

Further Research into the Role of the Caliper Piston in Brake Roughness

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ABSTRACT

Previously published research [1] covering the role of piston material properties in brake torque variation sensitivity and roughness concluded that phenolic pistons have significantly higher low-pressure range compliance than steel pistons, which promotes lower roughness propensity. It also determined that this property could be successfully characterized using a modern generation of direct-acting servo hydraulically actuated brake component compression test stands.

This paper covers a subsequent block of research into the role of the caliper piston in brake torque variation sensitivity (BTV sensitivity) and thermal roughness of a brake corner. It includes measurements of hydraulic stiffness of pistons in a "wet" fixture, both with and without a brake pad and multi-layer bonded noise shim. Inertia dynamometer measurements of BTV sensitivity against a machined 20-micron thickness variation rotor with multiple (prototype) phenolic piston geometries, as well as steel pistons with two different piston to bore clearance levels followed. Finally, higher-temperature thermal roughness testing of multiple phenolic piston geometries and steel pistons with tighter and looser piston to bore clearances were run.

The results of these studies give some insight into the role of the caliper piston - its material properties, geometry, and clearance to the bore - in brake roughness and performance.

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INTRODUCTION

Many questions were left at the conclusion of the authors' first study. Can the role of piston to bore clearance be verified more directly (the previous study used anecdotal information from piston to inner pad interface pressure distribution measurements to conclude that the role was minimal)? Can the geometry of a phenolic piston be "tuned" to minimize brake torque variation without too steep a penalty in fluid consumption? Different piston geometries affect the interface to the elastomeric and somewhat compressible noise shim - how important is this interface? Is the effect of the piston on roughness temperature-dependent?

A series of studies were run in order to explore these questions further. This paper is presented in the following sections, which covers each of the studies as well as some supporting information. The rationale and/or theory being explored is described in the introduction to each section:

1. Description of Piston Samples
2. Piston Stiffness vs. Geometry in Wet Fixture - High Force

3. Wet Fixture Measurements at Low Force - Adding Pad and Shim
4. Brake Torque Variation Sensitivity - 1st Study
5. Brake Torque Variation Sensitivity - Design Of Experiments
6. Thermal Roughness Study
7. Further Analysis of Results
8. Summary and Conclusions
9. Further Research

1. DESCRIPTION OF PISTON SAMPLES

Prototype pistons were created to support the series of studies that were planned. Two steel pistons of identical design save for one achieving a clearance on the "tight" side of typical design range, and the other achieving clearance on the "loose" side of typical design range, were created to help isolate the effect of piston to bore clearance on brake torque variation. Twelve phenolic pistons, with three different wall thicknesses, and 4 different dome thicknesses, were created to allow for testing the extent to which piston geometry can affect stiffness and brake torque variation. All of the prototype

phenolic pistons were ground to a piston to bore clearance on the “loose” end of a typical design clearance range. The exact clearance values are not published in order to maintain some propriety, however, the piston to bore clearances for all pistons in all of the studies in this paper were within a “typical” design range.

For ease of reference, the phenolic piston samples were numbered. Two identical pistons were created of each geometry (to support studies being conducted at multiple locations), and there were numbered from 1 to 12. Pistons 1 through 4 were thick-walled, and ranged from maximum dome thickness on piston set 1 to minimum dome thickness on piston set 4. Pistons 5 through 8 repeated this trend with a medium wall thickness, and pistons 9 through 12 repeated it again with minimum wall thickness. All piston samples - steel and phenolic alike - used in this study are intended to fit a 60mm caliper piston bore (and were therefore dimensioned slightly below 60mm to provide the intended clearance).

Figure 1 shows the full set of phenolic pistons created to support the various studies, and figure 2 shows a representative steel piston.



Figure 1. Photograph of Prototype Phenolic Pistons of Various Geometries (2 samples of each, labeled “A” and “B”)



Figure 2. Photograph of Prototype Steel Piston (tight and loose specimens appear identical)

2. PISTON STIFFNESS VS. GEOMETRY IN WET FIXTURE - HIGH FORCE

The first phase of compression stiffness testing involved all 12 phenolic piston geometries, and both tight and loose steel pistons, and carried the goal of determining the role of piston geometry in its stiffness behavior. All testing took place on a Greening model 1140 pad compressibility test machine. The pistons were loaded into a “wet” fixture (see figure 3) that allowed the compression test machine to apply load against the piston face and support it at the backside of the cylinder (the cylinder backside was ground as flat as possible to minimize the effect of non-uniform contact against the machine on the measurement). The wet fixture was used so that the hydraulic load would be distributed realistically over the piston face, and the dome geometry of the piston would play a role in the compression stiffness.



Figure 3. Piston Compression Test Wet Fixture

The stiffness of each piston configuration was measured in a range of 2kN to 30kN (equivalent to the force exerted by about 667 kPa to about 10,000 kPa of hydraulic pressure). The rationale for the high peak force was to insure the maximum differentiation in stiffness could be observed, and that any small error due to unintended non-uniformities in the flatness of the contact faces would be outweighed by geometry effects. Measurements were conducted in random order. Stiffness were reported as the inverse of the slope of a linear fit of the deflection versus load data in the 2 kN to 30 kN range. Data are presented in two formats - figure 4 shows the compression stiffness by piston configuration, and figure 5 shows the compression stiffness of the phenolic pistons only versus the volume of material in the piston.

Of immediate note in figure 4 is the fact that although the steel pistons both exhibit higher stiffness than all of the phenolic pistons, it only about 40% more. This contrasts with the earlier measures in [1] suggesting up to a 300% difference. The different phenolic piston geometries have a reasonably intuitive effect on stiffness (with some exceptions), with the thick walled pistons having the highest stiffness, the thin walled pistons the lowest, and stiffness scaling with dome thickness for the medium and thin walled sets. Some interesting observations include:

- The thick-walled pistons (see [figure 5](#)) were only minimally affected by the dome thickness. This makes sense given the very low area of the dome in this set; clearly the piston walls were supporting most of the hydraulic load.
 - The thin walled pistons were highly affected by dome thickness; this indicates that the dome is supporting a large percentage of the hydraulic load.
 - The medium-walled pistons showed - not surprisingly - a medium sensitivity to dome thickness.
 - The full set of pistons showed a very high correlation of stiffness to total material volume in the piston (correlation coefficient > 91%)
 - The full solid piston was actually less stiff than the thick-wall specimens. This suggests that the influence of the small cavity in the thick-wall pistons on stiffness is negligible, and it was likely variation in individual specimen material properties and/or uncontrolled geometric non-conformities (such as contact face out-of flatness) dominating the stiffness behavior. Sample 4B of the thick-walled set, although having the least material of this set, was the stiffest phenolic specimen, which is likely explained by similar rationale.
- Armed with the re-assurance that the piston geometries had more or less the intended effect on stiffness, a second phase of compression testing took place (and is covered in the next section).

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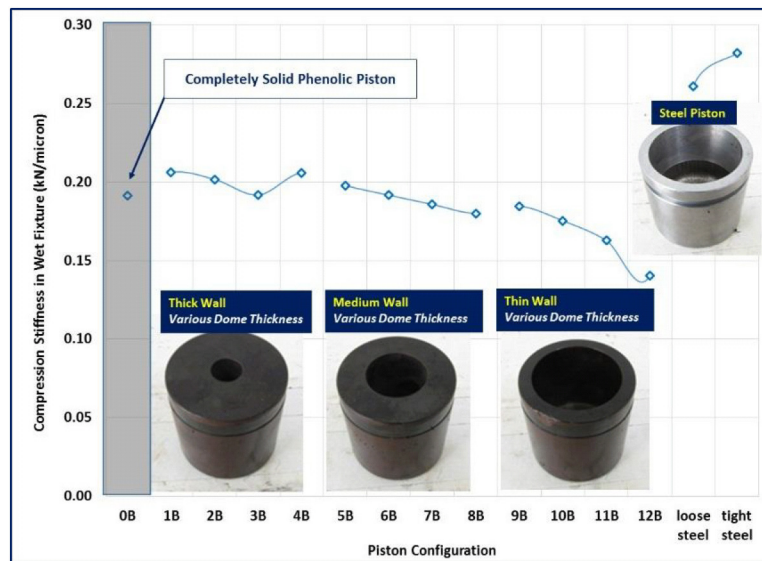


Figure 4. Summary of Piston Compression Stiffness Measurements, 2-30 kN Force Range

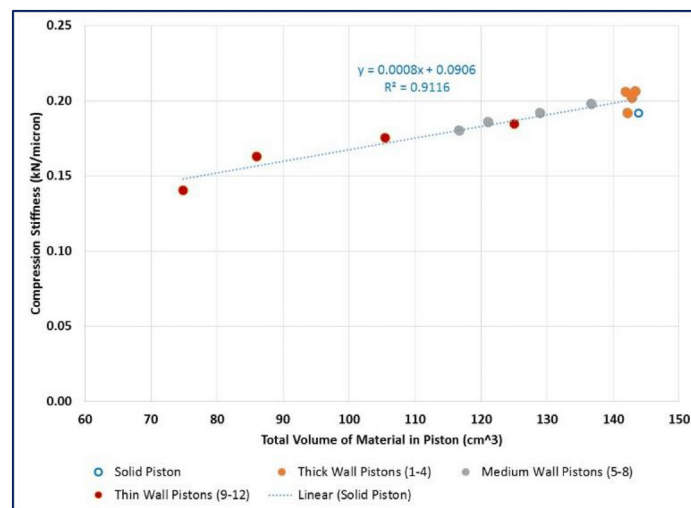


Figure 5. Summary of Phenolic Piston Compression Stiffness versus Total Material Volume, 2-30 kN Force Range

3. WET FIXTURE MEASUREMENTS AT LOW FORCE - ADDING PAD AND SHIM

The second phase of compression stiffness testing was targeted at a lower force range (with two load levels, one peaking at 3 kN and the other at 10 kN) to keep the loads in a realistic range for typical customer-usage braking (and a range in which most roughness events would tend to occur). A set of measurements was taken with a single brake pad with a bonded noise shim (noise insulator) added on top of the piston. It was recognized that there were potentially competing effects - as the piston wall thickness was reduced, the piston stiffness would be reduced, but at the same time the pressure on the noise shim would increase. The increase of pressure on the noise shim could decrease stiffness - or, if it increased compression beyond the initial non-linear range (much less stiff) and into the much stiffer linear range, it could have the opposite effect.

Five of the piston configurations (tight and loose steel, and phenolic pistons 1B, 7B, and 12B) were selected to be measured, as-assembled in a wet fixture and again with the wet fixture and a brake pad with a bonded noise insulator. For the measurements with the pads, each of the 5 pistons was tested with each of 5 pad samples in random order, and the average stiffness measurement is included in figure 6. The pads were allowed to relax overnight before a new set of measurements was taken with them. This was done in an attempt to minimize the effect of pad to pad variation, and also the effect of changes in pad compressibility due to previous compression cycles.

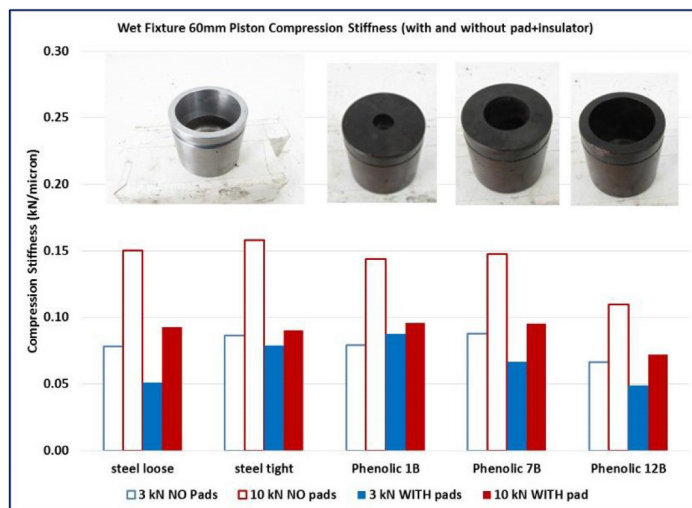


Figure 6. Summary of Piston Compliance in Wet Fixture (with and without pads)

The data in figure 6 support the following observations:

- Including the brake pad drops the net stiffness by about 25 to 50%. As intuition would suggest, the stiffest piston was affected the most (the steel piston), and the phenolic piston 12B the least.
- In the lower-load test condition (the maximum load these samples saw was 10 kN, versus a 30 kN peak for the pistons represented in figures 4 and 5), the steel pistons showed only marginally higher stiffness than the thickest-walled phenolic piston (1B) and even the medium-walled phenolic piston (7B).

- When the pads were added, the thick-walled phenolic piston (which had by far the largest contact area to the noise shim) was the stiffest of the group at low load (3 kN). The loose steel piston was actually the 2nd least stiff of the group at 3 kN! These data suggest a significant impact of piston face to pad noise insulator contact area on net stiffness, with higher area increasing stiffness.
- It is plausible at these load levels that differing levels of compression during installation and different levels of shear during actuation of the piston seal with changes in piston to bore clearance affected the overall system stiffness; this would tend to promote lower stiffness with higher piston to bore clearances (less compression/stiffness of the seal and lower stress in shear).

Measurements of piston and wet fixture hydraulic stiffness were also taken with a sinusoidal load, at 5, 10, and 15 Hz frequency. A nominal load of 5.56 kN was targeted, with a varying amplitude of ± 0.46 kN. This load range is appropriate for what the piston and hydraulic system might experience when operated in a judder condition, with elevated (20+ microns) of disc thickness variation. Figure 7 summarizes data by piston configuration and frequency.

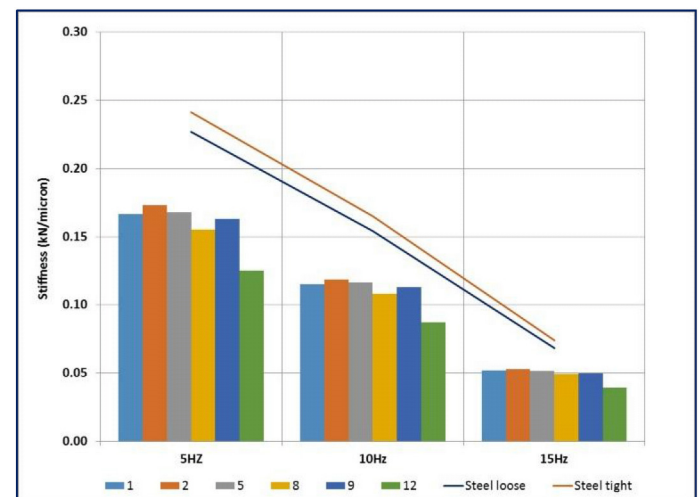


Figure 7. Summary of Piston Compliance in Wet Fixture versus Frequency (5.56 kN $\pm$ 0.46 kN amplitude, without pads)

Here again the same puzzling loss of stiffness with increased frequency of load application that was seen in material specimens during DMA testing and again with testing in a servo-hydraulic compression tester [1] was seen again, and was observed in both steel and phenolic specimens. Unlike the previous study, there was some consistency in stiffness measurements between high force (10 kN) ramp applies (see figure 6) and sinusoidal applies up to 5 Hz (figure 7), but stiffness dropped to less than a third of the 5 Hz value at 15 Hz. It is unknown how much of this is a real material characteristic versus a behavior of the test equipment and/or instrumentation. It should be noted that previous work by Jung et al [2] using micro-flown and capacitive sensors in a resonance method concluded that the stiffness modulus of steel was relatively independent of frequency in a range up to 800 Hz.