ABSTRACT

Thermo-elastic and thermo-plastic behaviour takes place with a disc brake during heavy braking and it is this aspect of braking that this paper considers. The work is concerned with working towards developing design advice that provides uniform heating of the disc, and equally important, even dissipation of heat from the disc blade.

The material presented emanates from a combination of modeling, on-vehicle testing but mainly laboratory observations and subsequent investigations. The experimental work makes use of a purpose built high speed brake dynamometer which incorporates the full vehicle suspension for controlled simulation of the brake and vehicle operating conditions. Advanced instrumentation allows dynamic measurement of brake pressure fluctuations, disc surface temperature and discrete vibration measurements. Disc run-out measurements using non-contacting displacement transducers show the disc taking up varying orders of deformation ranging from first to third order during high speed testing. This surface interrogation during braking identifies disc deformation including disc warping, “ripple” and the effects of “hot spotting”. The mechanical measurements are complemented by thermal imaging of the brake, these images showing the vane and vent patterns on the surface of the disc. The results also include static surface scanning, or topographical analysis, of the disc which is carried out at appropriate stages during testing. The work includes stress relieving of finished discs and subsequent dynamometer testing. This identifies that in-service stress relieving, due to high heat input during braking, is a strong possibility for the cause of disc “warping”. It is also seen that an elastic wave is established during a braking event, the disc returning to its original form on release of the brake.

INTRODUCTION

Whole vehicle NVH refinement has been extremely successful over the past few years and as a result the re-emergence of effort towards finding a solution for problematic brake instabilities has again become the focus of increasing attention. Research towards understanding brake noise has never really remained static and so the advanced analytical techniques developed over the last 20 years, combined with the improved practical investigative skills and the sophisticated test facilities, have all enabled the mechanisms involved to be better understood. As such the technical competence in controlling dynamically induced vibrations, such as brake noise, has grown considerably so allowing this area to enjoy similar successes as general vehicle NVH improvements. As a result of the continuous efforts in the areas of brake noise there are some improved design guidelines available that allow new brake designs to be created with a reduced propensity to generate noise and so the effort is stabilising a little.

This success in one area of braking NVH, noise, is not reflected in the mechanically induced vibrations of brake judder and as such this has not advanced at the same rate. Judder in one form or another is now often seen to be more problematic than noise as shown in Figure 1 where judder issues absorb 73% of new brake NVH research effort. Some
of the fundamental mechanisms are well understood and the practical investigation methodologies are well established, but these remain primarily vehicle based. This is an expensive approach as much equipment must be dedicated and carried on-board the vehicle as indicated in Figure 2 and even then the data that may be collected is restricted because of accessibility and the prevailing environment. In addition the available design guidelines remain restricted with manufacturing accuracy and brake materials forming part of the equation.

The proportion of noise to judder issues is not a constant and varies considerable from vehicle supplier to supplier. The heavier, high performance, vehicle manufacturers appear to suffer most from judder issues with the ratio being about 30% noise to 70% judder. Judder in this regard being as indicated in Figure 1 - Pad deposition, corrosion, thermal and off brake wear. On the other hand the proportion may change for the lighter “town cars” to become 70% noise 30% judder. In the latter case the types of judder will be the same but the proportions will differ from that in Figure 1. It is the demands for high speeds and braking performance on the edge of tyre friction limits that leads to such proportions and proportional changes. Off-brake wear and thermal judder will become more evident as speed increases and thus more prevalent with the higher performance cars. Clearly the speeds and energies involved are less for lighter cars and indeed the required braking forces are less. Brake judder, hot or cold, results from the degree of rotor distortion and the braking force applied. This combination results in a variable force being transmitted to the vehicle chassis and body - the heavier vehicle with the heavier braking being in a better position to generate these forces, hence judder.

Judder is not a brake problem it is a vehicle problem and the front corner assembly design can significantly influence the degree of judder experienced by the driver. Figure 3 [1] shows a typical front corner assembly where it is seen the knuckle holds the disc, calliper, suspension system and steering track rods; these therefore being mechanically linked to the vehicle, and through the tyre to the road. Under normal braking the knuckle will provide an appropriate reaction by transmitting the braking force to the vehicle body and road through any member attached to it and the vehicle. The braking torque would normally be steady so the reaction would be even. If there is any variation in braking torque the reaction force provided by the knuckle assembly will also vary and so the forces seen by the calliper, suspension, steering and tyre will also vary. The driver is also connected to the front corner assembly through the brake hydraulic system (foot pedal), the steering wheel (steering gear) and the cabin structure interface with the chassis. It is the interfaces that allows the driver to experience judder; through vibration of the foot pedal, vibration of the steering wheel and an audible noise. The frequency is not resonant but wheel speed related, similar to wheel imbalance, and in the region of around 100Hz. The effect of judder is not only an audible
experience; it is felt throughout the vehicle as whole body vibrations. This dynamic and often dramatic effect of vehicle vibration demands the focus to be moved from brake related noise problems towards brake judder - both hot and cold. It is the consideration of these two conditions that explain why brake torque may vary.

Cold judder is generally accepted to be all mechanically driven instabilities that are not thermally induced. These include off-brake wear, corrosion and uneven surface film transfer.

Off-brake wear is most obvious with discs having high disc runout or swash (when disc faces are not machined perpendicular to axis of shaft). A typical acceptable axial runout on a newly fitted disc may be in the region of 0.08 mm (80 micron). If the disc is fitted with a spec of rust or dirt in the region of 0.05mm (50 micron) then it is possible that the allowable runout will be exceeded. This swash allows the disc to make contact with the pads during normal running so leading to uneven off-braking rotor wear or disc thickness variation (DTV). On braking this DTV leads to a brake torque variation (BTV) with consequential brake calliper, steering and body vibrations. Figure 4 shows typical wear pattern of a disc with runout.

There may also be inherent DTV as a result of machining inaccuracies. Clearly this may be resolved through improved quality control and effective calliper seal roll-back. The former solution may result in higher base costs whilst the latter will result in enhanced pad retraction and will inevitably affect pedal feel.

Corrosion judder results from disc corrosion because of extended vehicle parking in a mild corrosive environment (dockside storage) and is generally temporary and will eventually reduce during normal vehicle use. The exposed area of the disc experiences rust whereas the area under the pad does not. Removal of the rust during initial braking results in DTV. With DTV the disc experiences uneven
heating, the thicker area exhibiting the higher temperature and expands more than the thinner area -a self serving condition is created.

**Surface film transfer**, or pad deposition, may also cause judder if the film is not evenly distributed. This uneven distribution of surface film causes a variable disc/pad friction value and thus brake torque variation (BTV). The transfer of material from pad to disc and disc to pad is a fact of life and any instability investigation should only be undertaken following a degree of brake bedding or conditioning. This period of conditioning allows an even surface film transfer to be established and so a more stable system regarding the disc/pad friction coefficient. If this transfer is uneven then the interface friction characteristics will also vary and as such BTV will result, hence judder. Control of surface film transfer is difficult and related to pad formulation which has additional demands such as a predictable compressibility, good wear characteristics, stable friction level, good clean-up characteristics and yet not be excessively aggressive. It is an area that requires a careful study of compatible material characteristics and is not the purpose of this paper.

**Hot (thermal) judder** is probably the least understood of all judder experiences because it involves both low and high frequency judder (drone). It occurs during heavy braking and the resulting deformations and service data may reduce and even disappear once the brake cools. The demand for high performance cars will always exist but the high speeds, combined with high deceleration, means the need for the dissipation of high levels of energy. That energy is absorbed and rejected by the brake and as a result thermal effects become the focus of concern. If the brake absorbs energy significantly faster than it can dissipate it then the disc experiences high temperatures and as such thermal deformation. This deformation may be recoverable and as such judder may only be temporary. The issue becomes more serious if the deformation does not recover and so the disc will hold a permanent wavelike shape resulting in classical judder.

It will be shown later that during a heavy braking stop the disc can experience a high temperature rise, in the region of 270°C. Clearly if the driver brakes again before the disc cools then the disc temperature will rise until the “soaking” temperature is reached - when the disc loses heat as fast as it is generated due to braking and a steady state temperature is achieved. With the uneven temperature caused by DTV there is a possibility that the temperature can reach values in the region of 650-700°C. At these temperatures the cast iron changes structure and Cementite is formed. This Cementite is harder than the base cast iron and so wears less and the DTV becomes worse. In addition the Cementite will run hotter as it is not a good “heat sink” and so penetration into the disc surface increases. If the discs are relatively new then the Cementite may be removed by re-machining the discs but if the penetration is too deep then the problem will quickly re-occur on refitting the discs.

It is common for high performance cars to use vented discs as these allow the heat to be dissipated more readily, but the overall purpose and effect is sometimes confused. Vented discs bring with them another form of thermal deformation in the form of “hot spotting” or “blue spotting”. Such heating effects are localised and distributed about the disc in an apparent random manner but if left to multiply then the surface of the disc becomes covered in these “hot spots” as shown in Figure 5. The hot spots may be localised regions of Cementite and as such the problems described above are equally relevant with these “hot spots”. The overall effect is high frequency judder leading to a droning sound caused by the multiple localised thermal deformations about the disc surface. It is tentatively believed that high frequency judder (drone) is caused by uneven temperature distribution on the disc surface, or within the brake rotor, causing both thermo elastic deformation and thermo elastic instabilities. Disc ripple also gives such effects. All these definitions and causes have been detailed in previous literature and therefore it is sufficient to only briefly mention them.

![Figure 5. Extreme case of “hot spotting”. Often appears to be random but may be a function of rotor speed.](image)

This paper reviews earlier work on the topic of thermal judder and focuses on establishing a disc design that avoids “heat sinks”, moving towards a more even heating and cooling of the disc surface and body. Such a design considers the geometry of the disc/pad interface and on the rib profiling. The purpose is not necessarily to increase the cooling capacity of the brake disc but to encourage a more temperature stable brake disc by adequate consideration of the heat distribution across, around and through the disc, in addition to overall even cooling of the disc body.
THERMAL ANALYSIS
To evaluate the thermal impact on a braking system it is possible to estimate the heat generated by a typical ABS stop for a typical vehicle. Consider values for the vehicle as indicated in Table 1:

Table 1. Characteristics of a typical high performance vehicle.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car mass</td>
<td>(m) 2600kg</td>
</tr>
<tr>
<td>Rotational Inertia</td>
<td>(Ieq) 126kgm²</td>
</tr>
<tr>
<td>Tyre radius</td>
<td>(r) 320mm</td>
</tr>
<tr>
<td>Mass equivalent</td>
<td>(meq) 1230kg</td>
</tr>
<tr>
<td>Initial speed</td>
<td>(u) 50m/s (180kph)</td>
</tr>
<tr>
<td>Final Speed</td>
<td>(v) zero</td>
</tr>
<tr>
<td>Uniform deceleration to zero</td>
<td>(d) -0.8g</td>
</tr>
<tr>
<td>Front disc mass</td>
<td>(mₙ) 15kg</td>
</tr>
</tbody>
</table>

Where “mass equivalent” (meq) represents the rotational inertia of all the rotating parts within the transmission and also the non-driven wheels. These rotational inertias are referred to the wheels and represented as a mass that is added to the vehicle mass (meq=Ieq/r²).

Ignoring drag and rolling resistance.

The Kinetic Energy (KE) = ½ (m + meq) v² = ½ x (2600 + 1230) 50²

KE = 4.79 x 10⁶ J

v = u + dt where t = braking time (s).

This is transposed to give t = u/d resulting in t = 50/(0.8 x 9.81) = 6.37 s

Power = Energy/ time = 4.79 x 10⁶/6.37 = 752 kW

This power/energy is absorbed by the discs and is seen as heat.

Assuming proportion to front brakes during the stop 85% (not unreasonable).

From

Q = mₙ C_p ΔT

For front brakes this gives

ΔT = Qfront / (mₙ C_p)

ΔT = 0.85 x 4.79 x 10⁶ / (2 x 15 x 502)

ΔT = 270 °C

for a single ABS stop.

This assumes all the disc has time to absorb the heat which will not be the case and the disc will experience high thermal gradients across its thickness with the surface temperature being well in excess of the 270°C. If the disc does “hold” all the heat uniformly then it will experience a thermal shock and will try to expand circumferentially. The time is insufficient and high mechanical stresses will be imposed on the disc. Consider the blade disc as a beam of section 97.5mm x 40mm thick. Let the disc be represented as a straight beam of a length equal to its effective circumference and constrained at its ends as shown in Figure 6.

The disc material characteristics are listed in Table 2.

Figure 6. Disc blade represented as a beam constrained at its ends.

Table 2. Characteristics of beam (disc):

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature rise (ΔT)</td>
<td>(°C)</td>
<td>228</td>
</tr>
<tr>
<td>Young’s Modulus (E)</td>
<td>MPa</td>
<td>86300</td>
</tr>
<tr>
<td>Poisson’s Ratio (ν)</td>
<td></td>
<td>0.25</td>
</tr>
<tr>
<td>Coefficient of Thermal expansion (γ)</td>
<td>1.00E-06/C</td>
<td>11.23</td>
</tr>
<tr>
<td>Specific Heat capacity (C_p)</td>
<td>J/kgK</td>
<td>502</td>
</tr>
<tr>
<td>Density (ρ)</td>
<td>Kg/m³</td>
<td>7250</td>
</tr>
</tbody>
</table>

The temperature rises over a very short space of time.

\[
\delta L = \gamma L \Delta T = 11.23 \times 10^{-6} \times 0.997 \times 270 = 3.027 \text{ mm}
\]

Thermal expansion

Giving Strain

\[
\varepsilon = \frac{\delta L}{L} = \frac{3.027}{997} = 3036 \mu \text{strain}
\]

and induced thermal stress = \( E \varepsilon = 86300 \times 10^6 \times 3036 \times 10^{-6} = 262 \text{ MN/m}^2 \)

Axial mechanical load within disc to such cause stress = Stress \times Area = 262 \times 10^6 \times 0.0975 \times 0.040 = 1.022 \text{ MN}

But buckling load of beam is given by Euler equation
THE FORMATION OF A WAVELIKE SHAPE DURING BRAKING.

With such loading considerations as shown by the thermal analysis it is concluded that the beam, or in this case of the disc blade, will buckle as the internal constrained/induced load is more than twice the allowable critical buckling load. This may be one reason for the disc to exhibit a wavelike shape. If the induced stress exceeds yield of the material then the deformation will remain to some degree. Such a disc shape may be seen in Figure 7 which shows runout (0.06mm) as supplied and the permanent wave after bedding and several braking events. It is seen that the disc has developed a permanent wavelike shape as predicted. It is seen that the wave will reach a limited maximum distortion of about 0.2mm and then stop to deform any further. If the disc is reground and retested it will once again continue to deform further. It is suspected that the continuing deformation is a purely mechanical (inbuilt stress) limitation, the regrinding removing that constraint. Runout measurements of the front and reverse side of the disc shown a thickness variation to be forming. This is seen on Figure 7 where the leading flank of the wave shown two distinct profiles.

This feature may be seen on most runout measurements. It is felt that this is due to the pad “ploughing” into the face resulting in high pressures at the interface and thus increased wear. The trailing flank does not exhibit the wear as the pressures are less. This type of contact also results in a brake torque and pressure variation. Such a typical recording of BPV is as shown in Figure 9 where the pressure fluctuation is around 0.1 MPa peak to peak with a mean pressure of about 0.44 MPa. The disc temperature during the recording was 150°C. This pressure fluctuation (measured on a test rig during earlier work) [2] may be compared to on-vehicle pressure measurements of an entirely different brake - Figure 10.

![Figure 8. Shows pad negotiating the disc wave. The leading face causes an increase in pressure but also results in increased wear of this face.](image)

The similarities are clear and may infer that there may be repeatability in the mechanism regardless of disc type.

![Figure 9. Resulting brake pressure fluctuation resulting from the permanent set wavellike shape.](image)
MANUFACTURING CONSIDERATIONS

CASTING CONCERNS

Casting is not an exact science and during dynamometer testing there was evidence of asymmetric heating of a disc during braking. Such thermal gradients may cause asymmetric thermal expansion of the disc, hence differences in deformation, and may contribute to the formation of the wavelike shape. The temperature variation tended to be related to the balance groove and a simple test was undertaken to investigate the density of the disc about its diameter. The discs were cast vertically which may result in change in density across the diameter due to carbon migration. To test whether this was the case 8 sections were cut from a disc, machined individually to a common size and weighed [8]. The normalised result is shown in Figure 11. The change would not appear significant, it being only 4% overall, but the introduction of any temperature gradient may induce small changes that are exacerbated during the braking process. A secondary issue is the asymmetric stresses remaining after machining due to the density variation. It was this aspect of manufacturing that was investigated further.

STRESS RELIEVING EFFECTS

The formation of a wavelike shape in the disc may be due to thermal “buckling” as indicated earlier. Other factors have to be considered and as such the stress relieving process during braking was investigated. It was felt that the repeated heating and cooling of the brake disc during testing was relieving the unstable stress field which remained after the casting and machining processes, the uneven density adding to this issue.

Stress Relieving Measurement and Experimental Procedure

Earlier brake dynamometer testing had indicated a possible stress relieving process was occurring during hot running of the brake disc. To identify whether this was the case testing was conducted consecutively on both standard un-used brake discs and also on brake discs which had been manually stress relieved after machining. The process would be in stages, namely; to measure the flatness of the brake discs, to stress relieve, re-measure, test and finally to measure again.

Bare metal brake discs were delivered for measurement without a protective zinc flake coating applied; it was therefore possible to get the discs stress relieved, minimising any scaling or oxidisation on the disc surface. The stress relieving process involved heating the discs to 550°C in a vacuum followed by a controlled cool-down. Following this the discs were measured again to identify any deformation as a result of the stress relieving process.

The absence of the protective coating increased the accuracy of the measured results. All surface measurements were performed using a CMM with a probing error of 0.7µm, a scanning error of 1.3µm and a volumetric accuracy better than +/−3 microns.

Subsequent measurements of the installed run-out (cold) on the dynamometer were recorded using digital DTI with a resolution of 1µm and an accuracy of <3µm, whilst the dynamic hot disc shape was recorded using non-contacting capacitive displacement transducers with a capture frequency of 10 kHz. Each dynamometer test comprised a sequence of eight, eight second braking events each separated by four seconds. The dynamometer speed was a constant 1200r/min, whilst brake pressure was 16bar, equivalent to a brake torque of approximately 600Nm.

Stress Relieving Results and Discussion

Figure 12 shows the measurements of a disc that has followed only the manual stress relieving process. The images clearly show that following stress relieving the disc has deformed significantly with the right hand edge deflecting upwards by approximately 40 microns. The graph shown in Figure 13 shows a scan at the mean rubbing radius of the brake disc pre and post stress relieving. It is evident that the disc has
changed from low (15 microns) first order run-out to a large (65 microns) predominantly first order wave, with a secondary peak as shown.

The reverse (outboard) side of the disc was also measured and as expected this reflected the same tendency to deform. In this case the friction ring transformed from 18 microns to 57 microns of run-out with again a secondary peak being evident after stress relieving.

Surface CMM scans from a second disc showed a similar second order mode of deformation present within the disc following the stress relieving process. This basic test shows that during the manufacturing process troublesome stresses are retained within the disc material as a result of uneven removal of material or variable density of the basic casting which will also disrupt the distribution of the inbound stresses following casting and cooling.

It is believed that this stress relieving process is occurring during braking as a result of the elevated disc temperatures followed by cooling. It is unclear as to why the disc takes up second order deformation following stress relieving and the orientation of the deformation was different for the two discs tested. The differences may be due to density variation superimposed on the casting/core tolerances. Regardless this has clearly identified that the stress relieving process is occurring during the braking process.

**Dynamometer Testing of a Stress Relieved Disc**

A further test was undertaken to evaluate the deformation of a disc that had been stress relieved and then tested on a dynamometer. The dynamometer test was carried out using a stress relieved disc to identify if there was any further change in the distortion, or evidence of further thermo-plastic deformation during braking. It would be expected that there would be little or no change in the post stress relieved deformation during dynamometer testing if the stress relieving theory was valid. It was felt that the stress relieving process would remove any tendency for the brake to deform if all the manufacturing stresses had already been removed. The result of such a test is shown in Figure 14. This graph shows that there is little change between the pre and post dynamometer test run-out of the brake disc in both magnitude and shape. The minor changes that do take place may be argued to being the result of residual stresses still being held in the disc, the stress relieving process being out of the control of the authors. It would be expected that if the thermal deformation was as a result of the braking then the deformation would occur regardless of stress relieving. In general it may be concluded that deformation of a disc does occur when the disc is stress relieved, either prior to machining or during braking.

In summary, if this stress relieving takes place during braking the disc will exhibit a second or higher made shape and hence judder. If the stress relieving takes place during the manual stress relieving process the disc deforms as expected and
before testing and during testing it does not change significantly. If this is the case then thorough stress relieving before machining could help reduce in-braking stress relieving and subsequent thermal distortion.

Confirmation of the test may be seen by comparing Figure 14 and the two shapes in Figure 15. These show a standard disc front set (as supplied) which was brake tested in the same manner as the disc in Figure 14 (a sequence of eight, eight second braking events each separated by four seconds). The marked amplitude changes are more obvious which shows the effect of in-braking stress relieving.

**Summary and Comment**

On-vehicle and dynamometer testing show the disc to establish a wavelike shape during braking. A similar wavelike shape occurs if the disc is simply stress relieved. It is felt that this is not a coincidence and that the continuous wavelike shapes are due primarily to in-braking stress relieving. It is believed that this has the effect of changing the disc distortion from first order ‘cold’ deformation to second order deformation during brake testing. Stress relieving a pair of un-used discs has supported this speculation, identifying that there are significant stresses retained in the disc following casting and machining, and that the removal of these stresses allows the disc to deform significantly. In both cases the mode of disc deformation was second order. It is proposed that repeat testing is carried out to fully substantiate this “de-stressing” process. It is also necessary to absolutely confirm this element of distortion by testing a disc which has been thoroughly stress relieved prior to machining. It has been shown that stress relieving a disc after machining allows the disc to deform in a manner according to the degree of stress remaining after processing. The resulting run-out of the disc becomes quite large leaving the disc susceptible to cold low frequency judder in addition to off brake wear, which will eventually lead to drone. It must be mentioned that stress relieving after machining imposes a necessary degree of care that may prevent full stress relieving to be realized. Stress relieving prior to machining would remove this problem and also give a very good base disc to further analyse the “elastic” deformation of the brake disc during braking, without any other influencing factors.

**DYNAMOMETER TESTING AND TEST EQUIPMENT**

Elastic Behaviour: Some of the results of dynamometer testing have been shown earlier but these were thermo-plastic behaviour, typically Figures 13 and 14. Dynamometer testing...
here is to show the “elastic” behaviour that may occur during braking. The brake dynamometer used for the purpose of this testing is a bespoke design created especially for investigation of brake judder. The dynamometer comprises a brake rig and motor as shown in Figure 16. Briefly the design encompasses the following aspects:

A full quarter car suspension is identical to that used on the vehicle and is rigidly mounted to a substantial steel backplate where it would be mounted to the vehicle hard-points. The suspension geometry is aligned so that the driveshaft is in an optimal horizontal position as would be seen on the vehicle. Loading of the suspension is applied through the base of the upright using a rubber mount to simulate the tyre stiffness. The quarter car suspension allows investigations of brake judder into the transmission path from the brake assembly, through the suspension, to the vehicle structure.

It also allows excitation frequencies of the various components to be analysed. The brake disc is directly coupled to a 110kW motor using the original driveshaft and CV joints. This allows unconstrained movement of the suspension system and brake head. The motor can drive the discs at speeds of up to 2400rpm with a maximum brake torque of 700Nm. This brake torque is sufficient to replicate the typical high speed low deceleration braking which can cause brake judder. The brake is actuated via a pneumatic proportioning valve which controls hydraulic pressure in the brake master cylinder and allows for repeatable brake actuation. The brake assembly used for the purposes of brake judder investigations comprises of a single piece cast iron vented brake disc and two piston sliding fist type calipers. The disc is designed with 40 straight vanes and is 405 mm diameter. The brake disc itself is mounted to the hub bearing using the centre section of the vehicle wheel. This allows identical mounting conditions, including clamping area, stress and torque, to be replicated. The mating faces of both the disc and hub are cleaned prior to mounting, and equal torque is applied to each of the mounting bolts. In doing this the installed run-out of the disc, and therefore the possibility of cold judder due to off brake wear, is minimised. It has been argued [4] that a full suspension dynamometer presents the most realistic results when carrying out high speed judder tests, allowing excitation frequencies of various suspension components to be investigated.

Measurement of the dynamic disc thickness variation and surface run-out is made possible by the use of non-contacting capacitive displacement transducers connected to a high speed data acquisition device. The transducers are rigidly mounted to a thick steel plate to eliminate any unwanted vibrations, as shown in Figure 14, and are positioned at the mean rubbing radius of 167 mm. The steel plate forms an arc around the disc allowing measurements to be taken at different positions. Rubbing thermocouples are mounted on the disc surface to give reference temperature measurements for both displacement and pressure measurements. The thermocouples also allow for accurate calibration of the thermal camera to take place by allowing for emissivity changes of the disc surface. Brake line pressure is measured at the caliper using a pressure transducer.
ELASTIC & THERMO-ELASTIC EFFECTS

Dynamometer testing shows the distortion of the brake disc to be highly dynamic in nature, with quite dramatic changes during and after braking. The mode order of brake disc distortion can change rapidly during the duration of a single braking event. The disc may be seen to revert back to lower orders of distortion immediately upon release of the brake disc, generally 1\textsuperscript{st} order. A typical dynamometer test sequence is shown in Figure 15, which highlights the application of the brake pressure. Each dynamometer test involves eight second braking applications at 16 bar and 1200 rpm.

From cold the brake disc being used for this example exhibits 1\textsuperscript{st} order run-out as shown in Figure 16.

When the brake is applied during the first braking application the disc shape is still predominantly 1\textsuperscript{st} order, however a superimposed higher order of displacement is also more apparent as shown in Figure 17. This may be caused by expansion of the raised regions, seen in Figure 16, due to non uniform contact and subsequent heating of the brake disc surface. The run-out of the disc at the initiation of braking also appears to be reduced due to the clamping action of the brake pads.

As braking progresses the order of deformation is seen to change from 1\textsuperscript{st} order to 3\textsuperscript{rd} order approximately mid way through the braking event. The data shown in Figure 18 is taken at $t = 7.5$ seconds into the braking event.
Figure 18. Graph showing disc deformation towards the end of the first braking application.

It highlights the change in disc deformation during braking, where three peaks are clearly seen per disc revolution. The relative magnitude of deformation of the disc has now doubled from the initiation of braking and is now the same magnitude as the cold disc.

Figure 19. Graph showing disc deformation immediately after the brake is released

Immediately following the release of the braking force the disc reverts back to dominant 1st order deformation shown in Figure 19. It is interesting to note that the waveform now has a slight “plateau” possibly due to wear at the peak of the waveform. There is evidence of secondary peaks on each “plateau” which would also indicate plastic deformation and the move towards higher mode orders. It must be noted that the sequence of events shown is a repeatable event on this brake system which is observed during every test on the brake dynamometer. These results indicate both plastic and elastic deformation occurs during a braking operation - both contributing to the occurrence of judder. The displacement traces shown in Figure 20 show the results of a subsequent test with the following sequence; no braking, start of braking, end of braking, brakes released. Again the “off braking” at the start and finish are the same indicating a degree of disc elastic deformation is taking place during braking. The images of elastic recovery are (Figure 20) not unusual but have only been able to be detected on the test rig because of the difficulties of on-vehicle testing. These elastic effects may have been recorded by the test engineers as some other form of judder as the deformation cannot be readily detected on the stationary vehicle.

Figure 20. Graphs of outboard side of disc displacement for the 1st braking event.

The results shown are for the first stop of the test and it highlights that the disc distortion is occurring from cold and almost instantaneously. Also of note is the apparent similarity of the waveform between both off-brake periods, this being an indication of elastic behaviour. As the braking sequence progresses and the temperature rises the “on-braking” deformation tends to become “set” and as such the disc does not fully recover back to its original shape - a permanent plastic deformation takes place. The above is a demonstration of the elastic/plastic behaviour of the disc during braking, the elastic possibly providing misleading information to the brake test engineers.

FE THERMAL MODELLING

For the purpose of this paper a simplified transient thermal model of a stationary brake disc with heat flux being applied to each friction ring was modeled. This was constructed using a mesh of tetrahedral elements and represented a single brake disc as used by the research vehicle. Transient simulations of three braking events from a vehicle test were chosen because the purpose of the investigation was to identify temperature distribution over time; with a steady state analysis only
constant heat flux loads could be applied, this was not representative of the actual system where the heat flux loads were nonlinear due to the aerodynamic drag force on the vehicle which was proportional to the square of the vehicle speed and aided vehicle deceleration. With a transient model the heat flux loads were non linear and therefore more representative of the actual system.

This allowed for investigation of heat flow through the brake disc due to the transient heat flux loads, whilst identifying any areas of concern. Whilst this was not a like for like simulation, the results presented demonstrate that this method has good correlation with measured data.

The thermal FEA simulations were validated against measured data from on-vehicle testing with a material property sensitivity analysis to improve the correlation of the results with the measured data. This validation is shown in Figure 21. Simulations were then performed to identify the heat distribution through the brake disc and a new design was created to reduce the brake disc operating temperature and to improve the heat distribution through the brake disc to reduce the propensity towards hot spotting and thermal distortion.

The FEA models used blanket convective cooling coefficients. It can be seen that the simulated disc temperatures in the final cooling phase fell quicker than the measured data (Figure 21). This indicated that the linear cooling curve used for the simulation did not fully represent the actual vehicle conditions in the final cooling phase. However the FE study was mainly focused upon the heating phase and the simulated data showed good correlation for the three braking events therefore no modification to the convective heat transfer coefficients was required.

The FE analysis showed that vent geometry influences the surface temperature profile causing a rippling effect on the disc surface - Figure 22. A small section of the analysis is shown in Figure 23 where the thermal ripple is more evident. This ripple will be the same as the disc “wavelike” shape but at a lower level of amplitude. The effect though would be the same, high thermal areas, the formation of hot areas leading to possible “hot spotting”, the formation of Cementite and so on. This is a demonstration for the need to provide a uniform heating and cooling design.

The FE modeling of disc revised design

Current work is concerned with developing a vane geometry to reduce disc maximum temperature and at the same time provide uniform heating and cooling. The predicted thermal result from such a preliminary vent design is shown in Figure 24 where evidence of the vane thermal profile is minimal. The prototype design centered on modifications to the vent profile of the standard brake disc to achieve an improved temperature profile and reduced disc temperature. The FEA prototype design incorporated two wavy profiles, both symmetrical about their own mid planes. The aim of this was to add mass midway between the vanes to try and draw heat more evenly from the surface of the disc and to avoid build-up of the hot regions on the surface and vent ceiling.
Figure 24. Principle of adding mass to inner vent face but “thinning” blade thickness to maintain overall mass.

The redesign was created with improved sculpting of the vent cross section to distribute the thermal, mass allowing for improved heat conduction path (Figure 24). The design of the vent is as shown in Figure 25. The general profile of the disc would then be as shown in Figure 26 which promotes more even heating of the disc surface, reduces the thermal gradient and improves heat distribution.

Several profiles were investigated as indicated in Table 3. It must be noted that N1, N2 and N3 refers to the geometry of the dimple protruding into the disc vent with N3 protruding the furthest. For the FEA simulations all discs (apart from the standard design) had the friction rings thinned so that the mass of the disc was identical to the standard design. This removed the effect that the added mass would have on the disc temperature. It can therefore be claimed that the temperature reduction is due to the redistribution of the mass in the vent and not an increased in disc bulk mass.

Such a modification reduces the thermal stress gradients and the potential for uneven thermal deformation. The design showed a 46°C improvement in maximum surface temperature following a 3 stop simulation for an equal disc mass. This reduction is equivalent to an 8% reduction in a disc temperature at 550°C.

Concurrent work continues with the improved aerodynamic design of the disc pillars and vents [9]. The objective in this case would be to avoid airflow separation from the disc walls and pillars. It is anticipated that combining the re-sculptured pillar with the improved aerodynamic geometry will further improve disc temperatures and cooling.

**SUMMARY/CONCLUSIONS**

- Thermal judder is associated with high performance cars and in particular heavy high performance cars. As such the

<table>
<thead>
<tr>
<th>Design</th>
<th>Wetted Area (single vent)</th>
<th>Cross Sectional Area (measured at disc outer radius)</th>
<th>Max. Disc Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard</td>
<td>0.00577m²</td>
<td>0.00027816</td>
<td>552°C</td>
</tr>
<tr>
<td>N1</td>
<td>0.00490m²</td>
<td>0.0002m²</td>
<td>544°C</td>
</tr>
<tr>
<td>N2</td>
<td>0.00497m²</td>
<td>0.00019m²</td>
<td>538°C</td>
</tr>
<tr>
<td>N3</td>
<td>0.00505m²</td>
<td>0.00016m²</td>
<td>506°C</td>
</tr>
</tbody>
</table>

Figure 25. Preliminary aerodynamic design of disc vent. The CFD analysis shows reduced separation and improved air flow [9].

Figure 26. Scrap view of predicted (FE) thermal pattern with the modified vents as shown in Figure 25. Profile replicates that in Figure 24.
brakes have to absorb, and dissipate, a considerable amount of heat. If the heat absorbed exceeds that which is dissipated then the disc will reach temperatures such that the disc material will undergo a phase change with the formation of localized Cementite and a plastic deformation will be imposed on the disc. There is no issue with understanding the energy that has to be absorbed - that is a scientific fact and the brake engineer can do little about that. The biggest problem of the brake engineer therefore reduces to heat management and better distribution of that heat.

- Judder is complex and may result from a combination of elastic and plastic disc deformation.
- The properties of the disc material may change due to excessive heating, Cementite being formed.
- The disc can relax back to differing orders of deformation upon cooling, usually first or second order. A disc which has previously taken up second order deformation upon cooling can relax back to first order deformation in follow-up testing.
- The pressure response leading to brake judder has shown to have a component at the same frequency as the disc deformation. These higher frequencies may be related to localized heating of vane “clusters”.
- For a sharp rise of only 300°C it would be expected that the disc would buckle and form a wavelike shape.

**MANUFACTURING SUGGESTIONS**

- Within this manufacturing process the design and production issues need to be addressed. At the outset the casting process priority must be to improve mass distribution. This may include improved core position control through effective tolerancing, casting orientation and stress relieving before machining.
- The first process in disc design phase is to reduce temperature variation during the braking process, that is to ensure overall even heating of the disc, acknowledge there is a difference between the inboard and outboard braking forces and arrange the design to account for that. The principal reason is to minimize inboard to outboard, radial, thickness or circumferential temperature gradients. The existence of large gradients will lead to localised deformation.
- The second challenge is to ensure even cooling of the disc - again the avoidance of large thermal gradients to more acceptable levels. The cooling needs to be lost as quickly as it is gained but under extreme braking that is difficult so the primary concern should be towards even cooling. The process is therefore air management around and through the disc by way of optimized vent design, care regarding cast surface finishes and well placed turbulence generators. The generation of heat generation radially needs to be recognized and the mass distributed accordingly to account for that. Such design considerations will maximise heat transfer through the body of the disc, with even cooling being paramount.
- The third challenge is to accept that temperature gradients will always exist - they will be unavoidable over the myriad range of braking situations possible. It was mentioned very early that judder is not a braking issue - it is a vehicle issue. As a result pressure fluctuations will be inevitable and the associated brake torque variations will always be present. The choice then is to collaborate with the suspension engineers in an attempt to desensitize the vehicles from the likely frequencies.
- If the forgoing proves unacceptable then the brake engineer needs to accept that the root causes of thermal judder and drone will always exist and accommodate for the brake pressure fluctuations at source. The elastic deformation, as measured and observed during the braking event, needs further investigation.

**REFERENCES**

1. http://media.photobucket.com/image/wheel%20knuckle/superex91/100_0538.jpg

**BIBLIOGRAPHY**

Afferrante, L., Ciavarella, M., “Thermo-elastic dynamic instability (TEDI) - how frictional heating excites the


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TERMINOLOGY

NVH
Noise Vibration & Harshness.

DTV
Disc Thickness Variation.

BPV
Brake Pressure Variation.

BTV
Brake Torque Variation.

BPF
Brake Pressure Fluctuation.

CMM
Coordinate Measuring Machine.