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INVESTIGATION OF AUTOMOTIVE CREEP GROAN NOISE WITH A DISTRIBUTED-SOURCE EXCITATION TECHNIQUE

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Creep groan is a high-intensity, low-frequency noise and vibration problem that affects road vehicles at very low speeds. It usually persists for short periods of time, but a "skilled" driver can deliberately make it last several seconds by tuning the force exerted on the brake pedal. The original cause is considered to be a self-induced vibration of the brake components, due to the friction material characteristics that make the system prone to a stick–slip behaviour. No clear evidence upon the creep groan and how it is perceived inside the passenger cockpit has yet been analyzed in the literature and no formal methods are yet available for its analysis. The present study focuses on the transmission of the vibration from the brake component regions to the ears of the vehicle occupants. Characterization of the calliper acceleration and noise inside the cockpit are described for a test vehicle. Distributed-source noise excitation via the standard vehicle hi-fi system is proposed as a practical but less rigorous particular application of the exact reciprocity method. Virtual groan (in which sound power is delivered by means of a loudspeaker) dismisses the airbone path and shows that the phenomenon is structure-borne. On the examined vehicle, front brakes contribute more strongly than rear. Groan frequency close to cavity acoustic resonance constitutes the worst case scenario, and has to be avoided.

1. INTRODUCTION

Noise and vibration problems are nowadays among the main concerns connected with road vehicle brake systems. This category of defect does not involve safety, but the loss of refinement and acoustic comfort introduces risk of customer dissatisfaction. Road vehicle vibration and brake noise can be classified according to its dominant frequency and its triggering conditions (such as decelerating or moving slowly while braking). Traditional categories are represented by judder, groan, moan and squeal, respectively corresponding to around 10 Hz, 100 Hz at very low vehicle speed (below 2 km/h), 100 Hz at higher speed (10–30 km/h) and more than 1 kHz vibration frequency phenomena [1–3].

Creep groan denotes unpleasant low-frequency vibration occurring on road vehicles at very low speed, generated by the brake systems, either accelerating or decelerating. The phenomenon may take place whenever only a light pressure is exerted by the driver on the brake pedal and some forces are acting on the vehicle. The modest brake torque does not completely inhibit the vehicle motion and a friction-induced vibration cycle arises at the contact interface between the stator (brake pads) and the rotor (brake discs). An automatic transmission equipped car waiting at a traffic light represents the typical picture. When "drive", and not "neutral", is set on the automatic transmission selector, the driving wheels

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are connected to the idling engine through the hydraulic torque converter, and the driver needs to operate the brake pedal, to not move forward. If the pedal is slightly released, creep groan prone conditions arise. The skilled driver can intentionally make the brakes groan in this manner for a prolonged period lasting even several seconds.

The first cause of the vibration is the stick–slip behaviour of the brake pads at the rotor surface and the friction coefficient velocity dependence [4–6], with static coefficient of friction higher than the dynamic value. It is well known that any effective brake friction material has the above-mentioned characteristics (coefficient of friction/velocity gradient) and it is important to provide a solution to the problem that can deal with these friction material properties. Noise with frequency around 100 Hz is highly vehicle dependent since it is usually influenced by chassis or body structural resonances.

This study focuses on the path followed by the vibration to reach the vehicle cockpit and the car occupants from the brake components. A set of experiments has been designed to show evidence of the structure-borne nature of the creep groan. The present work involved a test vehicle (a rear wheel drive luxury automatic saloon) and a specifically devised laboratory rig that includes the full front subframe of the target vehicle. The test vehicle features a four disc power-assisted brake system. Brake callipers, of the so-called floating type, consist of several parts; among them, major components are the torque plate directly bolted to the knuckle (hub-carrier), the calliper fist, a single brake piston and rubber coated pins. The calliper fist translates laterally when the brake fluid pressure pushes the inner pad against the internal side of the cast iron brake disc, and the outer pad against the external side, as a result of the reaction.

Creep groan is known to occur at the driving axle and/or at the undriven wheels. The mechanics of the low-frequency vibration involve rigid-body oscillations of the full assembly, which includes brake pads, calliper and knuckle [7].

2. EXPERIMENTAL RESULTS

Four independent experiments are reported herein, undertaken on the test vehicle:

- *Brake groan measured on the moving vehicle*: to assess noise levels in the passenger cockpit and acceleration levels of the brake components and their frequency characteristics.
- *Brake groan sound power emitted by the front brakes.*
- *Wheel arch to cockpit sound transmission loss*: with the aim of estimating the significance of the airborne path.
- *Simulated "airborne" groan by using a loudspeaker in the wheel arch*: a calibrated loudspeaker emits the previously estimated brake groan sound power. It produces a *sound only groan*, which takes place with no structural vibration. Comparison with the actual brake groan shows the relative unimportance of the airborne transmission.

2.1. BRAKE GROAN ON THE MOVING VEHICLE

A noise survey was carried out on the test vehicle in the free field. The vehicle had been fitted with acceleration transducers on front and rear callipers (locations K and L respectively) and two microphones, one at driver's ear location (A), the second at the back of the passenger cockpit (B), close to the rear screen, as described in Figure 1.

The groan phenomenon is obtained by diminishing slightly the foot pressure on the brake pedal and letting the car advance. The vibration transits through a few phases with different

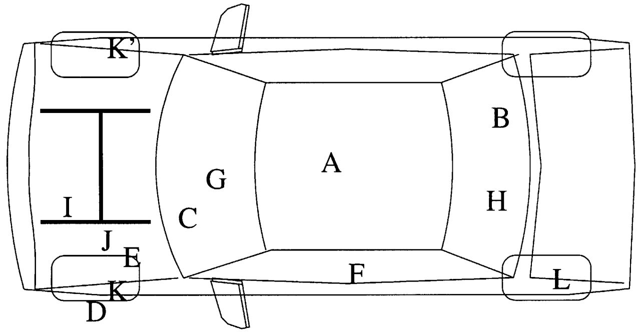

Figure 1. Transducer locations. Microphones: inside the cockpit, (A) driver's ear, (B) rear screen, (C) windscreen, and outside, (D) brake disc near field, (E) loudspeaker under the wheel arch. Accelerometers: (F) B-post, (G) windscreen, (H) rear screen, (I) subframe, (J) chassis, (K) front left calliper, (K′) front right calliper, (L) rear calliper.

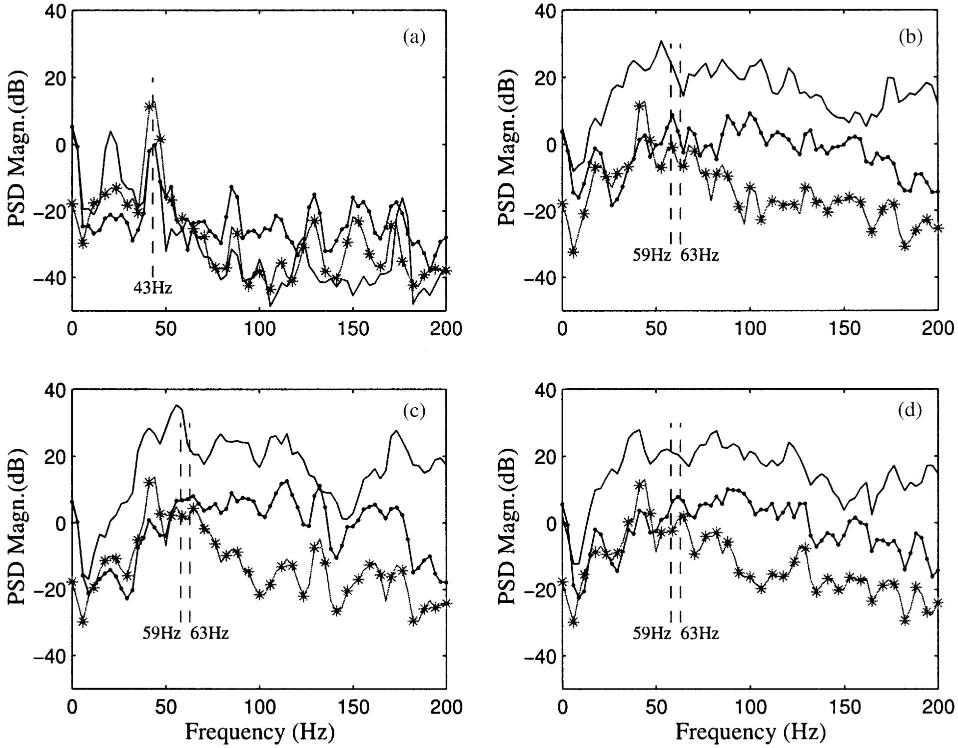

Figure 2. Comparison of different phases of the vibration. (a) No groan, (b) chaotic beginning, (c) tonal groan, (d) final phase. PSD functions of front calliper vertical acceleration (——, location K, reference 1 m/s^2), windscreen (●–●–●, G, 1 m/s^2) and recorded cockpit noise (∗–∗–∗, A, 1 Pa). Block length 0·3 s. Resolution 3 Hz.

characteristics, as illustrated by the short time PSD of the signals in Figure 2. They represent front calliper vertical acceleration (location K), windscreen acceleration (G, normal to the surface) and the recorded noise at driver's ear (A). While no groan is taking place, the noise at 43 Hz is due to the engine firing, (Figure 2(a)). At the beginning, the vibration of the front calliper (2(b)) is rather chaotic, then it settles (2(c)) to a reasonably

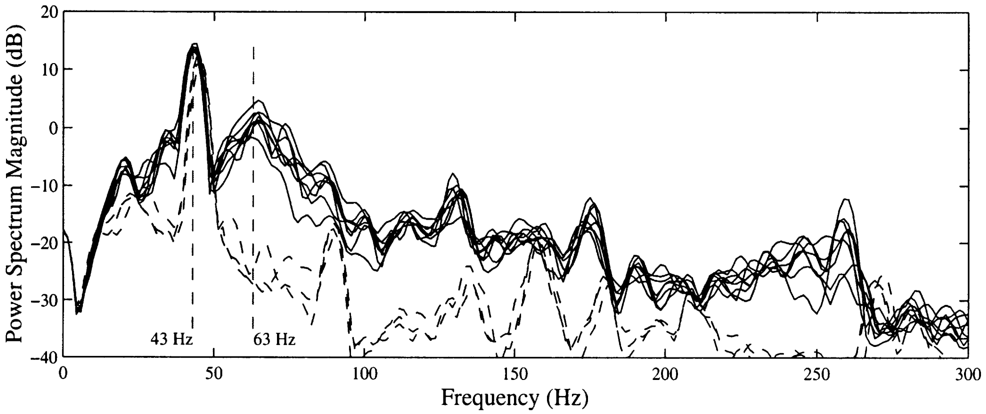

Figure 3. Recorded noise power spectral density function. Chaotic and tonal phases, several cases. Microphone at driver's ear (A). Comparison between idling engine only (full pressure on the brake pedal, – – –) and groan vibration case (——). Reference 1 Pa. Resolution 1·5 Hz.

regular frequency of 59 Hz. While the phenomenon decays (2(d)) its frequency content is similar to that of the beginning phase.

The excitation from the calliper is relatively high in the 49–63 Hz band (Figure 2(c)) and it is accompanied by high windscreen activity between 57 and 66 Hz. The calliper noise too has a broad peak in the neighbourhood of 59 Hz. This is the main problem as far as the passengers are concerned.

Power spectral density of the sound pressure (C-weighted) recorded at driver's ear (location A) is shown in Figure 3. Several groan events have been compared with the noise in normal idling engine conditions (dashed curves). Now, the PSD functions have been calculated taking into account the whole length of the phenomenon. This results in an overall average of the energy over the complete process. The spike at 43 Hz denotes the base firing frequency of the V8 internal combustion engine. Its harmonic is noticeable at 86 Hz. It can be observed that the groan is a very reproducible phenomenon and its frequency domain signature is clearly consistent. Measured sound pressure levels were 93 dB(C) at the source, calliper region near field (location D, 5 cm from the brake disc), 102 dB(C) inside the cockpit (rear screen, B), 83–84 dB(C) at the driver's ear position. The centre of the groan noise band falls at 63 Hz, including chaotic and tonal phases, Figure 3. Then a frequency range up to 170 Hz shows a remarkable difference between the groan noise and the engine only case.

From now on, data will refer to the sustained tonal phase of the event. Figure 4 contains the frequency analysis of signals very close to the vibration source. Power spectral density functions of vertical acceleration of front and rear callipers are plotted. Front left calliper (K) acceleration is characterized by a peak in correspondence of 59 Hz, while the main peak for the rear case (L) is at 133 Hz. Tonal groan occurs on the front right calliper at about 60 Hz. Velocities of the components, obtained from integration of the present signals, are displayed in Figure 5.

Component acceleration coherence functions (front calliper, rear calliper and windscreen) with the recorded noise are shown in Figure 6. Lack of coherence between the recorded noise and the rear calliper in the important region around 60 Hz (Figure 6) gives the impression that, with respect to the present vehicle, cockpit noise is mainly due to the brake vibration on the front end. Figure 7 shows acceleration levels recorded on the front suspension subframe (Figure 1, point I), on the chassis (J), on the windscreen (G) and on the

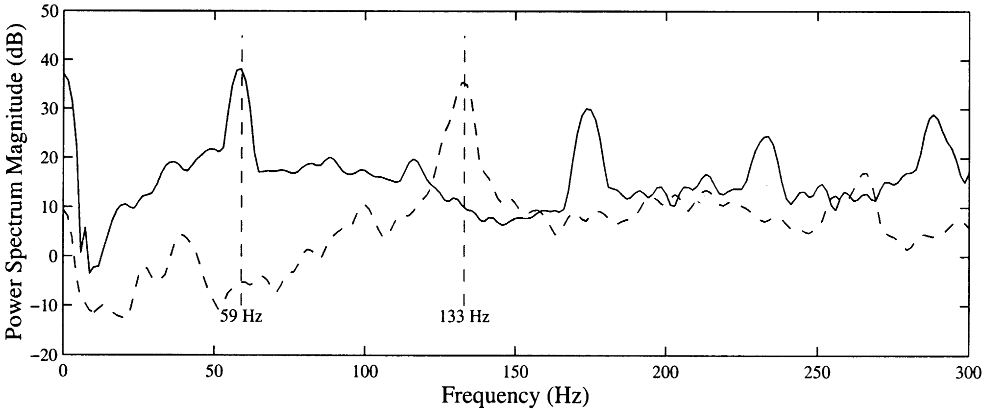

Figure 4. Power spectral density function. Vertical acceleration measured on front (——) and rear calliper (– – –) during creep groan tonal phase. Reference 1 m/s^2. Resolution 1·5 Hz.

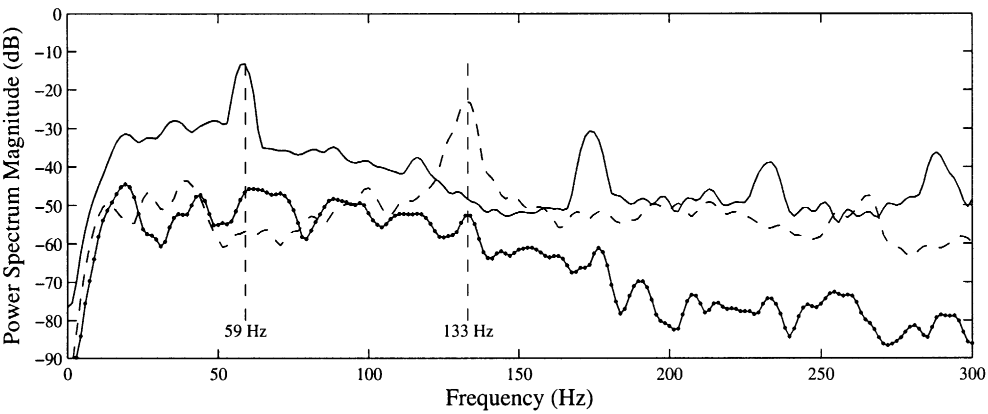

Figure 5. Power spectral density function. Velocity signals, tonal phase. Front calliper (——), rear calliper (– – –) and windscreen (●–●–●). Reference 1 m/s. Resolution 1·5 Hz.

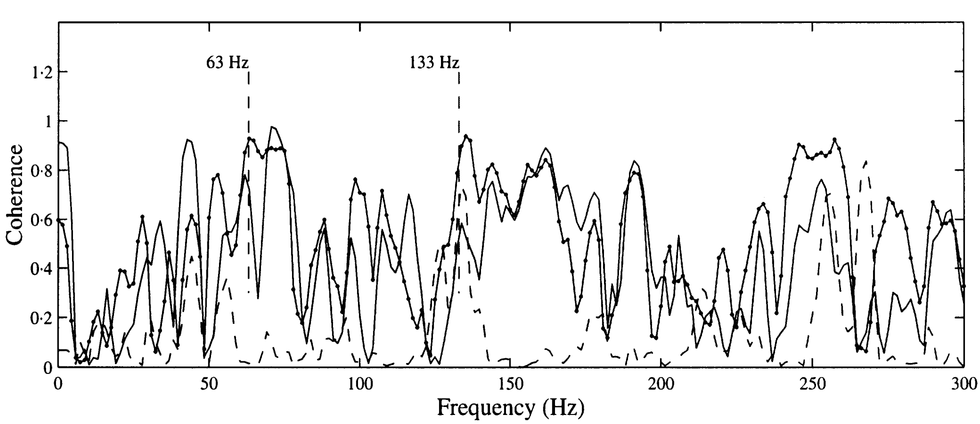

Figure 6. Front calliper (—), rear calliper (– – –) and windscreen (●–●–●) acceleration coherence function. Reference signal (input), recorded noise at driver's ear. Tonal phase. Averages: 18. Resolution 1·5 Hz.

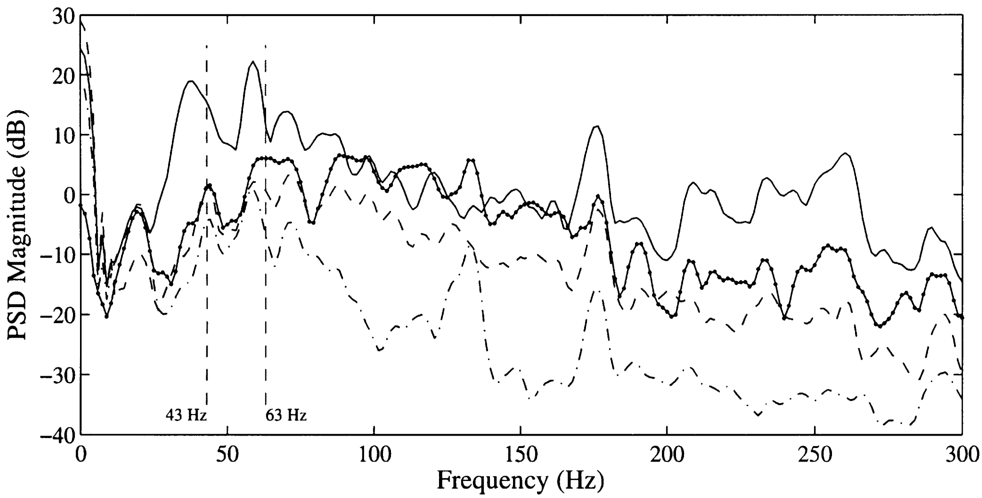

Figure 7. Power spectral density functions. Acceleration measured on front suspension subframe (location I, ——), chassis (J, – – –), windscreen (G, ●–●–●) and B-post (F, – · –). Location letter code in Figure 1. Tonal phase. Reference 1 m/s². Resolution 1·5 Hz.

B-post (F). The plots illustrate the transmission path of the friction-induced vibration. The isolation provided by the compliant subframe is effective, and almost 15 dB loss at 65 Hz is measured between the subframe and the chassis. B-post and windscreen acceleration levels closely match chassis amplitudes. Coherence between windwcreen acceleration and recorded noise is reaching 0·85–0·9 in the region 60–70 Hz (Figure 6).

It therefore appears that reasonably repetitive groan cycles occur independently at each front brake with base frequency varying around 60 Hz. Coherence between left front calliper vertical acceleration and cockpit noise is high in this region, in view of the unmeasured contribution from the right front brake. The latter will also contribute to the cockpit noise and will influence the calculated coherence to an extent that depends on the proximity of the frequency to that of the left brake and the resolution in the FFT processing, 1·5 Hz in this case. Coherence between the windscreen normal acceleration and the cockpit noise is particularly high between 60 and 70 Hz and the windscreen motions are vigorous in this range, showing that it is an important participant in the creep groan process.

2.2. BRAKE GROAN SOUND POWER ON THE VEHICLE

Sound power emitted by the test vehicle due to groan at a single wheel was measured in the free field, averaging the recorded sound pressure on a quarter sphere space delimited by the ground surface and the vehicle side. The sound power can be calculated [8] by using the following equation, where the term 5 dB is the constant referring to the case of junction of two planes (a list of nomenclature is given in Appendix A):

$$L_w = \bar{L}_p + 20\log_{10} r + 5\ \mathrm{dB}(\mathrm{dB}\ re\ 10^{-12}\mathrm{W}). \tag{1}$$

The set of N uniformly spaced measurement points p_i, at a distance r from the source, leads to the estimate of the spatial average:

$$\bar{L}_p = 10\log_{10}\left[\frac{1}{N}\sum_{i=1}^{N} 10^{L_{p_i}/10}\right]. \tag{2}$$

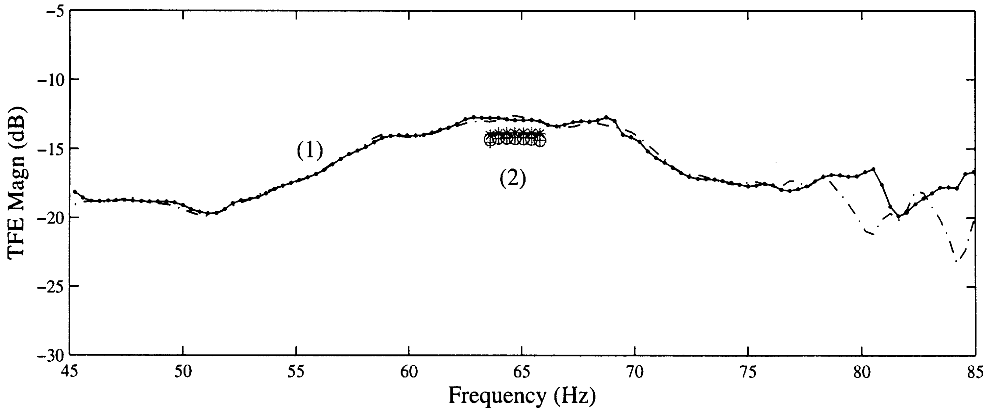

Figure 8. Airborne path loss between microphone in the calliper region and second microphone in the rear cockpit. Excitation given by a loudspeaker under the front wheel arch. (1) Sweeping sine wave, (2) 65 Hz sine wave. Several cases: 93–103 dB(C) *SPL*, measured in the front calliper near field. The case 93 dB(C) *SPL* corresponds to an acoustic power $L_w = 73$ dB, that is referred as the virtual "airborne" groan.

The vehicle front calliper acoustic power was estimated by taking $N = 7$ points on the quarter sphere (2 m radius) and filtering the acquired sound pressure signals with a band-pass 50–75 Hz digital filter, to cut off the engine noise, whose main signatures are 43 and 86 Hz. The estimated sound power value was $L_w = 73\ \mathrm{dB}(re\ 10^{-12}\ \mathrm{W})$.

2.3. WHEEL ARCH TO COCKPIT SOUND TRANSMISSION LOSS

It is necessary to access the relative transmission path properties between front brake callipers and cockpit noise. In principle, airborne and structure-borne paths are both present. The airborne noise path was tested from the calliper region to the passenger cockpit. Microphones were placed in locations B and D (Figure 1). The first one (B) is in the near field of the rear screen, the second one in the near field of the brake disc and calliper.

With respect to the vehicle cockpit, where the far field does not exist, near field sound pressure measurement is the only option. The location of the microphone, near to the rear windscreen (reflective surface), minimizes the cockpit acoustic mode effect. A similar boundary condition is necessary for the microphone in the brake component region. Near field measurements can lead to some inaccuracy, since particle motion close to the source is not directly related to sound pressure. On the other hand, a grid of points (30 cm interval) close to the rear screen gave the same results. Secondly, as will be shown in the next section, the calibratred loudspeaker, which delivers the same sound power as the actual brake groan, generates a sound pressure level in the near field of the calliper, which matches the actual groan case, giving confidence in the findings.

A sheet of metal on the ground, under the wheel arch, was employed to give the reflective surface boundary condition. Source of the generated noise was a calibrated loudspeaker, driven by a function generator, located in the wheel arch. Figure 8 shows the results, at different excitation levels. Cases of a single 65 Hz harmonic and a sweeping frequency (from 50 to 80 Hz) have been compared. A constant 14 dB loss is found around the 65 Hz typical groan frequency band, showing amplitude independence and implying linearity. The 14 dB transmission loss testifies to a reasonably weak airborne transmission.

2.4. SIMULATED "AIRBORNE" GROAN

A relevant particular case of the previously described experiment is when the source at the wheel arch is emitting noise whose properties match the characteristic of the groan. The estimate of the sound power generated by one wheel of the vehicle during groan (section 2.2) allows the reproduction of a simulated airborne groan, consisting in the emission of the same sound power by means of a calibrated loudspeaker, located under the wheel arch. What is perceived in the cockpit, with no mechanical excitation of the structure-borne path, is due to the airborne path only.

The loudspeaker driven by a 65 Hz sinusoidal input was calibrated to deliver an acoustic power of 73 dB. This corresponds to 93 dB(C) sound pressure level measured in the near field, unsurprisingly as obtained during the actual groan vibration cycle. As shown in Figure 8, a 14 dB transmission loss, between the source and the interior (rear screen) in the sound pressure level at 65 Hz has been measured. This value can be identified as the airborne transmission loss of the *simulated airborne* groan, and its result is the noise perceived in the cockpit due to an acoustic source of the equivalent power of the groan, but with no structural effect.

Close to the rear screen, 79 dB(C) *SPL* was detected during the "airborne" simulated groan, using only one loudspeaker. Employing two loudspeakers (front left and front right wheel arches) 82 dB(C) *SPL* would have been measured. No rear brake noise was considered at around 60 Hz, since the accelerations of the rear brake components, during the groan, exclude this possibility (section 2.1 and Figure 4).

The airborne groan 82 dB(C) *SPL* figure compares with the actual vehicle groan level of 102 dB(C) *SPL*, suggesting that the contribution made by the airborne noise path is 20 dB less than the total groan level, and therefore is insignificant.

3. DISTRIBUTED SOURCE EXCITATION VIA LOUDSPEAKERS IN THE COCKPIT

The result of a distributed-source excitation experiment is discussed here. It is proposed as a convenient and particular application of the exact reciprocity principle and the usual methods of noise path analysis, whose limitations and impracticalities, with respect to the creep groan, are described.

3.1. RECIPROCITY

With respect to structure-borne noise, the vehicle structure constitutes a transmission path between the excitation and the response. In the present case, the friction-induced mechanical forcing represents the excitation, while the response is the sound pressure in the passenger cockpit. The reciprocity theorem states that for a vibroacoustical coupled system, the inverse situation, where previous input and output are reversed, is absolutely equivalent [9], as formally written in the equation

$$\left.\frac{p_i}{F_j}\right|_{\dot{q}_i=0} = \left.\frac{-\ddot{x}_j}{\dot{q}_i}\right|_{F_j=0}. \tag{3}$$

The direct case is on the left side, where p_i is the measured pressure response and F_j the excitation force, when no acoustic source is active ($\dot{q}_i = 0$). On the right is the response $\ddot{x}_j$ when the system is driven by a compact monopole source whose volume velocity is denoted by q_i.

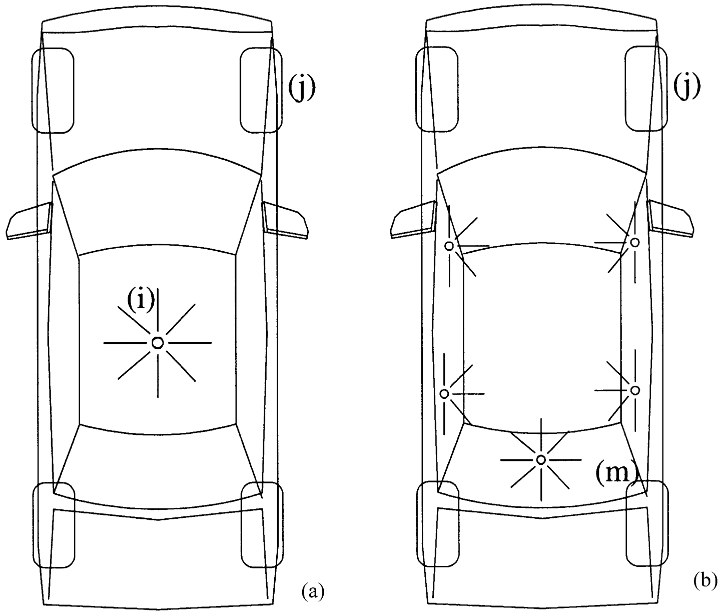

Figure 9. (a) Pure reciprocity experiment as defined in equation (3). (b) Distributed-source excitation experiment. *i* Driver's ear position, *j* brake calliper, *m* distributed-source experiment microphone location.

In the creep groan context (Figure 9(a)), the targets, denoted *i* and *j* in equation (3), are represented by a location in the passenger cockpit and a point belonging to the brake calliper respectively. It is intuitively obvious that vibrating the brake components using only a compact sound source in the vehicle interior is a difficult task. Rigorous application of the reciprocity principle requires a sound source placed in the cavity being omni-directional and approximately a point source (monopole), as described in reference [10]. High sound output levels at low frequency (65 Hz for the groan), in order to generate sufficient vibration on the calliper imply a large source and results in a device difficult to arrange. Considering the size, measurements in the vehicle cockpit will be affected by its presence.

The rigorous monopole source can be made by using a loudspeaker in a box with a very small hole, compared to the target wavelength. A simpler alternative is to use an approximation of the monopole source that can be called a non-compact single source. For low-frequency investigation, as for the groan problem, the non-compact single source may be provided by a single loudspeaker. It was attempted to excite the calliper by using a commercial loudspeaker (7 in diameter, embedded in a $30 \times 30 \times 16$ cm box), but without success. The pure reciprocity principle as written in equation (3) has been successfully used by Cornish [9] to show the structural path between the cockpit and the engine mounts. Reaching the brake calliper by following the whole path of the subframe, the rubber bushes and suspension arms up to the brake components, requires a much stronger source.

With respect to low-frequency problems, the target device is a powerful non-compact single source that allows an easy evaluation of its volume velocity q_i. One could imagine a 20 cm diameter pipe closed, at one end, by a loudspeaker. A pipe length of a few diameters is sufficient for a good approximation of a plane wave field, that simplifies the measurement of the source volume velocity, using sound intensity techniques. A resonant condition in

Acoustic methods				Vibration path analysis
Less desirable ↓	(1)	**True reciprocity**	(3)	↑ Less desirable
	(2)	**Non-compact single source**	(2)	
	(3)	**Distributed-source**	(1)	

Figure 10. Experiment comparison.

a rigid wall duct is obtained when

$$\frac{4Lf}{c} = n, \quad n = 1, 3, 5, 7, \ldots, \tag{4}$$

L being the pipe length, f the fluctuation frequency and c the speed of sound. Therefore, the pipe length that achieves the amplification, required to deliver the maximum power, matching the resonance condition, results in $L = \frac{1}{4}\, nc/f = \lambda/4$, a quarter of a wavelength ($\lambda = c/f = 5{\cdot}6$ m at 60 Hz), in the most convenient case $n = 1$. This leads to a big device that has to be a variable geometry system as it is required to sweep over a frequency range, and not to work on a single frequency. An alternative is excitation via a distributed-source, and the natural way of performing it, chosen here, is by means of the vehicle hi-fi system (Figure 9(b)).

3.2. CANDIDATE EXPERIMENTS

Figure 10 compares the possible candidate experiments, analysing their advantages and limitations. The true reciprocity directly returns an estimate of the $p_i/F_j|_{\dot{q}_i=0}$ quantity. It follows that a theoretical dynamic brake model, that calculates F_j, allows one to predict the cockpit noise level. The sound pressure p_i, in this case, is a function of F_j and the noise path, which includes acoustic and vibration resonances. The reciprocity method can easily take into account the effect of the cockpit modes, simply varying the location of the monopole sound source q_i. The result is a response function $-\ddot{x}_j/\dot{q}_i|_{F_j=0}$ whose parameter is the position of the noise source. Its main drawbacks are the impracticality of the monopole sound source at low frequency and the weak signal-to-noise ratio. On the opposite side of the picture, there is the distributed-source experiment. It is not rigorous, but its peculiarity is the good signal-to-noise ratio over a wide frequency band.

Usual methods of noise path analysis involve excitation by shaker at the vibration source in the real problem, and measuring directly a noise transfer function, the $H(f) = p_i/F_j$ ratio This is not practical with respect to brake callipers, due to the difficulties in attaching shakers to the brake components and in reproducing an excitation similar to the vibration occurring during the normal braking. The second possibility is the assessment of the force during the actual groan generation, while the vehicle is in motion. It is feasible, for example with force transducers attached to the engine mounts, but again many difficulties arise when the target is represented by a brake calliper.

3.3. SINGLE-INPUT/SINGLE-OUTPUT SYSTEM

In order to better understand the relevance of signal-to-noise ratio, it is useful to consider a single-input/single-output system, as proposed by Halvorsen and Bendat [11] and

$$
\begin{array}{ccccccccc}
 & & n(t) & & & & m(t) & & \\
 & & \downarrow & & & & \downarrow & & \\
x(t) & \longleftarrow & \oplus & \longleftrightarrow & \boxed{\mathrm{H}(f)} & \longrightarrow & \oplus & \longrightarrow & y(t) \\
 & & & u(t) & & v(t) & & &
\end{array}
$$

Figure 11. Single input/single output problem as described by Halvorsen and Bendat [11]. System transfer function $H(f)$. Measured signals, $x(t)$, $y(t)$. True signals, $u(t)$, $v(t)$ and unavoidable noise $n(t)$, $m(t)$.

represented in Figure 11. The measured signals $x(t)$, $y(t)$ are affected by the unavoidable noise $n(t)$, $m(t)$, that introduce disturbances to the true quantities $u(t)$ and $v(t)$. Regarding the creep groan, the method can be applied when the source is the actual brake vibration or, in a reciprocity like equivalent, when the source is sound pressure in the cockpit. The criterion to judge the most suitable layout is the estimated coherence. It indicates whether the experiment takes place correctly, i.e., with an acceptable signal-to-noise ratio. While the groan vibration is taking place $u(t)$ (input) denotes the acceleration of one calliper, $v(t)$ the sound pressure in the cockpit. Obvious principal disturbances $n(t)$ (altering the input) are introduced by the engine and the *other* brake calliper; $m(t)$ is mainly affected by external noise. In the reversed situation, input given by noise excitation inside the cockpit resulting in acceleration measured on the brake calliper, the engine can be switched off and other brake perturbations immediately disappear, as the vehicle is stationary. This is the driving idea behind the selection of the distributed-source noise excitation via the standard vehicle hi-fi system. Distributed-source provides high power that should guarantee high coherence levels, with a stationary vehicle reducing other noise signals.

The transfer functions between a microphone $u(t)$ (passenger cockpit, close to the rear screen, location B) and the acceleration signal output $v(t)$ (recorded on each front calliper, K and K′) were estimated, as illustrated in Figure 12. Despite the method not being rigorous, it was possible to generate sufficient signal-to-noise ratio to obtain high coherence and then reliably measure peaks corresponding to the groaning frequency, in the transfer function magnitude, associated with a phase shift, corresponding to resonance. Consistent behaviour was detected comparing the results obtained from the left and the right front callipers.

3.4. DISTRIBUTED-SOURCE EXCITATION RESULTS

The present experiment requires a very powerful source and sensitive acceleration transducers. Excitation was provided by the standard saloon hi-fi, and a triaxial accelerometer (500 mV/g) was fitted alternatively on the front left and right callipers. The sum of a series of sinusoidal signals with random phases and frequencies, ranging from 50 to 80 Hz, was created with Matlab$^{\mathrm{TM}}$, saved in *wav* format and written on a *CD-audio*, ready to be played on the vehicle hi-fi system. The noise source, being driven by an artificially generated signal, can be designed to contain energy in a large band of frequency, returning conveniently the transfer function across the desired frequency range, during a single test. This is a great advantage in comparison with the variable geometry device, described regarding the pure reciprocity experiment, in section 3.1.

It is relevant to observe the peaks in the magnitude of the transfer function between the recorded noise and calliper lateral acceleration (Figure 12, top) corresponding to frequencies of 59 and 63 Hz. The asymmetry differences can be considered as normal in

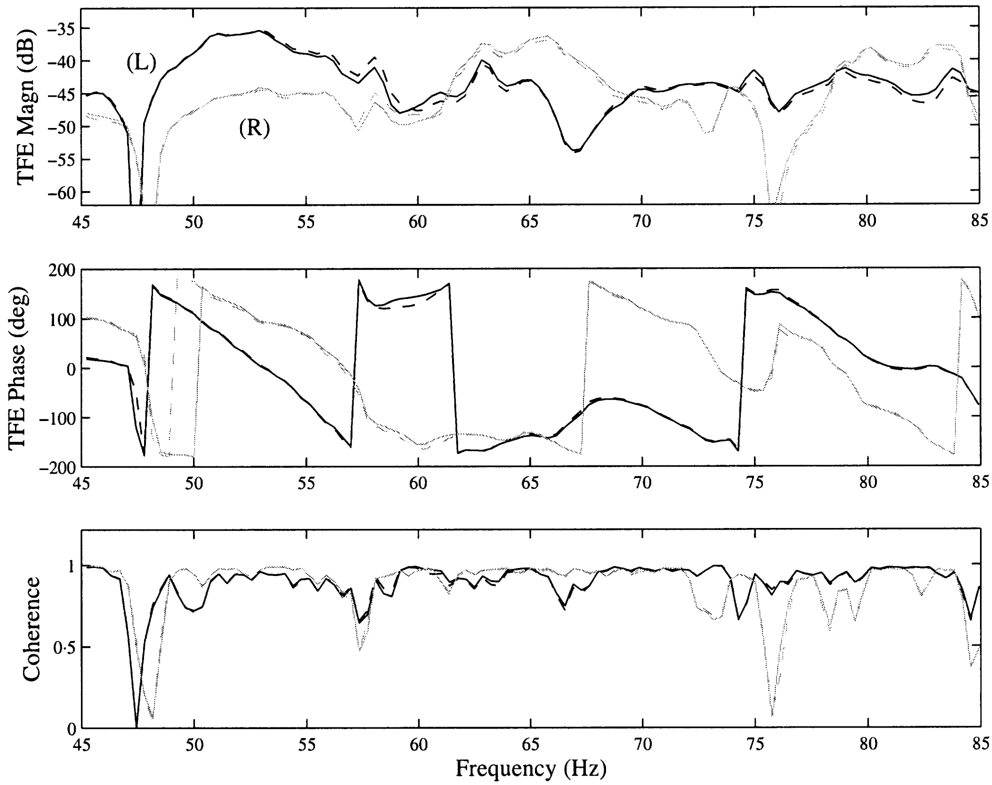

Figure 12. Distributed-source vehicle hi-fi excitation. Transfer function estimate magnitude (top) between recorded noise at rear cockpit location (Pa) taken as input signal and acceleration of the calliper (m/s^2), output signal. Transfer function phase plot (middle), and coherence (bottom, averages: 30). Dark curves refer to front left calliper, light grey to right calliper. Dashed lines denote measurements obtained while operating the brake pedal.

a standard production vehicle. The implications are that calliper resonances close to 60 Hz (the frequencies to which the vibrating cycle tunes during groan) can be excited by means of a distributed-source noise emission from the cockpit.

Figure 12 refers to either a case in which no brake pedal pressure was applied (solid line) or a case where the brake pedal was operated (dashed curves). The differences can hardly be noticed, and the implication is that brake fluid pressure affects the system as a spring preload, having no influence on the frequency response properties. Since the brake pads are always in contact with the brake disc, this is not an unexpected result.

Figure 12 shows evidence of an almost unitary measured coherence function for the current experiment, in which $x(t)$ and $y(t)$ correspond to cockpit sound pressure and calliper acceleration respectively. This states that the signal measured by the acceleration transducer is almost completely due to the sound pressure effect generated by the loudspeakers inside the vehicle passenger cockpit and that the experiment took place correctly, with a sufficient signal-to-noise ratio (which is not intuitive, shaking the calliper by means of hi-fi loudspeakers). The coherence is high over the whole range of frequency, from 50 to 85 Hz. This is apparently in contrast with Figure 6 (regarding front calliper acceleration coherence with the recorded noise), where the coherence plot follows an oscillating pattern, definitely not as regular as that of the distributed-source experiment. The two plots do not disagree, if it is considered that coherence is high where the signal-to-noise ratio is high. In Figure 12 the excitation was provided by the loudspeaker playing the artificial noise, uniformly distributed. While the brake groan constitutes the forcing phenomenon, energy density is concentrated only in correspondence of the particular groan frequency, and only there, it is expected to measure high coherence levels.

Distributed-source excitation can be used to provide a noise transfer function between the brake calliper and the cockpit. Placing the microphone at the rear screen minimizes the influence of cockpit modes on the result. Resonances in the structure-borne paths are seen as peaks in the transfer function. High and uniform coherence gives confidence in the results. The peaks found correspond to the groan frequency, confirming the role of the structural resonance in the groan phenomenon.

4. BRAKE GROAN SOUND POWER ON THE LABORATORY RIG

The measurement of the sound power on the laboratory rig was based on the following algorithm that gives the estimate of the radiated power in a semi-reverberant room [8]; sound pressure is recorded on the surface of two hemispheres whose centre is the noise source and whose surface areas are S_1 (inner hemisphere, close to the source) and S_2 (outer hemisphere):

$$L_w = \bar{L}_{p_2} - 10\log_{10}(S_1^{-1} - S_2^{-1}) - 10\log_{10}(10^{(\bar{L}_{p_1} - \bar{L}_{p_2})/10} - 1). \qquad (5)$$

Sound pressure levels $\bar{L}_{p_1}$ and $\bar{L}_{p_2}$ are again computed by using equation (2), p now denoting p_1 and p_2, respectively, where $\bar{L}_{p_1}$ applies to the points belonging to the inner hemisphere.

The laboratory rig includes the full vehicle front subframe and brake components. It is driven by a geared AC electric motor powered by a flux vector controller. The rig was operated to give the same acceleration levels as in the tested vehicle and the sound power was measured according to equation (5), that applies to semi-reverberant rooms. The number of measurement points was chosen to be 12 on each hemisphere. The two hemispheres selected for the measurements had their centres in correspondence of the brake disc centre and radii of $r_1 = 0{\cdot}8$ m and $r_2 = 1{\cdot}7$ m respectively. This does not satisfy the required condition of a quarter of a wavelength, but the room characteristics did not allow any better layout. The calculated value is $L_w = 54$ dB(*re* 10^{-12} W).

The sound power difference between the car generated noise and the laboratory rig is so large that it cannot be explained by measurement errors due to the incompletely satisfactory environment. The rig is a local approximation of the brake and suspension components only and it is excellent to study the mechanical behaviour close to the details of the source [7], but it is not producing the same noise as the vehicle. In the near field of the calliper a sound pressure level of around 93 dB(C) is measured on the car, and only about 80 dB(C) on the rig.

Since rig vibration component levels (calliper, pad, suspension arms) are reproducing closely those found on the car, the differences have to be searched in the car body and wheel arch interior panels, that are not present in the rig and that show high acceleration levels during the groan process. These are very effective radiators, considering their large surface areas. The implication is that vehicle body panels play a role in the exterior groan noise level.

5. SUMMARY

During the actual vehicle groan generation a sound pressure level of 83–84 dB(C) was measured in the cockpit at the driver's ear, and 102 dB(C), again in the cockpit, but close to

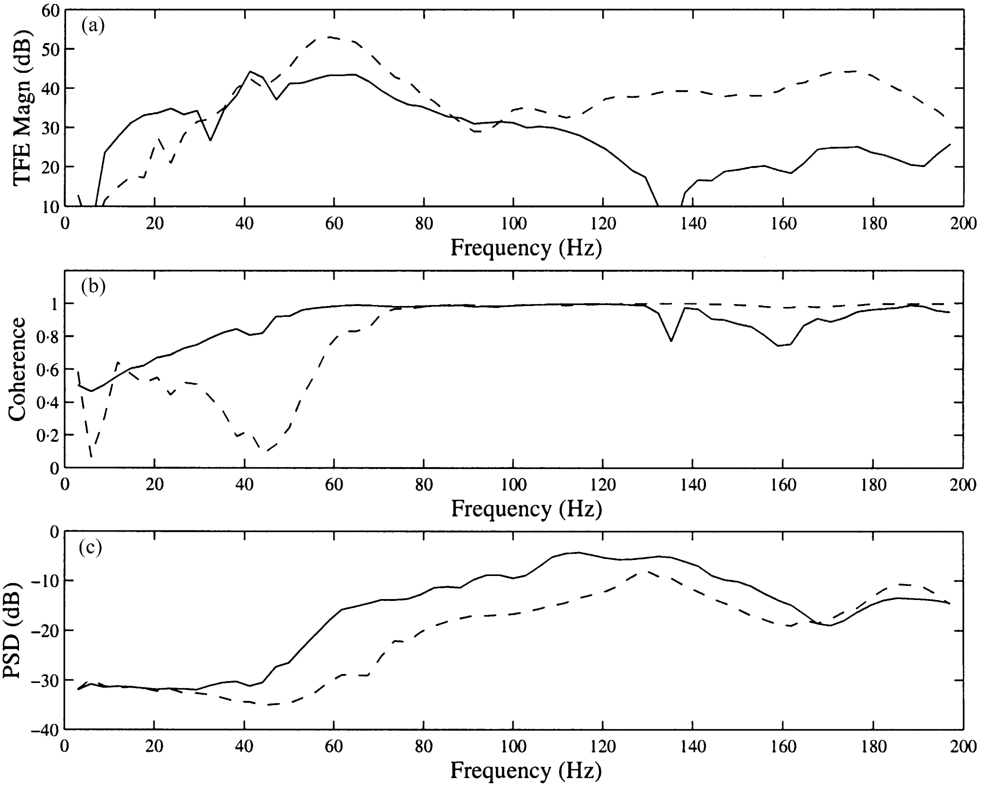

Figure 13. Impulses on front and rear (dashed lines) windscreen. (a) Transfer function estimate between windscreen acceleration (input) and near field recorded sound pressure (response). (b) Coherence evaluated between excitation (screen acceleration) and sound pressure (averages: 8). (c) Acceleration power spectral density function. (front windscreen ——, rear screen - - -).

the rear screen (reflective surface). The peak in the frequency domain corresponds to 63 Hz, and noise is mostly due to the action of the front brake components. The difference of 18 dB *SPL* in two different positions in the passenger cockpit must be due to the action of acoustic modes; only 3–6 dB difference can be due to the reflection of the rear screen.

The transfer function magnitude between the sound pressure recorded by a microphone inside the cockpit and one outside, the airborne transmission path, features higher gain between 62 and 65 Hz (Figure 8). This suggests the presence of a cockpit mode. As already anticipated there is a discrepancy between the first source of the vibration (calliper acceleration, 59 Hz peak) and the recorded noise (63 Hz). This can be explained considering the properties of the excitation source. As mentioned previously, despite the calliper acceleration PSD presents a peak at 59 Hz, it shows a broadband activity. Its PSD magnitude is still remarkably high at 63 Hz. This determines relevant acceleration magnitude of the windscreen at 63 Hz, confirmed by its almost unitary coherence with the recorded cockpit noise (driver's ear, location A).

Impact tests have been performed on front and rear screens in turn. The transfer function magnitudes are plotted (Figure 13(a)) between the acceleration recorded on the glass panel and the sound pressure in the near field for each of the front and rear screens. Mild peaks occur at around 60 Hz. Coherence (b) and the power spectral density function of the acceleration (c) are also displayed. Figure 13(c) suggests that screen natural modes are far from the frequency range of interest, and the fundamental aspect is the higher gain of the transfer function, Figure 13(a), that falls in the band that is relevant for the groan noise.

The presence of the cockpit mode is raising an issue regarding the participation that the windscreen exhibits during creep groan. In section 2.1, high coherence between normal windscreen acceleration and recorded noise was illustrated. This only means that they are correlated, thus providing no information on which one is the first cause and which is the response. A closer observation to the PSD diagram of the windscreen acceleration (Figure 7) shows that it is characterized by broad band activity from 57 to 66 Hz. This is remarkably similar to the acceleration PSDs recorded on other components, such as the calliper, the subframe and the chassis. The result is compatible with a scenario that involves vibrations generated at the callipers, reaching the windscreen and then producing noise in the passenger cockpit.

To the contrary, in the case the windscreen vibrations were produced by pressure fluctuations in the cockpit due to its cavity mode, the spectrum of the windscreen acceleration would have presented a narrow peak at the acoustic cavity frequency of 60 Hz.

The sound power emitted by the vehicle during the groan has been estimated at $L_w = 73$ dB (*re* 10^{-12} W). Subsequently, a loudspeaker in the wheel arch calibrated to emit the same sound power was adopted to generate what can be defined as the "airborne" groan. This resulted in 93 dB(C) in the calliper near field, as for the actual vehicle groan, but in 79 dB(C) only (82 dB(C) if two loudspeakers would have been employed) in the passenger cockpit, close to the rear screen. This confirms the weakness of the airborne sound transmission path.

6. CONCLUSION

An experimental survey on a test vehicle, by using acceleration transducers and microphones, suggested that the front callipers are the main source of the creep groan vibration. The measured sound power significantly differs between the vehicle and the laboratory rig. The rig includes the whole vehicle suspension and brake components, but not the body panels. Their vibration plays a relevant role in the exterior noise emission on the actual vehicle.

Sound transmission loss trials between front wheel arch and cockpit pointed to the weakness of the airborne noise path. This was confirmed by study of the so-called "airborne" groan (same acoustic power as the actual groan, but emitted by a loudspeaker, no structural vibration associated). Its measured sound pressure level in the rear cockpit is 20 dB(C) below that of the vehicle brake groan.

The standard vehicle hi-fi system performing a devised *CD audio* was employed as a convenient experimental method for studying the transmission path between brake calliper and cavity noise. The experiment is less rigorous than the acoustic reciprocity principle application (which requires an extremely powerful monopole source), but is simple and applicable in a situation where the standard noise path analysis methods are not practical (shaker attached to brake components).

With respect to structure-borne noise, the hi-fi excitation highlights easily any relevant system resonance between the sound pressure in the cockpit and calliper vibration, as a peak in the transfer function. The magnitude of the transfer function between noise in the cockpit (input) and component acceleration (output) allows the comparison of different design solutions. High coherence (showing sufficient signal-to-noise ratio) is required to validate the experimental results. Distributed-source excitation circumvents the limitations of the reciprocity method, which requires a high power monopole source, inconvenient in size for low-frequency problems. The application discussed here is a significant example, since the achievement of brake component excitation by means of the cockpit loudspeakers is not intuitively likely.

The present experiments imply that the creep groan phenomenon is a structure-borne vibration. This agrees with the subjective feeling that the noise outside the vehicle is somewhat different from the groan vibration perceived while seated in the passenger cockpit. Body panels have been shown to be important radiators of the sound to the outside of the car.

For the case treated, creep groan occurs at both the front and rear but the acoustic problem in the cockpit mainly derived from the front brakes. The important differences appear to lie in the brake calliper mount structural details, which determine the characteristic frequency of the vibrations and the structural path to the cockpit. In particular, it is desirable to detune the creep groan vibrations from the acoustic resonances of the cockpit and to minimize the transmissibility of the structural path, in order to minimize the quality loss through creep groan.

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REFERENCES

1. A. LEE and D. C. SHERIDAN 1997 *ISATA Magazine*, 8–10, January 1997. Diagnosis of noise and vibration problems during braking.
2. T. MIBE 1988 *JSAE paper* 881020. Friction characteristic analyses of brake-pad material at low speeds.
3. N. V. NACK and A. M. JOSHI 1995 *SAE paper* 951095. Friction induced vibration: brake moan.
4. M. T. BENGISU and A. AKAY 1994 *Journal of Sound and Vibration* **171**, 557–570. Stability of friction-induced vibrations in multi-degree-of-freedom systems.
5. M. K. ABDELHAMID 1995 *SAE paper* 951282. Creep groan of disc brakes.
6. M. BETTELLA and R. S. SHARP 2001 *ATA paper* 01*A*1009. Multi-body approach to creep groan vibration modelling.
7. M. BETTELLA 2002 *Ph.D. Thesis*, *Cranfield University*. The mechanics of the Automotive Brake Groan, in printing.
8. D. V. BIES and C. H. HANSEN 1996 *Engineering noise control—Theory and practice*. London: E&FN Spon, 2nd edition.
9. R. CORNISH 2000 *Journal of Automobile Engineering* **214**, 685–693. A novel approach to optimizing and stabilizing interior noise quality in vehicles.
10. P. J. G. VAN DER LINDEN and J. K. FUN 1993 *SAE paper* 931340. Using mechanical–acoustic reciprocity for diagnosis of structure borne sound in vehicles.
11. W. G. HALVORSEN and J. S. BENDAT 1975 *Sound and Vibration* **9**(8), 15–24. Noise source identification using coherent output power spectra.

APPENDIX A: NOMENCLATURE

Symbol	Description	Units
c	sound velocity in air	340 m/s
f	frequency	Hz

F_j	excitation force at point j	N
L_p, L_{p_i}	sound pressure level (SPL)	dB (*re* 20×10^{-6} Pa)
$\bar{L}_p$, $\bar{L}_{p_i}$	*SPL*, spatial average	dB (*re* 20×10^{-6} Pa)
L_w	acoustic power level	dB (*re* 10^{-12} W)
p, p_i	sound pressure	Pa
q_i	monopole sound source volume velocity	m^3/s
r, r_1, r_2	radii	m
L	length	m
N	number of measurement points	—
S_1, S_2	hemisphere surface	m^2
$\ddot{x}_j$	acceleration at point j	m/s
λ	wavelength $\lambda = c/f$	m