

The Measurement and Analysis of the Disc/Pad Interface Dynamic Centre of Pressure and Its Influence on Brake Noise

John D. Fieldhouse, Naveed Ashraf and Chris Talbot
The University of Huddersfield

Copyright © 2008 SAE International

ABSTRACT

This paper discusses the measurement of the dynamic centre of pressure (CoP) of a brake pad during a normal braking event using a modified 12-piston opposed calliper. The modifications allow the centre of pressure to be controlled both radially and along the length of the pad, inducing a leading or trailing centre of pressure as desired. The technique is unique in its design and implementation. Both the centre of pressures of the in-board and out-board pads are recorded simultaneously with varying pressures and speeds. The results, which include pressure and force maps, show the position of the centre of pressure to vary considerably during a braking event, both radially and axially along the pad. The CoP offset is related to the calliper mounting geometry which is subsequently compared to the effective "spragging angle" and the generation of brake noise. It is seen that by inducing a leading offset noise may be generated and subsequently eliminated if a trailing centre is then induced. The results suggest that by careful selection of the backplate abutment friction level the CoP may be controlled to always fall within the "stable envelope" region and so resist noise generation.

INTRODUCTION

Disc brake noise is as problematic as ever but has to some degree been surpassed by brake judder. This is shown in Figure 1. This shift is not due to diminishing noise problems but more to the thermal instabilities experienced in high performance cars – the problems of noise are as prevalent as ever, at around 2300Hz. The study of brake noise is a complex and fascinating subject for investigation. Research may be approached using the sophisticated software available such as FEA or complex eigenvalue analysis but in general such approaches need validation through experimentation. During noise the resulting vibration characteristics of the system are well understood, but regardless of the vast range of literature already published on the subject, the causes are not fully clear. It has been shown many times

that in all cases of disc brake noise, the dynamic noise frequency may be related directly to the free mode vibration characteristic of the disc, the noise frequency being slightly less than disc free mode frequency.

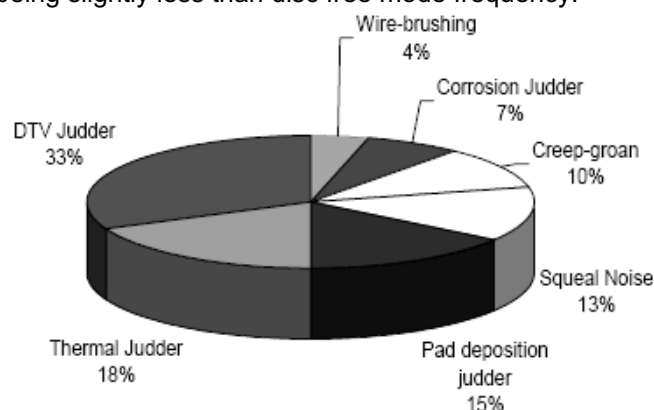


Figure 1 – General breakdown of current brake instabilities.

Despite the variety of mathematical models that have been proposed to date, no single model is currently capable of expressing the differing mechanisms involved over the frequency ranges 1200Hz to 20,000Hz – the range is simply too broad. Regardless, the mathematical models do allow the designer to define basic parameters and criteria at the design stage in an attempt to reduce the propensity of a brake to generate noise. Such criteria include disc/pad interface geometry, pad chamfers, grooves in pads, low coefficients of friction, compressibility of the pad and damping in the form of visco-elastic shims or interlayer damping in addition to material selection. It is accepted by most investigators, and drivers aware of noise, that noise occurs at low speeds and very low brake pressures. Noise will disappear if the braking force is increased and may be sustained if light braking is applied on a downhill run. In addition it is generally accepted that the higher the coefficient of friction at the pad/disc interface, the greater the tendency will be for the brake to promote noise, hence the saying "a noisy brake is a good brake". This increase in friction coefficient increases the disc/pad interface force for a given pressure, which in turn results

in a greater braking torque and greater contact forces (typically pad abutment) within the brake assembly. These frictional forces tend to be circumferential (parallel to disc surface) which does not explain why there is a tendency for the disc to be excited out-of-plane, with a diametrical mode order. Indeed it does not explain why there is any vibration at all. In 1971 Spurr [1] suggested spragging as the cause for a variable out-of-plane force, the sprag force related to the coefficient of friction at the disc/pad interface. The mechanism was represented by a semi-rigid strut (P O') which was inclined to a rubbing surface and pushed horizontal to the surface as shown in Figure 2. The lever O'P is considered to be rigid and pivoted at O'. Flexibility within the system is provided by a second cantilever O'O'' which allows pivot O' to move under load. The lever PO', of length ∂ , is loaded against a moving surface AB with a normal force L.

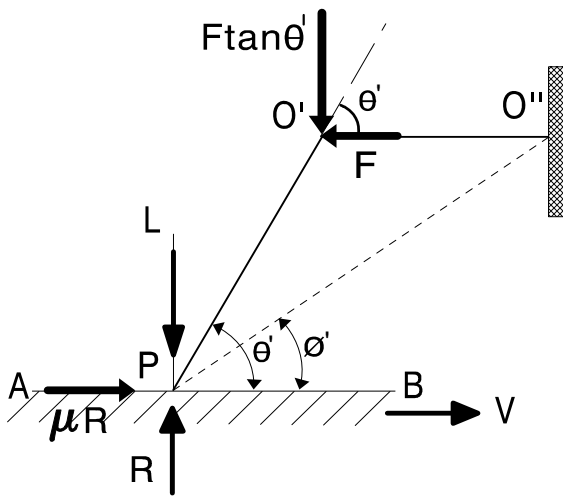


Figure 2 – Sprag-slip model as proposed by Spurr.

Resolving vertically gives:

$$R = L + F \tan \theta'$$

and taking moments about O' gives:

$$L \partial \cos \theta' + \mu R \partial \sin \theta' - R \partial \cos \theta' = 0$$

which reduces to:

$$L + \mu R \tan \theta' - R = 0$$

substituting R from equation (1.5) in the above gives:

$$L + \mu(L + F \tan \theta') \tan \theta' - (L + F \tan \theta') = 0$$

resulting in:

$$\mu L + \mu F \tan \theta' - F = 0$$

and rearranging gives:

$$\mu L = F(1 - \mu \tan \theta')$$

so that:

$$F = \frac{\mu L}{(1 - \mu \tan \theta')}$$

with F approaching infinity as μ tends towards $\cot \theta'$, when spragging will occur. When the inclination angle was set at the "sprag angle" of $\tan^{-1} \mu$, or greater, the strut would "dig-in". The normal force to the rubbing

surface would then increase until flexure of the system allowed a secondary strut arrangement (PO'') to be established whereby the sprag angle was reduced, the normal force would reduce and the strut would then continue to slide.

This work was continued on discs with Jarvis and Mills [2] and Earles et. al. [3-7]. These works are generally referred to as cantilever & disc and pin & disc models as shown in Figures 3 and 4 respectively.

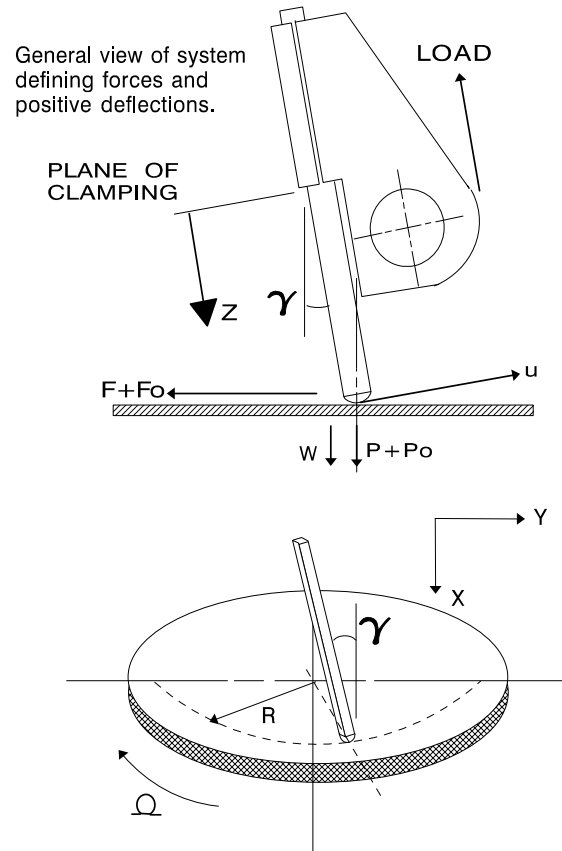


Figure 3 – Test Rig of Cantilever and Disc Model.

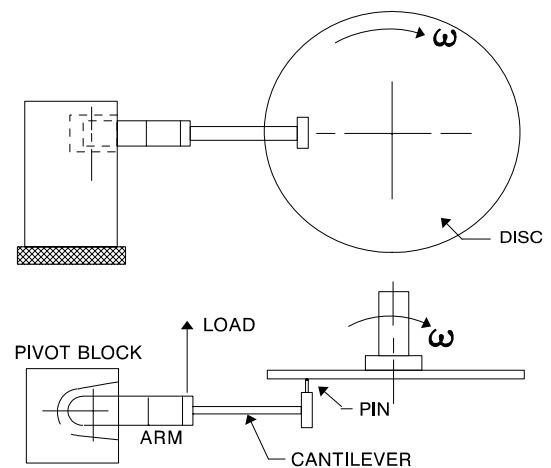


Figure 4 – Test rig for Pin & disc Model.

Clearly because of the construction of the test rigs, and the in-built flexibility of the members, the system was able to establish more than one sprag angle for it to work. The caliper is a similar system with a multiple of "sprag angles" and it is this mechanism, in relation to the

coefficient of friction at the pad/disc interface, which this paper examines. Such a comparable system is shown in Figure 5 when it was shown that noise was readily generated when a wire was inserted between the piston and the backplate to give a leading offset of between 12 & 15mm [8]. The results are shown in Figure 6.

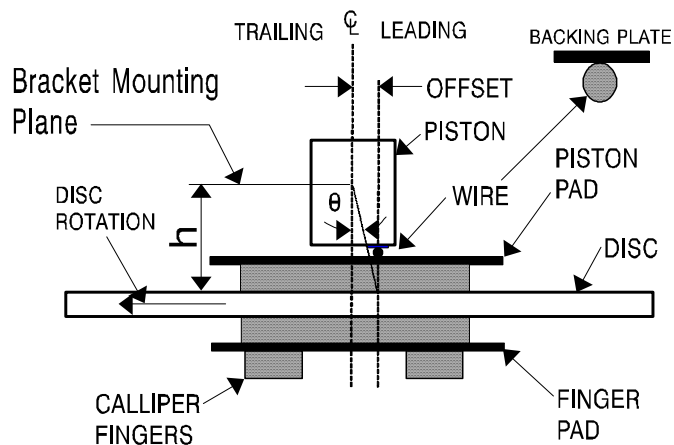


Figure 5 - Diagram showing wire offset contact position

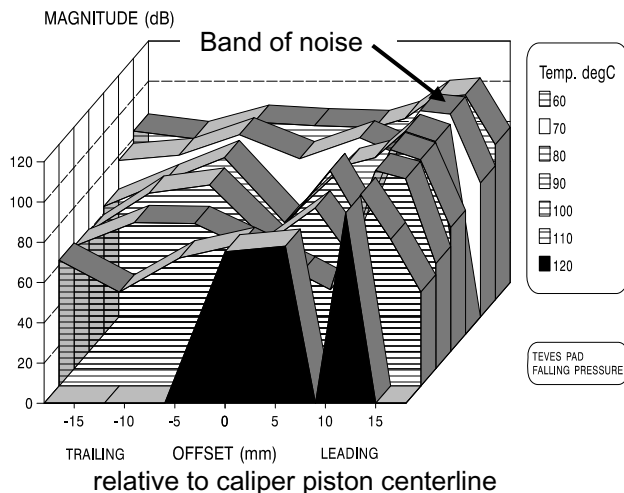


Figure 6 – Offset/Noise Magnitude ($L_{eq}(tot)$) for a Range of Temperatures

MAIN SECTION

HISTORICAL REVIEW

A study of brake excitation cascading from a very high frequency of 31.4kHz down to an audible frequency of 8480Hz indicated that the trailing end of the piston pad (inboard) preceded all the other components in modulating to the lower frequency. The features studied using piezoelectric beams were piston pad (leading and trailing end), finger pad (leading and trailing end), noise (using noise level meter), disc rim and "top-hat" section of the disc. Detail of the investigation is shown in earlier work (9) which includes theoretical reasoning explaining why vibration of the pad could result in out-of-plane vibration. It was suggested, following this study, that possibly the onset of noise is always preceded by a very high frequency vibration which develops by convenient

mechanical coupling of components until all are excited at a lower, audible, frequency - the "trigger" being the trailing end of the piston pad. Possibly it is damping, or frictional forces at its abutment face, that causes such a major frequency change. This may also answer the question why noise propensity increases as the brake force is reduced, or noise is most pronounced during light braking. Conversely, brake noise reduces as the braking force increases – the pad ceases to vibrate. If this is the case there may be a relationship between the braking force and the pad abutment force leading to some form of spragging as proposed earlier.

In addition it was further suggested that this may offer an explanation for why dogs chase cars; the car has an unstable brake system which is excited in a state of high frequency excitation. The dog hears this frequency and responds accordingly to the annoyance. Since the higher frequency is not detected by the human ear it is believed to be a quiet, stable brake.

For a caliper arrangement to be capable of spragging it is a requirement that the centre of pressure providing the reaction force normal to the disc surface, as generated by the caliper pistons, would need to be offset from the centerline of the caliper mounting arrangement as indicated in Figure 5. This normal force when combined with the in-plane frictional force would provide the resultant spragging force and thus an out-of-plane excitation force. This work considers an empirical approach to examine the effects of an imposed offset center of pressure and the resulting noise propensity.

MEASUREMENT OF CENTER OF PRESSURE DURING BRAKING

Although the static pressure distribution may be calculated, and measured (pressure sensitive films), there is little information regarding its calculated or measured position during a dynamic braking event. The initial work, with Bosch Braking Systems, indicated that the centre of pressure did move noticeably during the application of the brake but the work was not extended to consider how it moved during a brake application. The unique method employed in the investigations allows the dynamic center of pressure to be measured during a normal noisy brake application. This novel technique uses an embedded pressure sensitive film within the pad. To allow this the pad is machined to provide a recess as shown in Figure 7. A matching plug was then machined and ground on its film face to ensure good area contact. To ensure good area contact at the base of the recess a ground pressure plate was inset, again as shown in Figure 7. The film laminate was then fitted to the recess and the "plug" used to form a sandwich as shown in Figure 8. The pad was machined to give a level rubbing surface and the assembly used to measure the dynamic center of pressure using an opposed 12 piston caliper as shown in Figure 9 and Figure 10.