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OBSERVATIONS OF THE STRUCTURAL ACOUSTICS OF AUTOMOBILES

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ABSTRACT
Three structural acoustic databases are discussed. The first is an illustration of the variability of the structural acoustics of nominally identical automobiles due to naturally occurring environmental conditions and to normal manufacturing conditions. The second is a database that illustrates the range of tire/vehicle transfer characteristics that occur in various models of automobiles. The third is a two-dimensional measurement of the vibration pattern of a tire that shows the wave behavior of the tire and suggests a relatively simple model of the tire for modeling purposes. The implications of these results to future design philosophy and better interior noise in automobiles is discussed.

1 - INTRODUCTION
The structural acoustics of automobiles are an important element of automobile noise generation and transmission characteristics. From personal observation, the relative importance of noise sources inside new vehicles have changed in recent years and much of this is due to strategies designed to make automobile structures stronger and lighter to improve fuel mileage. Thus, some noise sources that have not been important in the past are now noticeable. As these trends continue it will become increasingly more important to develop a better understanding of the structural acoustics of automobiles. In this paper, several measurements of automobile structural acoustics will be described with the intention of drawing some perspective about how automobile systems behave and where structural acoustic characteristics may play a role in the future.
It is expected that the observations made for automobiles apply as well to the structural acoustics of other transportation vehicles such as aircraft, trucks, and trains, although the frequency range of applicability will be shifted higher or lower due to the relative stiffness and mass of the vehicle.

2 - VARIATION OF AUTOMOBILE CHARACTERISTICS
A very important element of understanding automobile structural acoustics is the sensitivity of the behavior to typical variations. Most prominent are the sensitivity and the resulting variations that occur due to normal environmental conditions. But also, since typical design models are assumed to be robust for the entire manufactured ensemble of vehicles, it is important to understand the variations that occur under typical conditions of normal manufacture and use. Several large data sets have been collected which are believed to be representative of these variations.

2.1 - ENVIRONMENTAL SENSITIVITY
Tests were done on two large samples of Isuzu vehicles, 57 Isuzu pickup trucks and 98 Isuzu Rodeos (a sport utility vehicle), at the Subaru-Isuzu Automotive automobile assembly plant close to Purdue University. Details of the tests are described by Kompella and Bernhard [1]. During these tests, one reference vehicle of each model was chosen and tested repeatedly throughout the test period. Most conditions of the test were carefully controlled except for weather conditions. Thus, variations in the measured response of the reference vehicle are primarily due to the effects of the environment, particularly temperature and humidity.
The data shown in this paper will be for impulse hammer excitation of the steel wheel of the pickup truck near the spindle. The data was high pass filtered at 50 Hz. A frequency response function (FRF)

between the sound pressure level at the driver's ear location and the excitation force was calculated. The ensemble of all of the data for the reference vehicle is shown in Figure 1. The variation of the transfer function data is as great as 4 dB near 100 Hz and 10 dB near 500 Hz. Although it is difficult to follow a single frequency response function, close inspection shows that the FRFs shift fairly uniformly across the frequency band. Thus, an automobile becomes more or less noisy as climatic conditions change.

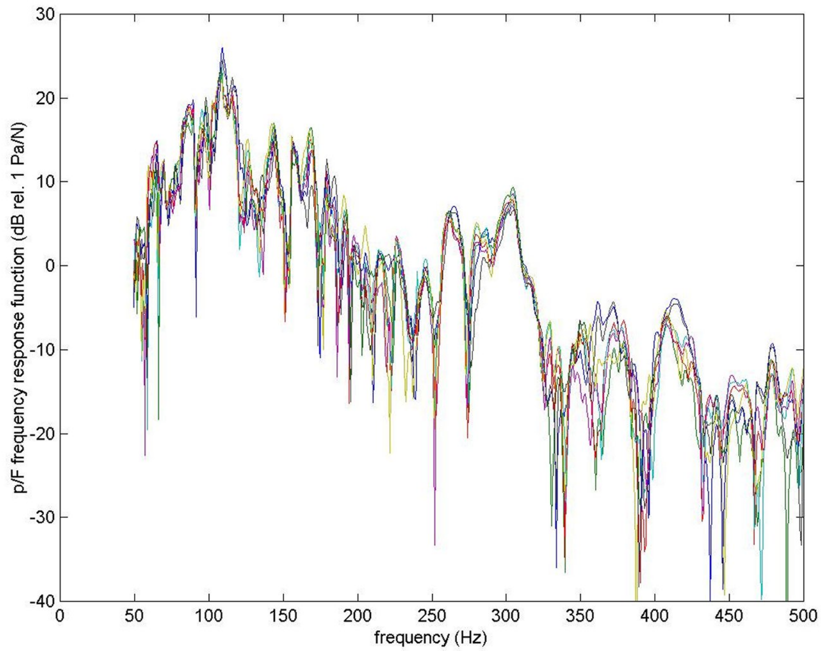

Figure 1: Seven structural acoustic frequency response function measurements of the Isuzu pickup trucks.

These same FRF data were reduced to one-third-octave band format. The same seven structural acoustic FRFs are shown in one-third-octave band format in Figure 2. In this case it is easier to follow a single FRF measurement. The responses are generally uniformly higher or lower across the entire frequency band. The variation is approximately 2 dB near 1000 Hz and 5 dB near 500 Hz.

The calculated standard deviation of the FRFs for narrow band and one-third octave band formats for the reference pickup truck are shown in Figure 3. Except where there are low levels of response in the narrow band data, which cause poor estimates of the FRF and high standard deviations, the standard deviations of the narrow band and one-third octave band results are similar. The frequency averaging associated with computing the frequency response in one-third-octave band format does not change the variation. The variation of this vehicle due to environmental change is a systematic change with temperature.

2.2 - MANUFACTURING VARIABILITY

Fifty-seven nominally identical Isuzu pickup trucks were tested during the same test period using the same test method. The results for all 57 vehicles for the FRF between the force excitation at the wheel/spindle and the sound pressure response at the driver's ear location are shown in Figure 4. The variation in these measurements is quite high. Typically in the lower frequency part of this frequency band, resonant responses are apparent even in the ensemble of all of the data. Variation is approximately 10 dB. At higher frequencies there is substantial mode shifting between vehicles. The ensemble of all measurements shows no distinctive modal responses although each vehicle has strong resonant behavior. Variations across the entire ensemble at high frequency are higher than 20 dB at many frequencies.

The same data are shown in Figure 5 in one-third-octave band format. When the primary cause of narrow band variations is due to mode shifting, frequency averaging across the frequency band where the modes shift will reduce the variation. The variations for the data in one-third-octave band format are significantly reduced. The variations range from 5 dB near 100 Hz to 8 dB near 500 Hz.

A comparison of the standard deviations of the narrow band data and the one-third-octave band data is

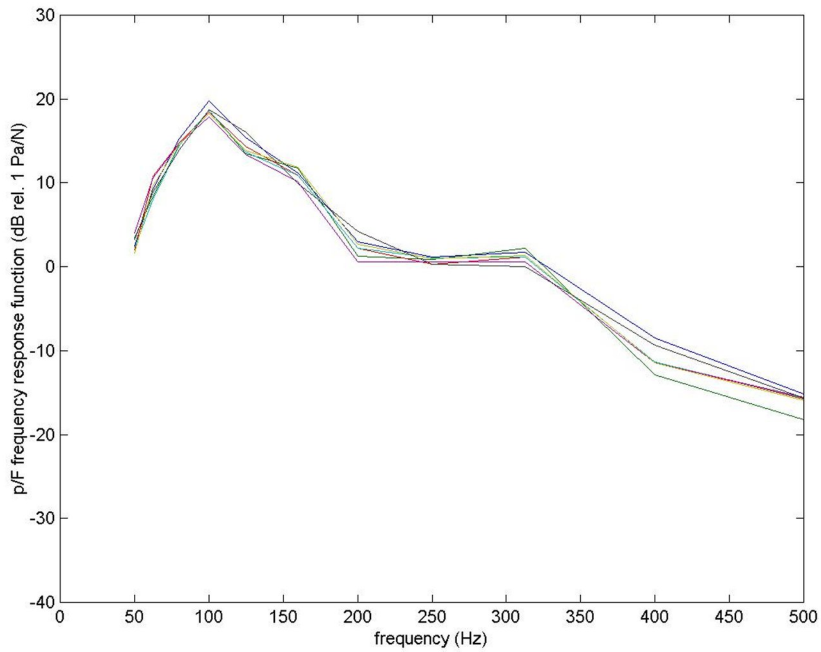

Figure 2: Seven structural acoustic frequency response function measurements of the Isuzu pickup trucks in one-third-octave band format.

shown in Figure 6. The variation of the FRFs in one-third-octave band format is significantly less than the variation of the narrow band measurements.

These data show that manufacturing causes greater variations than environmental conditions. Also, the character of these variations is different at low frequency than at high frequency. At low frequency, modes are apparent even in the ensemble average. Thus, a modal avoidance strategy is reasonable in this frequency range. At high frequency, the ensemble averages are smooth. Ensemble noise reductions will require a strategy that reduces the modal response in general. The results illustrated here are believed to be typical of the sensitivity of automobiles to manufacturing variations. Also, the variations shown here are expected to be smaller than would be realized under less controlled conditions typical of applications.

2.3 - DIFFERENT MODEL VARIABILITY

A complementary series of tests has been conducted recently. In this case, different vehicle models have been tested to investigate the variation that occurs between different models. In this case, structural acoustic transfer functions were measured between impulse hammer excitation of the tire and the resultant a-weighted sound pressure at the right and left ear of a binaural head/torso system located in the passenger seat of the vehicle. Excitation of the tire was done in both the fore-aft direction at the center of the tire and vertically through a hole in the tire support.

The results in one-third-octave band format for four of the vehicles for excitation in the vertical direction at the right front tire are shown in Figure 7. Both the right and left ear frequency response functions are shown. Similar results for horizontal excitation are shown in Figure 8.

The difference between the vehicles is significant, almost 10 dB near 100 Hz and as much as 20 dB near 500 Hz. The magnitude of these differences is somewhat surprising, particularly in light of the fact that all four of these vehicles are considered to be high quality automobiles. Road tests are being conducted to determine if the differences in the measured FRFs correlate with differences in measured interior noise spectra when tire/pavement interaction noise is dominant.

Significant differences were also measured between the left and right ear, particularly at high frequency. These data illustrate the variations that occur spatially in the automobile cabin, even for relatively small changes of location.

Another interesting issue is the importance of the horizontal excitation to interior noise. In an unpublished experimental study with data collected on a chassis dynamometer, it was found that the fore-aft dynamic loads at the spindle in the 100-500 Hz frequency range are as high as the vertical dynamic loads at the spindle. These excitation forces are believed to be due to non- vertical force excitation at the

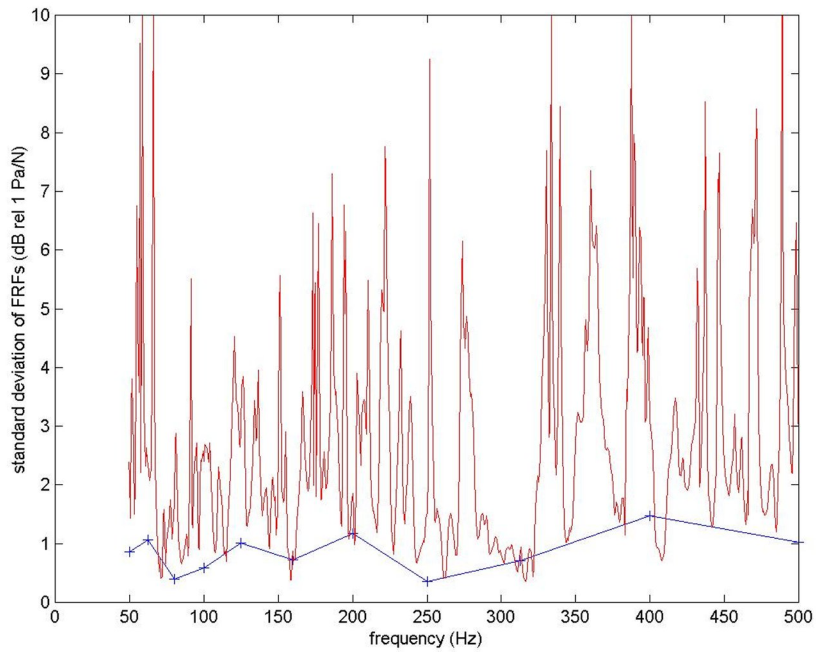

Figure 3: Comparison of the standard deviation of the structural acoustic frequency response functions in narrow band format and one-third-octave band format.

contact patch and inertial loads due to inhomogeneities in the tire. Thus the FRF due to horizontal excitation could be as important to noise generation in the passenger compartment as the vertical FRF. The vertical and horizontal FRF's for the premium minivan are shown in Figure 9. In this case the horizontal FRF is 10 dB higher than the vertical FRF for much of the frequency band from 20-500 Hz. If fore-aft excitation is significant, this path will provide significant transmission of tire/road noise to the passenger compartment.

2.4 - CONCLUSIONS

The implications of the variations that were found in these measurements to the design process are significant. A series of issues arise:

- What is an appropriate performance metric in light of the variations that occur? Should the results of test or prediction be frequency averaged? How do the responses correlate with customer perceptions? Should different metrics be used for different excitations (e.g., broadband road noise or harmonic engine noise)
- How can we predict the performance of the vehicles against these metrics in light of the variations we know occur? Kompella and Bernhard have suggested a probabilistic approach [2]. However, significantly more information about the variation of automobiles and the design objectives are required to implement this process.
- Is there such a thing as a typical car under typical conditions? In light of the type of variations found in these studies, performance evaluations should always be done on a sample of vehicles to understand the type of variation that will occur in that vehicle model under different conditions.

Resolution of these issues is critical to improving our design process and our vehicles for structure-borne noise, particularly from the tire.

3 - TIRE VIBRATION

For some time, investigators of tire vibrations have described tire vibration in terms of modal response. This is an effective description at lower frequencies where the effects of damping are small and standing wave patterns are evident. However, as frequency increases, the effect of damping is sufficient enough that propagating waves in the tire are dissipated and the wave interference effect that results in resonant

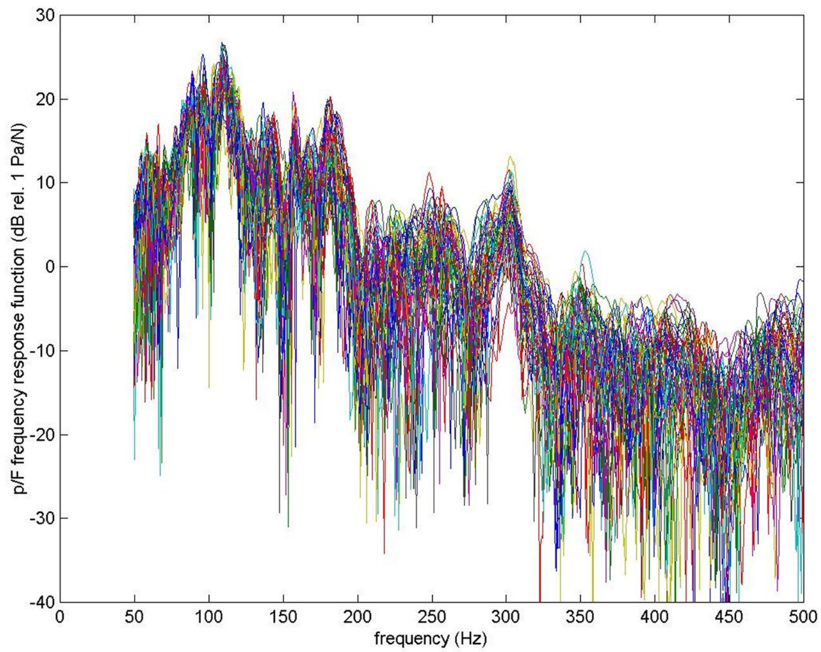

Figure 4: Structural acoustic frequency response function measurements of the 57 nominally identical Isuzu pickup trucks.

behavior is no longer apparent. Thus, a modal description is no longer a good characterization of the vibration of the tire. In addition, the number of modes in the tire is large and difficult to utilize at high frequency. Bolton et al. illustrated the experimental evaluation of tires using a waveguide characterization [3]. In this format, the vibration of tires is described by propagating waves in the circumferential direction and modes in the meridian direction. This type of description is well adapted to heavily damped systems with one short dimension and one long dimension.

Mathematically, the waveguide model is characterized in terms of wavenumber, the radian frequency in space. Typically these measurements are made on the tire for point excitation and transformed into an array of frequency response functions associated with an array of response measurement locations. A spatial Fourier transform of the FRF data is done on the array of responses at each frequency. The spatial Fourier components will indicate which waves are propagating in the tire at that frequency.

A typical circumferential wavenumber decomposition of a stationary tire is shown in Figure 10. At 100 Hz, one meridian "mode" cuts on. As frequency increases the circumferential wavenumber of this mode increases (circumferential wavelength decreases). At approximately 300 Hz another meridian mode cuts on. As frequency increases, this mode merges with the first mode, indicating these modes belong to the same family of waves. In this case both modes are flexural modes. At 500 Hz and 700 Hz, additional flexural modes associated with higher order meridian modes cut on. At approximately 500 Hz, the first meridian mode of another family of modes is cut on. These modes all merge onto another mode family. These modes are in-plane modes and have small circumferential wavenumber. These are fast modes and capable of efficient sound radiation.

In a recent complementary measurement, a two-dimensional array of vibration responses was measured on a treadless tire. The marked tire is shown in Figure 11. The tire was excited using a small shaker. The response was measured using a laser vibrometer on a 2-cm by 2-cm grid of points. The data is displayed by unwrapping the tire as shown in Figure 12.

The results indicate three distinctive frequency ranges. In the low frequency range, damping effects are not significant and modal response is strong. In this range, the tire tread band appears to vibrate primarily as a ring supported by the sidewall as a spring. The two-dimensional, unwrapped vibration pattern of the tire at 109 Hz is shown in Figure 13.

The two-dimensional spatial Fourier transform of the data shown in Figure 13 is shown in Figure 14. These results are very typical of the two-dimensional wavenumber transform of the vibration of a single mode in a rectangular plate. There are four dominant wavenumber components of the vibration pattern

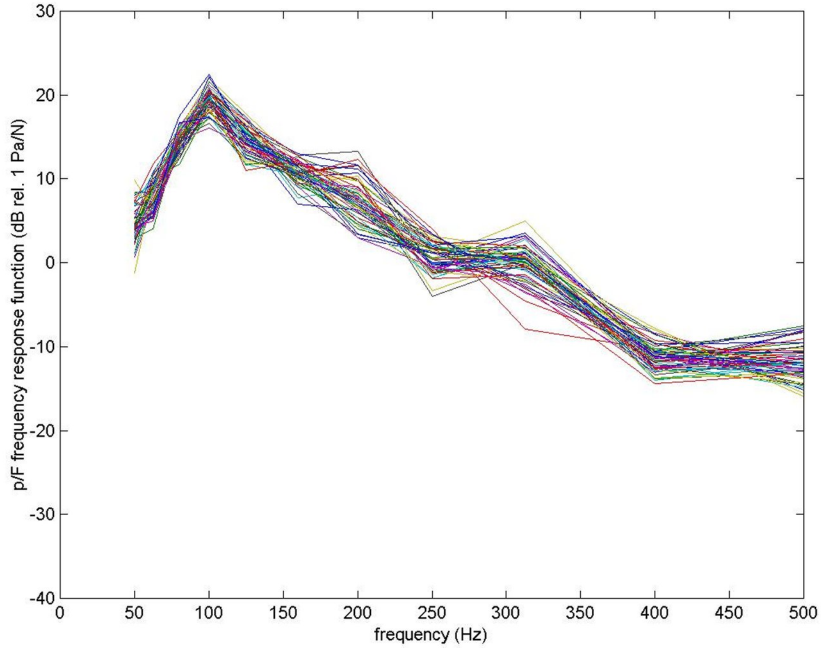

Figure 5: Structural acoustic frequency response function measurements of the 57 nominally identical Isuzu pickup trucks shown in one-third-octave band format.

associated with positive and negative propagating meridian wave components and positive and negative propagating circumferential propagating wave components.

In the mid-frequency range, waveguide behavior is evident. At many frequencies, multiple waves are cut on. Some decay rapidly and others set up standing waves. Above 600 Hz there is evidence of both short wavelength flexural waves, which decay rapidly, and long wavelength in-plane waves, which do not decay significantly. The two-dimensional vibration pattern at 653 Hz is shown in Figure 15. At this frequency, the tread/sidewall interface is apparent. The flexural waves behave almost as cylindrical propagating waves. The in-plane waves set up resonant modes.

The two-dimensional wavenumber transform of the response of the tire at 653 Hz is shown in Figure 16. The outer ellipse of significant wavenumbers is the wavenumber decomposition of the flexural waves. The elliptical nature of these results is due to the orthotropic properties of the tire. Damping causes smearing of the wavenumber components. The inner ellipse of wave number components is due to the in-plane waves.

At high frequency the waves in the tire propagate essentially as cylindrical waves. The damping in the tire dissipates the energy fast enough that no standing waves are created. The frequency response functions for the tire at 1231 Hz are shown in Figure 17. The spatially circular pattern of the waves is dominant.

The two-dimensional wavenumber transform of the frequency response functions at 1231 is shown in Figure 18. These results resemble the wavenumber transform of a cylindrical wave except for the elliptical nature of the results, which are due to the orthotropic properties of the tire.

The tire descriptions available through waveguide models are relatively "compact". It is possible to describe tire vibration with relatively few parameters. This approach seems useful for both understanding the tire system and building appropriate design models.

4 - CONCLUSIONS

The structural acoustics of automobiles are becoming more important due to the competing issues of increased customer sensitivity to interior noise and increased propensity of automobiles to transmit mechanical energy to the interior as noise. The survey measurements reported in this paper indicate there is a long way to go to fully characterize the systems we are working with and understand what performance target(s) we want to achieve. Successful resolution of these issues will require a significant experimental effort and a focus on resolution of the tough issues of customer perception and performance

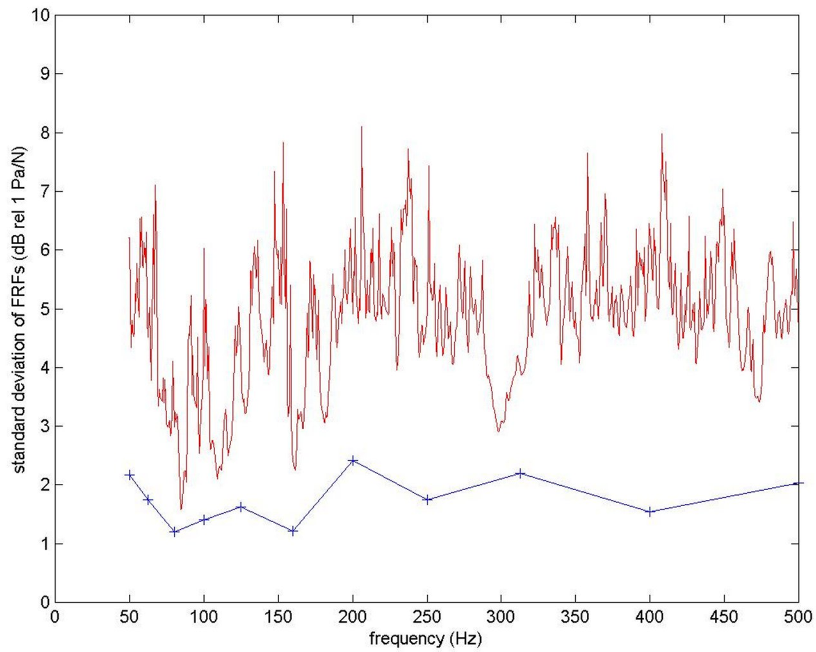

Figure 6: Comparison of the standard deviation of the structural acoustic FRFs of the 57 nominally identical Isuzu pickup trucks for narrow band and one-third-octave band format.

target setting. However, as the tire experiments indicate, there are new tools that can be utilized to address these problems in an efficient manner.

ACKNOWLEDGEMENTS

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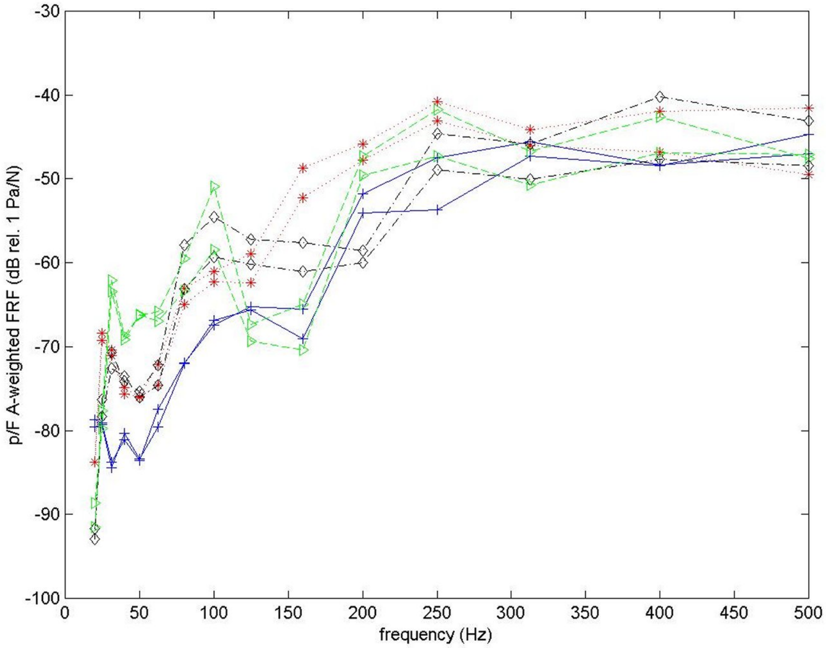

Figure 7: Comparison of the frequency response function between a-weighted sound pressure at the right and left ear locations in the passenger seat location and force excitation of the right front tire in the vertical direction; + premium sport utility vehicle, * premium minivan, ◇ premium mid-size sedan, and > mid-size sedan.

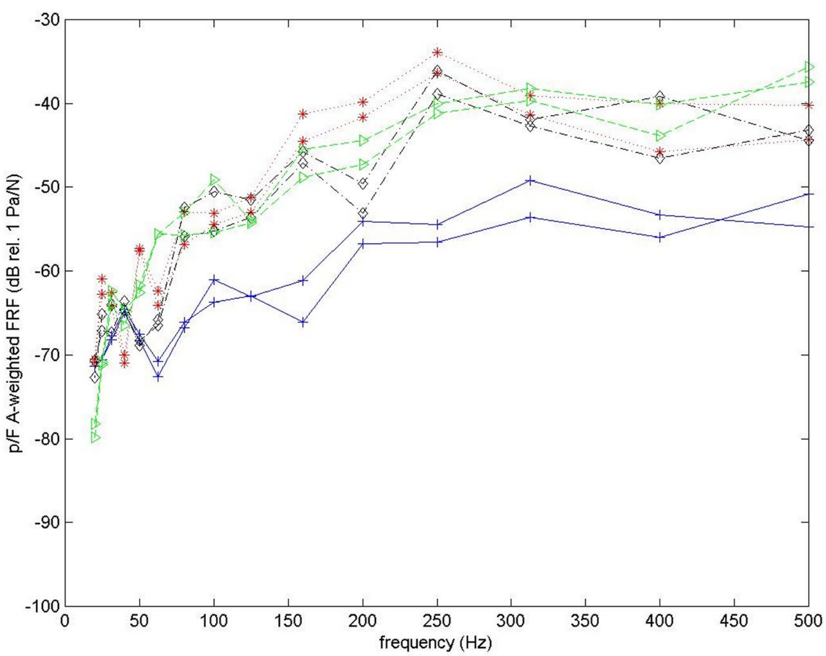

Figure 8: Comparison of the frequency response function between a-weighted sound pressure at the right and left ear locations in the passenger seat location and force excitation of the right front tire in the horizontal direction; + premium sport utility vehicle, * premium minivan, ◇ premium mid-size sedan, and > mid-size sedan.

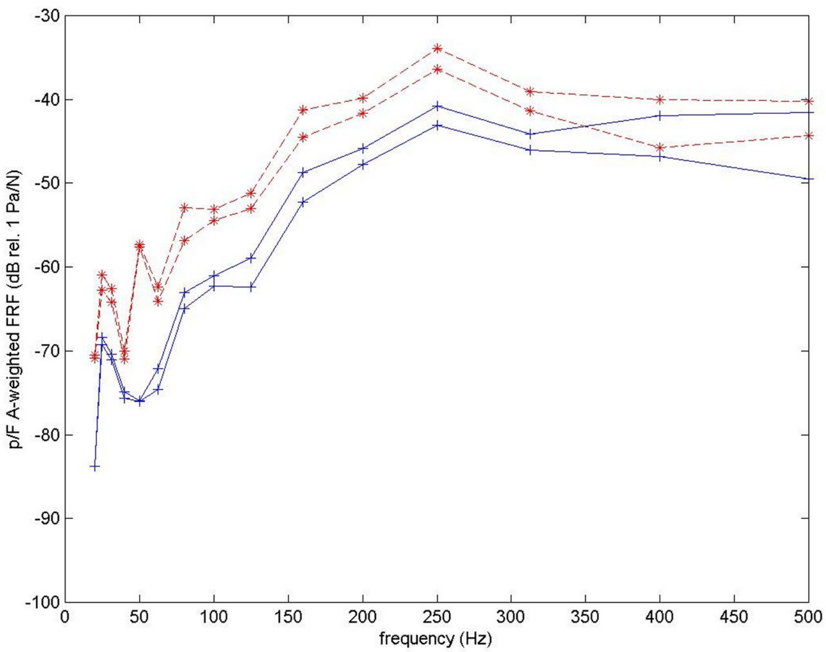

Figure 9: Comparison of the frequency response function between a-weighted sound pressure at the right and left ear locations in the passenger seat location and force excitation of the right front tire in both the vertical and horizontal direction for the premium minivan; + vertical excitation, * horizontal excitation.

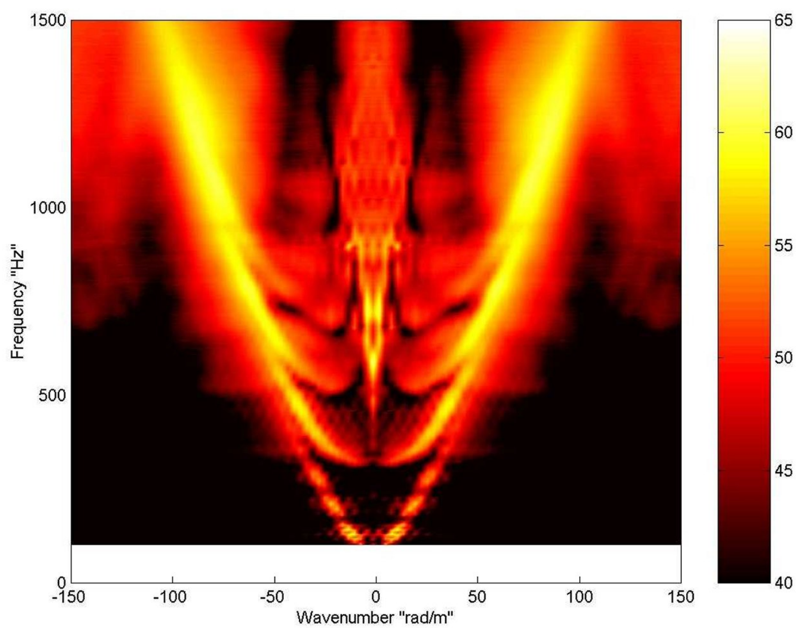

Figure 10: Wavenumber decomposition of the frequency response functions of vibration response along the circumferential direction of a tire.

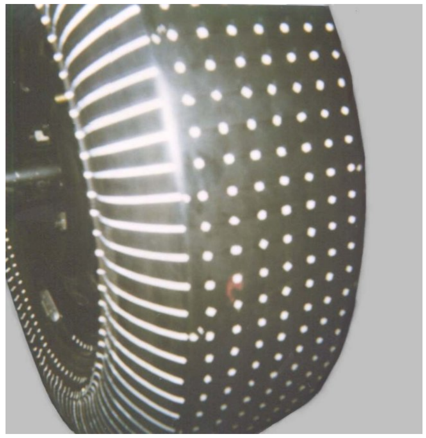

Figure 11: Treadless tire marked for vibration measurement on a two-dimensional array at 2-cm intervals.

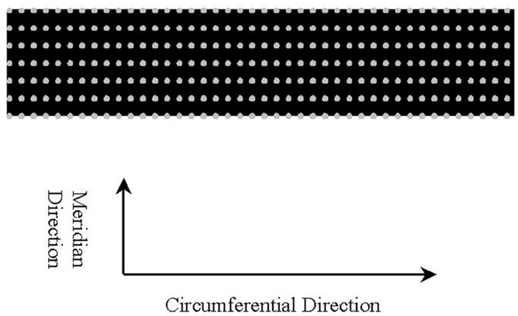

Figure 12: Tire measurement locations unwrapped for display on a two-dimensional plane.

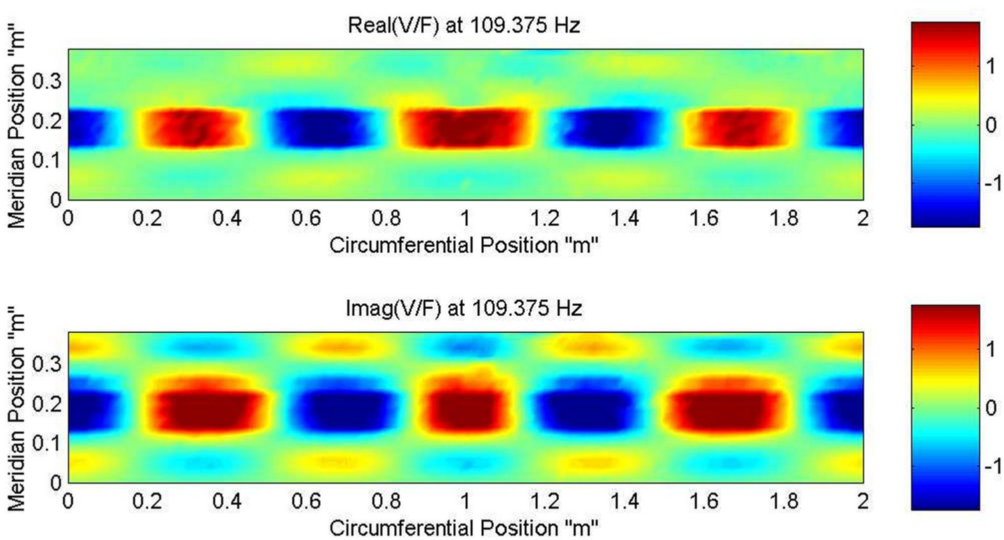

Figure 13: Two-dimensional display of the real and imaginary parts of the frequency response functions for the tire at 109 Hz.

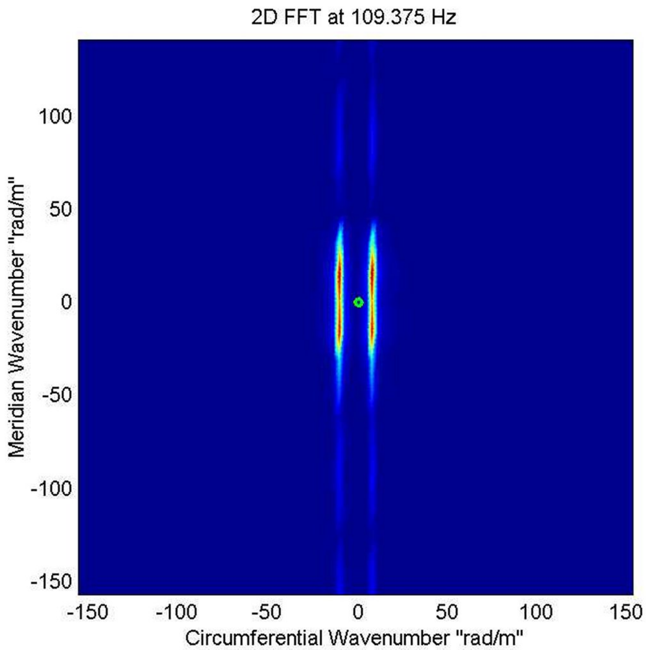

Figure 14: Two-dimensional wavenumber transform of the frequency response functions of the tire at 109 Hz.

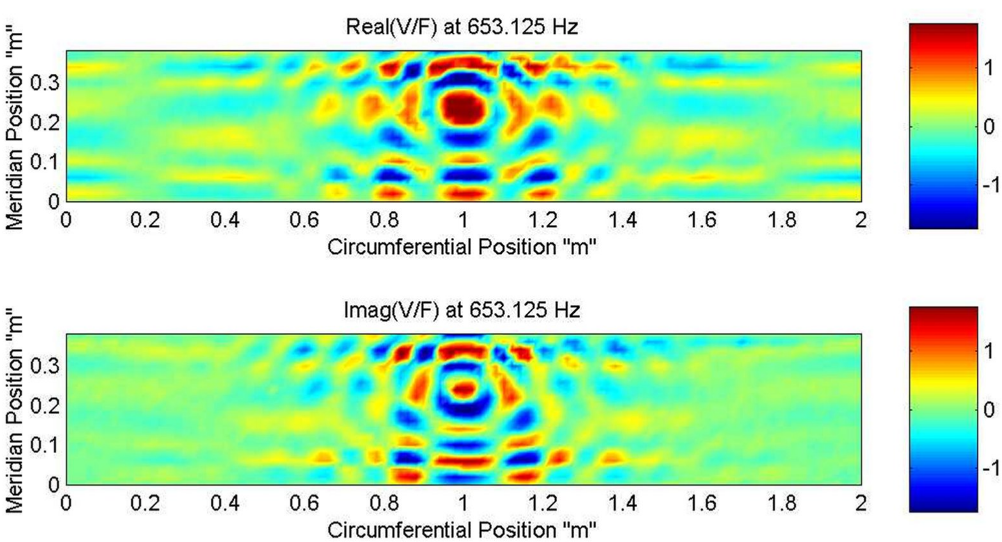

Figure 15: Two-dimensional display of the real and imaginary parts of the frequency response functions for the tire at 653 Hz.

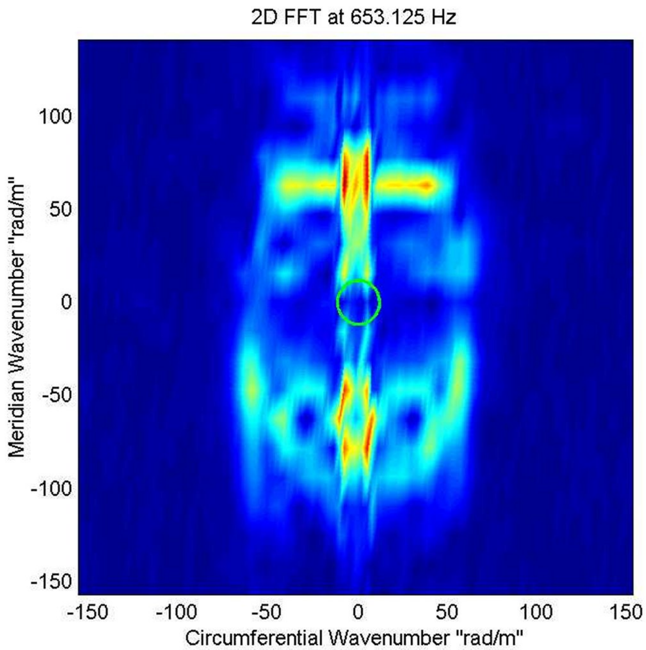

Figure 16: Two-dimensional wavenumber transform of the frequency response functions for the tire at 653 Hz.

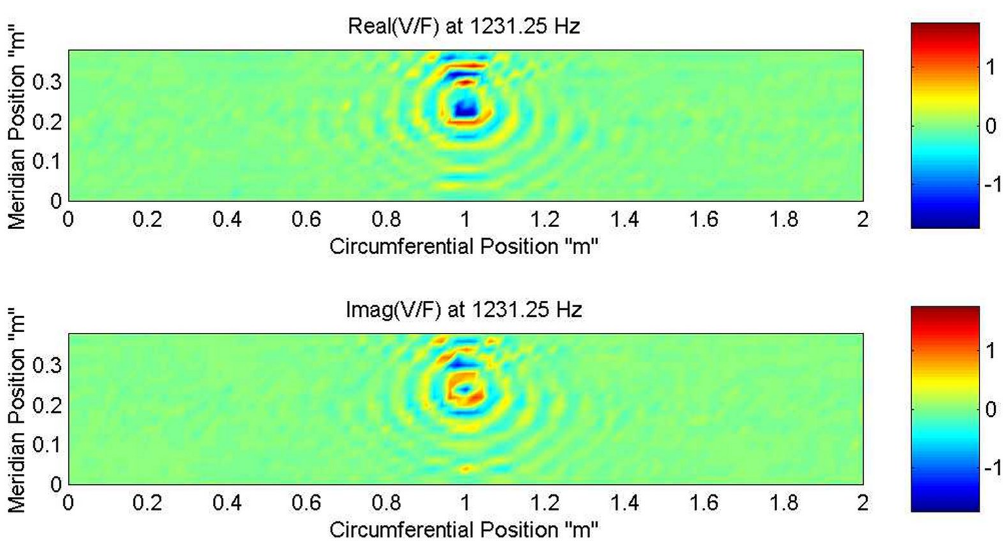

Figure 17: Two-dimensional display of the real and imaginary parts of the frequency response functions for the tire at 1231 Hz.

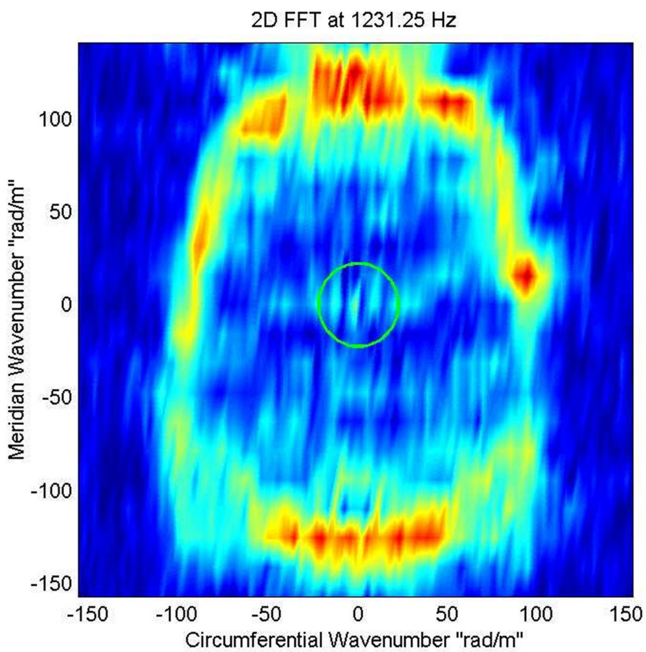

Figure 18: Two-dimensional wavenumber transform of the frequency response functions for the tire at 1231 Hz.