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

John D Fieldhouse, Naveed Ashraf & Chris Talbot

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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.

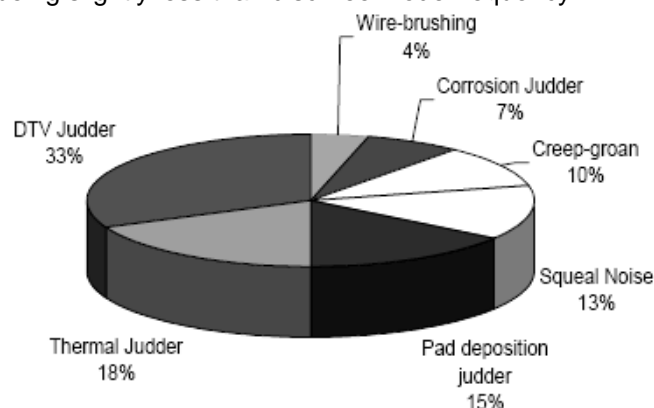


Figure 1 – General breakdown of current brake instabilities.

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.

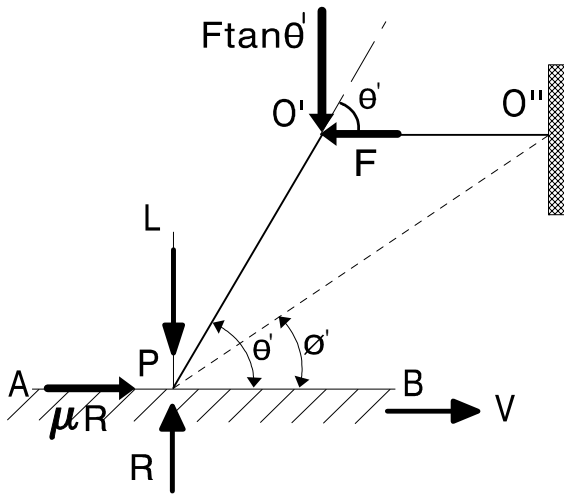


Figure 2 – Sprag-slip model as proposed by Spurr.

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.

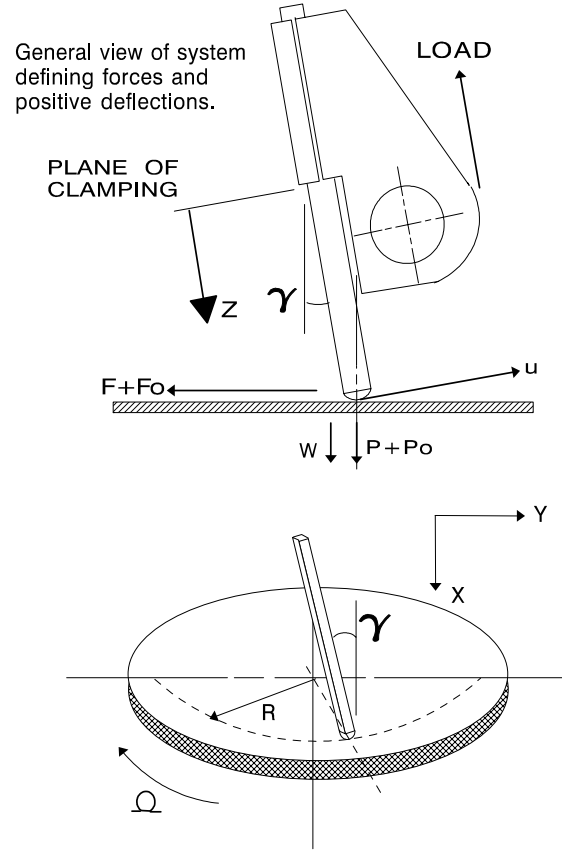


Figure 3 – Test Rig of Cantilever and Disc Model.

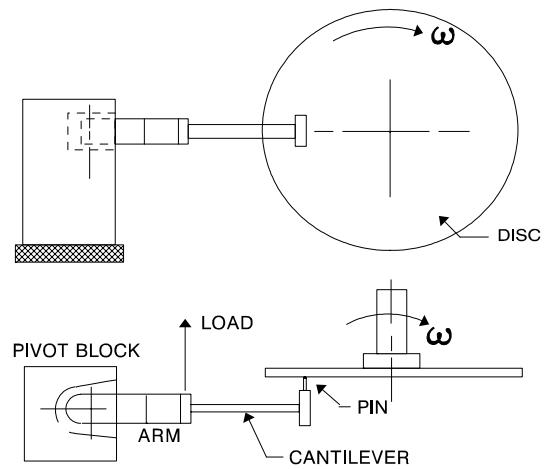


Figure 4 – Test rig for Pin & disc Model.

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.

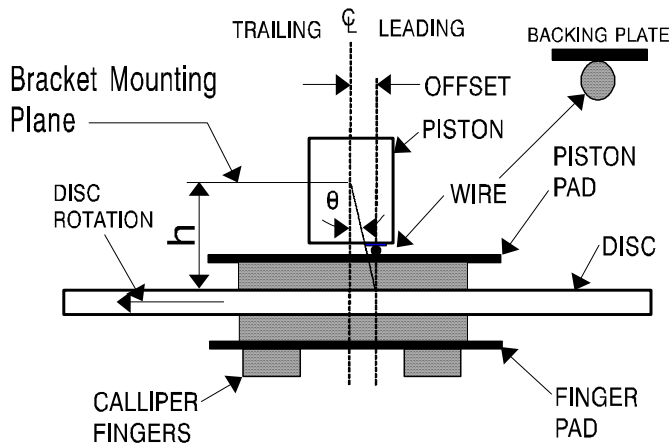


Figure 5 - Diagram showing wire offset contact position

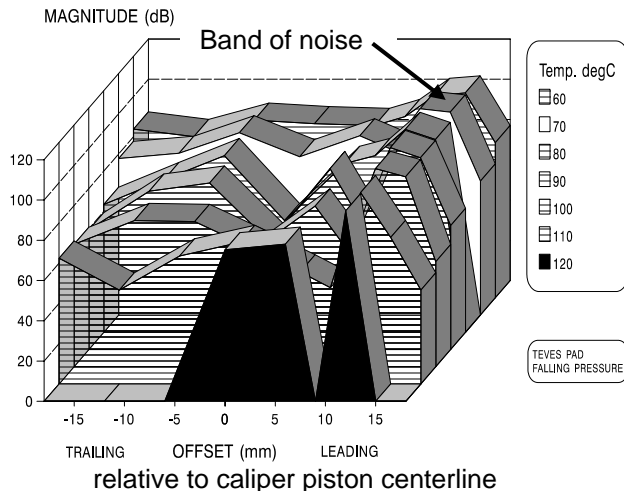


Figure 6 – Offset/Noise Magnitude ($L_{eq(tot)}$) for a Range of Temperatures

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.



Figure 7 – View of pad and insert before pressure film is fitted. Note smooth metallic face for film bed.



Figure 8 – View of pad with embedded pressure film.

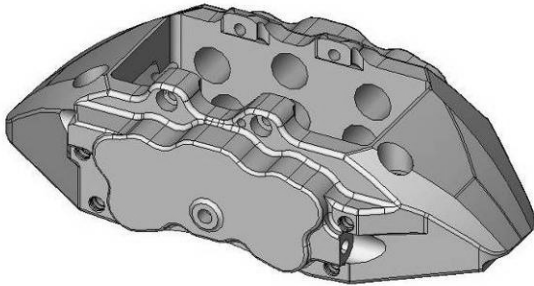


Figure 9 – General view of opposed 12 piston caliper.

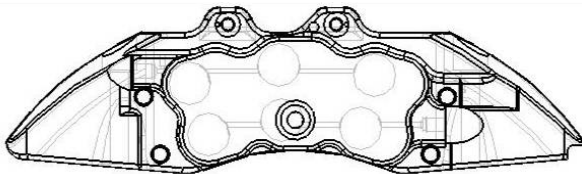


Figure 10 - Drawings of 12 piston opposed caliper.

It must be noted that care has to be taken during testing to ensure the pad temperature does not exceed the melting point of the film.

The test rig is shown in Figure 11 where it is seen the caliper pistons are connected to a series of master

cylinders and pressure gauges. The connection of the pistons to the gauges and master cylinders are shown in Figure 12.

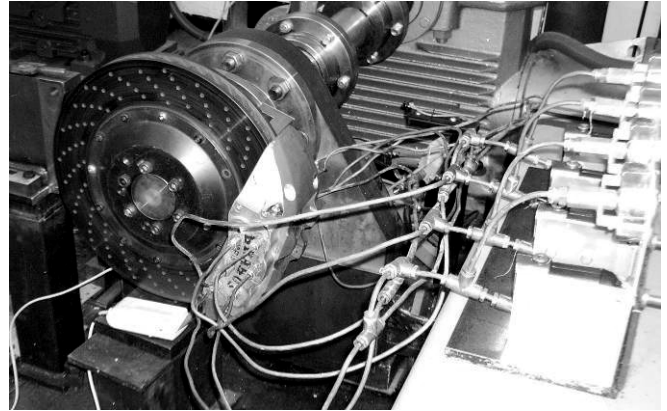


Figure 11 – General view of test rig. A 12 piston opposed caliper giving pressure control along its length and radially.

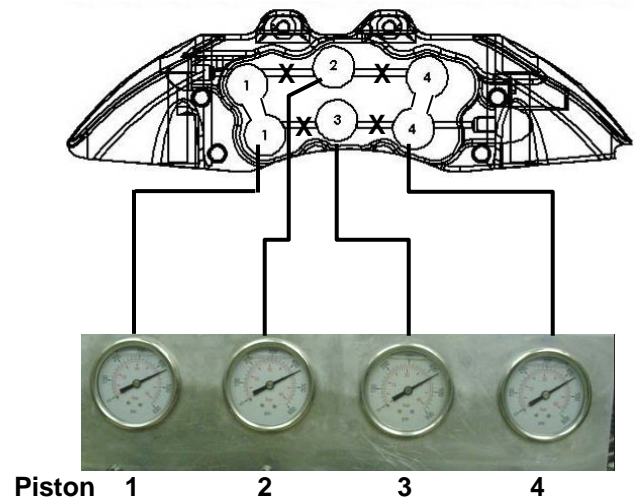


Figure 12 - Arrangement of piston(s) and position of pressure adjustment.

Normally all the pistons are connected together, the leading and trailing pair being connected by casting voids. The two central pistons are not connected together but are connected to the leading and trailing pair by the cross-drillings as indicated.

To provide individual adjustment of the pistons the cross drillings were blocked, as indicated by the crosses "X" in Figure 12. With such an arrangement it was possible to change the leading pairs (gauge 1), the outer central piston (gauge 2), the inner central piston (gauge 3) and the trailing pair (gauge 4), all independently, with their own master cylinders.

Figure 13 shows the master cylinder screw adjusters and their associated pressure gauges.

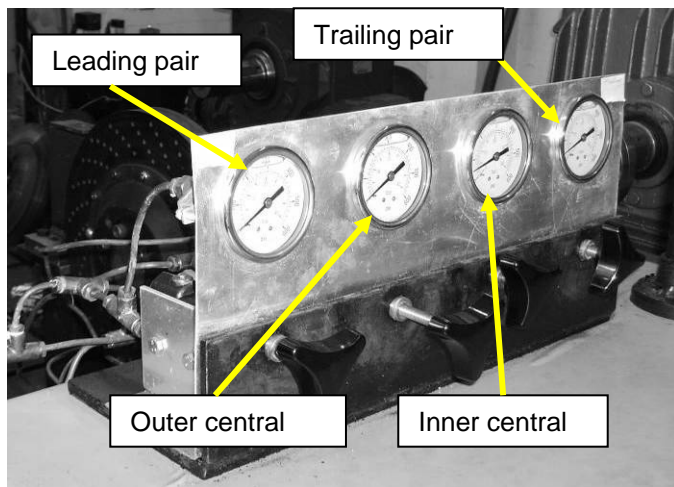


Figure 13 - Pressure gauges and adjusters used to control caliper piston pressures as indicated.

RESULTS

UNIFORM BUT VARYING PRESSURES

Tests were carried out at varying speeds and with varying (but uniform) pressures along the length of the pad. The graphical results for variation in longitudinal CoP for both pads are as shown in Figures 14 and 15. In all cases the centre of pressure is measured from the centre of the pad, with positive the leading direction. Pressures at each stage (1 to 7) are shown in Table 1.

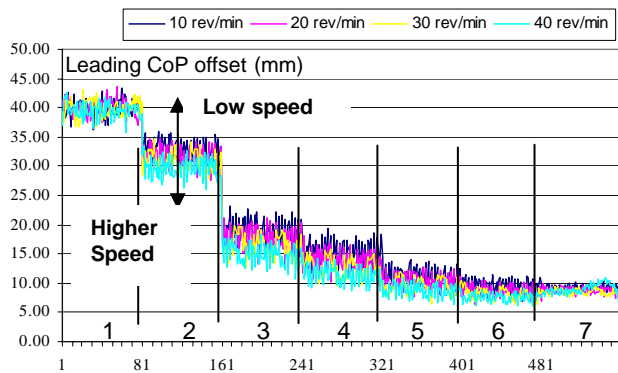


Figure 14 – Movement of **longitudinal** centre of pressure with varying pressure (**Inboard Pad**)

The variation in the position of the radial CoP is shown in Figures 16 and 17 for the inboard and outboard pads, respectively. In all cases the centre of the pad is the reference (zero) position with positive being measured towards to outer edge and negative radially inwards, towards the inner edge – or towards the disc centre.

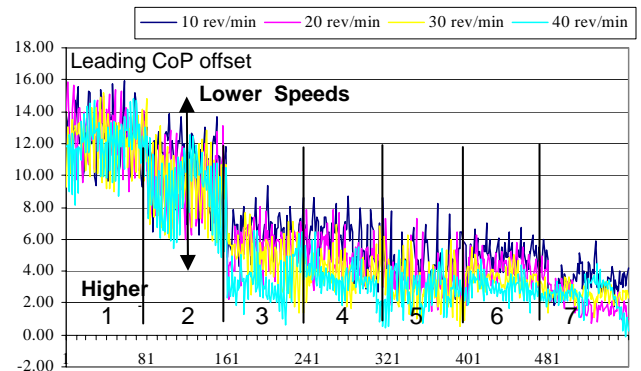


Figure 15 – Movement of **longitudinal** centre of pressure with varying pressure (**Outboard Pad**).

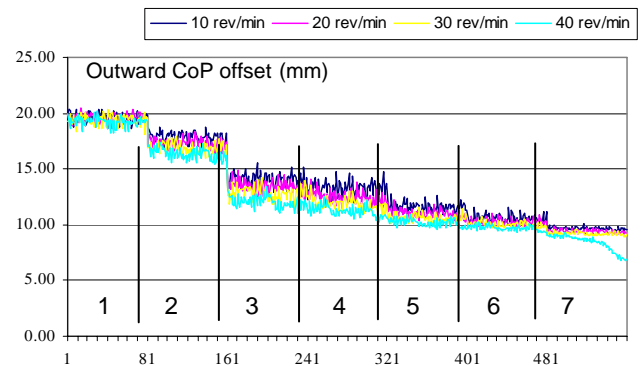


Figure 16 – Movement of **radial** centre of pressure with pressure (**Inboard Pad**).

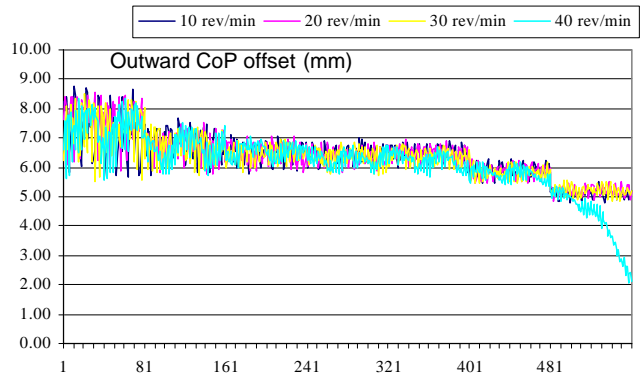


Figure 17 – Movement of **radial** centre of pressure with pressure (**Outboard Pad**).

Table 1 – Uniform pressure setting for each stage. Speed is constant at 10 rev/min.

Stage	Pressure (MPa)
1	0.2
2	0.4
3	0.6
4	0.8
5	1.0
6	1.5
7	2.0

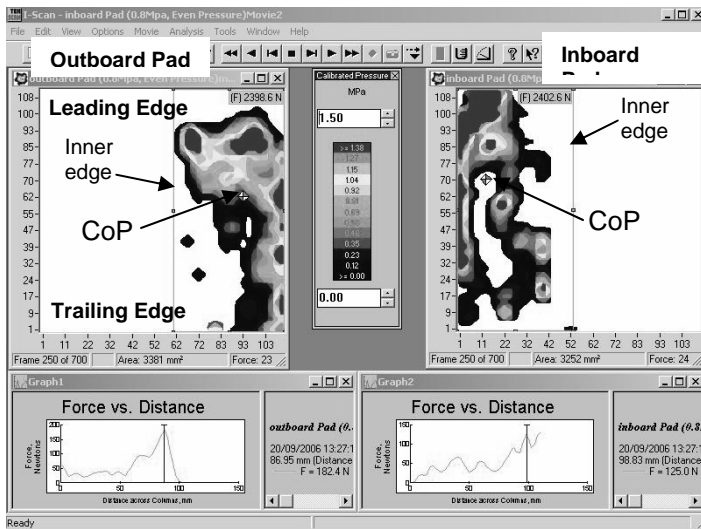


Figure 18 – Typical display of a force distribution map reading for both outboard (LHS) and inboard (RHS) pad. Centre of pressure is indicated by shaded diamond.

A typical force distribution map is shown in Figure 18, the pressure map being similar.

If the results of each stage in Figure 14 are averaged and plotted then a series of speed curves may be plotted, as shown in Figure 19. These curves take the form $y = ax^3 + bx^2 + cx + d$ where a,b,c,d are constants and x is the pressure in bar. The table of constants are shown in Table 2.

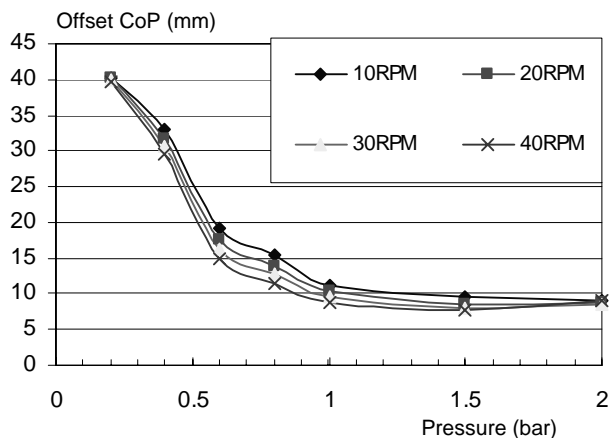


Figure 19 – Averaged values of pressure curves as presented in Figure 14.

Table 2 – Constants for the 4 speed curves shown in Figure 19.

	10 rpm	20 rpm	30 rpm	40 rpm
a	-10.1614	-11.9543	-13.3197	-14.6811
b	51.8579	58.9742	64.1285	69.8372
c	-86.7794	-94.6533	-99.8906	-105.951
d	56.7561	57.9346	58.3971	58.9785

If the values of the 4 curves are averaged and a single curve is plotted, the result is shown in Figure 20. The curve is again of cubic form with equation,

$$\text{Offset} = -12.4265x^3 + 60.8625x^2 - 96.3752x + 57.862$$

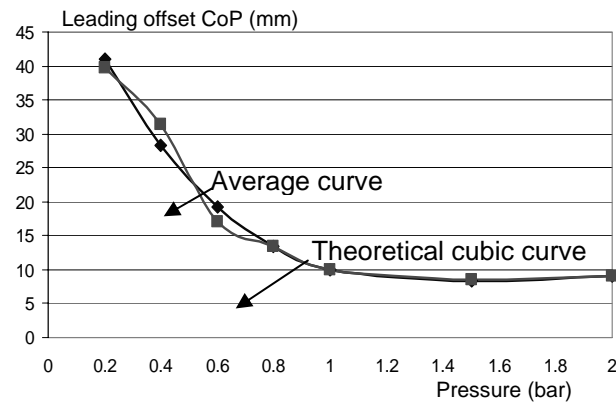


Figure 20 – Average speed curve for the results presented in Figure 19. The curve takes the form of a cubic equation and is plotted against the average curve.

The values of the constants in Table 2 may be plotted and from this it may be concluded that speed has little effect on the CoP. This is also shown in the effects of speed, shown in Figure 21. It is seen that speed has limited effect whereas with increasing pressure there is a tendency for the CoP to move towards the trailing, more stable position.

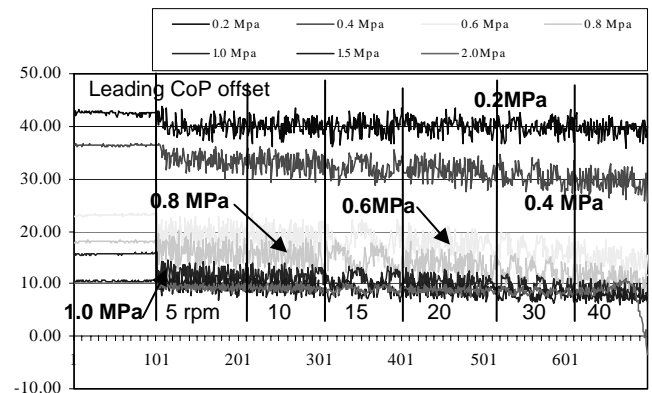


Figure 21 - Variation of longitudinal centre of pressure with speed and pressure (**Inboard pad**). Outboard pad exhibits similar characteristics.

VARIABLE PRESSURES

This test allows the pressures to be varied along the length of the pad and also varied radially. A benchmark is established with uniform pressure but then all results varied to give either a leading or trailing offset. All tests were recorded at a fixed speed of 10rpm.

Longitudinal results - The piston arrangements (sets) are shown in Figure 12. The general settings are shown in Table 3 where loudest squeal was created at stage 3 - with a deliberately induced leading centre pressure.

Table 3 - Details of pressure variation at each cylinder position and the effect on noise propensity.

Stage	Pressure (MPa) – for each piston set				Comments
	1	2	3	4	
1	1.4	1.4	0.1	0.1	Static
2	1.4	1.4	0.1	0.1	Squealing
3	1.4	1.4	1.4	0.1	Very loud squealing
4	1.4	1.4	1.4	1.4	Intermittent noise
5	1.4	1.4	1.4	2.0	Quiet
6	1.4	1.4	0	1.4	Intermittent noise
7	1.4	1.4	2.0	2.0	Quiet
8	0.7	0.7	2.0	2.0	Quiet

The quietest setting was stage 5 with a deliberately induced trailing centre of pressure. These results are shown most clearly in Figure 22 where the leading centre of pressure position is shown on the “y axis”, and each stage on the “x axis”. Figure 23 shows the radial CoP position.

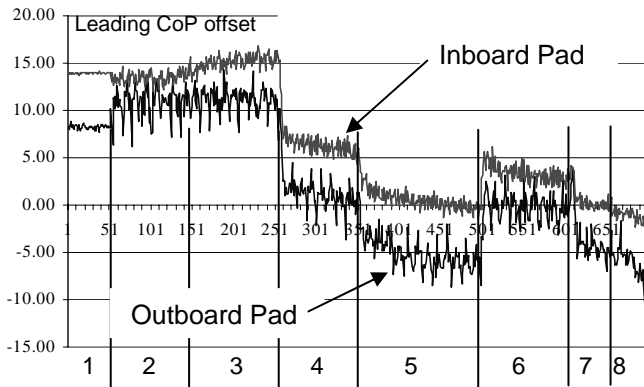


Figure 22 – Centre of pressure along pad with varying pressure along and a radially across pad. Initially static but then measurement taken at 10 rpm. See Table 3.

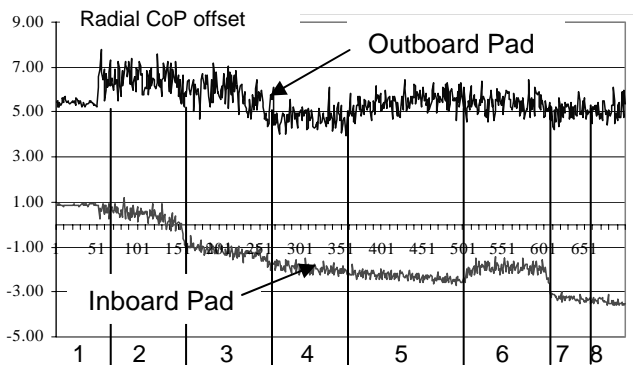


Figure 23 – Radial centre of pressure across pad with varying pressure along and a radially. Initially static but then measurement taken at 10 rpm. Refer to Table 3.

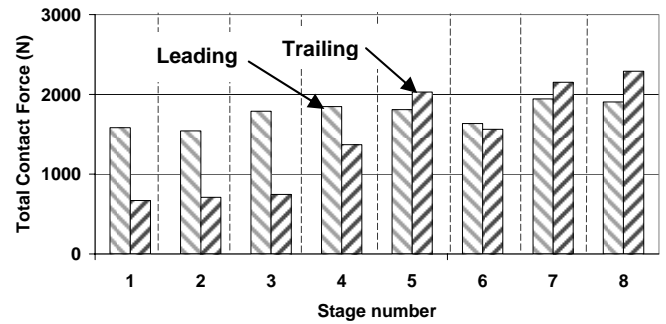


Figure 24 – Force distribution for inboard pad under conditions given in Table 3. Condition 3 - Very loud squeal, condition 5, 7 & 8 quiet.

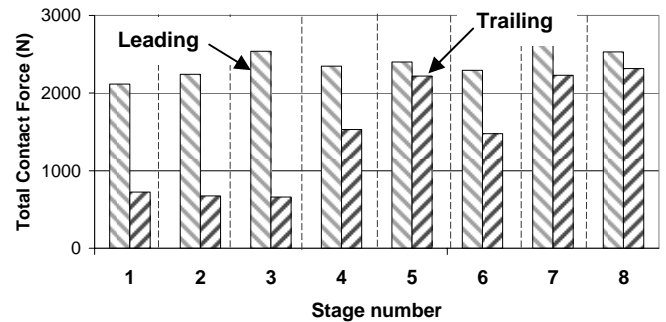


Figure 25 – Force distribution for outboard pad under conditions given in Table 3. Condition 3 - Very loud squeal, condition 5, 7 & 8 quiet.

To complement the CoP results in Figure 22 it is possible to show the disc/pad interface contact forces of the leading and trailing ends of both the inboard and outboard pads. These results are shown in Figures 24 and 25 respectively. Again it must be remembered that the brake is very noisy at condition 3 and quiet at conditions 5, 7, and 8. It is particularly noticeable in Figure 24 that when the contact force is greatest within the trailing section the brake is quiet. When marginal, as in condition 6, Figure 24, the noise is intermittent.

Radial results –

The general pressure settings are shown in Table 4 with the radial offset shown in Figure 26. It is seen the loudest squeal was experienced at stages 3, 4 and 7. In the case of stages 3 & 4 the CoP was deliberately induced towards a zero or negative radial offset, radially towards the centre. The offset in the case of stage 7 was not so apparent but the audible results were still obvious. The quietest arrangement was when the CoP was generally close to 2mm positive offset (radially outwards). As the CoP migrates further radially outwards the noise begins to become re-established. The reason for this is still unclear but may be related to rotational instability of the pad. Overall it is seen the outboard pad is less sensitive to pressure variations.

Table 4 - Details of variation of pressure at each cylinder position and affect on noise propensity – emphasis on radial variation (Figure 26)

Stage	Pressure (MPa) – for each piston set				Comments
	1	2	3	4	
1	1.0	1.0	0.4	0.3	Static
2	1.0	1.0	0.4	0.3	Squeal
3	1.0	1.0	4.0	0.3	Very loud squealing
4	1.0	1.0	2.0	0.3	Less noisy
5	1.0	1.0	0	0.3	Squeal
6	1.0	1.0	0.5	0.3	Squeal
7	1.0	1.0	4.0	0.3	Very loud squealing
8	1.0	1.0	0.4	0.7	Intermittent
9	1.0	1.0	0.4	1.4	Quiet

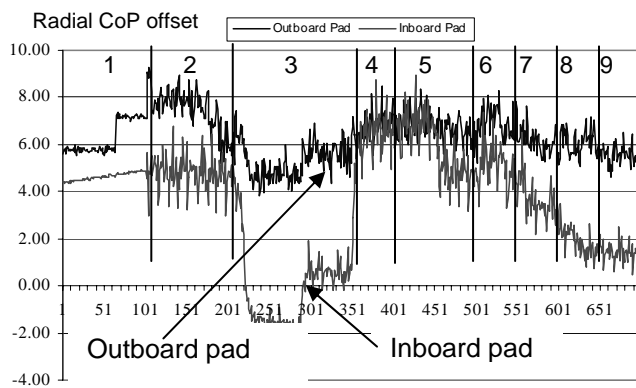


Figure 26 – Radial centre of pressure across pad with varying pressure along and a radially - emphasis on radial pressure adjustment. Refer to Table 3.

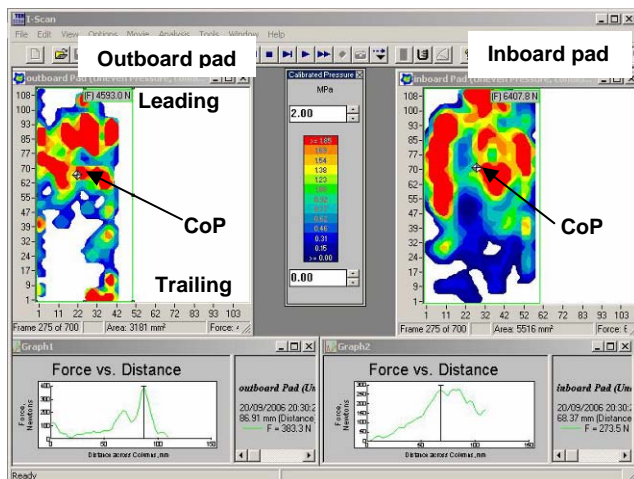


Figure 27 – Force map of outboard and inboard pad Pressure 1.0 (1), 1.0 (2), 4.0 (3), 0.3 (4) MPa. Very loud noise. Refer to Table3, Stage 3.

The load map of both pads is shown in Figure 27 where it is seen the inboard pad appears to carry significantly more load than the outboard pad. Such an observation

leads to possible thermal issues and temperature gradients across the disc blade.

DISCUSSION OF RESULTS

Original work [8] indicated that a leading centre of pressure (CoP) tended to have a high propensity to generate noise. This was explained as being the result of the resultant force at the disc/pad interface being offset such that spragging could occur. It was also shown that with a very large leading offset the brake would be quiet and with a near zero offset the brake would again be quiet (Figure 6). It was suggested that within a “critical” region, where possible spragging would occur, then the brake would be noisy. If this situation of a moving center of pressure is accepted then it may be related back to caliper mounting as follows. Figure 28 represents the free body diagram of the assembly, where the pad, pistons and caliper are considered as a rigid body. The position of the normal force to the disc/pad interface is variable dependant on direction of the pad/caliper abutment frictional force and so the position of the resultant of the normal force and the disc/pad friction force will also vary.

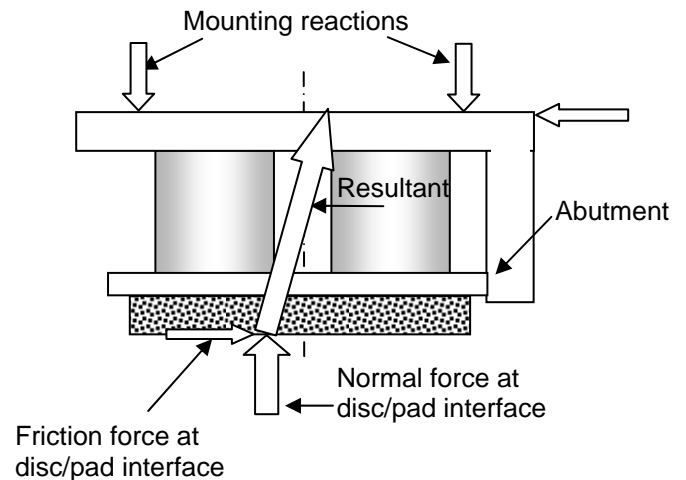


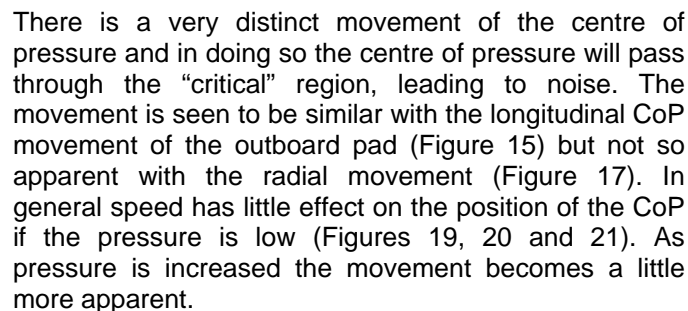
Figure 28 – Free body diagram of caliper assembly.

Figures 29a-c show the force diagrams of the pad, pistons and carrier/caliper mounting. The disc is moving left to right and the co-planar forces are indicated. The caliper/carrier mounting points are represented by a spring.

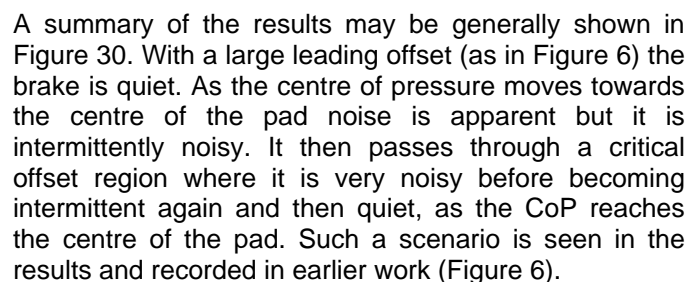
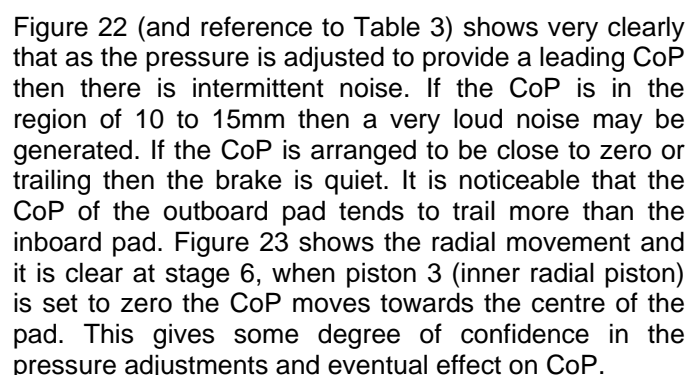
In Figure 29a the center of pressure is leading and as such the resultant force causes the mounting to rotate clockwise. If the center of pressure is less leading then the resulting force causes the mounting to rotate counter-clockwise as indicated in Figure 29b.

Clearly if the center of pressure “hunts” between these two extremes it will pass a point where the resultant will cause the mounting to physically move axially (vertical in diagram) as shown in Figure 29c – spragging at its worst. In this case the normal force at the disc/pad

pressure will move towards the centre of the pad and at the same time move radially inwards (Figures 14 & 16).



The typical load distribution map as shown in Figure 18 indicates the CoP and the force distribution. There is an also accompanying area map which looks similar but indicates pressure distribution (not shown).



The diagram shows a cross-section of a pad with a central rectangular hole. A vertical dashed line divides the pad into two regions. The left region is labeled 'Stable region' and 'Stability'. The right region is labeled 'Intermittent noise' and 'Area of maximum instability'. The 'Pad profile (leading end)' is indicated on the right. Dimensions are provided: a vertical distance of 8 mm from the top edge to the top of the hole, a vertical distance of 2 mm from the bottom edge to the bottom of the hole, a horizontal distance of 10 mm from the left edge to the left side of the hole, and a horizontal distance of 15 mm from the right edge to the right side of the hole. A horizontal dashed line passes through the center of the hole.

Figure 30 – General areas of instability, intermittent noise and stable region.

The radial position of the CoP also plays a part and so a more noisy brake is more likely when the CoP tends towards the centerline of the pad (Figure 30).

CONCLUSION

Earlier theoretical co-planar analysis of a brake abutment arrangement indicated the co-planar forces would tend to promote a leading offset. It was established that the degree of offset would be influenced by the coefficient of friction between disc and pad and coefficient of friction between pad abutment and caliper. It was also shown earlier that the position of the mounting plane for the caliper carrier bracket is important because of its influence over the spragging angle. It needs to be as close to the plane of the disc-rubbing surface as possible. The work concluded by suggesting that to promote stability a disc brake requires a low friction coefficient between pad abutment and caliper mounting bracket and a low friction material coefficient. Clearly the latter is not ideal to stop the car effectively.

The work supports the earlier theoretical approach and clearly shows that when a leading center of pressure is introduced at the disc/pad interface then noise is more likely. It is also demonstrated that as pressure increases the CoP tends to move towards the central region of the pad, leading to stability. This is borne out by general observation - when the brake pressure is increased during normal braking, noise reduces.

It is seen that under light braking, with a uniform pressure setting, the centre of pressure will always tend to be leading, hence an increased propensity to generate noise. Whether this will actually result in noise depends on the coefficient of friction, pad wear and brake mounting geometry. Both rig and vehicle testing demonstrate that a trailing centre of pressure will tend towards a stable, quiet brake. The degree of offset does not appear to be influenced too much by the speed of disc rotation.

The leading centre of pressure results in a dynamic instability that is caused by "spragging" of the system. It is also observed that noise will be more likely if the CoP moves radially inwards towards the center of the pad as shown in Figure 24.

The ideal position for the center of pressure of the pad would be longitudinally central to trailing and radially out from the pad centerline, see Figure 28. It is recognized that pad wear is also a consideration in such an arrangement which may prevent such action.

To cater for this wear issue it may be possible to induce a permanent trailing offset by "hanging" the calipers and arranging the caliper (or carrier) mounting points (bolts) to be in a leading arrangement. This would induce a permanent trailing CoP offset with respect to the mounting geometry, regardless of the CoP at the disc/pad interface. In essence the pads would be "dragging" behind the mounting fixings. It may be suggested that a trailing CoP may solve noise issues in

the forward direction but it would cause problems in reverse. To answer the question of braking in reverse, it has been observed that if the leading offset is excessive (Figure 6) then the brake would again be quiet. It may be possible to study the situation and arrange the caliper/carrier mounting points to create such an appropriate leading offset. This arrangement is the focus of ongoing research. The geometry of the pad/caliper abutment interface is also under investigation.

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