

Characterization of Caliper Piston Material Stiffness and Damping

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ABSTRACT

The brake caliper piston plays a key role in caliper function, taking significant responsibility for qualities such as fluid consumption, insulation of the brake fluid from heat, seal rollback function, and brake torque variation sensitivity to disc thickness variation. It operates in a strenuous environment, being routinely subjected to high stresses and elevated temperatures. Given all of the demands on this safety-critical component (strength, stiffness, wear resistance, stable friction against rubber, thermal stability, machinability, manageable thermal conductivity, and more), there are actually relatively few engineering materials suitable for use as a caliper piston, and designs tend to be limited to steel, aluminum, and engineered plastics (phenolic composites). The lattermost - phenolic composites - has been of especial interest recently due to mass savings and possible reduction in brake corner judder sensitivity to disc thickness variation. This paper focuses on characterizing two important mechanical characteristics, stiffness and damping, of the most common piston materials, steel and phenolic. Data are shown first suggesting the effect of piston material on brake performance, and then stiffness and damping data from different methodologies are presented. From these data, a preferred methodology is recommended and results are reconciled with brake corner subsystem performance and modeling.

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INTRODUCTION

When the modern brake caliper is compared to its ancestors of one to two decades ago with a casual eye, the comparison defies the years of evolution incorporated into the modern design. Indeed, the basic function remains unchanged. However, modern demands of crisp, short-travel pedal feel, ultra-low residual drag, light weight, and trouble-free operation (no noise, no judder) have forced an extremely high level of optimization, showing up in the subtle details of almost every aspect of design. Optimization of the caliper piston (geometry and material selection) has been included in the quest for the most optimal caliper design. Within the

realm of piston design such questions arise as "what is the contribution of the piston itself to caliper fluid consumption", "does the piston itself have any appreciable damping characteristics". The piston is a compelling component to try to influence brake torque variation sensitivity to disc thickness variation, for reasons including its central location to the pad (stiffness can be tuned without degradation to pad to rotor pressure distribution), and the fact that it is not "consumable" like the brake pad assembly - its characteristics exist for the life of the caliper.

The caliper piston is a very stiff component; its deflection characteristics under typical operating loads are comparable

in magnitude to that of most devices that could be used to measure them. Furthermore, the caliper piston interacts with the hydraulic brake fluid on one side, which generates a distributed load over the piston crown. This in turn causes both bending (at the dome) and compression (of the walls) of the piston material, which makes it important to generate a realistic loading condition to get accurate deflection.

The interest in measuring piston stiffness and damping began with the observation of a measurable reduction in brake torque variation observed during a test for thermal roughness. This paper will start with a further elaboration of the testing that led to this interest, and cover pressure distribution and piston temperature tests designed to determine the operating conditions of the piston during thermal roughness testing. It will then detail a series of 4 different methods of characterizing these characteristics, including (1) Dynamic Mechanical Analysis (DMA), (2) an Elastomer Test Machine, (3) an air over hydraulic pad compressibility machine, and (4) a direct-acting hydraulic brake component compressibility machine. Following this, a brake torque variation sensitivity model is used to verify the significance of the observations. Finally, the results are summarized and some conclusive remarks offered.

The authors are well aware of the differences in parameters (and corresponding units of measure) selected and reported for both damping and stiffness characteristics of the phenolic and steel pistons evaluated. The effect of parameter selection on the damping and stiffness values reported have been considered at some length without reaching a definitive resolution. It is fully expected that it depends upon the experimental method employed and the specific test conditions evaluated. Throughout this paper, like methods and like units within one measurement technique were used for both materials to enable comparisons, however, caution should be exercised in comparing data between different measurement techniques.

BACKGROUND

Thermal Roughness

Thermal roughness is a term used to describe unstable growth of disc thickness variation (DTV) and the resultant brake torque variation (BTV). Simplistically, the unstable growth of DTV occurs when the growth of relative thick regions of the brake disc due to thermal expansion incurred while absorbing the energy input from braking exceeds the mitigating effect of heat rejection to the surrounding regions. Thermal roughness (BTV) behavior is driven by both the propensity of the brake corner to increase DTV during a braking schedule, and the brake corner's sensitivity to DTV. Even with the highly simplified mechanism described above, one may readily see how parameters such as rotating speed, braking pressure, simulated vehicle mass (inertia), pad and rotor thermal conductivity, pad length, and pressure distribution can affect DTV growth. Other parameters such as pad and insulator compressibility, piston compressibility,

housing stiffness, and pad to piston/housing contact conditions can affect BTV sensitivity to the DTV that develops. Many of the parameters that affect BTV sensitivity can also affect DTV growth, through affecting the pressure distribution.

Thermal roughness in the field typically manifests itself as an objectionable vibration of the steering wheel and/or brake pedal, and vibration or low frequency audible noise from the body structure, all in response to the torque variation at the brake corner. It will occur with varying degrees of ease in customer vehicles - with rotors damaged by corrosion or rough usage, that have a high level of DTV to start, thermal roughness can start with just a few braking events and slight temperature increase of the discs. With new, low DTV discs, it can require aggressive or extensive braking, such as repeated braking from highway speeds or extended braking down a long grade. In market regions that have frequent higher speed braking events or commuting routes with extensive elevation changes, thermal roughness is believed to contribute significantly to pad and rotor replacement warranty and customer complaints due to pulsation.

Thermal roughness (or more generally, thermo elastic instability in brakes) has been studied and reported on extensively [1,2,3,4,5]. However, relatively little of the research has been focused on factors affecting the BTV sensitivity aspect, with the work by Leslie [6] being one of the few.

The authors' interest in the contribution of the caliper piston to roughness behavior began with observation of consistently different thermal roughness behavior between two very similar brake corner designs, both using single piston, cast iron sliding calipers and identical friction material formulation on cast iron discs of very similar geometry. The thermal roughness behavior was measured on an inertia dynamometer test. Although the caliper hydraulic piston sizes were the same, the caliper designs differed slightly and the piston materials differed as well (one was steel and the other phenolic).

An inertia dyno-based test for thermal roughness was used that consisted of a simple sequence of repeated light snubs (not complete stops) from highway speeds with a defined cycle time. Four (4) sets of fifteen (15) snubs with complete cooling in between sets comprise the test. [Figure 1](#) below shows the thermal roughness behavior between the two brake corners tested at identical inertial loads. Brake corner A (with caliper A) exhibited consistently low BTV (less than 45 Nm on this sequence), while brake corner B (with caliper B) exhibited significant thermo elastic instability, with BTV growing to peaks of 85 Nm.

It should be noted that the above behavior is reflected in testing on many samples of both corner configurations, which the described comparison test summarized nicely. A series of component swaps was performed between the two brake corners to try to identify why the behavior was so different between very similar designs with identical friction materials. For brevity, the full extent of the study will not be

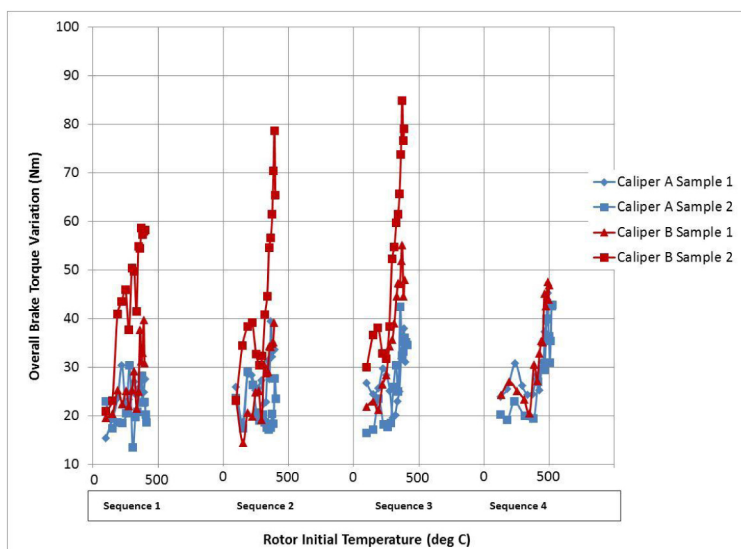


Figure 1. Comparison of Thermal Roughness Behavior on Inertia Dyno Test between Two Similar Brake Corners

summarized here. However, the study did identify the caliper piston as being one of the key factors explaining the difference in behavior. This was evidenced by a swap of the phenolic piston present in Caliper A with the steel piston present in Caliper B and re-testing both. Results of this are summarized in [figure 2](#):

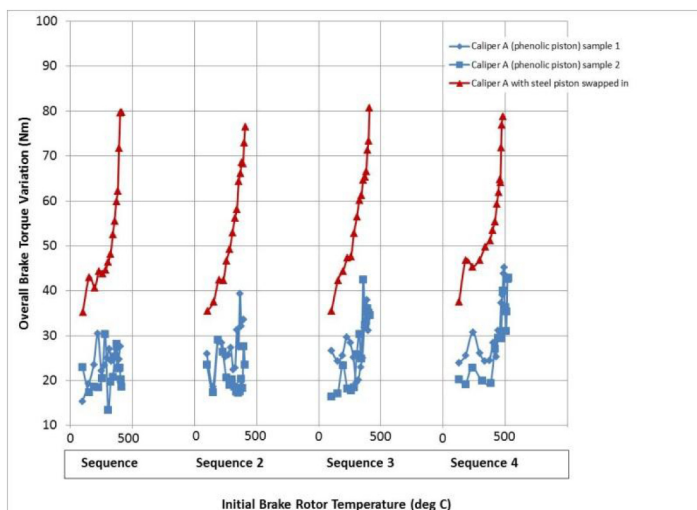


Figure 2. Thermal Roughness of Caliper A with steel vs. baseline phenolic piston

The caliper fluid consumption and pad to rotor friction levels were monitored through all testing. It was of great interest to note that brake corner A - with the lower BTV response - actually had higher overall hydraulic stiffness. This was evidenced by lower fluid consumption in caliper A and similar friction levels.

The compelling difference in thermal roughness propensity between the two piston materials in two similar brake corners combined with the lack of explanation in a

simple hydraulic stiffness comparison lead to the conclusion that there was a more profound explanation to be had. This led to an effort to understand the parameter or parameters that best characterized the contribution of the piston to roughness. To start, it was deemed helpful to understand what the operating conditions of the piston (temperature and loading condition) during typical thermal roughness events were.

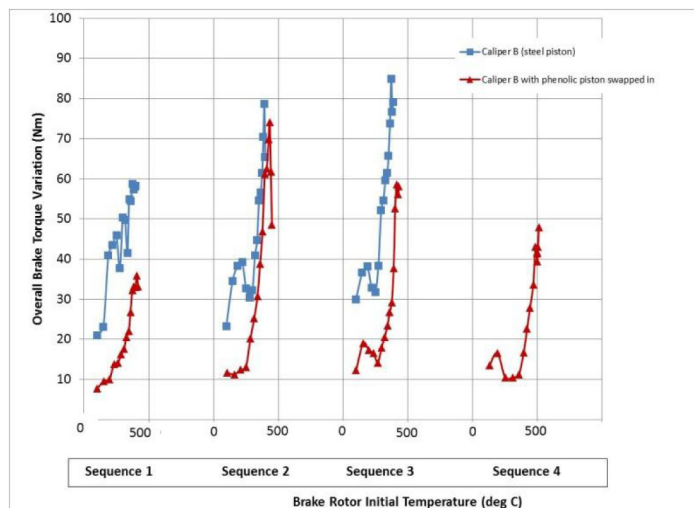


Figure 3. Thermal Roughness of Caliper B with baseline steel vs. phenolic piston

Operating conditions of piston

Two measurements were completed to help define the operating conditions of the piston during the cyclical thermal roughness test described in the previous section. First, a thermocouple was installed into a drilled hole located in the piston at the dust boot groove near the pad backing plate. This was used to monitor operating temperature near the pad backing plate. Second, after the test was complete and the

brake components cooled back to room temperature, a dynamic pressure distribution measurement between the piston and inner pad was made while performing one of the braking events in the cyclical thermal roughness test (the temperature was kept below 150 deg C on the brake rotor for this measurement to protect the integrity of the thin-film pressure sensors).

The plot below shows the measured piston and brake rotor in-stop maximum temperatures during the cyclical thermal roughness test. The piston reached peak temperatures of approximately 112 deg C with a corresponding rotor maximum temperature of close to 600 deg C. A test was also completed with a low-met lining, during which piston temperatures peaked at 146 deg C.

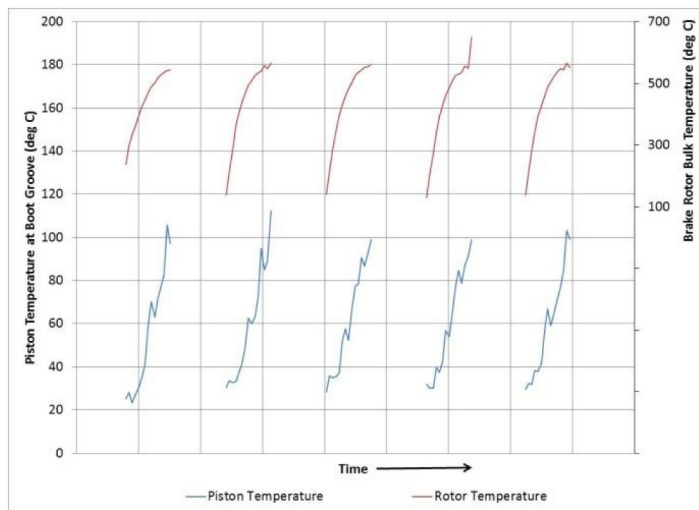


Figure 4. Measured Maximum Piston and Rotor Temperatures during Cyclical Thermal Roughness Test with NAO Lining

Figures 5 - 6 below show the force magnitude and center of force measured by thin-film electronic pressure sensors placed between the piston/finger and pad interfaces. Due to nonlinearity in the individual sensors in the film, the peak loads recorded are contact-area sensitive, and to compensate for this, the data on the finger side were normalized to the same peak as the data on the piston side.

Interestingly, the finger interface built up to peak force more quickly than the piston interface, which is only possible through the piston exerting some form of leverage via a rocking motion of the housing. Indeed, as the piston applies against taper-worn pads, the force at the pad interface will tend to rock and bind the piston within its bore, leading to frictional losses. The housing may still 'rock' into contact with the outer pad. As the apply completes, a very regular pattern of force variation develops in both interfaces, conforming that the variation is not mere noise, but a measurement of the variation driven by rotor LRO and DTV.

The center of force measurements confirm that the force transitions from near centered in the off-brake condition, to a trailing edge and radially inboard bias (which is driven by the

very typical inboard pad taper wear pattern of thicker towards the trailing and radial inner edges). The piston does become side-loaded - the offset in the center of force from center requires the side to loads develop between the piston and the bore to balance this offset. By observing the "orbit" of the center of force during the apply, one may see that it never crosses center - the piston remains side-loaded in roughly the same position throughout the apply, and simply orbits in a tight elliptical pattern during the apply.

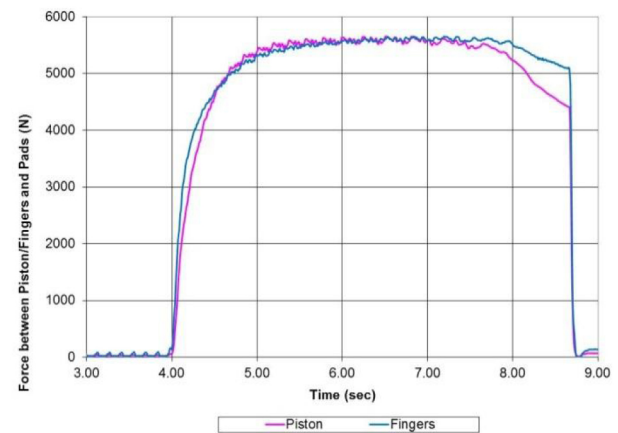


Figure 5. Measured Piston and Fingers to Pad Interface Force during Highway Speed, Light Pressure Braking Event

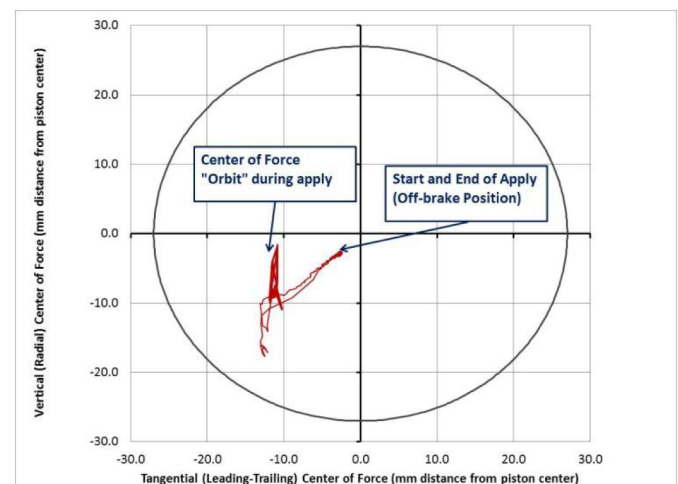


Figure 6. Measured Piston to Inner Pad Center of Force during Highway Speed, Light Pressure Braking Event (NOTE: Leading-Trailing COF is positive towards the leading edge, Radial COF is positive towards the rotor OD)

The significance of this is that it strongly suggests that the difference in BTV sensitivity between the steel and phenolic piston is not due simply to the higher piston to bore clearances inherent in phenolic piston designs. Between the elevated temperatures during thermal roughness and greater

thermal expansion of the phenolic piston, and the fact that with pad taper wear the piston remains tilted in roughly the same spot and only rocks slightly around the contact point to the bore, it became clear that the source of the difference in BTV sensitivity was more likely related to the different material characteristics. It is acknowledged that the contact conditions may change when the brake is hot - however, the piston to bore clearances between the two piston materials also tend to equalize more with increasing temperature as well. Therefore, the balance of the investigation centered on finding and characterizing the material property or characteristics that best explain the difference.

The next four sections detail the different methodologies used in attempts to characterize the piston material and component. It will be noted here that all four efforts attempted, within the capabilities of the test machines employed, to determine the stiffness and the damping characteristics of the piston materials and/or component in a frequency range up to 30 Hz (slightly above the typical 1st order frequency with respect to wheel rotation for fast highway travel). The prevailing hypothesis was that the phenolic material especially, being a polymeric composite, may have some rate-dependent mechanical characteristics. Testing with all four methods focused on comparisons between steel and phenolic materials/pistons, since data of the type generated were not readily available with either material, and the comparison would be important to evaluate the significance of the measurement.

REVIEW OF MEASUREMENT TECHNIQUES AND RESULTS

1. Dynamic Mechanical Analysis (DMA) testing of storage and loss moduli on sample coupons

The first attempt to characterize key characteristics of piston materials was dynamic mechanical analysis, or DMA testing performed by Professor Deborah Chung of the University of Buffalo for Sumitomo (then Durez automotive). In this testing, $25 \times 6.35 \times 3.18$ mm (for flexure, with a 20mm span) and $6.35 \times 3.18 \times 3.18$ mm (for compression, with the compressive load born by the 3×18 mm square surface) sized specimens were subjected to a low-amplitude (1N), sinusoidal load. Testing was completed across a matrix of temperature and frequency levels in both flexure and compression. The DMA testing reported as results the storage modulus (representing the real, or purely elastic portion of the complex modulus) and a loss modulus (representing the viscous, or imaginary portion of the complex modulus). The loss modulus is the product of the storage modulus and the loss tangent (with the loss tangent in turn representing the phase lag between the stress and strain waves). The purpose of testing in both flexure and compression is because in the actual piston component, both loading conditions are present, with the walls of the piston bearing a mostly compressive

load, and the dome of the piston bearing both compressive and bending loads.

Selected results are shown in the following figures. Figure 7 shows the storage moduli for steel and phenolic samples in flexure (read the left hand y-axis) and in compression (read the right-hand y-axis), measured at 20 Hz frequency across a temperature range from ambient to 150 deg C. The ambient (25 deg C) condition was repeated

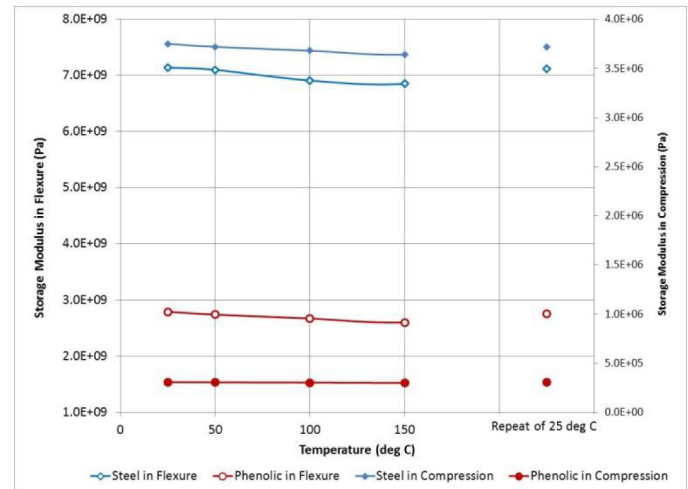


Figure 7. DMA Measured Storage Moduli vs. Temperature at 20 Hz for Steel and Phenolic Specimens

Some notable observations are apparent in the above data. Comparing steel and the phenolic material, the steel was roughly 2.5 times more stiff in flexure, and over 7 times as stiff in compression. Furthermore, the storage moduli for both materials were approximately 3 orders of magnitude higher in flexure versus compression. Figure 8 shows the corresponding loss moduli for the materials in the same test conditions as above.

Comparing now the loss moduli, it may be observed that the phenolic material had approximately 3.5 times the loss moduli as the steel in flexure, but were very close in compression. The loss moduli in flexure for both materials were 4 orders of magnitude higher in flexure versus in compression. The final chart, in figure 9 below, shows both the storage and loss moduli for both materials plotted versus frequency (all at ambient temperature, or 25 deg C).

Both the storage and loss moduli of both materials tended to decrease with increasing frequency, with the loss modulus of the phenolic material showing the most significant drop between 1 and 10 Hz. The storage modulus of the phenolic material remained at roughly 30 to 50% of that of steel throughout the frequency range. The loss modulus of the phenolic material was higher than steel at 1 and 10 Hz, but dropped below that of steel at higher frequencies, which was oddly inconsistent with the results in figure 8 at constant frequency.

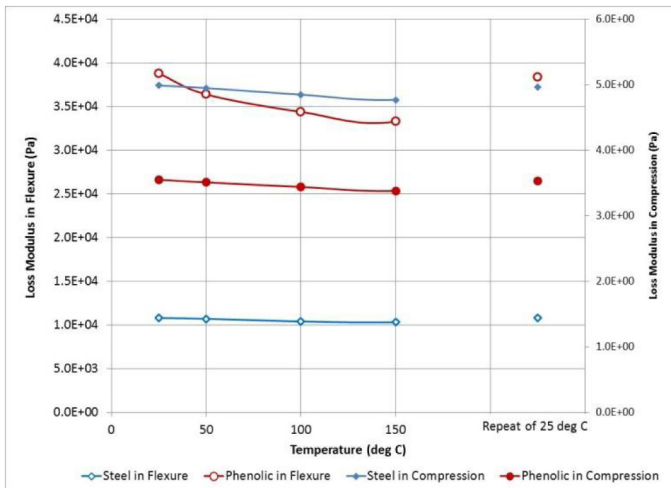


Figure 8. DMA Measured Loss Moduli vs. Temperature at 20 Hz for Steel and Phenolic Specimens

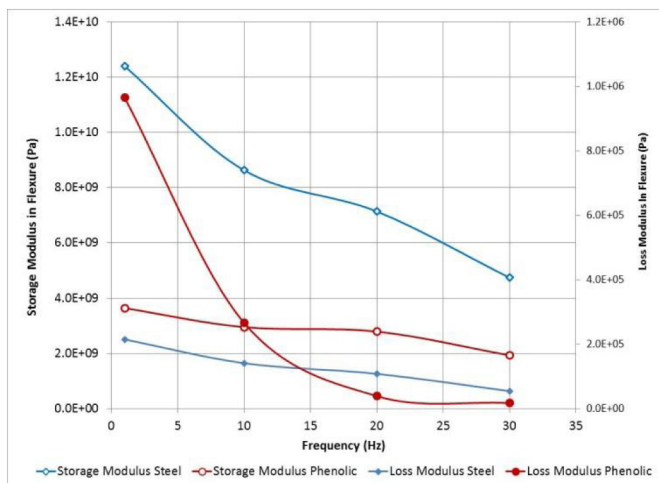


Figure 9. DMA Measured Storage and Loss Moduli vs. Frequency at 25 deg C for Steel and Phenolic Specimens

The results suggested that the phenolic material's dynamic stiffness was significantly lower than that of steel throughout the range of conceivable temperature and frequency conditions for operation, and also suggested that there could be a difference in damping that favors the phenolic material. However, there remained concerns about the low load range (1N), the knowledge that composite materials often exhibit substantial non-linearity in mechanical characteristics versus load, and the inability to draw firm conclusions about the relative damping of the two materials. Furthermore, a hypothesis remained that the mechanical characteristics of a piston component could have a strong geometry effect present, and testing on a bar-shaped sample could not capture these effects.

The results of the DMA testing were noted as useful clues, but the quest for deeper understanding continued. The next major effort was focused on characterizations made with

an elastomer test machine, of the type used to characterize rubber suspension bushings.

2. Stiffness and damping measurements of piston samples on elastomer test machine

The elastomer test machine was operated by the General Motors elastomer test lab at the Proving Grounds in Milford, Michigan. The test specimens were finished piston components, fully machined and installation-ready into a caliper. A 3/8" hose was drilled into the piston domes through which a bolt could be fitted, and an area surrounding the bolt hole was milled flat in an end mill to provide a flat area for a steel washer to fit. A bolt was installed through the hole with a steel washer on either side; the bolt provided an interface for the elastomer test machine head to fasten to. On the pad end of the piston, the piston was gripped in a collapsible collar which very closely matched the diameter of the piston (within tenths of a mm). The collar gripped the piston about halfway up, which was done to prevent damage to the piston from the collar (it is acknowledged that it would have been preferable to grip the piston as close to the end as possible). Every effort was made to minimize the impact of the fixturing on the stiffness and damping characteristics of the piston; however, due to the high stiffness of the piston, it is still likely that the fixturing had some influence.

The photo in figure 10 illustrates how the piston (phenolic piston is shown) was fixtured into the test machine. The machine had an actuator which moved the collar end of the piston, and a frame above the test specimen supported a load cell that measured the reaction load at the bolted end. The dynamic characteristics of the machine were measured and compensated (the compensation occurred automatically in the data processing software), but it was unknown how effective this compensation would be while measuring a specimen of comparable stiffness to the machine itself.



Figure 10. Phenolic Piston Sample fixture into Elastomer Test Machine

The pistons were given a mean load of 10,000N (selected to prevent control issues with the test machine, which was