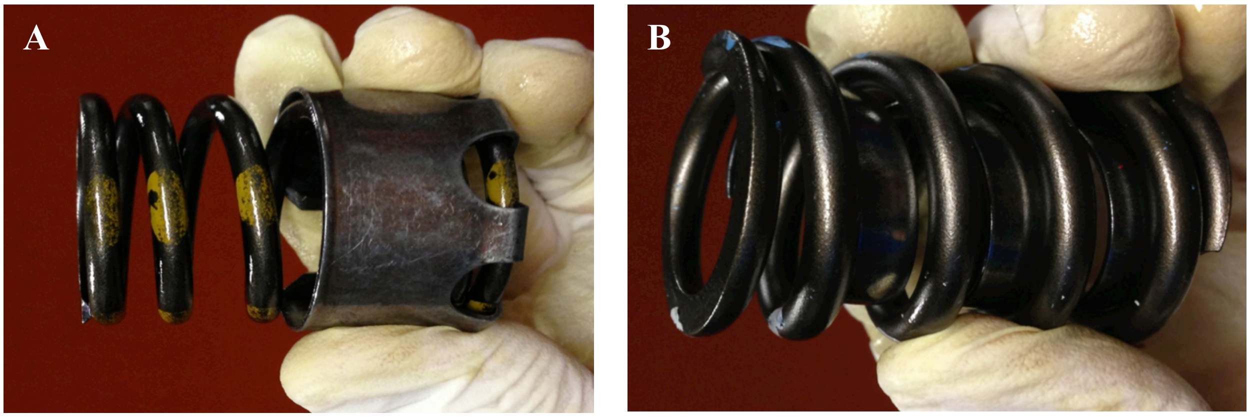


Figure 2A-B: Images of installed cup(A) and wire(B) dampeners.

The VFD used in this experiment, shown below in Figure 3, can adjust the amount of friction force applied to the exterior of a spring. The VFD is composed of a pipe clamp fitted around a regular cup dampener. The amount of dampening can be increased by tightening the pipe clamp's screw. When designing this experiment, it is expected that there will be a linear relationship between tightness and performance: spring surge and bounce failure modes should fall off as the spring becomes "stiffer" with the tightening of the VFD.

Figure 3: Image of VFD installed on Ford Spring

Experimental Procedure

For this experiment, the VFD is tested on a Ford engine spring. Initially, low resolution trials are performed to determine the general effect of the VFD on valve float, valve bounce, and spring surge at different RPMs. After this first round of experimentation, higher resolution trials are performed to identify the characteristics of valve float more successfully. In addition the pipe clamp is re-adjusted for the higher resolution trial to allow for a greater degree of tightening over the course of the trial.

For the low resolution trial, the pipe clamp on the VFD is initially tightened to the point where there is sliding friction between the spring and the dampener. This means that there is just enough friction between the spring and the dampener such that they can slide along each other with minimal frictional resistance. A quasi-static spring tester is then used to measure the force-displacement curve when compressed.

Once the force-displacement curve is obtained, the assembly is put together by inserting the spring (with dampener), valve guide, valve seal, and rocker arm in that order. After all the components are in place, the cover is placed over the assembly and the shaft encoder is attached. The oil is turned on and cycled under the cover to ensure the entire camshaft assembly is lubricated. The oil pressure is monitored and adjusted using the pressure knob to ensure it stays at or near a constant 60 psi throughout testing. The main power is turned on and the camshaft assembly is started by pushing the start button. The camshaft begins at a rotational frequency of 1241 RPM. A LabVIEW program is then used to take measurements and graphs of the lifter force, spring force, and linear variable differential transformer (LVDT) voltage. Once the data is saved, the wheel is used to increase the rotational frequency of the camshaft by 200 RPM and data is then taken again. This process is repeated for every 200 RPM increment until the onset of complete valve float is observed or until a rotational speed of 4900 RPM is reached. The engine then is subsequently lowered back to 1241 RPM and shut down.

The assembly is then taken apart and the test is repeated with the pipe clamp on the VFD tightened by a one-eighth turn. All of the previous steps are repeated until the pipe clamp is tightened by one full turn.

For the higher resolution trial, the procedures are repeated in the same manner, except the VFD is tightened by quarter turns (instead of one-eighth turns) and the rotational frequency of the camshaft is increased at roughly 30 RPM increments from 2000 RPM to 3000 RPM. This is done to better identify the valve float onset speed.

For post-processing the data, force-displacement curves from the spring tester are generated and used to convert the raw voltage data gathered from the piezoelectric force sensors, positioned under the lifter and spring, to units of force (Newton). It is assumed that the springs' responses under the quasi-static testing unit will be similar to their responses in the engine at a low engine speed, 1241 RPM. Linear scaling factors, m, and vertical adjustment factors, a, which brought the spring force data from the engine tests at 1241 RPM, V_{1241}, into agreement with the spring tester data (Figure 4) were found using Equations 7 and 8. These are applied to valve force sensor data at subsequent engine speeds, V_{valve}, to calculate the valve spring force in Newtons.

[Eq. 7] $Spring\ Tester\ Force = m(a + V_{1241})$

[Eq. 8] $Valve\ Spring\ Force = m(a + V_{valve})$

Results

For the first round of experimentation, lifter reaction forces, spring reaction forces, and valve displacements as a function of camshaft rotation were successfully collected in the low resolution trial. After a review of the data, lifter reaction forces, spring reaction forces, and valve displacements were re-taken for the high resolution trial with varying dampener friction forces, obtaining data that more accurately indicated valve float onset speed. A qualitative description of the effects of tightening the VFD on the general response of the spring will be presented. Subsequent analysis of the effects on valve float and valve bounce failure mechanisms will be given in the discussion.

Spring Tester:

As described in the procedures section, prior to engine testing, a quasi-static spring tester was used to estimate the sliding friction force applied by the VFD to the Ford spring. Presented below in Figure 4 are force displacement-curves for the VFD-Ford spring at increasing levels of tightness, T0 to T7. The curves have been laterally shifted such that zero displacement corresponds to the complete seating of the valve in cup dampener after the sliding friction force measurement. The spring was tested up until the point of maximum compression, at which point the spring coils bind and the stiffness greatly increases.

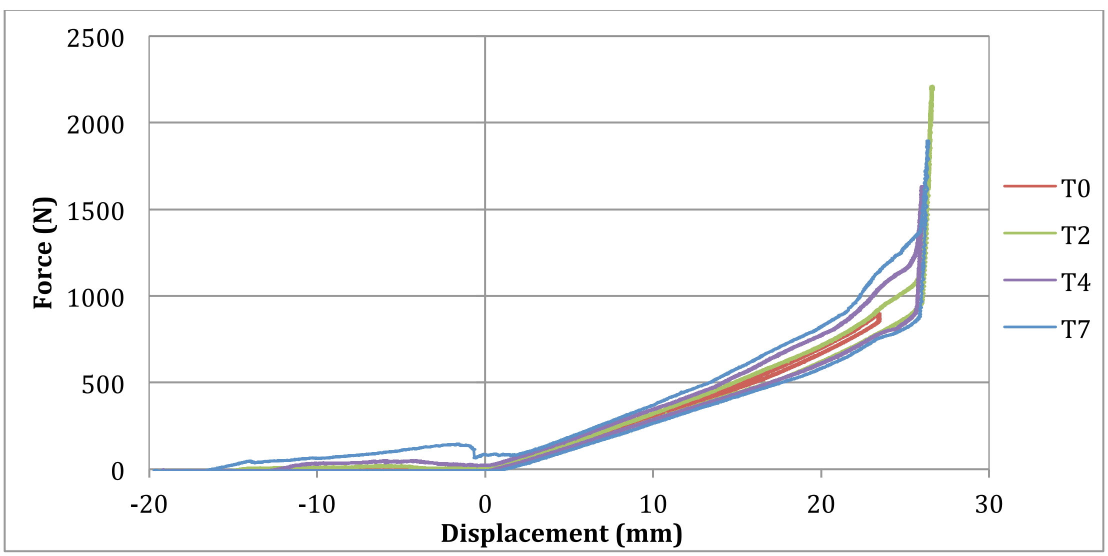


Figure 4: Spring tester force-displacement curves for varying levels of VFD tightness.

As the VFD is tightened, T0-T7, the friction force applied to the spring increases. As expected the size of the hysteresis loop likewise increases indicating more energy is lost during the loading-unloading cycle for the spring. The apparent stiffness of the spring also seems to increase, with higher levels of tightening resulting in higher spring forces. This can be attributed to the spring coils encountering greater resistance to their compression as the VFD is tightened.

From these spring tester curves, the dynamic friction force of the spring tester was also estimated. As mentioned previously, the negative displacement range in Figure 4 corresponds to the dynamic friction test, where the unseated spring is pushed into the dampener. A dynamic friction value was obtained by averaging the spring force values for this region and assuming the spring did not deflect while being seated. A focused graph of the friction test region for varying levels of VFD tightness is presented below in Figure 5. It can be seen that at lower VFD tightness levels the measured force is nearly linear. At higher levels of VFD tightening however, the force value is non-constant as the spring is pushed into the dampener. This variation does not disqualify the use of average spring force values for evaluating the dynamic friction force but does indicate that the measured friction values are relative rather then absolute.

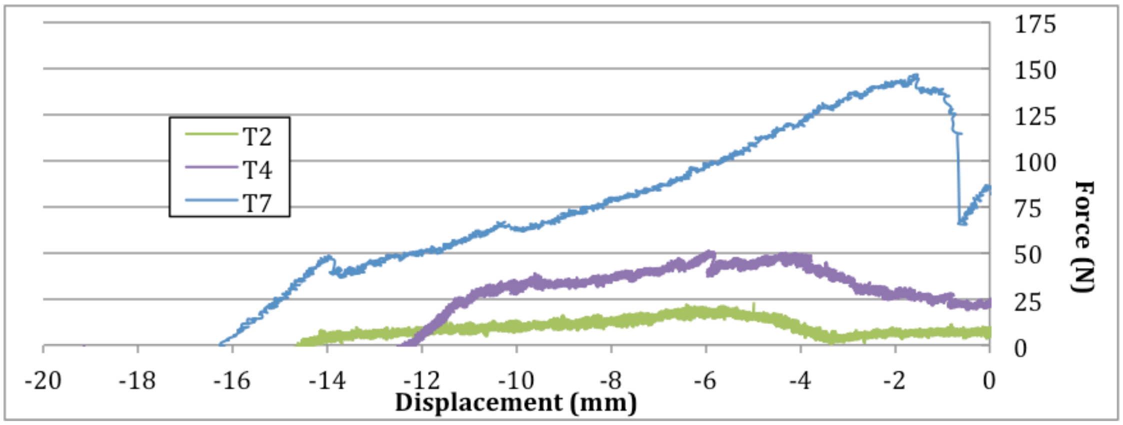


Figure 5: Force-displacement curves used for dynamic friction evaluation.

Presented in Figure 5 are the dynamic friction force values collected for the first experiment, T1-T7, and the second experiment, TA1-TA7. As mentioned previously, in the second experiment, the VFD's pipe clamp was re-adjusted to allow for greater tightening. The T1-T7 values were taken at 1/8 turns of the valve clamp screw, while TA1-TA7 were taken at 1/4 turns. It is clear that a significant range of friction forces can be achieved using this method. However, it is also clear that for the VFD setup used in this experiment, the relationship between the degree of tightening of the pipe clamp screw and the friction force applied is nonlinear.

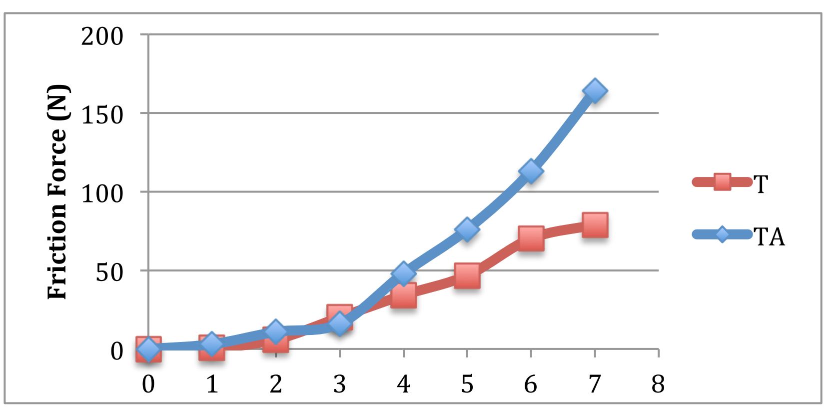
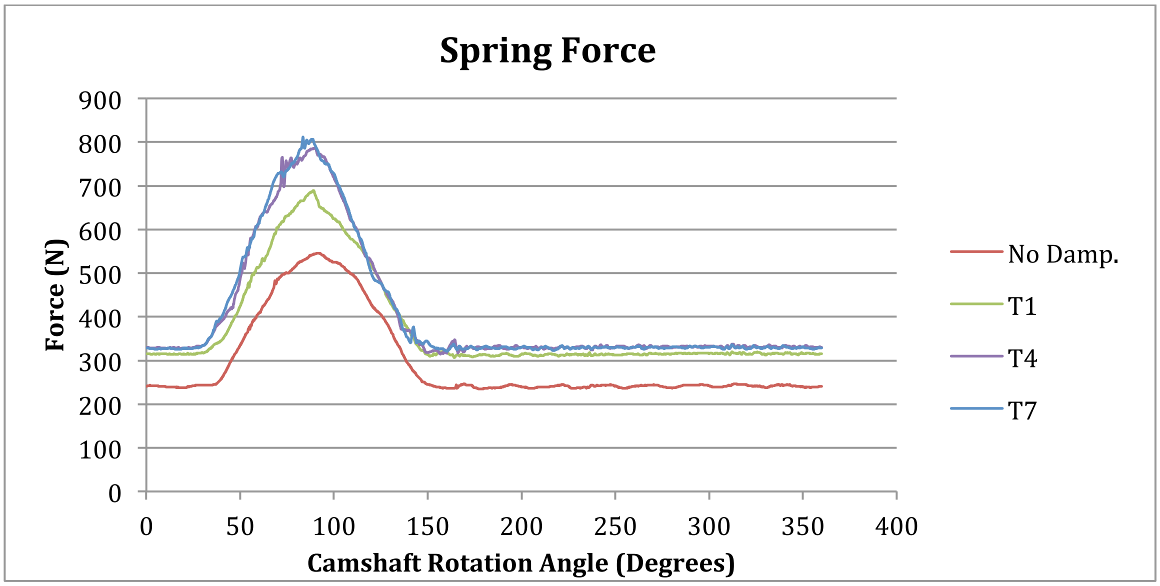


Figure 6: VFD friction force for varying levels of VFD tightening.

Engine Test:
Presented below in Figures 7A and 7B are spring and lifter force curves for varying VFD friction levels along with the base case of no dampener. It is readily clear that the application of the dampener to the Ford Spring has a greater impact on the valve train response then the tightening of the VFD. This is especially visible in the spring force graph. The application of the cup dampener to the un-damped Ford Spring reduces the high-frequency force oscillations and significantly increases the spring force. Increasing the VFD tightness, and hence the friction applied to the spring, produces only a minor increase in the spring force. The effect of increasing the VFD friction force on the lifter force profile is even more difficult to determine from a visual comparison.



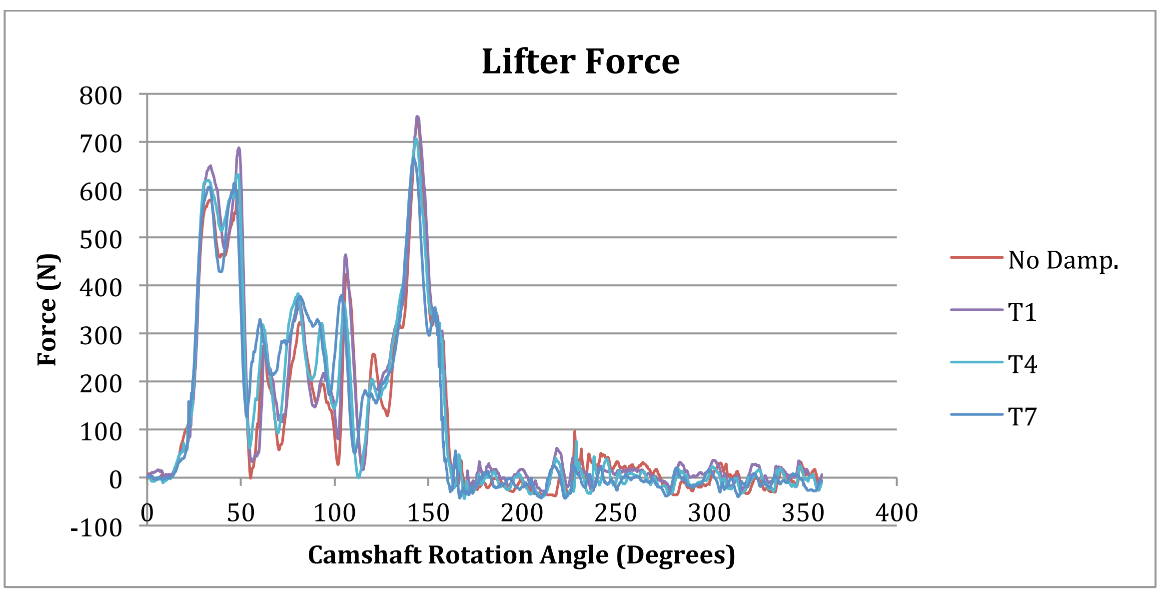


Figure 7A-B: Lifter and Spring force curves as a function of camshaft angle for various levels of VFD tightening.

Experimental Error

A brief discussion of the possible sources of error within this experiment will be presented. A first possible source of error is the assumption that the dynamics of the valve train achieve steady state responses at a specific engine speed. For most of the experiments, force and displacement curves were taken over only one complete revolution of the camshaft, instead of over multiple cycles and then averaged. Differences between the force and displacement curves taken over three non-consecutive cycles was determined to be insignificant. This of course may not be the case for different spring and dampener combinations.

A second source of error is noise in the force and LVDT sensor lines. Without understanding the specifics of the DAQ system used for this experiment it is difficult to analyze the level of noise in the system. However it appears that the noise must be fairly insignificant for the afore mentioned reason that force and displacement curves taken over three non-consecutive cycles for the Ford spring with no shim were in near perfect agreement. If an appreciable level of noise existed one would expect the results to conflict with each other.

It is also possible that thermal effects might be skewing the results of the tests. Although oil cooling is applied during the experiment, an increase in temperature during experimentation caused by a rise in internal stresses as the engine speed is raised is apparent. In addition as various engine tests were carried out in a successive order, the average temperature of the oil increased over the duration of experimentation. This temperature increase could affect the response of the system, for example by expanding components and increasing contact stresses between them as well as altering the dynamics of the system.

Discussion

The two major failure mechanisms identified for this experiment and presented earlier in the theory section are valve float and valve bounce. Qualitative and quantitative examinations of the failure criteria for each are presented below.

Valve Float Failure:

The onset of valve float occurs progressively as the camshaft rotation speed is increased. It is therefore difficult to ascertain at which point an appreciable level of valve float has occurred hence for this experiment we were interested in identifying the valve float onset speed as well as the point at which valve float occurs for the entire cam lift-cycle.

An indicator of valve float is the emergence of large impulse forces at the lifter point and in the valve spring. These forces result from the “collision” of the rocker arm surface and initial lifting ramp of the cam profile. A second less apparent indicator is the zeroing of the lifter force during valve float. An analysis of the dynamics of the system shows that during valve float, the rocker arm acts like a lever with its fulcrum removed and therefore nearly zero force is applied to the lifter point. Figure 8 below shows a representative lifter force vs. camshaft angle curve during complete valve float. The negative forces on the lifter force curve results from the pre-loading of the piezoelectric force sensor.

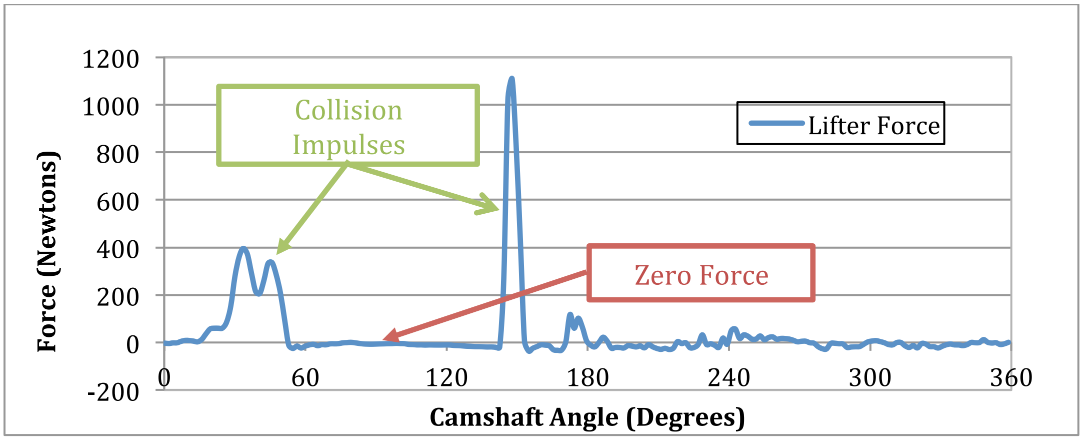
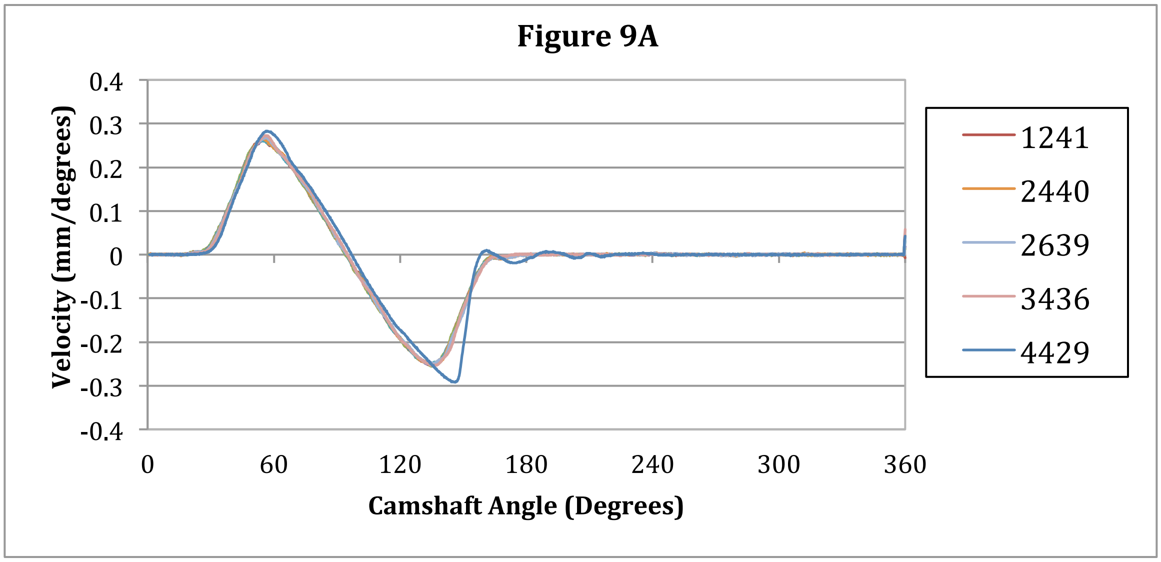


Figure 8: Representative Lifter Force vs. Camshaft Angle Curve for Complete Valve Float

With this definition, it is clear from the lifter force profiles in Figure 8 that the Ford Spring with the VFD never achieved complete valve float. From a visual comparison of the lifter force curves it is difficult to ascertain how increasing the VFD tightness affects the risk of achieving complete valve float. It is certain however that significantly increasing the friction force on the springs does not significantly increase the likelihood that complete valve float will occur.

The engine speed corresponding to the onset of valve float, known as the liftoff speed, can be most readily identified by examining valve velocity curves over a range of camshaft speeds and identifying at which camshaft speeds the velocity curve begins to deviate from its expected profile. To exemplify this method, valve displacement and velocity for an un-damped Ford spring are graphed on the same plots, Figures 9A-B. Figure 9B presents a focused view of the velocity curves at the point of maximum velocity. A running average, with a sampling size of seven data points, was applied to the velocity curve to smooth out the profile. It is clear from these figures that at higher camshaft speeds the valve displacement and velocity curves deviate from their original, ideal paths. These deviations, which are indications that the cam profile is no longer dictating the motion of the valve, are signs of a separation between the cam and rocker surfaces. The increasing valve displacements and higher valve velocities shown on the curves confirm that higher camshafts speeds result in greater and more prolonged valve lift. As it can be seen in Figure 9B, deviation from the original velocity profile first occurs at 2639 RPM.



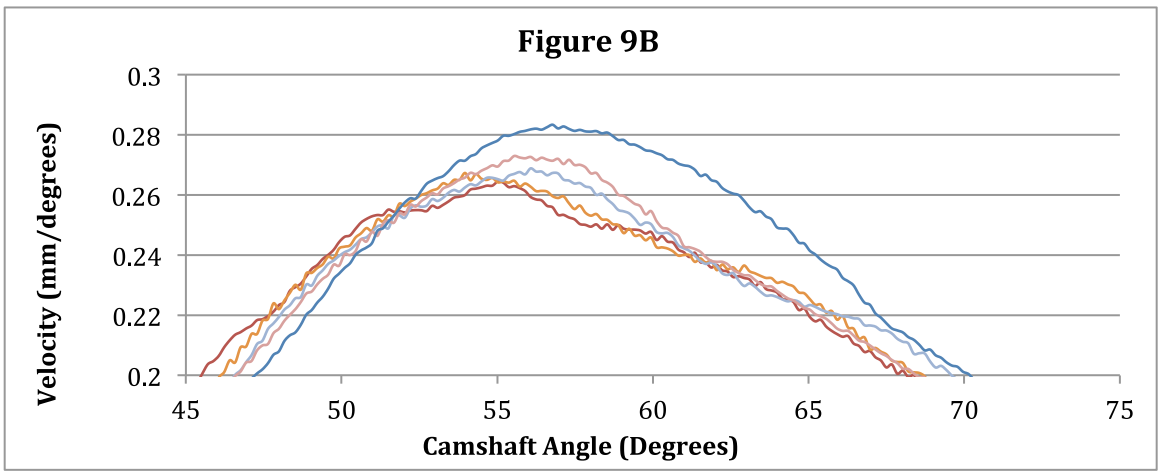


Figure 9A-B: Plots of valve velocity as a function of camshaft rotation. Figure 5B represents a focused view of the velocity curve at the point of maximum spring velocity.

The same method used to identify the valve float onset speed for the un-damped Ford spring can be applied to the Ford springs with an attached VFD. The valve float liftoff engine speeds for varying VFD friction forces values are displayed below in Figure 10. It is clear that as the dampener friction force is increased, by tightening the VFD clamp, the spring's resistance to valve float is increased. This relationship can be explained by noting that as the VFD is tightened, the spring's apparent stiffness increases, as seen by the higher spring forces in Figure 4 above. The "stiffer" springs can provide greater resistance to the acceleration of the valve by the cam-head and therefore greater engine rotational speeds, which produce greater valve accelerations, are necessary to produce valve float. It is interesting that the relationship between friction force and liftoff speed is nonlinear. More experimentation is needed to discover if this trend is localized to the Ford spring or shared amongst other springs.

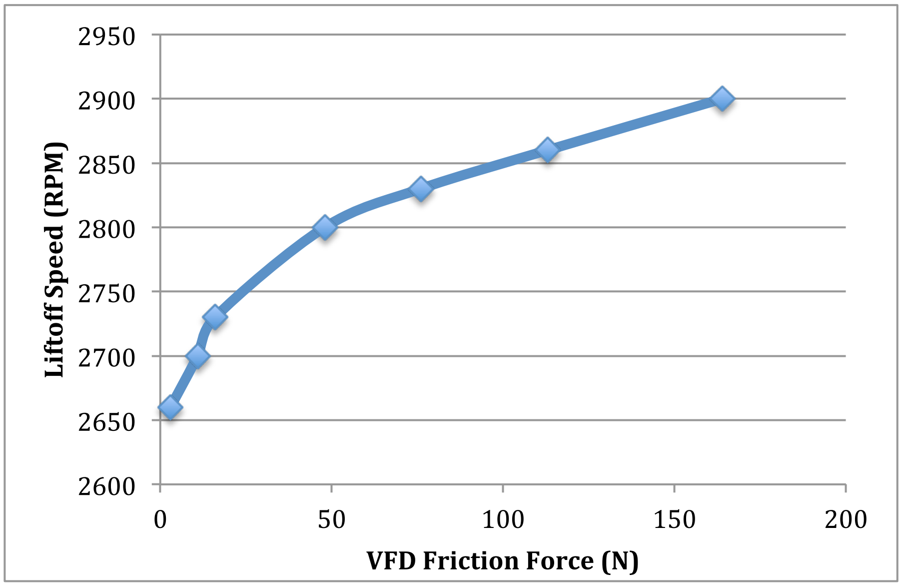


Figure 10: Valve float onset speeds for varying VFD force levels.

Valve Bounce Failure:

Under strong accelerations and insufficient spring force, a valve may bounce off its seat one or more times at the conclusion of the lift cycle. This is an extremely undesirable valve train failure mode known as valve bounce. Unlike valve float, valve bounce causes gas intakes and exhausts to be open out of turn. This can reduce fuel intake or evacuate pressure prematurely from a cylinder. Valve bounce as low as 0.125mm can cost up to 10 horsepower.[4] In addition, valve bounce significantly increases the rate of wear on valve components through repetitive impact stresses. A conservative failure criterion for valve bounce is therefore 0.05mm. A representative example of valve bounce is shown in Figure 11, which plots valve bouncing of the un-damped base spring. In this example, the valve bounces somewhere between 0.06 and 0.22mm before returning to its seat.

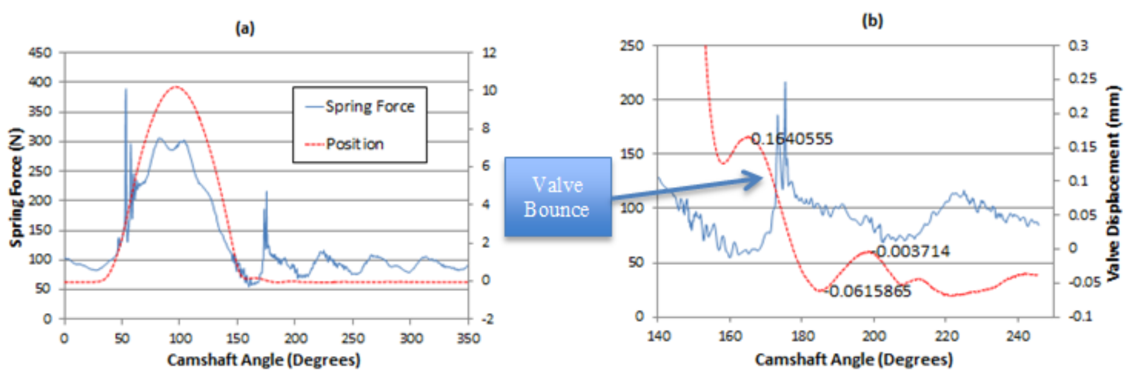


Figure 11: Base Spring, No Shim, Valve Bounce at 4429 RPM, Overview and Detail. The detailed view shows that the valve clearly bounces twice before coming to rest.

During the VFD trials, the quality of the Ford spring prevented strong valve bounce from occurring at the maximum engine speed of 4900 RPM. The majority of tightness levels demonstrated valve bounce under the safety threshold of 0.05mm. Interestingly, the valve bounce mitigation did not go directly as the tightness of the VFD. Figure 12 shows that the optimal setting for valve bounce mitigation was in the middle of the range of tightness settings.

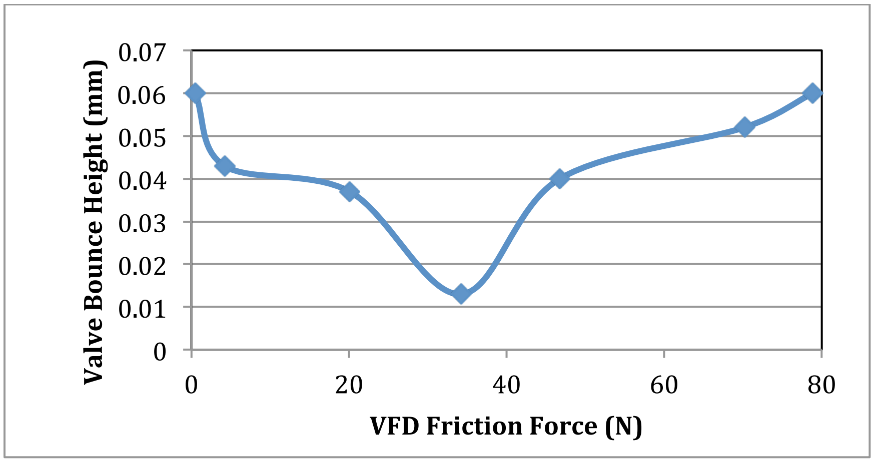


Figure 12: Valve Bounce Height Vs. VFD Tightness Indirect Relationship, Low Resolution Trials, 4900 RPM Engine Speed

This relationship disagrees with the initial hypothesis that engine performance would improve directly with the dampening effects (A direct relationship between valve train performance and shimming effects was demonstrated in previous investigations). However, the relationship is corroborated by a consideration of the valve's free body diagram (Fig. 13).

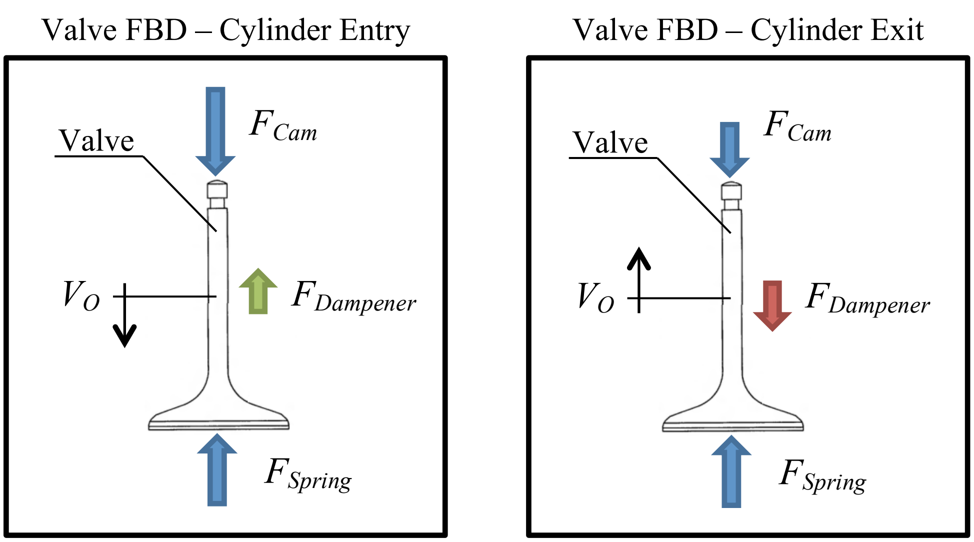


Figure 13: Constructive and Destructive Interference of Dampener Force with Spring Force.

Valve bounce occurs at the end of the lift cycle, as the valve is exiting the cylinder. At this part of the cycle, the dampener resists the spring force, limiting the spring's ability to hold the valve in the valve seat. Relatively low amounts of dampening mitigate valve bounce by mitigating the detrimental effects of spring surge. Therefore, the dampener is shown to have both positive and negative effects in regards to valve bounce mitigation. Figure 12 shows that after 35 N of VFD friction force, the detrimental effects begin to outweigh the positive and valve bounce starts to increase in magnitude with the tightness of the VFD. Therefore for the purpose of valve bounce mitigation in this system, the ideal dampener is an analogue of a VFD with 35N of friction force.

Conclusion

It has been show that the VFD can help negate the onset of valve float as well as valve bounce failure. Increased pressure on the outer coils of the spring in the cup dampener resulted in higher friction forces working against the motion of the spring, which effectively created a “stiffer” spring. The higher stiffness of the spring allowed the cam to follow its ideal path more closely as well as offset the tendency for the cam head and rocker arm surfaces to separate at higher rotational speeds. However it was discovered that if the VFD was over-tightened, the increased “stiffness” could have the detrimental effect of retarding the spring’s motion at the end of the lift cycle preventing it from containing spring bounce. In conclusion, if adjusted properly, the VFD is an effective dampener for negating specific, undesirable failure modes in the camshaft. Although, its practicality in a real engine is uncertain, the VFD can be used to detect the ideal dampening force for a specific valve train, allowing for optimized static cup dampeners to be made and implemented.

Future Work

The set of experiments run with the VFD have highlighted an issue with its current implementation and sparked new ideas in regard to dampener design. The major issue that is found with the current VFD is that the resulting friction force is nonlinear with the clamping tightness. For future experimentation, it would be desirable to have a method in which the additional friction and effective stiffness could be increased in constant increments. On the other hand, the VFD is currently designed to increase friction on the outer coils of the tested spring; however, it could be worthwhile to see how a VFD that increases friction to the inner coils of a spring compares. This could be achieved by modifying a flat-wire dampener to expand after being placed inside the inner coil of the spring. Lastly, the VFD could be combined with other dampeners, including the proposed design for a VFD that targeted the inner coils of the spring, to see if the compounded frictional effects further improve the performance of the camshaft dynamics.

Furthermore, the set of experiments that were run thus far have helped to develop a qualitative measure for the onset of valve float and valve bounce. Although many different configurations of springs were tested, there are other experiments that may be useful in order to fine-tune the current criteria for failure. Possible future work could include changing factors outside the main spring valve in the standard procedure and improving data collection methods. Other factors to be tested in the standard procedure include changes to the remaining seven springs in the valve train and testing of different grade oils. Changing the remaining seven springs in the valve train would give insight to the effects of compounding vibrations that could be transmitted to the primary spring which is the only spring monitored during testing. Testing different grade oils in the system would give insight into the concern of heat buildup and its effect on cam dynamics as touched on in the experimental error section. Improved data collection methods could include changes to the LabVIEW GUI’s and additional procedures in order to collect more data on the valve train efficiency. The LabVIEW programs used can be updated to include scaling factors for the sensors used in the system which currently only output raw voltage levels, rather than calibrated force values. Collecting data on valve train efficiency, by calculating friction forces through the time elapsed before the system comes to a complete stop after cutting off power, could provide another criteria for spring evaluation.

References

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Appendix A: Ford Spring Measurements

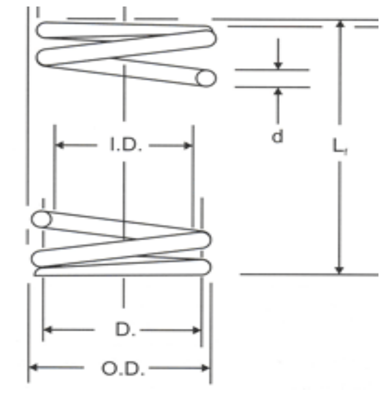

Figure A.1: Schematic of spring with outer diameter (O.D.), inner diameter (I.D.), spring diameter (d), and free length (listed).[5]

Parameter	**ID (mm)**	**OD (mm)**	**Wire Diam. (mm)**	**Free Length (mm)**	**Installed Height (mm)**	**Number of Coils**
Value	25.73	34.29	4.37	49.43	39.93	6

Table A.1: Measured Ford spring dimensions