Identification and reduction of sound sources in car wheel suspensions

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Abstract

The identification and reduction of sound sources in car wheel suspensions is one of today's major issues in the Automotive industry. The last five years a lot of effort has been spent identifying shock absorber noise. These investigations have now been extended towards complete suspensions. Due to its strong non-linear characteristic, the shock absorber generates high frequency vibrations (100-900 Hz). First, the relationship between the structure borne noise and the damper behaviour needs to be defined by means of a vehicle evaluation. Next the components that need to be modified are identified by comparing time-domain simulations and measurements. To deal with full suspension noise source identification and reduction some specific time-frequency techniques are applied. Additionally some new techniques such as the 3D coherence function are proposed. Because of the transient character of suspension components, these techniques can give some insight into operating mechanisms and reduction potential of car wheel suspensions.

1. Introduction

The ride control division of Tenneco Automotive manufactures shock absorbers under the brand name Monroe. In the past few years, it has developed analytical and test-based methods in order to improve the NVH behavior of its products.

However, even if noise sources, internal to the shock absorber have been identified, analyzed and the noise level improved by using the developed methodology, in a lot of cases, the structural performance of the shock absorber can not be dissociated from the structural characteristic of the vehicle suspension and vehicle body for which it is developed.

Furthermore, because of the difficulty localizing noise sources, the shock absorber is sometime wrongly accused of being the cause of a given noise problem.

Finally, the role of suppliers for the automotive industry has evolved from component to system supplier and requires a full understanding of all parameters that influence suspension performance, even when the component itself is only a small fraction of a larger puzzle.

That is why Tenneco Automotive has extended the focus of its NVH research to the vehicle suspension.

This paper is divided into two parts. The first part describes the methods that are applied for the study of the noise generated at shock absorber level. The second part considers the suspension in which the shock absorber is a structural component.

2 Noise at shock absorber level

2.1 The shock absorber as a source of structure borne noise

The classical telescopic shock absorber may have different configurations, but the basic principle remains identical. The motion of the shock absorber rod forces oil to flow from one chamber to another through valve assemblies. This creates differential pressures over the piston and a damping force results.

These valve assemblies consist of a set of orifices, discs and springs tuned together in order to obtain the desired characteristics. The flow-pressure characteristic is chosen to achieve the best compromise between comfort, ride and handling.

The distinction between structure borne and air borne noise is sometimes difficult to define. The most convenient definition is that air borne noise is clearly audible when the shock absorber is excited on a test rig. The vehicle itself may shield this noise or on the contrary amplify it.

In the case of structure borne noise, the shock absorber produces high frequent forces (100-900 Hz) that are not or hardly audible at the shock absorber level. The specific vehicle structure will determine the actual noise level.

The frequency region to which both types of noise apply can be clearly distinguished in most cases.

Structure born noise may have different causes: play between components, force lag due to improper balance between valve assemblies, or impact of travel stops or hydraulical transitions. The first two causes are only due to a deficient design and/or manufacturing process. However transitions are intrinsic to every damper due to the opening and closing of oil paths, resulting in high frequent hydraulical perturbations, i.e. high frequent forces. The hydraulical noise due to transitions is generally known as chuckle noise.

2.2 Methodology (figurel)

The methodology is developed in two complementary stages (figure 1).

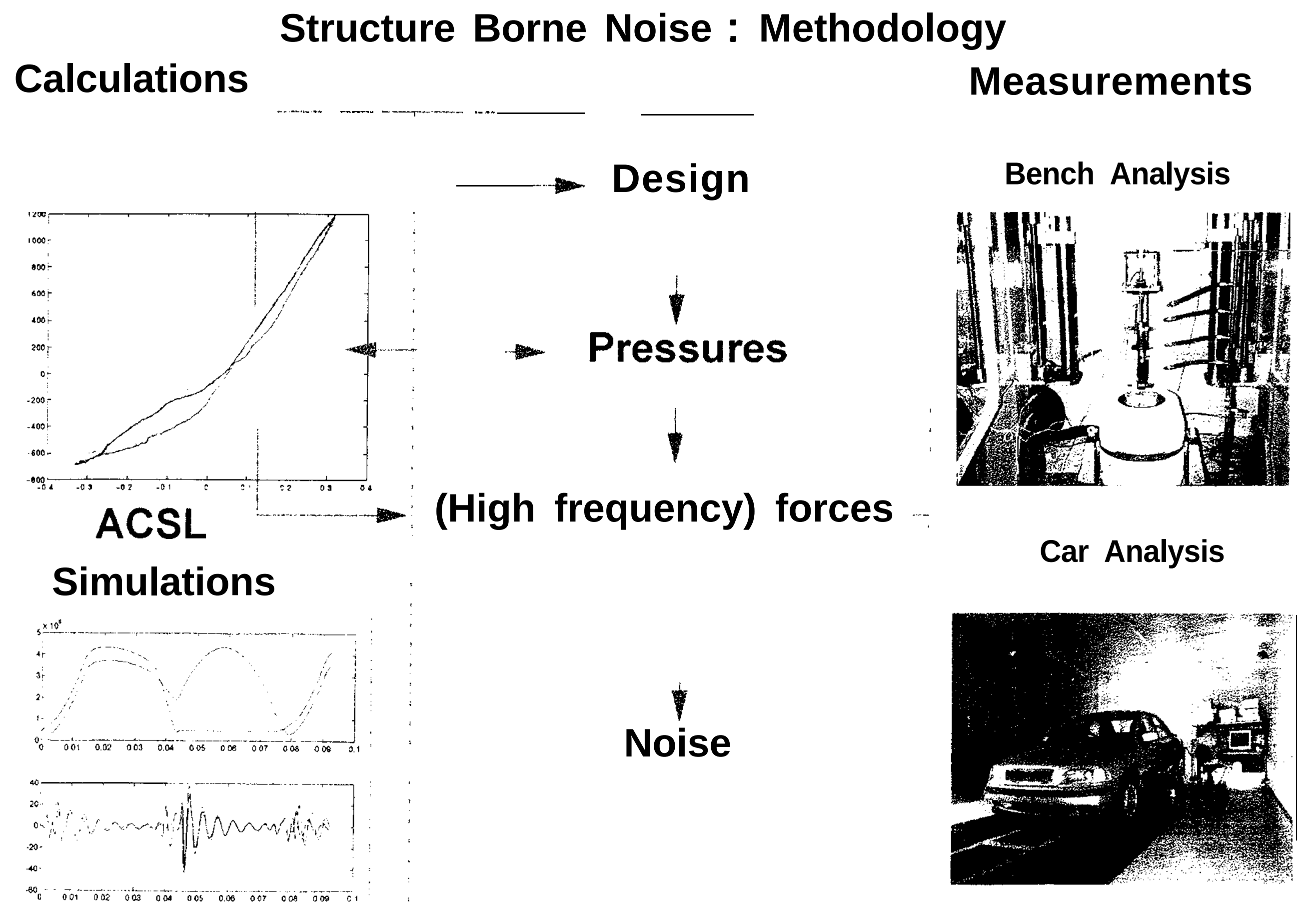

figure 1
Methodology for the identification and reduction of structure borne noise

In the first and the final analysis, noise is evaluated on the vehicle. In order to focus on possible causes, a study is then engaged by means of simulation and measurement at shock absorber level.

2.2.1 **Evaluation at component level**

A better understanding can be achieved by simulating the internal behavior of the shock absorber. Most models developed in the past were dedicated to the low frequent behavior [0-30 Hz] and were intentionally built in a way that allowed vehicle manufacturers to use such models in their vehicle simulation pertaining to ride & handling. Those models are descriptive in the sense that they can predict the ride & handling behavior of a given shock absorber when integrated in a suspension; after that a few key parameters are identified with an easy to use identification algorithm [1].

In order to simulate the high frequency behavior of the shock absorber, a more detailed model was needed. This model needs each component described by a set of algebraic and differential equations which in turn, is solved numerically.

This model, written in ACSL, can not only reproduce the high frequency characteristic of the shock absorber but is also well suited to understand which factors influence it and in which direction the design needs to be modified [2].

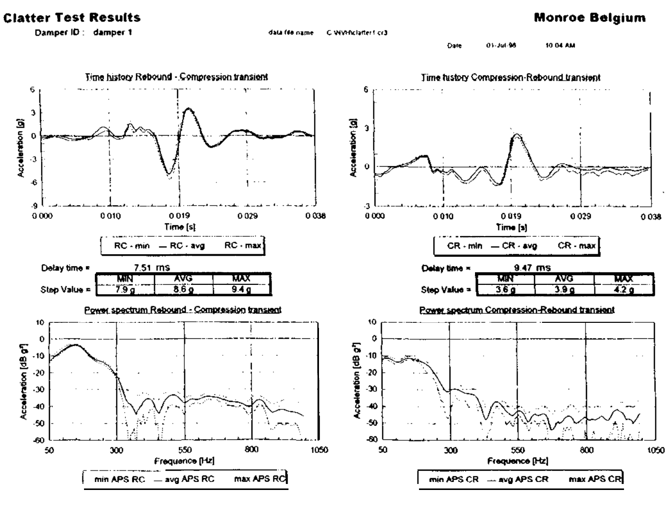

figure 2
Clatter test : time history and power spectrum of the shock absorber acceleration for harmonic excitation

The basic test used to validate the model is the “clatter test”. The shock absorber is mounted with its top and bottom mounting. It is submitted to a monochromatic displacement and the rod acceleration is recorded in the time domain and processed in the frequency domain (figure 2).

Though the comparison test vs. simulation is suited to identify the causes of high frequency forces and to see how the shock absorber can be improved, it fails to address the question whether the change will diminish the noise in the passenger compartment.

2.2.2 **Evaluation at vehicle level**

The god to avoid any structure borne noise due to the shock absorber in the vehicle is an impossible mission due to the fact that hydraulical high frequent forces produced by the shock absorber are not to be dissociated from the working principle. Therefore at some point, even if Tenneco Automotive improves its shock absorbers by better designs that reduce those perturbations without modifying the low frequent behavior, a trade-off between different design alternatives exists. The supplier’s know-how consists in choosing the design that is best suited for a given application.

It is also not obvious to dispose over the vehicle structural characteristic at the earliest stage of suspension development. Therefore a vehicle evaluation of the NVH shock performance remains an essential part of the design procedure.

Vehicle evaluation is a difficult task, not only because of the subjective process of the human perception, but also because of the transient character of the hydraulic noise. The first problem is addressed by performing a full vehicle evaluation in a semi-anechoic chamber. The wheel is excited in the vertical direction with a road signal chosen to generate chuckle noise. Not only the relationship between sound pressure level and high frequent rod forces can be analyzed, but more importantly, the effect of design changes can be evaluated in the same conditions.

The transient character of high frequent rod force is a supplementary difficulty. The time analysis is essential to identify which type of transient does cause noise, and hence pinpoint the components which need to be modified when detailing with a shock absorber analysis. However an objective evaluation is based on power spectrum densities of the sound pressure level and acceleration signals. Therefore a time-frequency analysis gives a more complete view of the process.

Vehicle evaluation, even in that condition, remains a time consuming process. Therefore, when the relationship between sound pressure level and rod acceleration has been established, and when it is defined which level in which frequency bands correspond to a critical level, a bench test at the shock absorber level can be established. It is similar to the previously mentioned clatter test (figure3). The aim of this test is not to identify the high frequency force sources anymore, but to evaluate their contribution in the total sound pressure level. The excitation signal is also equal to the one the shock absorber is submitted to in the vehicle.

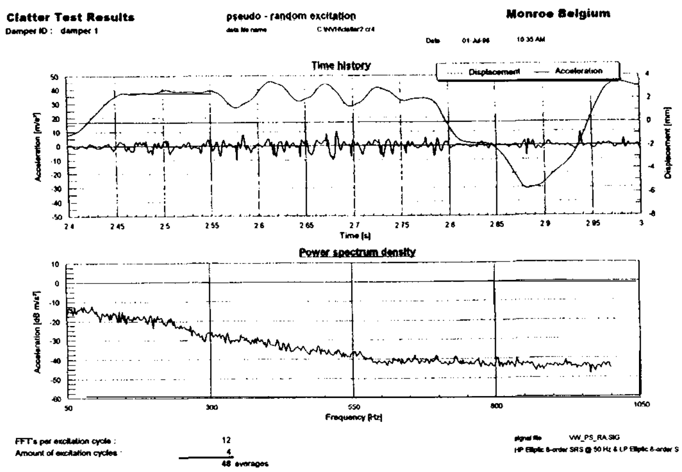

figure 3
Clatter test : time history and power spectral density of the shock absorber acceleration for pseudo-random excitation

3. Noise generated at suspension level investigated by means of time-frequency analysis

In order to identify noise sources and noise propagation paths in the vehicle wheel suspension, the special characteristics of the involved elements and their operating mechanisms are to be taken into account. As discussed above in detail, the shock absorber and many other noise generating phenomena act as transient noise sources, the operation of which is based on highly non-linear phenomena. Apart from non-linear sound generators, there are non-linear sound transmitting elements in the suspension too, namely those resilient bushings which connect the various supporting elements (suspension arms, shock absorber/coil spring assembly etc.) to the chassis. As a result, traditional source and/or path identification techniques are not really apt to give meaningful results without special precautions.

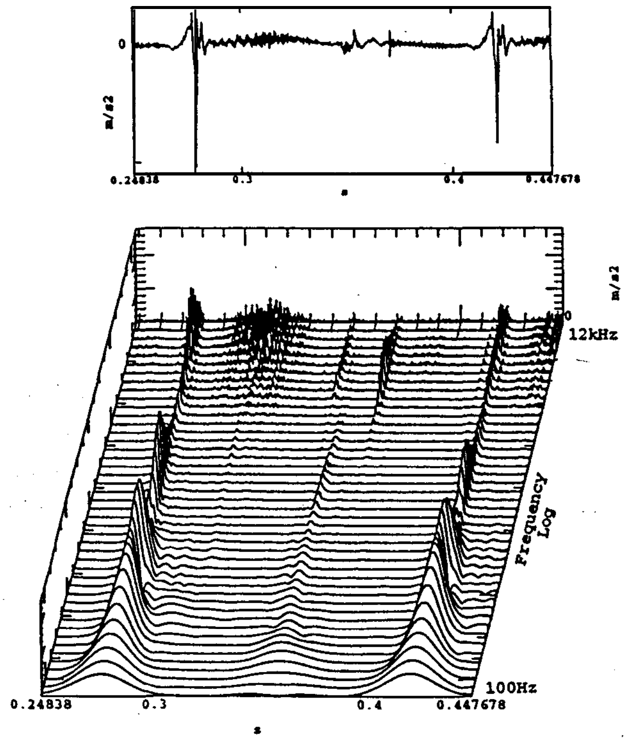

figure 4a-b
Time history (upper part) and wavelet transform (lower part) of the acceleration signal, measured on the shock absorber top mount for sinusoidal wheel displacement

Fig. 4a. shows the time history of a typical shock absorber acceleration signal, measured on the top mount of the rear left shock absorber in a car model under laboratory conditions. Note that the wheel of the car was excited by sinusoidal displacement signal. (The applied test rig and the operating conditions are described in detail in [3].) The time scale of the figure covers about one and a half period of the excitation signal, in which strong peaks and random components rather than true sinusoidal variation can be observed. One can expect that the spectral content of the excitation does significantly vary with time too. In order to describe the behavior of such noise sources in detail, the use of some sort of time-frequency analysis technique is essential. In the following sections the application of a few time-frequency analysis techniques are discussed, with special regard to the possibilities of source and propagation path identification in vehicle wheel suspensions.

3.1 Source identification by means of time-frequency analysis

Time-frequency analysis techniques are plentiful, and are being used for objective and subjective evaluation of sound signals for quite some time (a good evaluation can be found in [6 and 7]). Unfortunately, there is a clear trade-off between the accuracy of identification in the time and frequency domain: the better the identification in time, the worse in the frequency domain and vice versa. [8]. Therefore, the most appropriate option has been selected on the basis of a comparative study, evaluated from the transient noise source analysis' point of view. Four analysis techniques were compared:

- the short time Fourier transform method (STFT);
- the Wigner-Ville distribution (WVD);
- the Wigner distribution (WD; as a matter of fact, the WD is a special case of the Wigner-Ville distribution with no smoothing applied);
- the wavelet transform (WT).

As can be expected, the WD and WVD techniques result in components (the so called cross terms) which were not present in the original signal. Dealing with strong non-linearities anyway, this non-linear effect would be rather disturbing for the analysis in concern, and hence unacceptable to our purpose. The STFT method was found to give well balanced but rather poor time and frequency resolution. Considering that we are more interested in time-domain than frequency-domain discrimination of the measured signal components, the WT was selected as the most appropriate technique.

As an example, Fig. 4b. shows the wavelet transform of the time history of Fig. 4a. (The applied transform is the Morlet wavelet corresponding to a constant relative bandwidth, or more specifically, a third-octave band filter set, which enables improving time resolution towards higher frequencies.) One can clearly distinguish between the impulses separating the compression and rebound phases of the shock absorber, and the appearance of a random high frequency component caused by cavitation is obvious too.

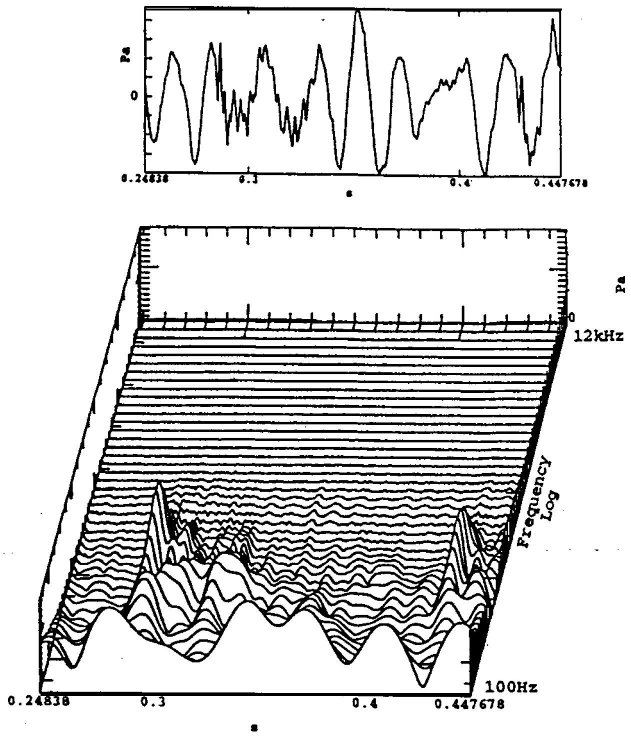

figure 5a-b
Time history (upper part) and wavelet transform (lower part) of the sound pressure, measured in the car interior for the same sinusoidal wheel displacement as in Fig. 4

Just as it is the case in traditional frequency analysis, the identification of noise sources in terms of time-frequency analysis functions consists in comparing the analysis pattern of the observed output signal to the pattern of the suspected noise source(s). Fig. 5a. shows the time history of the sound pressure as measured in the car interior for the same excitation signal as shown in Fig. 4a. Apart from some localized random behavior, the microphone signal is rather regular and a direct relationship with the presumed source is highly doubtful at first sight. Nevertheless, the comparison of the two wavelets in the lower frequency range clearly reveals that the same frequency components appear at the same time (disregarding a systematic time delay of 7 ms). Note that no agreement can be found for higher frequencies, simply because very low signal amplitudes can only be measured in the microphone signal.

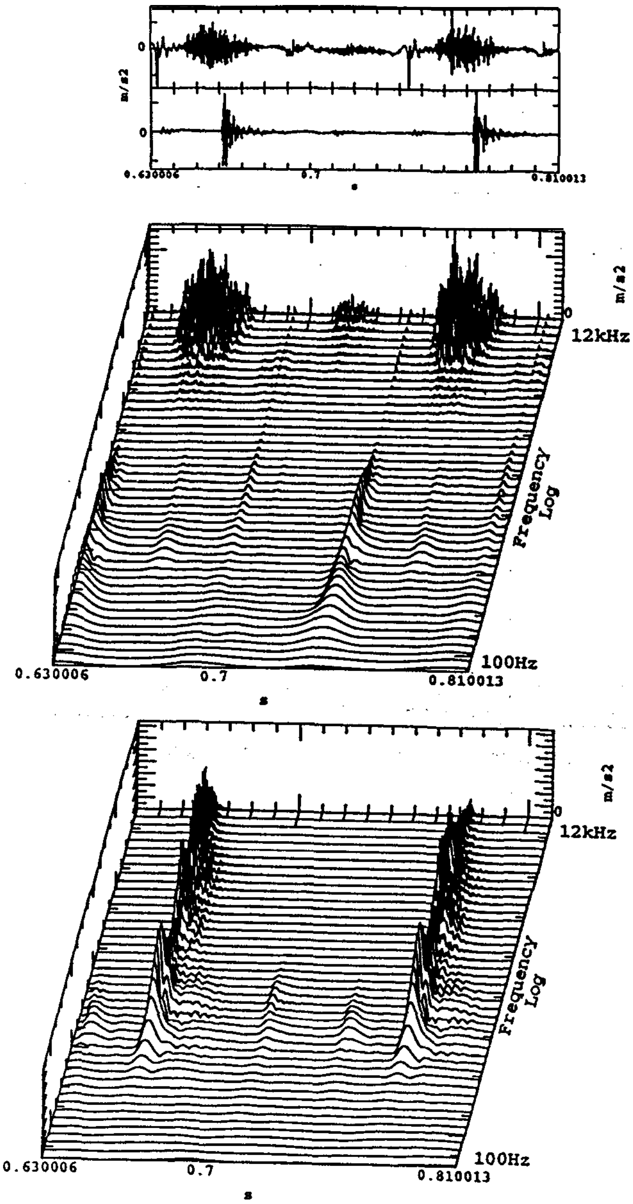

figure 6a-c
Time histories (upper part) and wavelet transforms (middle and lower part) of acceleration signals, measured on the shock absorber top mount and on the stabilizer rod for sinusoidal wheel displacement.

The usefulness of the technique is even more apparent if more than one noise sources are to be identified. We have produced a simple but realistic test case by removing the rubber bushing from the joint of the stabilizer cross rod, thereby creating a secondary noise source in the wheel suspension. The measured time histories are depicted in Fig. 6a. and the corresponding wavelet transforms in Fig. 6b. and 6c. The obtained wavelet transforms show characteristic differences between the two sources.

Though less obvious, the time-frequency pattern of the two sources can also be found back in the measured microphone signal, Fig. 7.

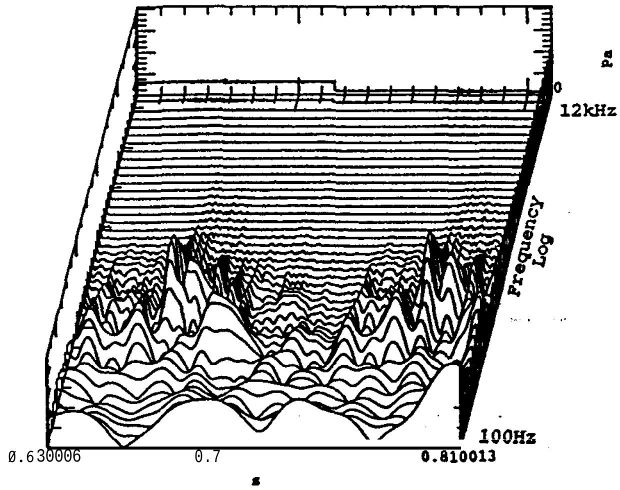

figure 7
Wavelet transform of the sound pressure for the same excitation as in Fig. 6.

3.2. Investigation of noise transmission by means of time-frequency analysis

Once the potential noise sources have been identified – or often a *priori known* -, *the* real issue for the designer is how to reduce their effect, how to modify the construction of the suspension. The problem can only be solved in an effective way if the contribution of the various propagation paths are quantified and the dominant path can be pinpointed. One way to do that is to measure transfer functions between the suspected inputs (e.g. acceleration or force) and the observation point output (most often sound pressure). Once again, the transient nature of the sources and the non-linear propagation paths require special treatment.
As seen above, the frequency content of typical vehicle suspension noise sources varies strongly with time. Not only the accuracy of transfer function estimation but the transfer function itself can be dependent on the applied excitation, due to non-linear elements in the transmission. Instead of measuring the transfer function of the suspension by means of continuous (measured or simulated) road signal, a much more clear picture of noise propagation phenomena can be obtained if the measurements are extended to cover both time and frequency dependence.

The applied technique is based on short time auto- and cross spectrum measurements, by using appropriate windowing. In the course of laboratory tests the measurements are recorded in pure time signal format. (In principle, any excitation signal is appropriate which can bring about those source effects we are going to investigate. Nevertheless, the easiest test signal was found to be a pure sinusoid.) In the subsequent analysis both the input and output signals are multiplied by a short time window which is long enough to comprise the full transient but not too long with respect to the operating cycle, in order to enable good time separation between various effects. The auto- and cross spectra are calculated, then the windows are shifted along the whole time record and the procedure is repeated and averaged for a couple of operating cycles to get reasonably smooth results. As a result, a 3D representation of the FRF and coherence functions vs. both frequency and time are obtained.

Fig. 8a. depicts the coherence function between the acceleration signals measured on two sides of a bushing of a suspension arm and depicted as a contour plot vs. both time and frequency. The coherence is close to maximum in the whole frequency range for the approximate time of the strong valve impulses (denoted by tl and t2), but rather low in between. This means that an FRF estimation makes sense for these high coherence signal parts only. Fig. 8b. shows these two FRFs, together with the corresponding coherence functions in the usual frequency representation. Even though FRF(t1) is a smoother function and there is a peak in FRF(t2) caused by bad coherence, there is no essential difference between the two FRFs. This in turn suggests that the investigated mechanical part, while showing some traces of non-linearity, operates essentially as a time invariant transmission element.

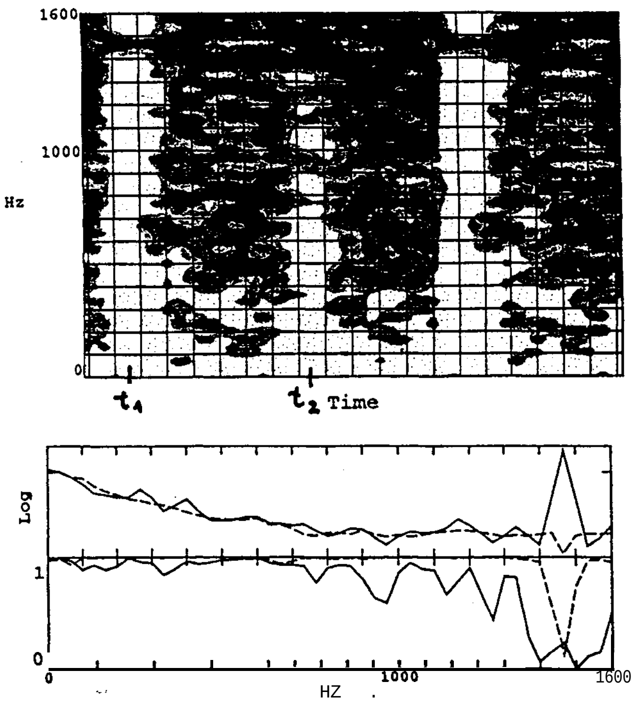

figure 8a-b
Upper part: 3D coherence plot vs. time (horizontal axle) and frequency (vertical axle) between acceleration signals measured on two sides of a suspension arm bushing. Lower part: measured mechanical FRFs and ordinary coherence functions for the denoted times t 1 and t2

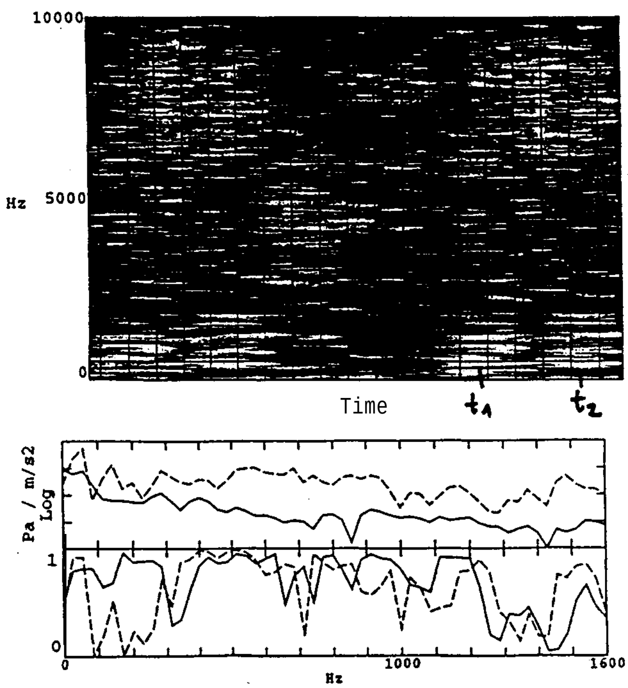

figure 9a-b
Upper part: 3D coherence plot vs. time (horizontal axle) and frequency (vertical axle) of the sound pressure, referenced to the shock absorber top mount acceleration signal. Lower part: measured vibro-acoustical FRFs and ordinary coherence functions for the denoted times t 1 and t2

Fig. 9. in turn refers to a full vibro-acoustic transmission system, excited by two, in principle independent sources as discussed above. The 3D coherence plot (Fig. 9a) between the shock absorber top mount acceleration and the microphone signal reveals many subtle details of the relationships between the signals and the operating mechanisms represented by them. As we have already seen in Fig. 6, the two sources excite partly different frequency ranges at different times. High frequency cavitation components are to be identified for t2 only, but up to 1600 Hz (see also the lower part of Fig. 9b.) the coherence functions are somewhat similar. Evaluating the transfer function measurements in this frequency range for different times tl and t2, one can observe rather different FRFs. The reason for this difference can be interpreted in different ways: it can be caused by time-variant behavior of the shock absorber mount, but the same effect can be caused by source contamination too. Other investigations have revealed that this is indeed the case: the stabilizer source also appears in the top mount signal as a weak but clear reference component and results in good coherence and a seemingly high transfer function.

4 Conclusion

It has become clear that a shock absorber nvh characteristic needs to be designed application by application. For most vehicles, generic design improvements are sufficient to avoid any shock absorber noise. Other vehicles require a more detailed analysis. The shock absorber internal design is then tuned for the particular application. This process requires vehicle evaluation. This has the disadvantage that such procedure occurs in a later phase of the development process. That is why Tenneco Automotive is also involved in the development of analytical predictive tools at the suspension level [3].

As to the measurement methodologies, the transient nature of typical shock absorber and vehicle suspension signals require special signal processing techniques. Well established time-frequency evaluation techniques (e.g. wavelet transform methods) have been applied and some new techniques (such as the 3D coherence function) have been developed for noise source and propagation path identification. One can expect that these methods, possibly combined with standard methods such as partial coherence analysis give a good insight into the operating mechanisms and noise reduction potential of vehicle wheel suspensions.

Acknowledgements

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