

VEHICLE BOOMING NOISE INVESTIGATION

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Department : Mechanical Engineering

Programme: Automotive

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PREFACE

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ABBREVIATIONS

SPL	: Sound Pressure Level
RPM	: Rotation Per Minute
RMS	: Root Mean Square
NVH	: Noise Vibration Harshness
FFT	: Fast Fourier Transform
CPB	: Constant Percentage Bandwidth
SPC	: Source Path Contribution
SPR	: Source Path Receiver
FRF	: Frequency Response Function
B&K	: Brüel&Kjaer

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SYMBOL LIST

p	: Sound pressure
L_p	: Sound pressure level
dB	: Decibel
dB(A)	: Decibel A-weighted
p_o	: Standardized reference level
I	: Sound intensity
r	: Density of air
c	: Speed of sound
λ	: Wavelength
f	: Frequency
L	: Level
B	: Bandwidth
D	: Displacement amplitude
ω	: Angular of frequency
T	: Period
k	: Spring constant
m	: Mass
t	: Time
t_r	: The time delay between the dual impacts
H	: Magnitude of Frequency Response Function
X, Q	: Functions
Q_i	: Volume velocity
F	: Force
K	: Mount stiffness
V	: Velocity
P	: Sound power
φ	: Phase
n	: Integer
T	: The effective duration of transient

VEHICLE BOOMING NOISE INVESTIGATION

SUMMARY

In recent years, vehicle comfort has become one of the most important parameters in choosing a vehicle. When noise and vibration are considered as an aspect of comfort, the main objective in the automotive NVH studies is to reduce the undesired noise which comes from several sources like road, engine noise and gearbox. Low interior sound pressure level and vibrations that affect both the driver and passengers make the vehicle preferable in the market like its design and performance.

In this study, the sources and transfer paths of the booming noise which causes an unpleasant effect have been described and the methods of NVH testing in automotive industry have been explained.

In a certain engine speed range, for several gearbox positions booming noise inside the cabin of a commercial vehicle has been investigated by using NVH testing equipments and software. Moreover, several noise and vibration tests in a semi-anechoic room have been performed in order to identify the booming noise with its sources and the transfer paths ending at the driver right ear position.

Body panel vibrations, structural resonances and the locations inside the cabin where the sound pressure levels are higher than the average have been measured via the experimental techniques which are explained in the further chapters. The engine block was thought to be a solid structure and its structural modes have been obtained by using a modal exciter. The vibrations through both the engine and gearbox mounts cause the resonances of the rear panel and doors. Because of this situation, an unpleasant sound that can be called as booming noise has been formed.

The frequency range which the booming noise inside the cabin appears, was detected and the reasons of this unpleasant phenomenon were investigated. Furthermore, some suggestions were made in order to overcome this problem.

ARAÇLARDA UĞULTU TARZI GÜRÜLTÜNÜN ARAŞTIRILMASI

ÖZET

Son yıllarda araç seçiminde dikkat edilebilecek kriterlerinin en başında konfor gelmektedir. Ses ve titreşim açısından konfor denildiği zaman, yoldan, motordan ve diğer kaynaklardan meydana gelen gürültünün sürücü ve yolcuları en az seviyede etkilemesi hedeflenmektedir. NVH çalışmalarında hedeflenen kabin içi düşük ses basıncı ve titreşim seviyesi dizayn, görünüş ve performans gibi aracın tercih edilmesinde öne çıkan önemli bir parametredir.

Bu çalışmada, araçlarda hoşnutsuzluğa neden olan uğultu tipi gürültü ve kaynak patikaları analizi konularında bilgi verilmiş, ayrıca taşıtlarda NVH çalışmaları kapsamında gerçekleştirilen deneysel yöntemler anlatılmıştır.

Çalışmanın deneysel kısmında, ticari bir taşıtta belli bir devir sayısında, çeşitli vites konumlarında araç kabininde oluşan rahatsız edici uğultu şeklindeki gürültü, deneysel ekipman ve bilgisayar programı yardımıyla ölçülmüş ve değerlendirilmiştir. Ayrıca, bu uğultu şeklindeki gürültünün kaynakları ve sürücü kulağına kadar izlediği patikalar ileri ölçüm yöntemleri ile yarı anakoik odada yapılan testlerde tesbit edilmiştir.

Araç panellerinin titreşim modları ve ses basınç seviyesinin kabin içinde yüksek olduğu bölgeler çalışmanın ileriki bölümlerinde anlatılan deney tekniklerine bağlı kalınarak ölçülmüştür. Motor bir katı cisim gibi düşünülerek titreşim modları araştırılmıştır. Şanzıman takozundan geçen titreşimler, kabin içinde arka panelin ve kapıların rezonansa girmesine neden olmaktadır. Bununla birlikte motor, bağlı olduğu takozlarda gürültünün olduğu frekansta rezonansa girmekte ve oluşan titreşimler kabin içi kavite rezonansına sebep olmaktadır. Bu etkilerin sonucunda kabin içinde uğultu şeklinde rahatsız edici gürültü oluşmaktadır.

Gürültünün olduğu frekans bölgesi tesbit edilerek oluşan problemin nedenleri araştırılmış ve bu sorunu giderme doğrultusunda çalışmalar yapılmıştır.

1. INTRODUCTION

1.1 The Introduction and the Aim of Study

Any vehicle operation is accompanied by a variety of vibrations and noises. But if a particular noise is unexpected, it will not only be unpleasant to the drivers or passengers, but it will make them doubt the integrity of the vehicle itself. This is true even if that unusual sound does not affect the overall mechanical functioning. For this reason, causes of these vibrations and noises thoroughly investigated and repaired.

As vehicles are comprised of numerous components such as the engine, transmission, suspension, body and other parts, the way vibrations and sounds are generated and how they resonate among themselves are diverse. If these vibrations and sounds are acceptable to the people who ride the vehicles, all is well. However, with the advent of smoother roads and technologically advanced vehicles, customers are demanding a more comfortable ride and even a slight vibration or noise is often a source of concern to them.

The one of the strongest sources of NVH in an automobile is the engine. The combustion stroke of a piston is the only engine activity which applies a tuning force to the crankshaft. The remaining movements do not contribute to the tuning force; instead, they apply resistance to the rotational movement of the crankshaft. This creates torque fluctuation in the crankshaft which is transmitted by way of the clutch to the transmission and the driveshaft.

The powertrain is also a source of vibration, since it is subject to bending and twisting forces such as the torque fluctuations created by the engine and the vibrating forces transmitted from the tires.

In order to reduce these vibrations and noises, several tests and measurements are made and vehicle booming noise has been investigated. Experiments have been done in Istanbul Technical University Automotive Department Laboratory Semi-Anechoic Room.

In the second chapter, basic concepts of sound and vibration subjects are mentioned. Sound pressure, frequency, sound intensity, sound level meter weightings, filter types, scales and vibration fundamentals are some of the themes that explained in this part.

In the third chapter, NVH testing in automotive industry is taking part. The explanation of NVH, noise and vibration testing equipments, system and signal analysis, hammer and shaker tests are the main topics of this chapter.

In the fourth chapter, source path contribution technique is explained. The basis of the SPC technique is to perform a phased summation of partial responses from all noise and vibration paths to give total tactile and acoustic responses under specific operating loads at a given frequency or RPM.

In the fifth chapter, the booming noise is mentioned. In a certain frequency range the booming noise originated from powertrain system cause an unpleasant effect for both the driver and passengers. As the booming noise is the main topic of this study, further information and detailed figures about the subject were given.

In the sixth chapter, tests and measurements in a semi-anechoic room were made for identifying the interior noise problem. Run-up acoustic performance tests using octave analysis and orders versus engine RPM were performed. In addition, run-up vibration diagnostics tests to identify the structural resonances and noise generated by vibrations, determination of critical speeds and resonances. Finally, several additional tests like hammer, sound intensity, engine mounts performance, and the shaker tests were made for exposing the reasons of the problem.

In conclusion, the main problem in the test vehicle is stated as the booming noise. There is an unpleasant noise in the commercial test vehicle interior cabin which gives clues of booming noise. The interior noise reaches to peak values at most commonly used engine speed ranges like 2500-3000 RPM. If engine speed harmonic coincides with a cavity resonance frequency in the passenger compartment, passengers will perceive an unpleasant dull booming sound in most of the compartment. With the detection of the frequency range which the booming noise inside the cabin appears several changes and improvements were suggested to prevent the booming noise which encountered in the vehicle.

2. BASIC CONCEPTS OF SOUND AND VIBRATION

2.1 Terminology of Sound

Sound is defined as any pressure variation that the ear can detect ranging from the weakest sounds to sound levels which can damage hearing.

A sound source will produce a certain amount of sound energy per unit time [Joule/sec], i.e. it has a certain sound power rating in W [Watt = Joule/sec]. This is a basic measure of how much acoustical energy it can produce, and is independent of its surroundings. The sound energy flows away from the source giving rise to a certain sound pressure in the room. When the sound pressure is measured this will not only depend on the power rating of the source and the distance between the source and the measurement point, but also on the amount of sound energy absorbed by the walls and the amount of sound energy transferred through the walls and windows etc [1].

2.1.1 The Frequency of the Sound

The number of pressure variations per second is called the frequency of the sound, and is measured in Hertz (Hz). The frequency of a sound produces its distinctive tone. Thus, the rumble of distant thunder has a low frequency, while a whistle has a high frequency. The normal range of hearing for a healthy young person extends from approximately 20 Hz up to 20 000 Hz (or 20 kHz). These pressure variations travel through any elastic medium (such as air) from the source of the sound to the listener's ears [5].

A sound which has only one frequency is known as a pure tone. In practice pure tones are seldom encountered and most sounds are made up of different frequencies. Most industrial noise consists of a wide mixture of frequencies known as broad band noise. If the noise has frequencies evenly distributed throughout the audible range it is known as white noise.

2.1.2 Sound Pressure and Sound Power

The sound pressure that we hear or measure with a microphone is dependent on the distance from the source and the acoustic environment (or sound field) in which

sound waves are present. This in turn depends on the size of the room and the sound absorption of the surfaces. The sound power must be found because the noise quantity is more or less independent of the environment and is the unique descriptor of the noisiness of a sound source [9].

A sound source radiates power and this radiation results in a sound pressure. Sound power is the cause. Any piece of machinery that vibrates radiates acoustical energy. Sound power is the rate at which energy is radiated (energy per unit time). Sound pressure is the effect. The relationship between sound power and sound pressure is similar. What we hear is sound pressure but it is caused by the sound power emitted from the source [9].

Too high a sound pressure may cause hearing damage. So when trying to quantify human response to sound, such as noise annoyance or the risk of hearing loss, pressure is the obvious quantity to measure. It is also relatively easy to measure: The pressure variations on the eardrum we perceive as sound are the same pressure variations which are detected on the diaphragm of a condenser microphone.

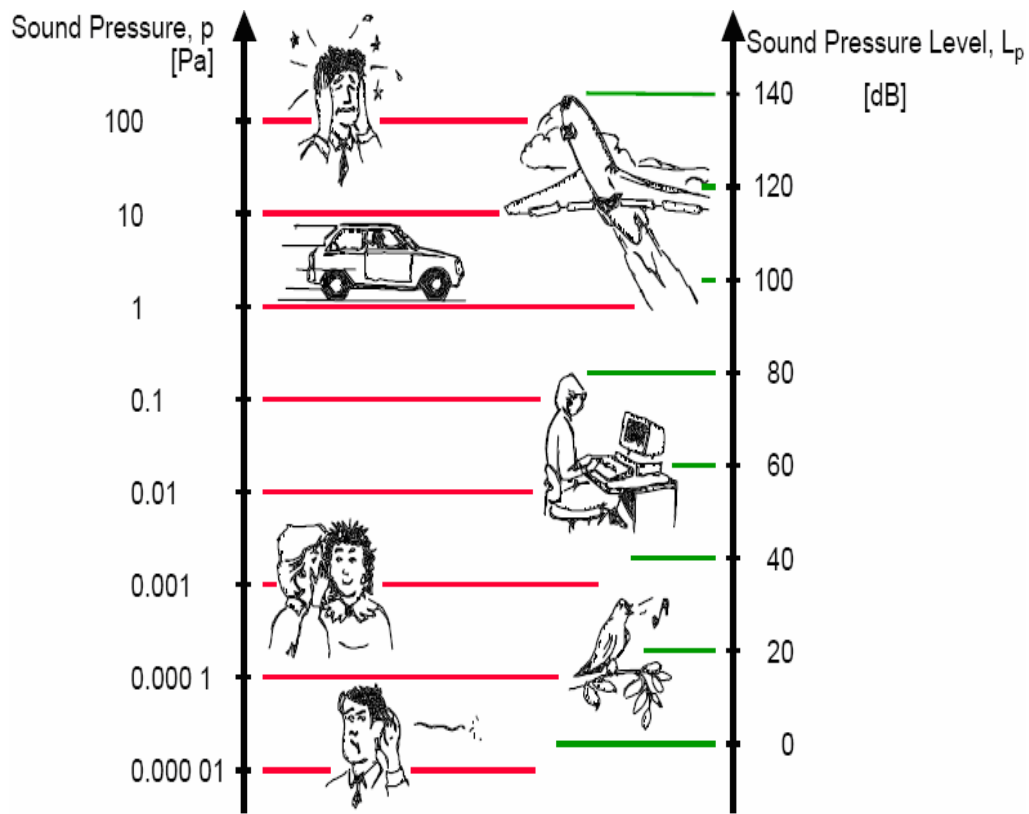


Figure 2.1 : Range of Sound Pressure Levels [1]

The direct application of linear scales, in Pa, to the measurement of sound pressure would therefore lead to the use of enormous and unwieldy numbers. Additionally, the ear responds not linearly but logarithmically to stimulus. For these reasons, it has been found more practical to express acoustic parameters as a logarithmic ratio of the measured value to a reference value a logarithmic ratio called a decibel or just dB.

The advantage of using dB's is clearly seen in Figure 2.1. The linear scale with its large and unwieldy numbers is converted into a much more manageable scale from 0 dB at the threshold of hearing (20 mPa) to 130 dB at the threshold of pain.

The sound pressure level, L_p , in dB's is defined as $20 \log p/p_0$, where p is the measured value in Pa, and p_0 is a standardized reference level of 20 mPa the threshold of hearing. The word level is added to sound pressure to indicate that the quantity has a certain level above the reference level, and the symbol for sound pressure level is L_p [1].

2.1.3 Perception of dB's

An increase of 3 dB in pressure (corresponding to 1.4 times) is just perceptible. A change of 10 dB or 3.16 times is perceived as twice as loud. There is no linear relationship between the loudness level in dB and the perception by man. Table 2.1 shows the approximate relation changes in spl and loudness.

Table 2.1: Approximate Relation Changes in SPL and Loudness

Change in Sound Level (dB)	Change in Perceived Loudness
3	Just perceptible
5	Noticeable difference
10	Twice (or 1/2) as loud
15	Large change
20	Four times (or 1/4) as loud

2.1.4 Frequency Weighting Curves

The apparent loudness of a sound varies with frequency as well as with the sound pressure, and that the variation of loudness with frequency also depends to some extent on the sound pressure. A measuring instrument can be designed to make some allowance for this by the use of a weighting network. The various standards organizations recommend the use of three weighting networks, as well as a linear

response. The A-weighting, which is now used almost exclusively, was originally designed to follow the response of the human ear at low sound levels [13].

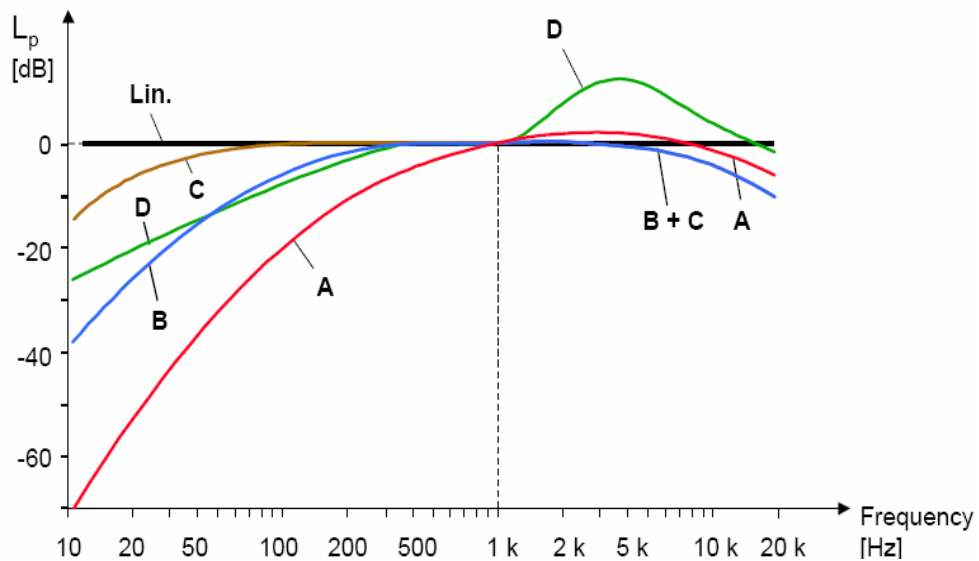


Figure 2.2 : The Internationally Standardized Weighting Curves for Sound Level Meters

Figure 2.2 shows the weighting curves for sound level meters. The A-weighting, B-weighting and C-weighting curves follow approximately the 40, 70 and 100 dB equal loudness curves respectively. D-weighting follows a special curve which gives extra emphasis to the frequencies in the range 1 kHz to 10 kHz.

2.1.5 Conversion to dB Using Charts

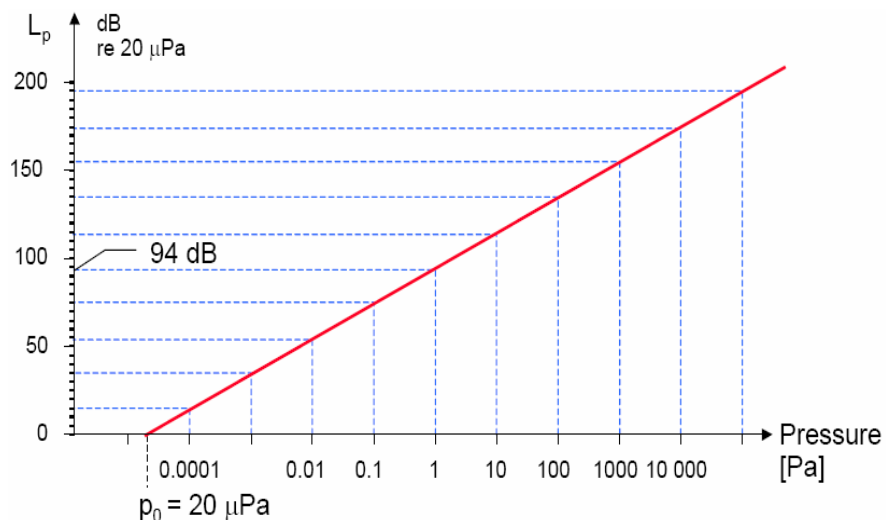


Figure 2.3 : Conversion Chart Pa to dB [1]

Instead of using the formula for conversion between pressure values and dB levels (or vice versa) it is possible to use a simple graph for conversion. The Figure 2.3 is based on dB values re 20 mPa and shows how 1 Pa converts to 94 dB [1].

2.1.6 The Sound Intensity

The Sound Intensity vector, I , describes the amount and direction of flow of acoustic energy at a given position.

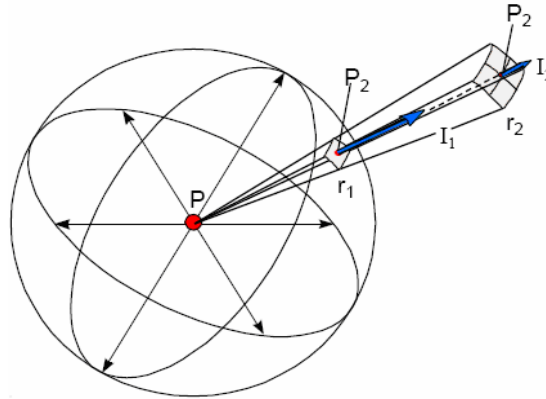


Figure 2.4 : Transfer of Energy from the Source [1]

Power: P [W]

Intensity: I [J/s/m^2] = W/m^2

Pressure: p [$\text{Pa} = \text{N/m}^2$]

$$I = \frac{P}{4\pi r^2} = \frac{p^2}{\rho c} \quad (2.1)$$

When sound is produced by a sound source with a sound power, P , a transfer of energy from the source to the adjacent air molecules takes place. This energy is transferred to outlying molecules. Thus the energy spreads away from the source rather like ripples on a pond. The rate at which this energy flows in a particular direction through a particular area is called the sound intensity, I . The energy passing a particular point in the area around the source will give rise to a sound pressure, p , at that point. ρ is the density of air, c is the speed of sound (2.1). Sound intensity is a vector quantity and it has magnitude as well as direction. Figure 2.4 shows the transfer of energy from the source [1].

Sound intensity gives a measure of direction as well as magnitude it is also very useful when locating sources of sound. Therefore the radiation patterns of complex vibrating machinery can be studied in situ.

Sound intensity and sound pressure can be measured directly by suitable instrumentation. Sound power can be calculated from measured values of sound pressure or sound intensity levels and knowledge of the area over which the measurements were made. The main use of sound power is for the noise rating of machines etc. and sound intensity is mainly used for location and rating of noise sources. When it comes to evaluation of the harmfulness and annoyance of noise sources, sound pressure is the important parameter [1].

2.1.6.1 Pressure and particle velocity

When a particle of air is displaced from its mean position there is a temporary increase in pressure. The pressure increase acts in two ways: to restore the particle to its original position, and to pass on the disturbance to the next particle. The cycle of pressure increases (compressions) and decreases (rarefactions) propagates through the medium as a sound wave. There are two important parameters in this process: the pressure (the local increases and decreases with respect to the ambient) and the velocity of the particles of air which oscillate about a fixed position. Sound intensity is the product of particle velocity and pressure. And, as can be seen from the transformation below (2.2), it is equivalent to the pressure times particle velocity [9].

$$\text{Intensity} = \text{Pressure} \times \text{Particle Velocity} \quad (2.2)$$

In an active field, pressure and particle velocity vary simultaneously. A peak in the pressure signal occurs at the same time as a peak in the particle velocity signal. They are therefore said to be in phase and the product of the two signals gives a net intensity.

In a reactive field the pressure and particle velocity are 90° out of phase. One is shifted a quarter of a wavelength with respect to the other. Multiplying the two signals together gives an instantaneous intensity signal varying sinusoidally about zero. Therefore the time-averaged intensity is zero. In a diffuse field the pressure and particle velocity phase vary at random and so the net intensity is zero. In figure 2.5 phase shifts and instantaneous intensities are shown.

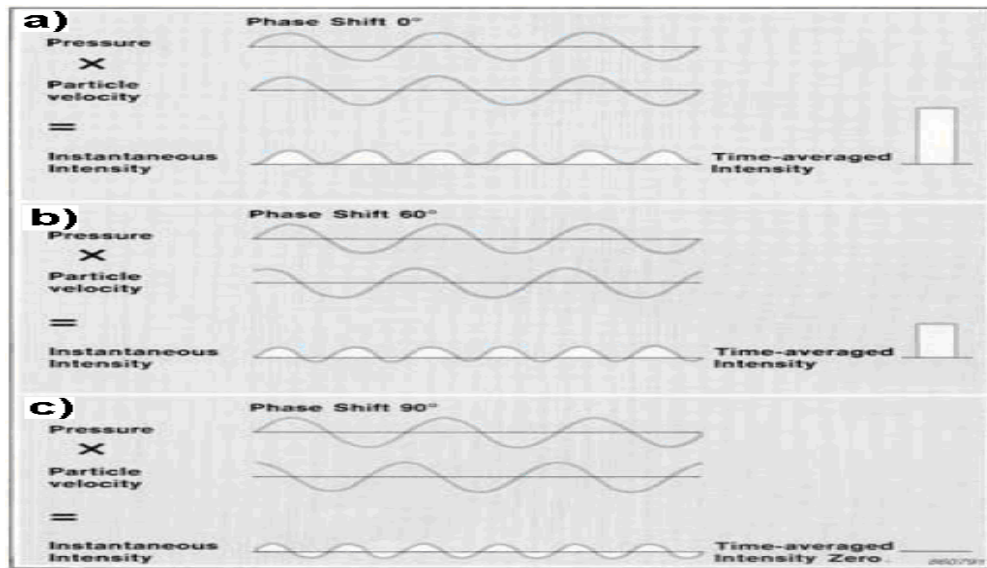


Figure 2.5 : Instantaneous Intensity (a) Phase Shift 0° (b) Phase Shift 60°
c) Phase Shift 90° [9]

2.2 Basic Frequency Analysis of Sound

2.2.1 Perception of Sound

The frequency span of the sounds that typically surround human beings vary considerably. Normally, young human beings can detect sounds ranging from 20 to 20000 Hz. Figure 2.6 shows the human beings sound range.

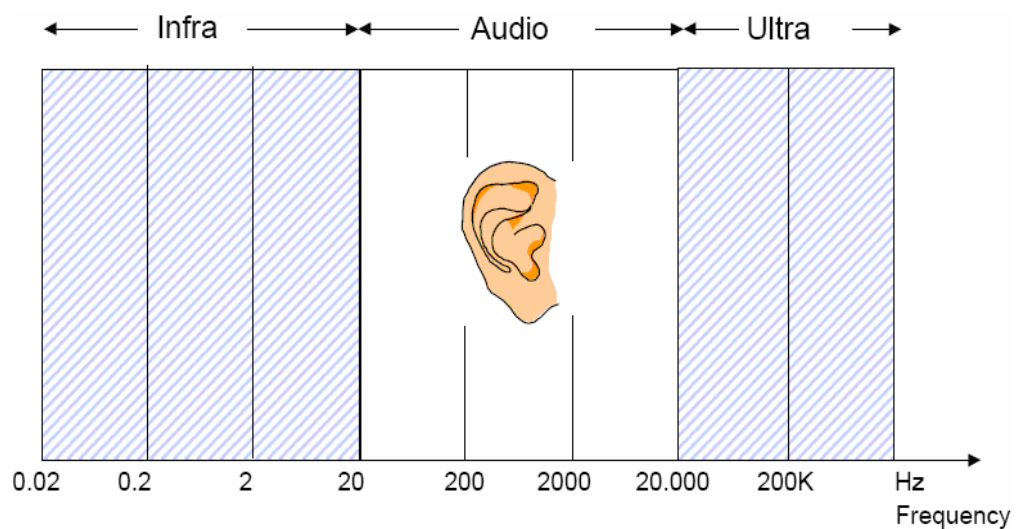


Figure 2.6 : Human Beings Sound Range [2]

2.2.1.1 Wavelength and frequency

A sound signal from a loudspeaker mounted at one end of a tube will produce a sound wave that propagates forward at a speed of 344 m/s. If the signal is a single sine signal the sound wave will consist of a number of pressure maxima and minima all separated by one wavelength. Representation of sound propagation is shown in Figure 2.7 [2].

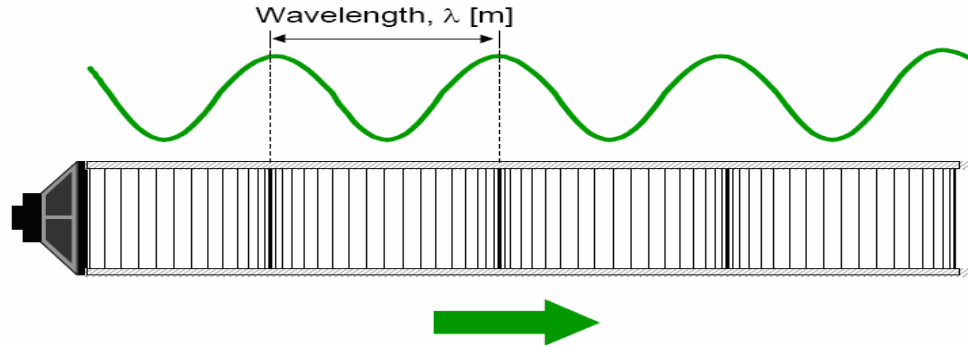


Figure 2.7 : Representation of Sound Propagation [2]

The wavelength, the speed of sound and the frequency are related according to the formula shown (2.3). It is useful to have a rough feeling for which wavelength corresponds to a given frequency. Figure 2.8 shows the relation between wavelength and frequency [2].

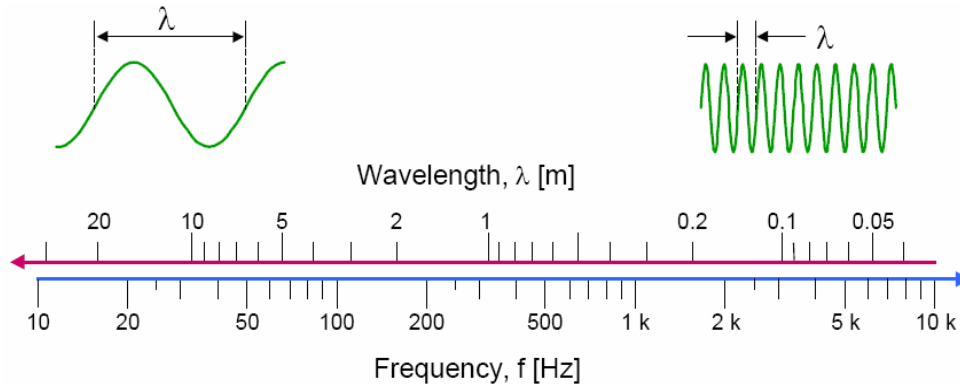


Figure 2.8 : The Relation between Wavelength and Frequency [2]

$$\lambda = c / f \quad (2.3)$$

At 1 kHz the wavelength is close to 34 cm or one foot, at 20 Hz it is close to 17 m, and only 1.7 cm at 20 kHz.

2.2.2 Reflection of Sound

When sound hits obstructions large in size compared to its wavelength, reflections take place. If the obstruction has very little absorption, all the reflected sound will have equal energy compared to the incoming sound. This is one of the important design principles used when constructing reverberant rooms. If almost all reflected energy is lost due to high absorption in the reflecting surfaces, the situation is close to what is found in an anechoic room. Figure 2.9 shows the reflection of sound in the surfaces [2].

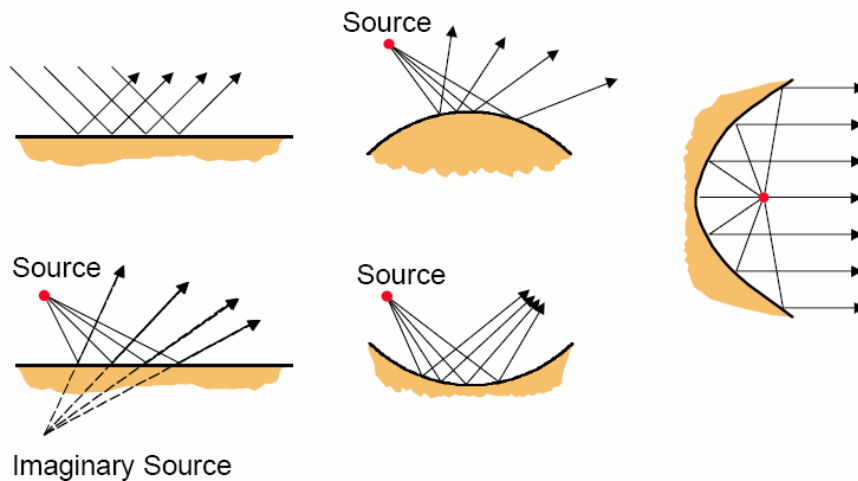


Figure 2.9 : Reflection of Sound in the Surfaces [2]

2.2.3 Frequency Analysis

2.2.3.1 Waveforms and frequencies

Three examples of the relationship between the waveform of a signal in the time domain compared to its spectrum in the frequency domain.

In the Figure 2.10 (a), a sine wave of large amplitude and wavelength is showing up as a single frequency with a high level at a low frequency.

In the Figure 2.10 (b), a low amplitude signal with small wavelength is seen to show up in the frequency domain as a high frequency with a low level.

At the Figure 2.10 (c), it is shown how a sum of the two signals above also in the frequency domain shows up as a sum [2].

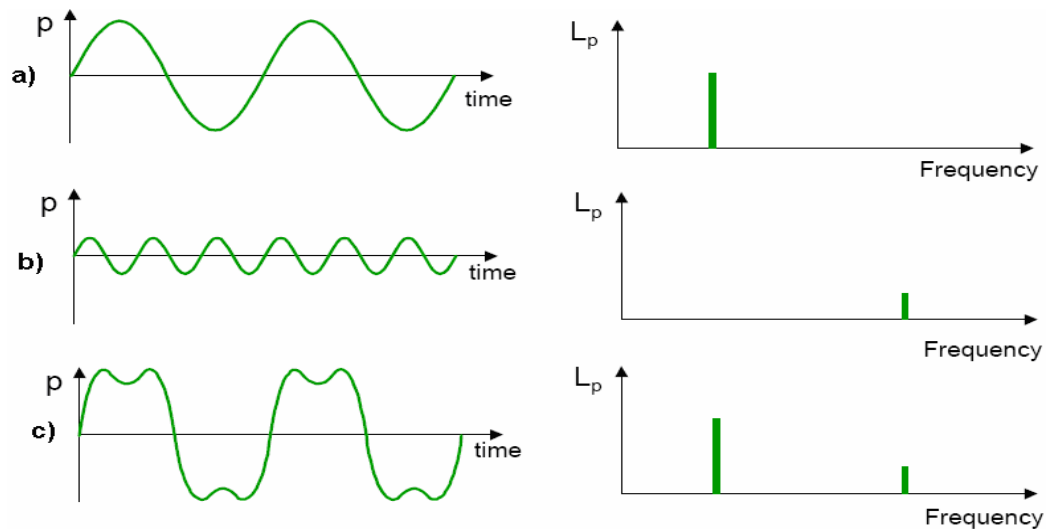


Figure 2. 10 : The Relationship between the Waveform and the Frequency of a Signal [2]

2.2.3.2 Filter types and frequency scales

The two most used filter banks are:

Filters that have the same bandwidth e.g. 400 Hz and displayed using a linear frequency scale. This is a result of a (FFT) Fast Fourier Transform analysis. Constant bandwidth filters are mainly used in connection with analysis of vibration signals [2].

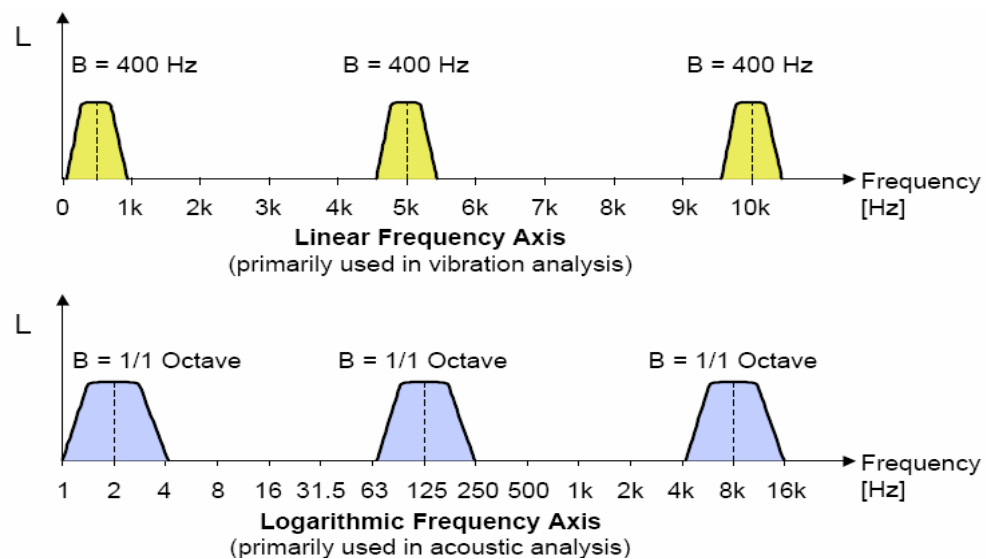


Figure 2.11 : Filter Types of Frequency [2]

Filters, which all have the same constant percentage bandwidth (CPB filters), e.g. 1/1 octave, are normally displayed on a logarithmic frequency scale. Sometimes these filters are also called relative bandwidth filters. Analysis with CPB filters (and logarithmic scales) is almost always used in connection with acoustic measurements, because it gives a fairly close approximation to how the human ear responds. Figure 2.11 shows the filter types of frequency [2].

The widest octave filter used has a bandwidth of 1 octave. However, many subdivisions into smaller bandwidths are often used. The filters are often labeled as “Constant Percentage Bandwidth” filters. A 1/1 octave filter has a bandwidth of close to 70% of its centre frequency (2.5). Figure 2.12 (a) shows the 1/1 octave filter [2].

$$f_2 = 2 \times f_1 \quad (2.4)$$

$$B = 0.7 \times f_0 \approx 70 \% \quad (2.5)$$

The most popular filters are perhaps those with 1/3 octave bandwidths. Figure 2.12 (b) shows the 1/3 octave filter. A 1/3 octave filter has a bandwidth of close to 23% of its centre frequency (2.7). One advantage is that this bandwidth at frequencies above 500 Hz corresponds well to the frequency selectivity of the human auditory system. Filter bandwidths down to 1/96 octave have been realized [2].

$$f_2 = \sqrt[3]{2} \times f_1 = 1.25 \times f_1 \quad (2.6)$$

$$B = 0.23 \times f_0 \approx 23 \% \quad (2.7)$$

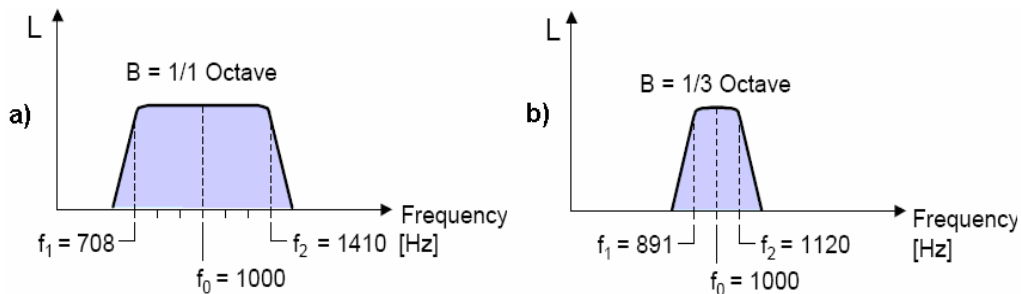


Figure 2.12 : (a) 1/1 Octave Filter (b) 1/3 Octave Filter [2]

A detailed signal with many frequency components show up with a filter shape as the dotted curve when subjected to an octave analysis. The solid curve shows the

increased resolution with more details when a 1/3 octave analysis is used. Figure 2.13 shows the difference between octave filters.

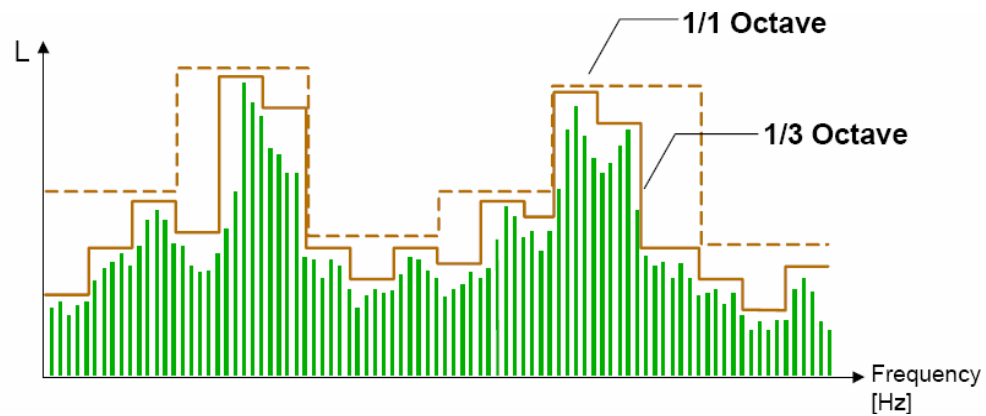


Figure 2.13 : The Difference between 1/1 and 1/3 Octave Filters [2]

2.3 Vibration Fundamentals

Vibration is an oscillation wherein the quantity is a parameter defining the motion of a mechanical system. Oscillation is the variation, usually with time, of the magnitude of a quantity with respect to a specified reference when the magnitude is alternately greater and smaller than the reference [3].

A body is said to vibrate when it describes an oscillating motion about a reference position. The number of times a complete motion cycle takes place during the period of one second is called the frequency and is measured in hertz (Hz).

Vibration is a result of dynamic forces in machines which have moving parts and in structures which are connected to the machine. Different parts of the machine will vibrate with various frequencies and amplitudes. Vibration causes wear and fatigue. It is often responsible for the ultimate breakdown of the machine.

All mechanical systems contain the three basic components: spring, damper, and mass. These are shown in Figure 2.14. When each of these in turn is exposed to a constant force they react with a constant displacement, a constant velocity and a constant acceleration respectively.

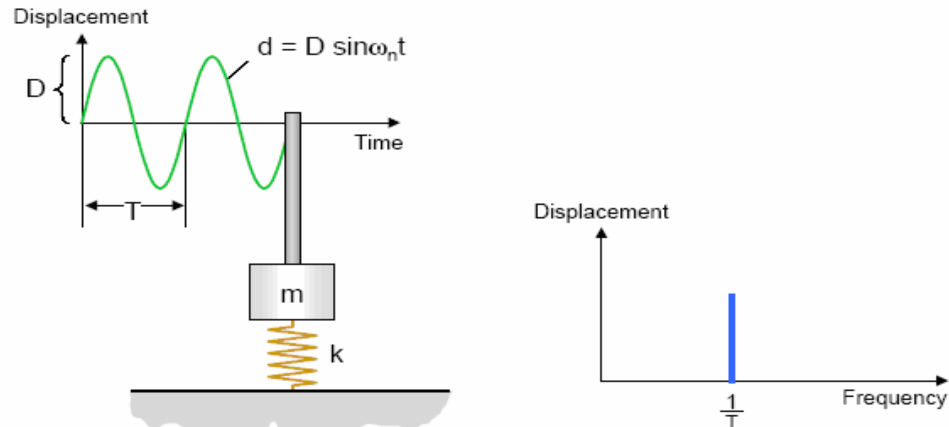


Figure 2.14 : Simplest Form of Vibration System [3]

Once a (theoretical) system of a mass and a spring is set in motion it will continue this motion with constant frequency and amplitude. The system is said to oscillate with a sinusoidal waveform [3].

The sine curve which emerges when a mass and a spring oscillate can be described by its amplitude (D) and period (T). Frequency is defined as the number of cycles per second and is equal to the reciprocal of the period. By multiplying the frequency by 2, the angular frequency is obtained, which is again proportional to the square root of spring constant k divided by mass m . The frequency of oscillation is called the natural frequency f_n [3].

2.3.1 Forces and Vibration

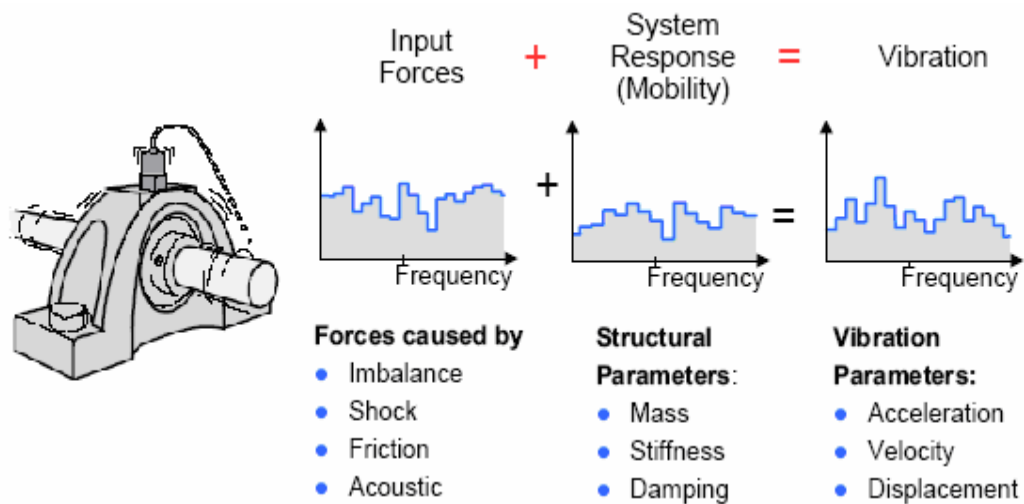


Figure 2.15 : Force and Vibration System [3]

A system will respond to an input force with a certain motion, depending on what we call the mobility of the system. Knowing the force and the mobility permits us to calculate the vibration. Figure 2.15 shows the force, mobility and the vibration [3].

Modal analysis or other methods are used to model systems. Once the model is created and calculated its mobility for a force input at a certain point to predict vibration at different locations. Such models can also in some cases be used to calculate the load on the structure to predict failure [3].

2.3.2 Signals

Basically a distinction between Stationary Signals and Non-stationary Signals has to be made. Stationary Signals can again be divided into Deterministic Signals and Random Signals, and Non-stationary Signals into Continuous and Transient signals. In Figure 2.16 signal types are shown [3].

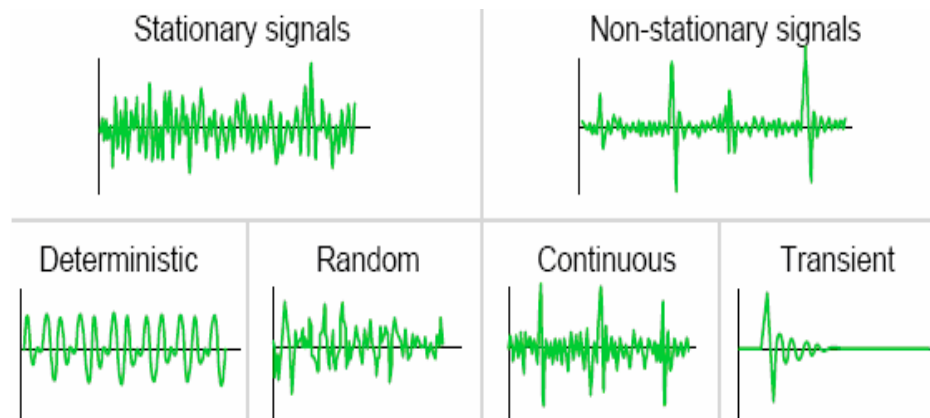


Figure 2.16 : Signal Types [3]

Stationary deterministic signals are made up entirely of sinusoidal components at discrete frequencies. Random signals are characterized by being signals where the instantaneous value cannot be predicted, but where the values can be characterized by a certain probability density function. Random signals have a frequency spectrum which is continuously distributed with frequency.

The continuous non-stationary signal has some similarities with both transient and stationary signals. During analysis continuous non-stationary signals should normally be treated as random signals or separated into the individual transient and treated as transients. Transient signals are defined as signals which commence and finish at a constant level, normally zero, within the analysis time.

2.3.2.1 Signal level descriptors

The level of vibration signal can be described in different ways. Peak and peak-to-peak values are used to describe the level of a vibration signal since they indicate the maximum excursion from equilibrium position. The RMS (Root Mean Square) level is a very good descriptor. It is a measure of the energy content of the vibration signal. The signal level descriptors are shown in Figure 2.17 [3].

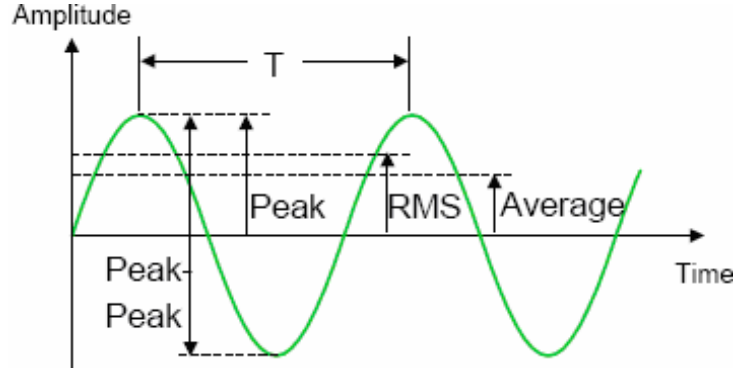


Figure 2.17 : Signal Level Descriptors [3]

$$\text{RMS} = \sqrt{\frac{1}{T} \int_0^T x^2(t) dt} \quad (2.8)$$

$$\text{Average} = \frac{1}{T} \int_0^T |x(t)| dt \quad (2.9)$$

2.3.3 Frequency Spectrum

The frequency spectrum gives in many cases detailed information about the signal sources which cannot be obtained from the time signal. The frequency content can be found in many different ways, using scanning filter, filter banks or a digital treatment of a record using Fourier Transformation.

The process of Frequency Analysis is as follows: By sending a signal through a filter and at the same time sweeping the filter over the frequency range of interest. It is possible to get a measure of the signal level at different frequencies. The result is called a Frequency Spectrum.

Therefore logarithmic scales are often used, sometimes for level using dBs or just with appropriate numbers at the tick marks, sometimes for the frequency maybe with decades or octaves as indications.

2.3.3.1 Scales for frequency spectrum

Both linear and logarithmic frequency scales are used in connection with vibration measurement. The linear frequency scale has the advantage that it is easy to identify harmonically related components in the signal.

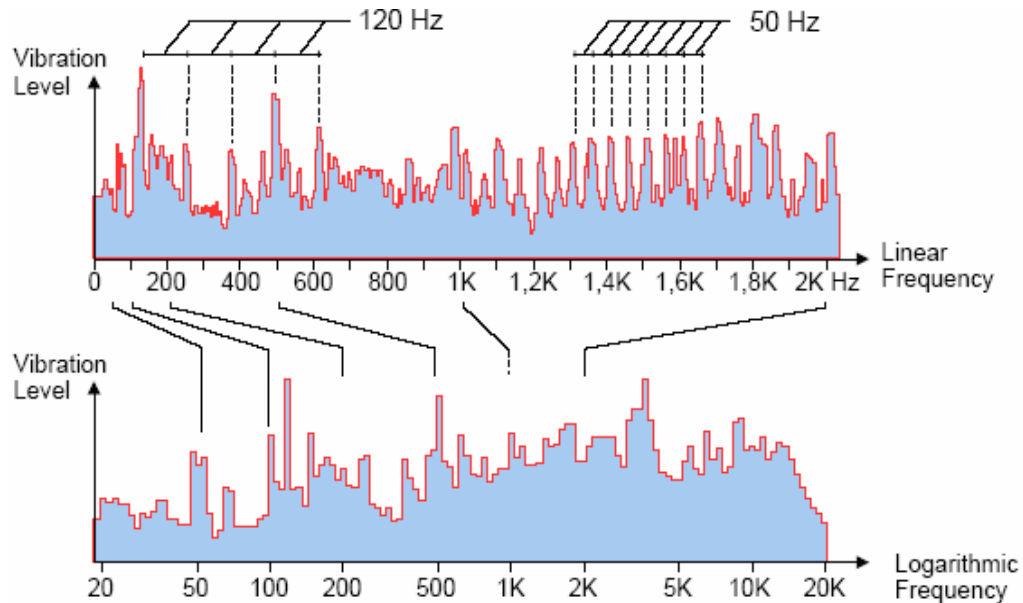


Figure 2.18 : Linear and Logarithmic Frequency [11]

The logarithmic scale, however, has the advantage that a much wider frequency range can be covered in a reasonable space and each decade is given the same emphasis. The signal shown in Figure 2.18 is the vibration signal with the two different scales. The harmonically related components in the signal are easily identified on the linear scale and the logarithmic scale gives many details in the low end while it covers a 10 times wider frequency range at the same time.

Analysis with constant bandwidth filters (and linear scales) is mainly used in connection with vibration measurements, because signals from mechanical structures often contain harmonic series and sideband structures. These are most easily identified on a linear frequency scale [11].

Analysis with CPB filters (and logarithmic scales) is almost always used in connection with acoustic measurements, because it gives a fairly close approximation to how the human ear responds. In connection with vibration measurements, the CPB bandwidth filter is used for measurement of structural responses.

2.3.3.2 Selecting a bandwidth for frequency spectrum

The filter bandwidth must be selected for distinguishing the important frequency components from one to another. If the narrower bandwidth is used, the more the information can be detailed. The more detailed (narrow band) analysis however requires a longer analysis time. Figure 2.19 shows the filter bandwidth and frequency spectrum.

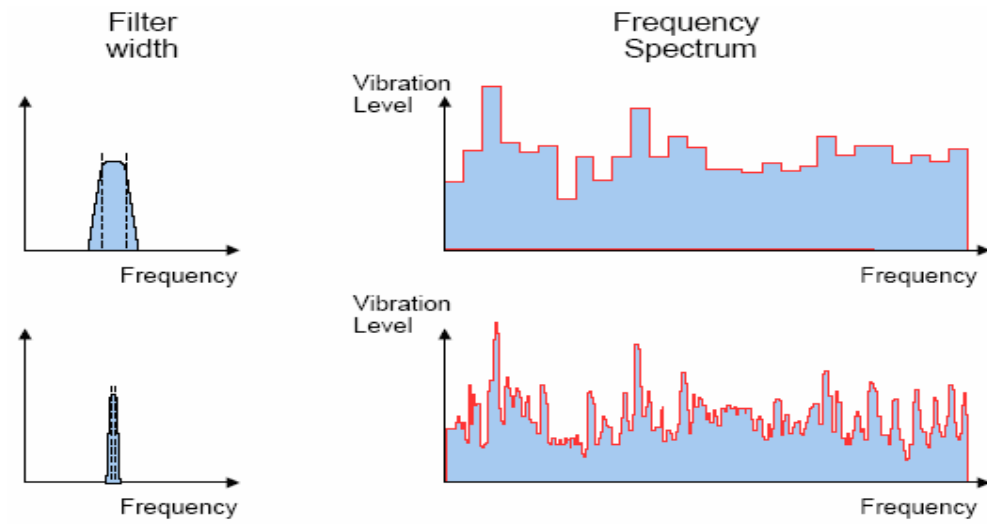


Figure 2.19 : Filter Width and Frequency Spectrum [11]

2.3.3.3 The dB scale for the frequency spectrum

The dB scale reduces the considerable numerical span of the normal log scale to a compact linear numbering system. The dB scale is such that a given percentage interval in acceleration level is represented by a given number of dB's. This is an advantage when dealing with vibration and interested in a percentage change in the vibration level rather than in the actual levels. Zero dB on the dB scale can be chosen for any vibration level. For acceleration, the reference level 10^{-6} ms^{-2} has been internationally chosen.

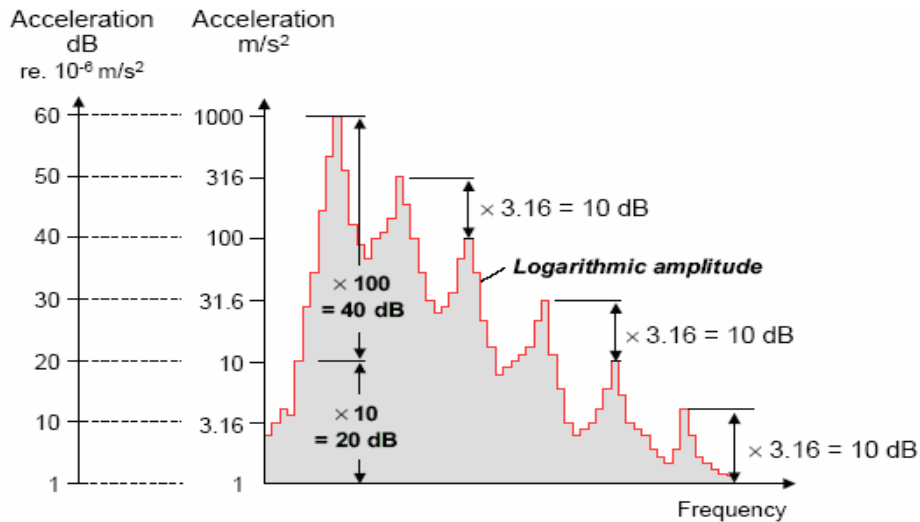


Figure 2.20 : Logarithmic and Linear Scale for Amplitude [11]

2.3.4 Vibration Parameter Choice

If the type of measurement being carried out does not call for a particular parameter to be measured, the general rule is that the parameter giving the flattest response over the frequency range of interest should be chosen. This will give the biggest dynamic range of the whole measurement set up. If the frequency response is not known, start by choosing velocity [11].

An advantage of the accelerometer is that its electrical output can be integrated to give velocity and displacement signals.

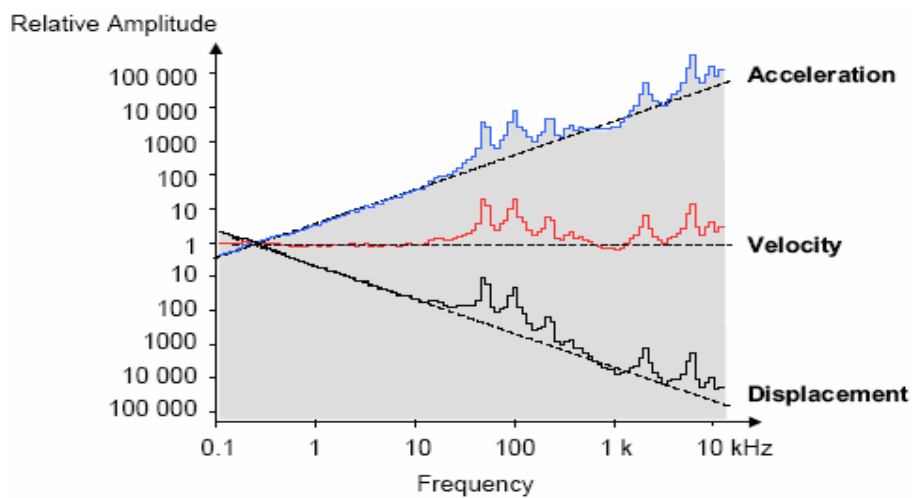


Figure 2.21 : Vibration Parameters in Amplitude and Frequency [11]

The best perform in the analysis on the signal is the flattest spectrum. If a spectrum is not reasonably flat, the contribution of components lying well below the mean level, will be less noticeable. In the case of overall measurements, smaller components might pass completely undetected.

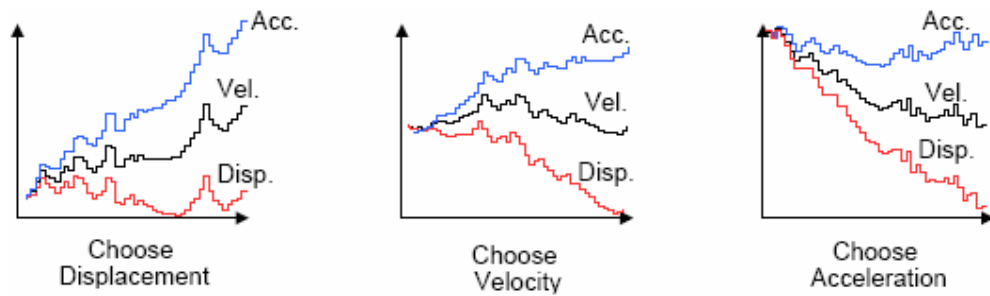


Figure 2.22 : The Detection Parameters of Vibration Measurements

3. NVH TESTING IN AUTOMOTIVE INDUSTRY

3.1 NVH Description

Noise and vibration are inherent occurrences in the operation of a motor vehicle. A vehicle does not make noise or vibration simply does not exist. NVH is an acronym for noise, vibration and harshness.

3.1.1 Noise

A running engine or a moving vehicle produces a variety of sounds. Whether these sounds are a noise or not depends more on how the listener perceives them than how loud they actually are. Generally, a noise is an inappropriate, unpleasant, or excessively loud sound.

3.1.2 Vibration

A running engine or a moving vehicle generates various vibrations, just as with noise. The driver or the passenger may consider these vibrations unpleasant, depending on where and how they occur.

3.1.3 Harshness

Harshness is a single and momentary sound created by a strong impact to a tire. It is like hitting the tire with a sledge hammer, and its impact is transmitted to the steering wheel and the floor.

3.2 The Phenomenon of Sound & Vibration

The main source of noise and vibration in a vehicle is engine. The engine produces two kinds of energy, while it's working. These are aerodynamic energy and mechanic energy. Figure 3.1 shows the radiation of sound and vibration.

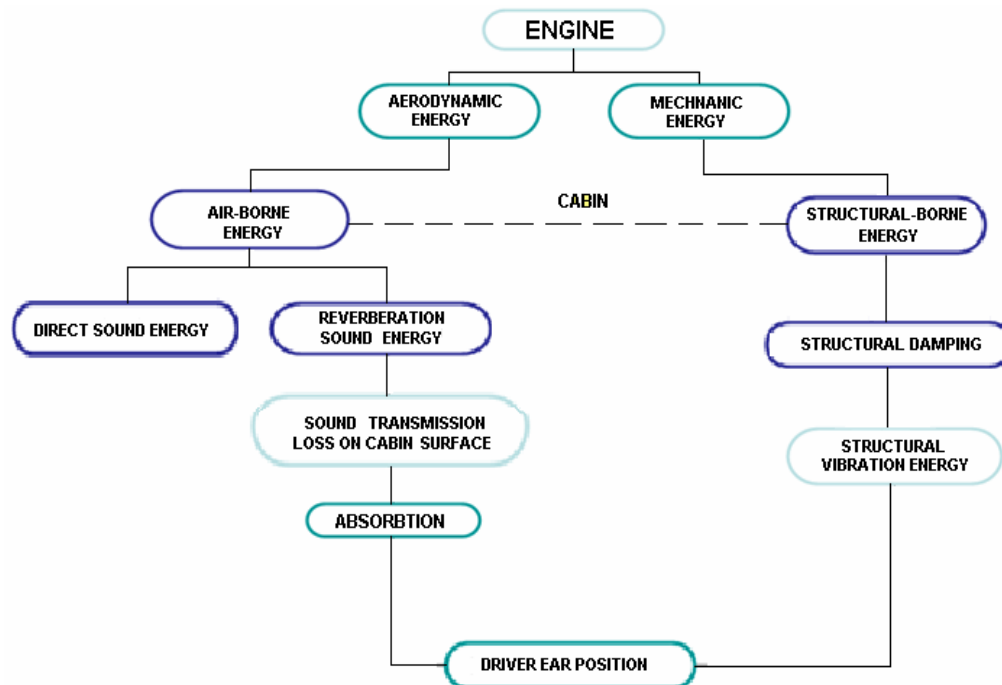


Figure 3.1 : The Sound and Vibration Radiation

The aerodynamic energy is air-borne energy. It radiates direct sound energy and reverberation sound energy. Reverberation sound energy is absorbed on cabin surface because of the sound transmission loss. It reaches to the driver ear position.

The mechanic energy is structural energy. This structural energy has been damped on mounts, houses and hangers. The structural vibration energy reaches to the driver ear position.

3.3 Vibration Frequency and Transmission

Although humans feel vibration through their tactile sense and hear sound through their auditory sense, the way a vibration or a sound reaches those senses is the same. Figure 3.3 shows the NVH frequency range.

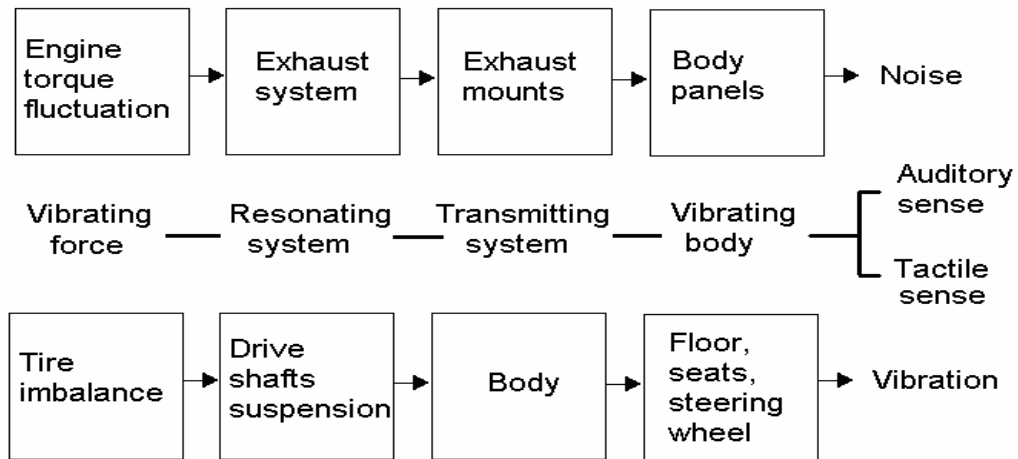


Figure 3.2 : The Noise and Vibration Cascading

To investigate the exhaust system noise and vibration; a vibration generated by the engine of a car causes the exhaust system to vibrate. The exhaust system then resonates with the engine amplifying the vibration. This vibration is transmitted through the exhaust mounts to the body, causing the body vibrate. So the vibration is heard or felt depending on its frequency. Figure 3.2 shows the noise and vibration cascading.

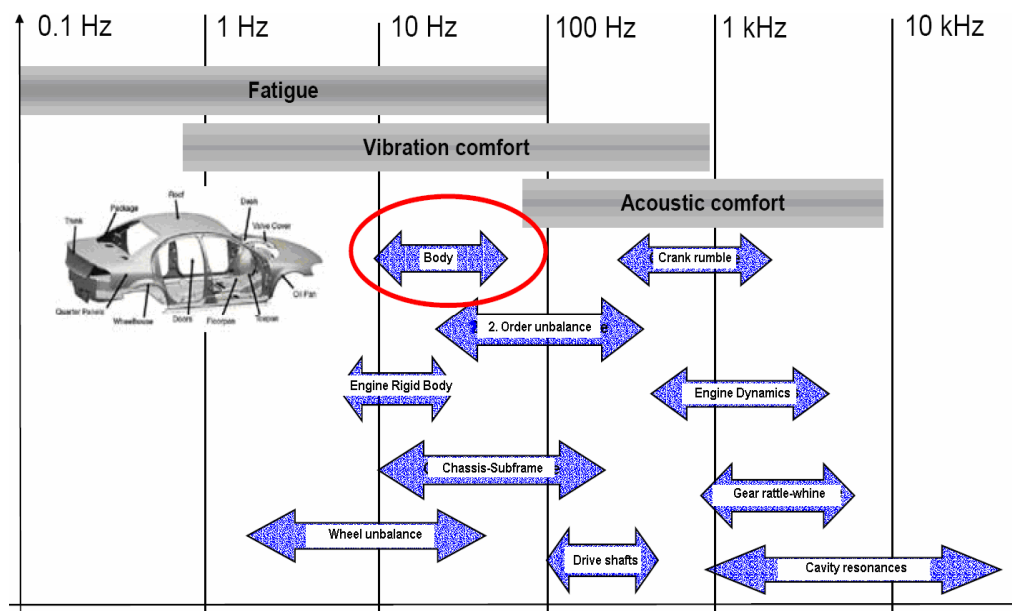


Figure 3.3 : NVH Frequency Range [19]

3.4 Noise and Vibration Measurement Equipments

3.4.1 Microphones and Preamplifiers

The condenser microphone converts the acoustical pressure variations into an electrical signal which thereafter is amplified in a preamplifier. The preamplifier must always be connected very close to the microphone since its main purpose is to convert the very high impedance of the microphone into a low output impedance permitting use of long cables and connection to instruments with relatively low input impedance. The low impedance ensures very little picks up of external electrical noise and this is especially important when using long cables. In Figure 3.4, a Brüel&Kjaer type 4189 microphone is shown [4].



Figure 3.4 : The Microphone

3.4.1.1 Types of Microphones

Microphones are divided into 3 types according to their response in the sound field. These are free field, pressure and random incidence microphone. Figure 3.5 shows the types of microphone [4].

Free field microphones have uniform frequency response for the sound pressure that existed before the microphone was introduced into the sound field. It is of importance to note that any microphone will disturb the sound field, but the free field microphone is designed to compensate for its own disturbing presence.

The pressure microphone is designed to have a uniform frequency response to the actual sound level present. When the pressure microphone is used for measurement in a free sound field, it should be oriented at a 90° angle to the direction of the sound propagation, so that the sound grazes the front of the microphone.

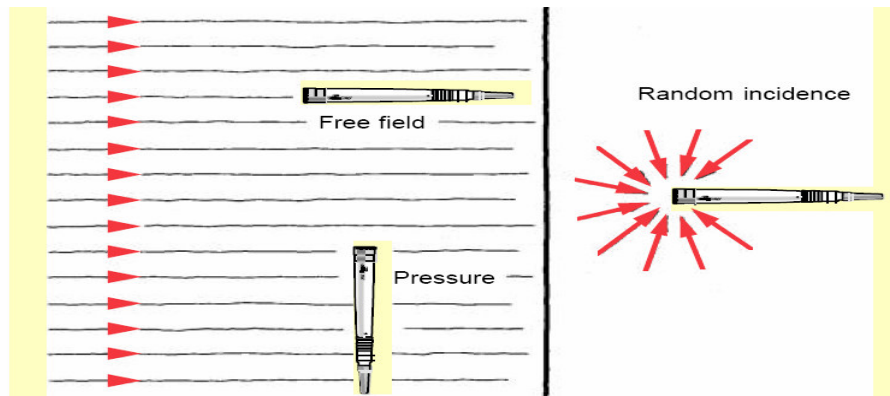


Figure 3.5 : Types of Microphones [4]

The random incidence microphone is designed to respond uniformly to signals arriving simultaneously from all angles. When used in a free field it should be oriented at an angle of $70^\circ - 80^\circ$ to the direction of propagation.

3.4.2 Response Transducers

3.4.2.1 Accelerometers

For response measurement, any of the motion parameters (displacement, velocity or acceleration) can be measured. Figure 3.6 shows the accelerometers.

The best choice of transducer is the piezoelectric accelerometer, for the following reasons:

- Good linearity
- Broad dynamic range (160 dB)
- Wide frequency range (0,2 Hz to over 10 kHz for better than 5% linearity)
- Simple mounting methods

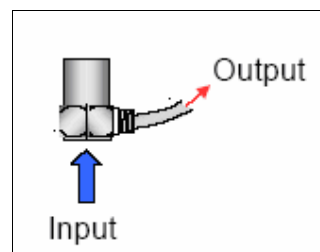


Figure 3.6 : Accelerometers

The velocity or displacement parameters can readily be obtained through electrical integration, either through a conditioning amplifier or using the post-processing facilities of the analyzer.

When a force is applied to a piezoelectric material in the direction of its polarization an electric charge is developed between its surfaces, giving rise to a potential difference on the output terminals. The charge (and voltage) is proportional to the force applied. The same phenomenon will occur if the force is applied to the material in the shear mode. Both modes are used in practical accelerometer design.

When the accelerometer is exposed to a constant level of acceleration it will give a constant output signal over a very wide frequency range up to frequencies near its resonance frequency. The sensitivity and frequency range of an accelerometer are related in general the bigger the accelerometer the higher its sensitivity and the smaller is its useful frequency range. In Figure 3.7, the relation between sensitivity and frequency range of an accelerometer is shown [12].

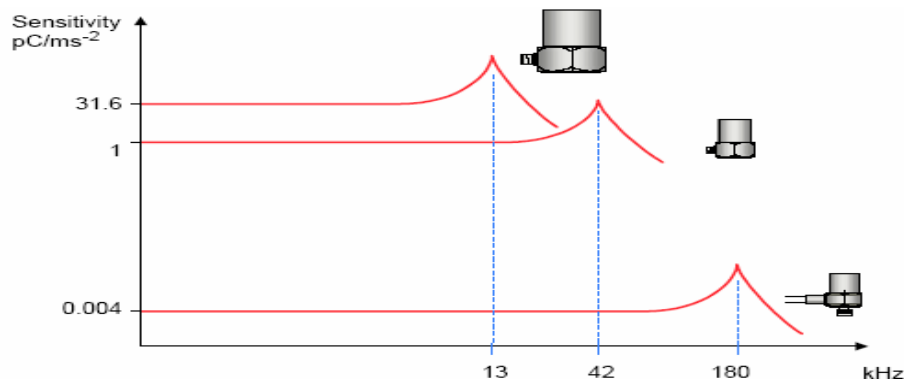


Figure 3.7 : Sensitivity and Frequency Range [12]

The accelerometer should be mounted so that the desired measuring direction coincides with the main sensitivity axis. Accelerometers are slightly sensitive to vibrations in the transverse direction, but this can normally be ignored as the maximum transverse sensitivity is typically only a few percent of the main axis sensitivity.

The reason for measuring vibration will normally dictate the position of the accelerometer. The accelerometer should be positioned to maintain a direct path for the vibration from the bearing.

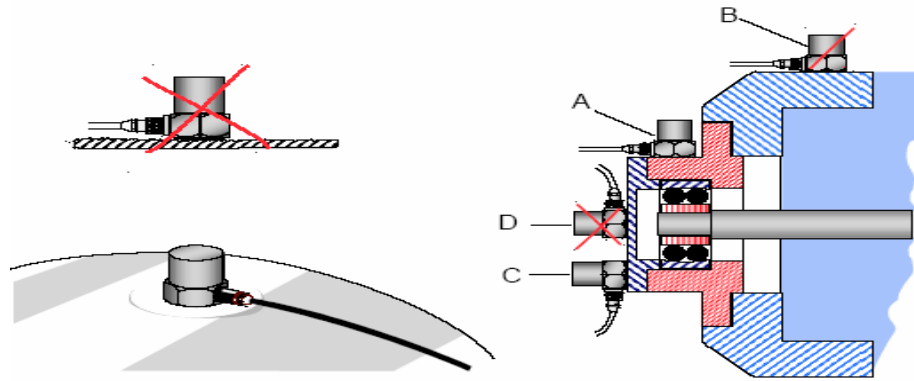


Figure 3.8 : Accelerometer Mounting Position [12]

In Figure 3.8, accelerometer “A” thus detects the vibration signal from the bearing predominant over vibrations from other parts of the machine, but accelerometer “B” receives the bearing vibration modified by transmission through a joint, mixed with signals from other parts of the machine. Likewise, accelerometer “C” is positioned in a more direct path than accelerometer “D” [12].

3.4.3 Force Transducer

Force transducers are used in mechanical-dynamics measurements together with accelerometers to determine the dynamic forces in a structure and the resulting vibratory motions. The parameters together describe the mechanical impedance of a structure.

By impacting or exciting a structure at different positions with an instrumented hammer and measuring the structural response, so called modal analysis can be made describing the total behavior of the structure as a system [12].

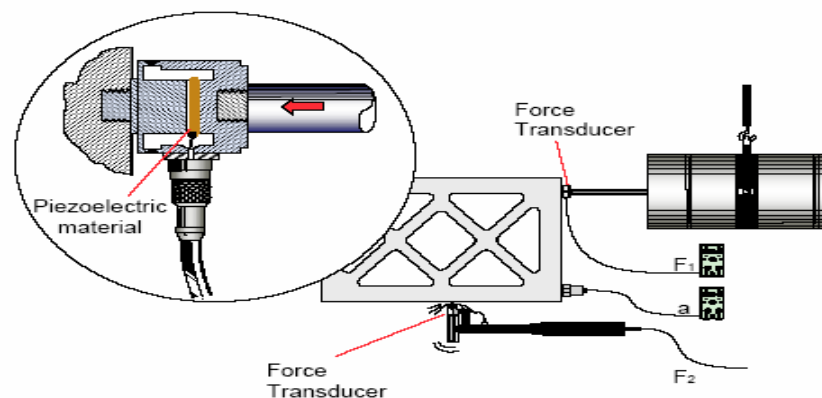


Figure 3.9 : The Force Transducer [12]

The force transducer also uses piezoelectric elements, but in this case the forces are directed directly to these elements. The instrumentation used with these transducers is identical to the instrumentation used with accelerometers. Figure 3.9 shows the force transducer.

Sometimes combined force transducers and accelerometers are used to measure the mechanical impedance of light structures.

3.4.4 Analysis System

3.4.4.1 Front-ends

Front-ends are multichannel data acquisition units for real-time measurements on more than 200 channels. Figure 3.10 shows the two types of front-ends. Signal and system analysis using all PULSE application for:

- Time data acquisition
- General noise and vibration measurements
- Basic and advance acoustics
- Structural Analysis
- Machine Diagnostics
- Electro acoustic testing



Figure 3.10 : Front-Ends **(a)** 5 Input Channels **(b)** 96 Input Channels

3.4.4.2 Analyzers

FFT and CPB analyzers provides general noise and vibration testing using real-time, multichannel analysis as well as general research and development, noise and vibration analysis using several analyzers and multiple frequency spans simultaneously. With user-definable measurement solutions, all basic requirements, including data acquisition, measurement, analysis, calibration, post-processing and reporting are convenient and manageable.

Display of functions in a range of graph types including:

Waterfall, Waterfall (step), Color contour, Bar, Line, Curve, Curve (step), Overlay, Overlay (all), Multi-value

Order analysis is a technique for analyzing imperfections in the moving parts of rotating and reciprocating machinery that cause unwanted noise and vibration. The engineer's or scientist's knowledge about machinery such as aircraft and automotive engines, power trains, pumps, compressors and electric motors is greatly improved by performing order analysis. This is because it allows measurements to be related to the revolutions of a rotating part [7].

In Run-up/down Acoustic Performance Test, projects are performed, for example, cabin noise or exhaust noise tests using octave analysis and orders versus engine RPM.

In Run-up/down Vibration Diagnostics, a number of projects for the determination of critical speeds, resonances and instabilities from measurements are performed with and without tracking.

In Run-up/down Tests without Tracking, to perform a run-up/down test without tracking, a tachometer and an FFT analyzer are used. The tachometer supplies triggers that allow FFT measurements to be made and stored in a multi-buffer when specified conditions relative to the change in speed of a shaft occur [7].

This is used for:

- Separation of rotational and structural noise and vibration phenomena
- Identification of noise generated by rotational vibrations
- Determination of critical speeds and resonances
- Investigation of instabilities in rotating machinery

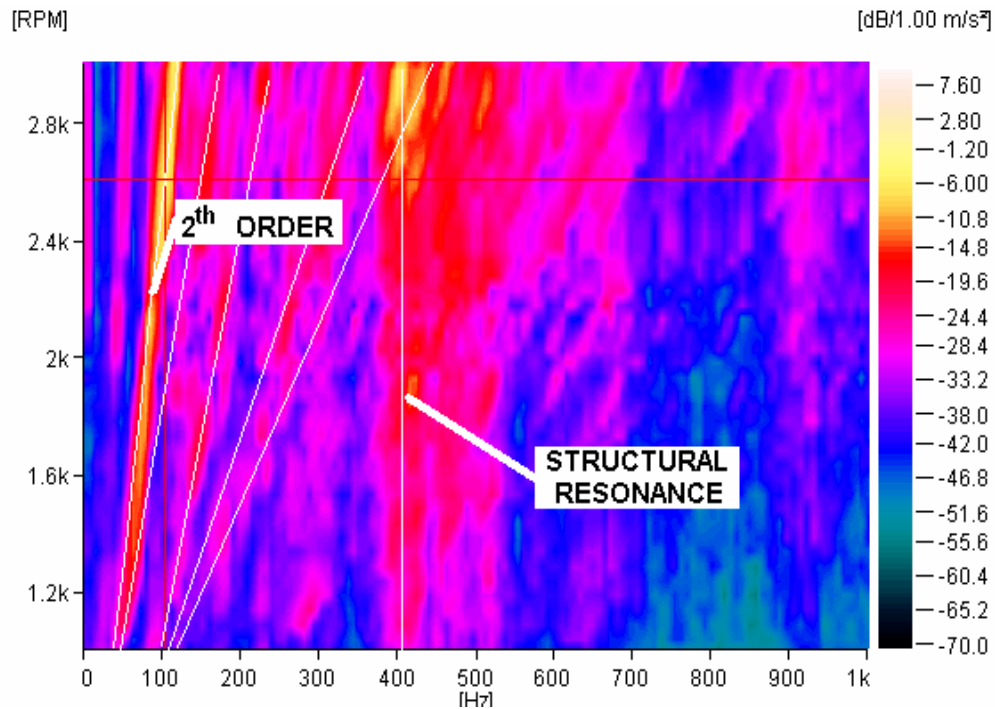


Figure 3.11 : The Contour Plot of Frequency Spectra versus RPM

A color contour plot of a run-up test has been performed without tracking. The line passing through the intersection of the X and Z cursors identifies the 2nd order. The harmonic cursor marks the first six orders for the extraction of order slices by making oblique cuts in the contour plot of frequency spectra versus RPM [7]. The contour plot of frequency spectra versus RPM is shown in Figure 3.11.

Displaying these stored FFT spectra allows the identification and extraction of orders and/or frequency bands (structural resonances). The amplitude and/or phase of the various orders as a function of RPM are obtained by cutting and extracting oblique slices from contour plots showing frequency spectra versus RPM.

Run-up/down tests without tracking are useful where frequency smearing is insignificant. An advantage is that it shows structural resonances at fixed frequencies parallel to the RPM axis.

For removal of jitter from a low-quality tacho signal, applying an averager. This provides a running average for the distance between tacho pulses. If the shaft under analysis in a machine is not the one supplying the tacho pulses, a tacho gearing can be defined as either a single factor, a ratio containing up to 4 fractions, or the product of these two methods [7].

3.5 The Sound Intensity

To measure sound intensity accurately using a two-microphone technique, a reliable sound intensity probe set containing a matched microphone pair to obtain information on both the instantaneous pressure and pressure gradient in the sound field.[15] The microphones are separated by a fixed distance in the sound field, and the microphone signals are fed to a sound intensity processor which calculates the sound intensity. The sound intensity is calculated from the time average of the sound pressure multiplied by the particle velocity (calculated from the measured pressure gradient). Such a system measures the component of the sound intensity along the probe axis and also indicates the direction of energy flow [17].

3.5.1 The Sound Intensity Probe

The sound intensity probe is constructed on a face-to-face design. It consists of a robust frame which holds the microphone preamplifier(s) and matched microphones in a face-to-face configuration. The sound intensity probe is shown in Figure 3.12.



Figure 3.12 : The Sound Intensity Probe [21]

The distance between the microphones is defined by solid plastic spacers which are held in place by threaded studs on the microphone grids. Sound is constrained to act on each microphone through a narrow slit between the spacer and the microphone grid. This arrangement gives well-defined acoustic separation of the microphones and minimizes shadow and reflection effects. The spacers used to separate the microphone pairs in the sound field are designed to give acoustic separations of 6 mm, 8.5 mm, 12mm and 50 mm. Their physical lengths in fact differ slightly from these values.

3.6 System Analysis with Run-up/Coast-down Test

Once the vibration source has been located and concentrated on the system. The properties of the transmission path, between the source and receiver, represent the inherent dynamic characteristics of the combined systems. A first step towards describing path properties is to make a run-up/coast-down test, during which the response (acceleration) is measured for different speeds. The response is then plotted against speed. This plot will give a qualitative indication of significant resonances in the operating frequency range, since excitation frequency is proportional to speed [10].

The run-up/coast-down technique can be extended to give three-dimensional plots. These can be plots of the spectrum vs. the speed (waterfall display), or the vibration level and frequency, for a number of harmonics, as a function of speed.

In the run-up/coast-down test, only the response to varying excitation frequency is measured, and the level of the excitation force varies without control. Our measurements can only therefore give coarse qualitative information about the system properties.

3.7 System and Signal Analysis

In most of the preceding slides it has been assumed that a vibration existed, generated in some way by forces present in the system itself. When this vibration signal is analyzed it is called Signal Analysis.

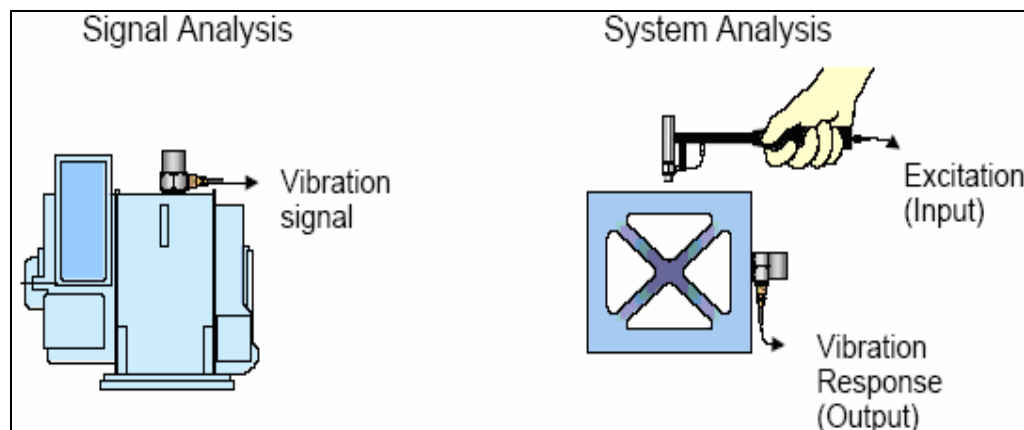


Figure 3.13 : System and Signal Analysis [25]

During development of new structures, and in some cases to analyze in detail existing structures, it is a requirement to try to make a model of the structure, in such a way that if input forces are given the output vibration can be calculated. These types of measurements are used to make a modal model of the structure, which can then be used to predict the behavior of the structure under given circumstances. The model can also be used to predict the effect of changes in the structure, especially if it is combined with Finite Element Modeling (FEM). This type of analysis is called System Analysis. Figure 3.13 shows the signal and system analysis.

3.7.1 The Frequency Response Function

One very efficient model of a linear system is a frequency domain model, where the output spectrum is expressed as the input spectrum weighted by a system descriptor.

$$X(\omega) = H(\omega) \times F(\omega) \quad (3.1)$$

This system descriptor $H(\omega)$ is called the Frequency Response Function (FRF), and is defined as:

$$H(\omega) \equiv X(\omega) / F(\omega) \quad (3.2)$$

It represents the complex ratio between output and input, as a function of frequency ω and the function has a magnitude $|H(\omega)|$ and a phase:

$$H(\omega) = \phi(\omega) \quad (3.3)$$

The physical interpretation of the FRF is that a sinusoidal input force, at a frequency ω , will produce a sinusoidal output motion at the same frequency. The output amplitude will be multiplied by $|H(\omega)|$, and the phase, between output and input, will be shifted by $H(\omega)$. Any input/output spectrum can be considered to be the sum of sinusoids. The FRF describes the dynamic properties of a system independent of the signal type used for the measurement. The FRF is therefore equally applicable to harmonic, transient and random excitation [10].

The definition of the FRF means that, in measuring a specific function, the measurements can be made sequentially at discrete frequencies or simultaneously

at several frequencies. A useful technique is to use a wide frequency bandwidth for the excitation force. This gives a dramatic reduction in measurement time, as compared to sinusoidal excitation where one frequency is measured at a time.

3.7.2 The Coherence Function

The bounds for the Coherence Function are 1, for no noise in the measurements, and 0 for pure noise in the measurements. The interpretation of the Coherence Function is that for each frequency ω it shows the degree of linear relationship between the measured input and output signals. The Coherence Function is analogous to the squared correlation coefficient used in statistics. When making mobility measurements, using this powerful property of the Coherence Function to detect a number of possible errors [10].

3.8 Hammer Test

3.8.1 Excitation

For mobility measurements the structure must be excited by a measurable dynamic force, but there is no theoretical restriction as to waveform, or to how the excitation is implemented.

3.8.1.1 Impact Excitation

The most popular excitation technique used for modal analysis is impact, or hammer excitation. The waveform produced by an impact is a transient (short duration) energy transfer event. The spectrum is continuous, with a maximum amplitude at 0 Hz and decaying amplitude with increasing frequency.

The spectrum has a periodic structure with zero force at frequencies at n/T intervals, where n is an integer and T is the effective duration of the transient. The useful frequency range is from 0 Hz to a frequency , at which point the spectrum magnitude has decayed by 10 to 20 dB. The duration, and thus the shape of the spectrum, of an impact is determined by the mass and stiffness of both the impactor and the structure. For a relatively small hammer used on a hard structure, the stiffness of the hammer tip determines the spectrum. The hammer tip acts as a mechanical filter [10].

3.8.2 Impact Testing and the Coherence Function

The deterministic character of impact excitation limits the use of the Coherence Function. The Coherence Function will show a perfect value of 1 unless:

There is antiresonance, where the signal-to-noise ratio is rather poor. No particular attention needs to be paid to this. Taking a number of averages should make the FRF curve smooth.

The person conducting the test impacts the structure in a scattered way, with respect to point and direction. This should be minimized so that the Coherence is higher than 0, 95 at the resonances. If the impact point is close to a node point the Coherence may be extremely low ($\approx 0, 1$). This is acceptable however, since the modal strength at this point is weak, and not important for the analysis [10].

3.8.3 Double hits

If the hammer is too heavy, the structure may rebound at the hammer producing double impacts. The occurrence of double hits also depends on the skill of the experimentalist. A double hit cannot be used since the spectrum will contain zeros with a spacing of n/t_r , where n is an integer and t_r is the time delay between the dual impacts. Double hits cannot be compensated for by using the transient window, and any Frequency Response Functions measured with a double hit will be erroneous and must be excluded from the data set.

3.8.4 Impact Hammers

Impact Hammers have been designed to excite and measure impact forces on small to medium structures such as engine blocks, car frames and automotive components. An accelerometer or laser velocity transducer) is used to measure the response of the structure. By using a multichannel FFT analyzer, such as the PULSE system, the frequency response function and mode shapes of the test structure can then be derived. Contrary to using an electrodynamic exciter, an impact hammer does not apply additional mass loading to the test object and it provides a very portable solution for excitation. Figure 3.14 shows the impact hammer [22].



Figure 3.14 : The Impact Hammer [22]

The impact hammer is supplied with three interchangeable impact tips of aluminum, plastic and rubber. The choice of impact tip determines the impulse shape (amplitude and duration) and the bandwidth of the excitation.

In Figure 3.15, impulse shapes for the hammer tips as a function of time showing in the pulse decay and peak value; Force spectrum of an impact on an aluminum plate.

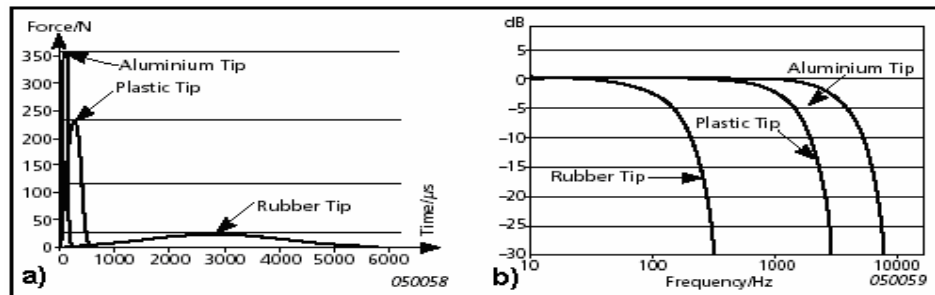


Figure 3.15 : Hammer Tips (a) The Pulse Decay and Peak Value (b) Force Spectrum [13]

3.9 Shaker Test

3.9.1 Implementing the Excitation

Excitation forces can be generated by many different kinds of devices. There are two types of exciters for broadband excitation. These are attached and non-attached exciters.

Examples of attached exciters are:

- Electromagnetic shakers
- Electrohydraulic shakers
- Eccentric rotating masses
- More exotic devices such as rockets or guns

Acoustic excitation cannot be used in modal analysis, since control of direction and excitation point is not possible. It can be used however, for checking modal frequencies, and for producing unsealed mode shapes. Figure 3.16 shows the electrodynamic exciter [10].

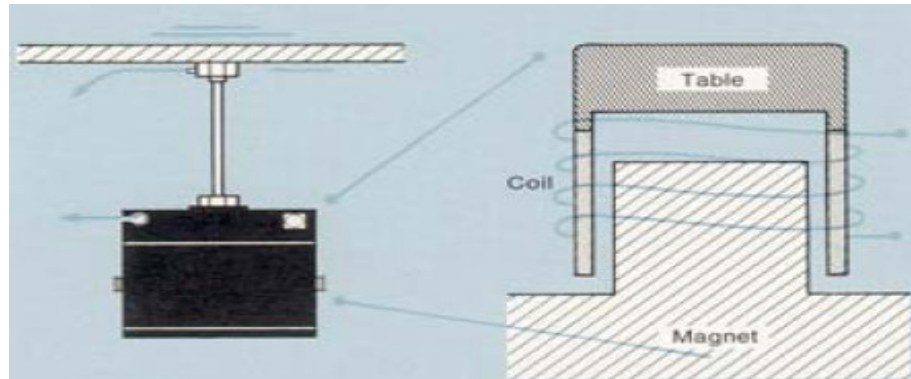


Figure 3.16 : Electrodynamic Exciter [10]

3.9.2 The Pseudo-Random Excitation

The pseudo-random waveform is a periodic signal that repeats itself with every record of the analysis. A single time record resembles a random waveform, with a Gaussian-like amplitude distribution. The spectral properties however are very different. Because the signal repeats itself with each record, or is periodic with a period equal to the record length, something dramatic happens to its spectrum. Figure 3.18 shows the pseudorandom waveform [10].

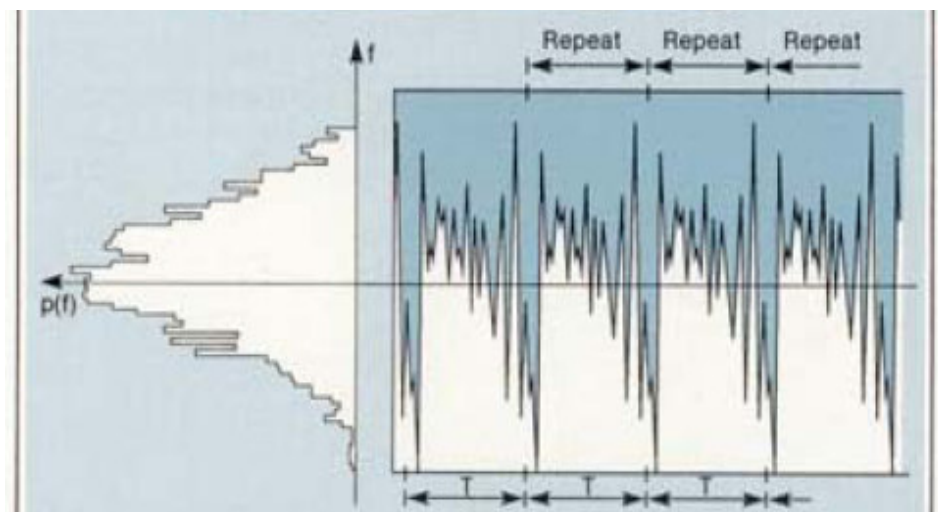


Figure 3.17 : The Pseudo-Random Waveform

3.9.3 Exciter Attachment

The exciter must be attached to the structure so that the excitation force acts only at the desired point, and in the desired direction. The structure must be free to vibrate in the other five degrees of freedom at that point, with no rotational or transverse constraints.

A good attachment technique is to connect the exciter to the force transducer with a slim push rod or stinger, as shown in Figure 3.18. This type of attachment has high axial stiffness but low transverse and rotational stiffness, giving good directional control of the excitation. An additional benefit is that the stinger acts as a mechanical fuse between the structure and the exciter, protecting both them and the transducer from destructive overloads [10].

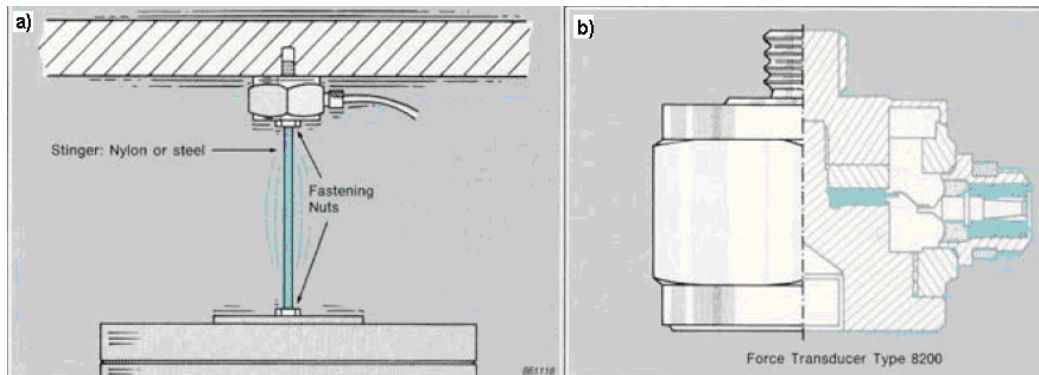


Figure 3.18 : Attachment Technique of Force Transducer with Stinger and Piezoelectric Force Transducer [10]

3.9.4 Triaxial Accelerometers

Normally, triaxial accelerometers consist of three individual accelerometers mounted in a single housing and positioned so that vibration can be measured in three mutually perpendicular directions.

This approach sets limits to reducing the size of the accelerometer and also means that the three axes have different points of reference. The OrthoShear design is built around a common seismic mass. This uni-mass design results in a very compact triaxial accelerometer where all the axes have the same point of reference. The design also ensures accurate and consistent measurements, even when the accelerometer is exposed to complex vibration patterns. The seismic mass is surrounded by a piezoelectric ring which is surrounded by four individually suspended, curved plates. Because of the suspension pins, different

sections are exposed to shear forces for different directions of acceleration. By appropriate summation of the signals, the outputs for the X, Y and Z axes are obtained. In the Figure 3.19, the assembly is clamped together by the outer ring. The preamplifiers, suspension pins and Microtech compatible connector constitute an integral part which is hermetically welded to the titanium housing. The risk of ground loops, which can be particularly troublesome in multichannel measurements, is therefore reduced considerably. The accelerometers are specifically designed for the automotive industry [16].

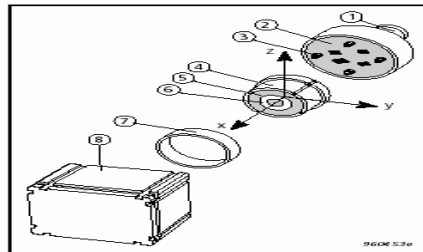


Figure 3.19 : The Assembly in a Single Housing

3.9.4.1 Triaxial accelerometers mounting

Special effort has been put into making mounting as flexible as possible. The accelerometer housing has slots that allow the use of mounting clips so that the accelerometers can be easily fitted to a number of different test objects, or removed, for example, for calibration. The mounting clips are glued to the object, or fitted with double sided, adhesive tape. A mounting clip with thick base is also available and can be filed down to suit your mounting surface. A mounting clip with swivel base is a third option. This makes it easy to align the accelerometer in order to retain the coordinate system. The temperature range of mounting clips is from -54° to $+50^{\circ}\text{C}$. Their material is glass reinforced polycarbonate. The plastic mounting clips are shown in Figure 3.20.

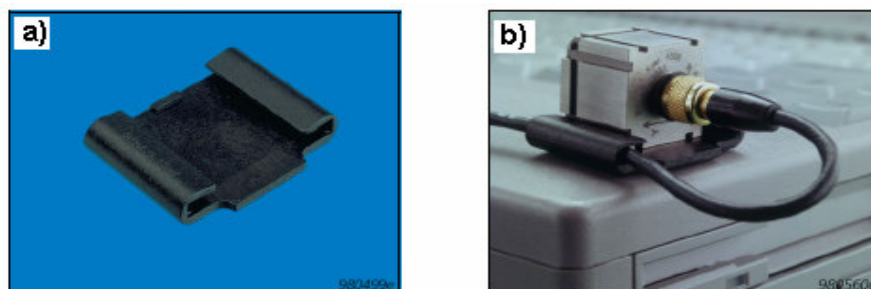


Figure 3.20 : Plastic Mounting Clips [16]

3.9.5 The Modal Exciter

The modal exciter, which is used to determine the natural frequencies and modes, excites the structure. According to the signal which is modified from the power amplifier, the structure or component is excited by a stringer. The force is measured by the force transducer. In the transverse directions and in torsion, the flexure system provides very high stiffness to counteract rotational movement of the test specimen. Also, through this configuration, the modal exciter can absorb high lateral forces without damage to the exciter's internal construction. The modal exciter is shown in Figure 3.21.



Figure 3.21 : Modal Exciter [14]

The hole-through design makes it possible to use tension wire stingers or traditional push/pull stingers with the exciter. Easy and fast attachments of both types of stingers are achieved with the chuck nut assembly (for use with tension wire stingers). Force rating is 100 N sine and through-hole design for choice of tension wire stingers or traditional stingers.

3.9.6 Power Amplifier

The power amplifier shown in the Figure 3.22 drives any modal or vibration exciter. It modifies the signal. Its features are:

- Low or high output impedance
- Low distortion over wide frequency range



Figure 3.22 : Power Amplifier [24]

4. SOURCE PATH CONTRIBUTION ANALYSIS

4.1 The SPC Technique

The basis of the SPC technique is to perform a phased summation of partial responses from all noise and vibration paths to give total tactile and acoustic responses under specific operating loads at a given frequency or RPM. The system allows to rate noise and vibration contributions using NVH analysis methods, such as the mount stiffness or the impedance matrix method for structure-generated phenomena and the source substitution method for airborne phenomena. Figure 4.1 shows the noise and vibration sources of a vehicle [18].

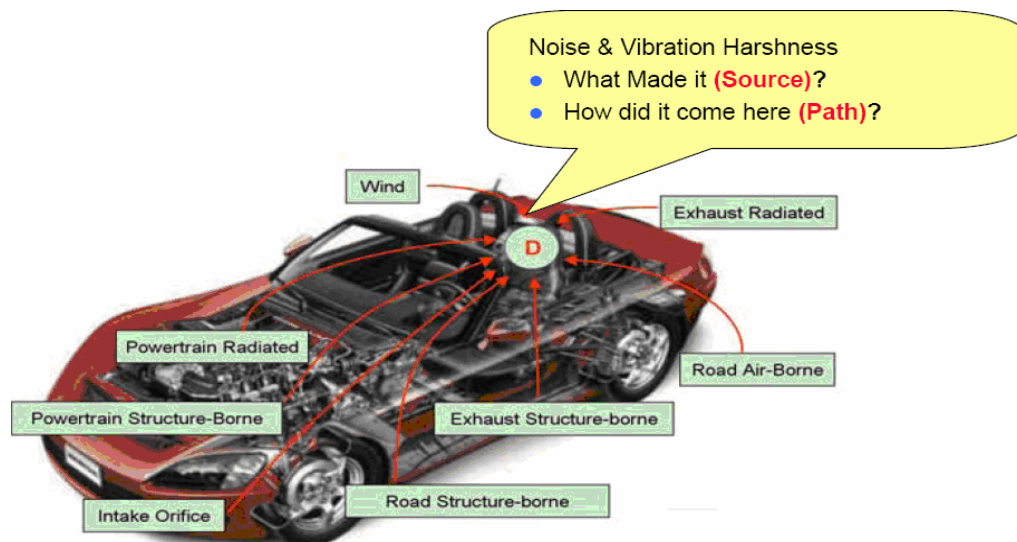


Figure 4.1 : Noise and Vibration Sources [19]

Source path contribution technique uses for:

- Perform standard source path contribution analysis (Transfer functions and operating conditions) for engine, powertrain or complete driveline assessment
- Combine vibro-acoustic path sensitivities and operating conditions to determine contributions

- Determine contributions in the frequency and order domains
- Contribution ranking
- Management of sources, paths and receivers in a structured model

4.2 Powertrain Noise Cascading

The main causes of the interior noise are wind noise, powertrain noise and road noise. The powertrain noise radiates structural and air borne noise. Structural noise depends on chassis dynamics, powertrain forces and the body sensitivity. Powertrain forces are the source of powertrain vibration. This vibration is related to the mounts transmission. This cascading is shown in Figure 4.2.

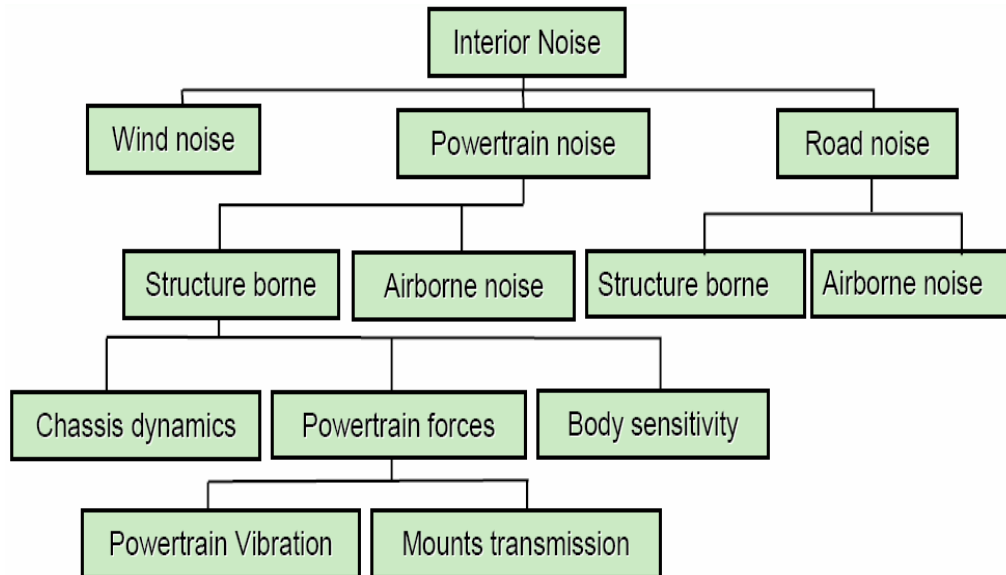


Figure 4.2 : Powertrain Noise Cascading

4.3 The Volume Velocity Source

The volume velocity source is used for measuring the output volume velocity and transfer function measurements. The adaptor contains two phase-matched microphones. Figure 4.3 shows the volume velocity source.



Figure 4.3 : The Volume Velocity Source

4.4 Creating a Model

Noise and vibration are transmitted through various paths of the automobile both structure-borne and airborne. Source Path Contribution (SPC) is a method for decomposing, evaluation and rating vehicle interior noise into contributing sources and transfer paths. A source path model is shown in Figure 4.4. Related methods:

- Transfer Path Analysis
- Noise Path Analysis

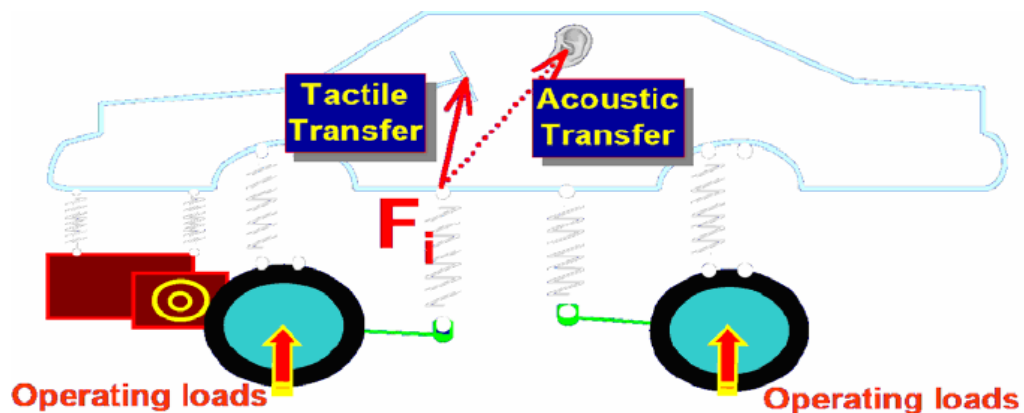


Figure 4.4 : Source Path Model [19]

4.4.1 Source Path Receiver Model

The SPC is built around a cascading Source Path Receiver (SPR) model. This model specifies all the sources, paths and receivers involved in the NVH test of a single vehicle or vehicle component. The number of source, receiver and indicator points, the analysis methods, type of data required, and relevant operating conditions are all determined.

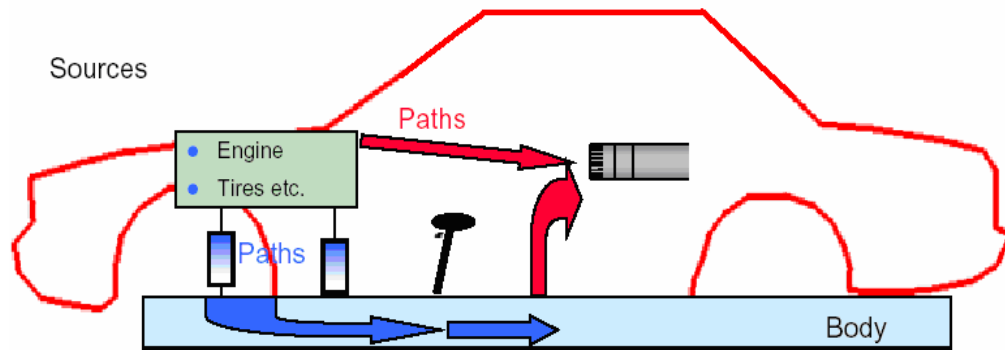


Figure 4.5 : Sources and Paths

Air borne energy is radiated from the acoustic source through air to a receiver and the structure borne energy is the vibration through solid (via air) to a receiver.

The receivers at the vehicle are sound pressure microphone at the driver's position and the vibration at the gear shift etc. Sources and paths are shown in Figure 4.5.

SPR models quantify the individual path contributions from all of the sources being important for the sound or vibration perceived by occupants in the vehicle. Sources, paths and receivers of a structured model are shown in Figure 4.6.

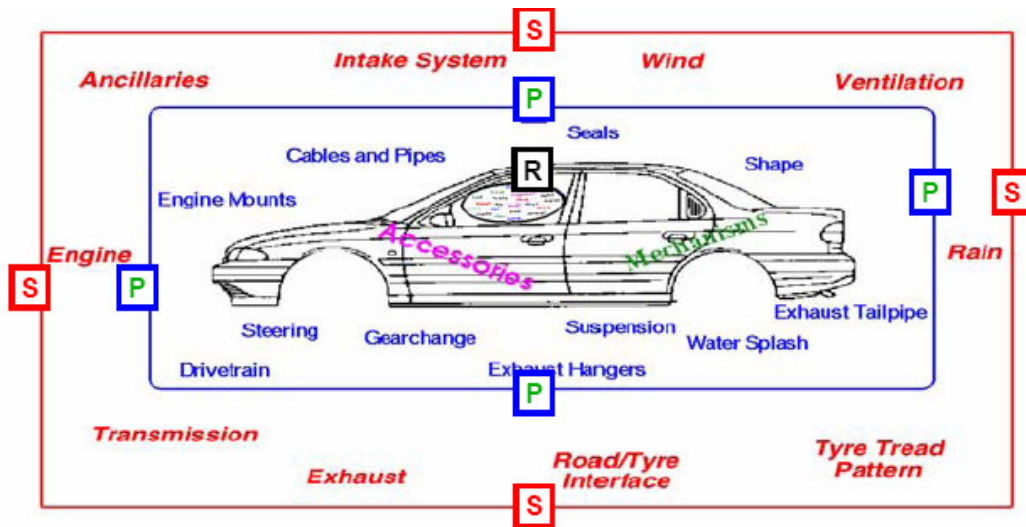


Figure 4.6 : Sources, Paths and Receivers in a Structured Model [19]

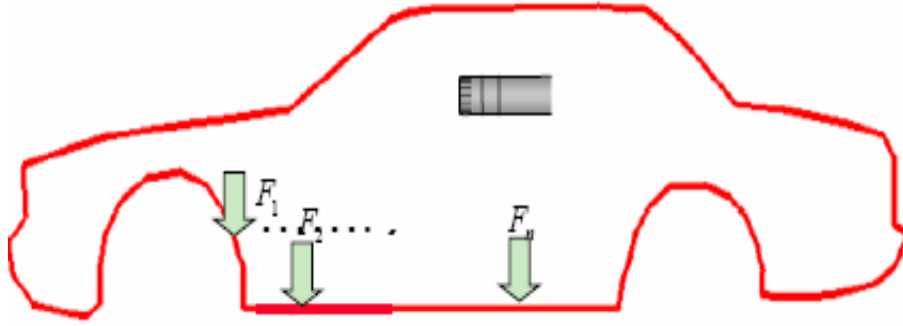


Figure 4.7 : Forces on Passive Side of Mounts

Total sound pressure level in the cabin can be calculated by the equation (4.1). F_n is the force on passive side of mounts and the H is the frequency response function from body to receiver (noise transfer function). The forces on passive side of mounts are shown in Figure 4.7.

$$P_{total} = \sum_{n=1}^n H_n^v \times F_n \quad (4.1)$$

4.4.2 Determination of the Force

Forces on the mounts are determined by the Mount Stiffness method. In this method, the complex stiffness is experimentally determined by measuring force and displacement and the force is calculated by using complex stiffness and operational displacement [19,23]. Figure 4.8 shows the mount stiffness method.

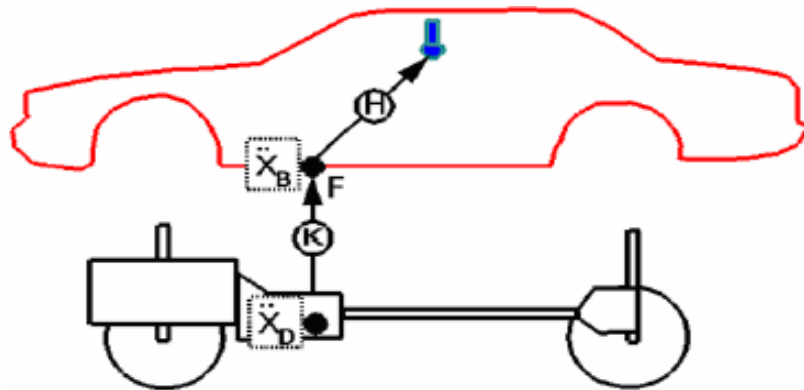


Figure 4.8 : Mount Stiffness Method

4.4.3 Airborne Reciprocity

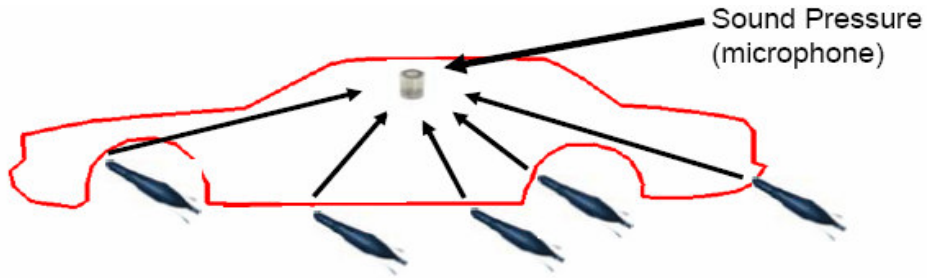


Figure 4.9 : Sound Pressure Microphone in Cabin [19]

Airborne source contributions can be measured reciprocally. In Figure 4.9, sound pressure microphone and the volume velocity source are located. H is the noise transfer function, Q is the volume velocity and the P is the sound pressure at the driver ear position. In Figure 4.10, the volume velocity source is inside the cabin and the microphones are located.

Airborne Contribution can be calculated by the formula (4.2).

$$P_{cabin} = \sum_i Q_i H_{icabin} \quad (4.2)$$

Reciprocal Acoustic Frequency Response Function is the equation (5.3).

$$H = \frac{P_{cabin}}{Q_i} = \frac{P_i}{Q_{cabin}} \quad (4.3)$$

Noise transfer functions also known as body sensitivities or P/Fs frequency response functions.

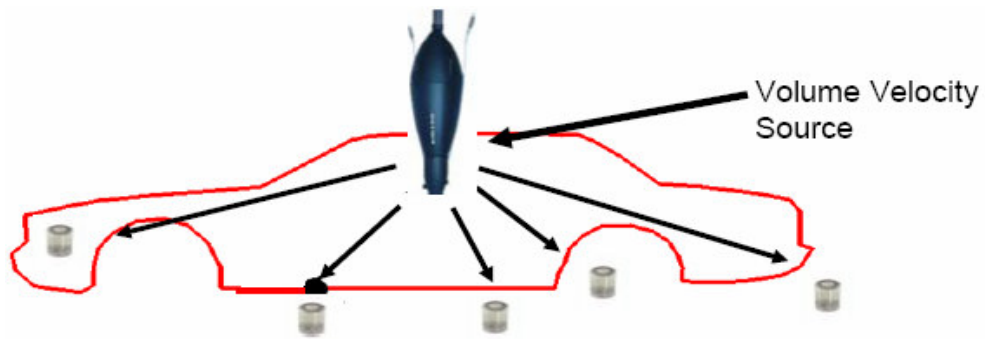


Figure 4.10 : Volume Velocity Source in Cabin with Using Microphones [19]

In Figure 4.11, accelerometers are used to measure response of FRF's for vibration (velocity).

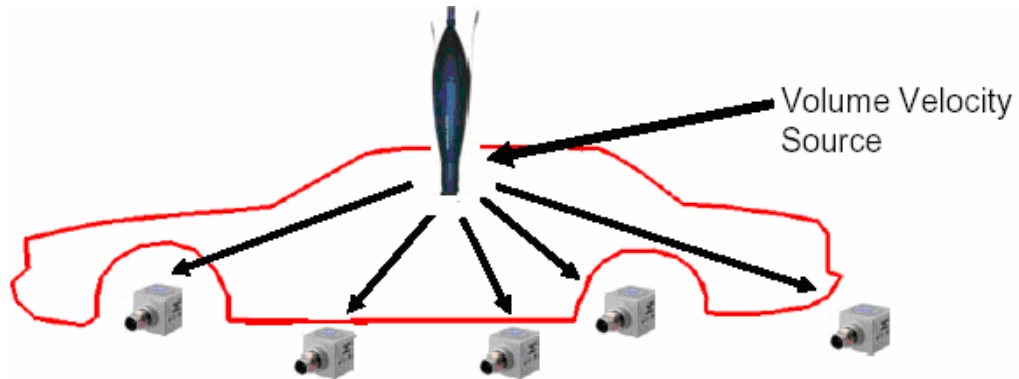


Figure 4.11 : Volume Velocity Source in Cabin with Using Accelerometers [19]

$$V_x = \sum_i Q_i H_{cabin.ix} \quad (4.4)$$

4.5 Contribution Analysis

During contribution analysis, input contributions to overall vehicle interior noise or vibration are evaluated and compared to total or measured values and deviations from set targets can be identified. Modifications to the vehicle are done here, and reassessment of noise and vibration contributions easily performed. Results can be manipulated to account for different loads, operating conditions, or other parameters for benchmarking and determining vehicle design alternatives.[18]

A test based technique that traces the flow of vibro-acoustic energy from multiple sources to a receiver. Applying accelerometers to each side of the body mounts and ran the vehicle on a dynamometer with a rough road profile at the same speed that the boom had been reported.

5. CHARACTERISTICS OF BOOMING NOISE IN A VEHICLE

5.1 Booming Noise

Booming noise is felt as a pressure in ear and its origin is often unknown. It increases in volume with the vehicle speed. It occurs within the relatively narrow vehicle speed range centered around 10 km/h, or around 50 RPM if associated with engine speed.

When climbing a high peak or driving through a tunnel at high speeds, rapid changes in the atmospheric pressure apply pressure to eardrums. This may cause discomfort in ears. Booming noise affect the driver as described above.

One particularity of four-cylinder internal combustion engines is that they vibrate markedly at frequencies corresponding to the second and fourth harmonics of the engine rotation speed. The resulting structural exctation is transferred to the chassis of the car; mostly via the engine mounts but also via the exhaust system [17].

If an engine speed harmonic coincides with a cavity resonance frequency in the passenger compartment, passengers will perceive an unpleasant dull booming sound in most of the compartment.

The paths along which acoustic vibrations can be transferred within the vehicle are many and complex. Usually, the engine is fixed to the chassis structure through three engine mounts. Each mount can transfer vibration to the chassis along three axes: x, y and z. The net acoustic signal observed at a given point in the passenger compartment (near the driver's ear) will be influenced by the vibrations transferred by each of the three axes. Each resultant excitation gives us a noise\force transfer function, which varies with the frequency [21].

The major noise and vibration sources in a vehicle are road surface, tire wheel, powertrain system and the brake system. The powertrain system is the main source of many noise and vibration problems. One of these problems can be stated as booming noise. Its frequency range is around 100 Hz. The frequency range of major noise and vibration systems, and the low speed, medium speed and high speed booming noise appears in the Figure 5.1.

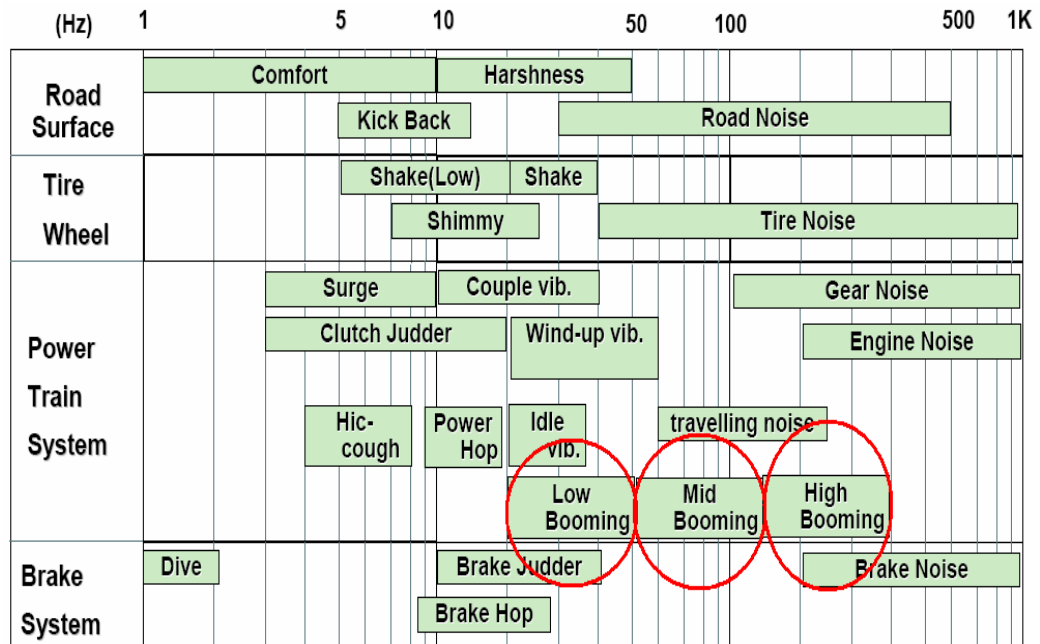


Figure 5.1 : The Frequency Range of Major Vibration and Noise Systems [19]

Main causes of booming noise are:

- Out-of-tune engine
- Inertia due to reciprocating motion of pistons, imbalance in engine
- Unbalanced propeller shaft
- Incorrect propeller shaft joint angle
- Resonances in the exhaust pipe
- Exhaust noise
- Resonances in auxiliary in engine components
- Vibration due to torsional stresses on the propeller shaft and drive shaft

Table 5.1 : Types of Booming Noise [19]

Phenomenon		Frequency Band	Major Sources	Resonance Systems
Booming	Low	30 – 60 Hz (- 50 kph)	- Engine torque variation	- Room light of interior - Chassis resonance
	Mid	60 – 100 Hz (50-80 kph)	- Engine torque variation - Penetration of air-intake	- Torsion of power-train - Bending of Power-train - Exhaust resonance - Membrane vibration of chassis panel - Beating of interior
	High	100–200 Hz (80 kph -)	- Engine torque variation - Penetration of air-intake - Tire unbalance	- Torsion of power-train - Bending of Power-train - Exhaust resonance - Membrane vibration of chassis panel - Beating of interior - Engine auxiliary resonance

There are three types of booming noise. These are at low speed booming noise, at medium speed booming noise and high speed boomig noise. Figure 5.2 shows the types of booming noise.

At low speed (below 50kph), the frequency band is 30 – 60Hz, major source is engine torque variation, and the major causes are the resonance between room interior and chassis.

At mid speed (50 – 80kph), the frequency band is 60 – 100Hz, major sources are engine torque variation and penetration of air intake system. It is caused by the torsion and bending vibration of powertrain, exhaust system resonance, membrane vibration of chassis panel, and beating of cabin interior.

At high speed (above 80kph), the frequency band is 100 – 200Hz, major sources are engine torque variation, penetration of air intake system, and tire unbalance. It is caused by engine auxiliary resonance.

5.2 Mechanism of Booming Noise

Booming noise can be investigated in three parts; vibration, transfer and radiation. In the first part, engine vibrations create inertia forces and torque variations. In transfer part, bending of powertrain transfers the vibrations to the body via the

engine mounts. In the radiation part, booming noise is caused by the body panel vibrations and the cavity resonances. Figure 5.3 shows the mechanism of booming noise.

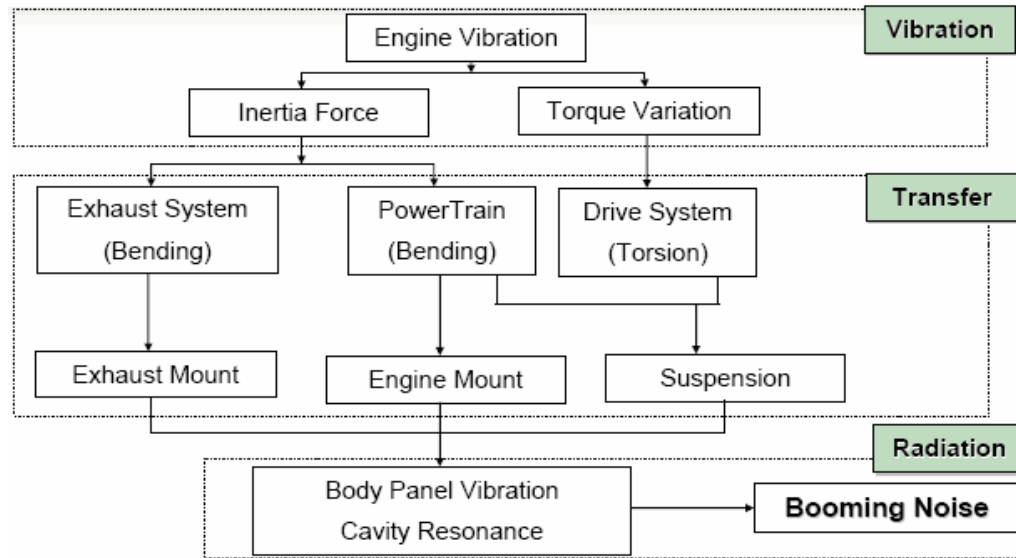


Figure 5.2 : Mechanism of Booming Noise

5.2.1 Mechanics Of Noise Development

5.2.1.1 Propeller shaft joint angle

When a joint angle exists in the propeller shaft, there are two torque fluctuations for every revolution of the propeller shaft. These fluctuations become larger as the joint angle increases.

At certain vehicle speeds these torque fluctuations vibrate the drive train, and are transmitted through the rear suspension arm bushings or springs, causing the body panels to vibrate. This results in the body booming noise [20].

5.2.1.2 Out-of-balance propeller shaft

When the propeller shaft is out of balance, the centrifugal force created by this imbalance attempts to cause the ends of the propeller shaft to bend outward and revolve in large circles around the centerline of the shaft. This does not actually happen because the propeller shaft is fixed at both ends. However, it does cause the shaft to vibrate once each time it rotates.

This vibratory force is further transmitted through the engine rear mounts, center bearing for the propeller shaft, rear suspension bushings, and to the body panels. The body panels then vibrate, generating the body booming noise [20].

5.2.1.3 Exhaust pipe vibrations

The exhaust pipe, being long and narrow, is easily vibrated. Another important factor which causes it to vibrate vigorously is that it is attached to the engine, the largest source of vibrations in a vehicle.

When the exhaust pipe resonates with engine vibration, the vibration is further amplified and transmitted by way of the O-rings and muffler clamps to the vehicle body, causing body booming noise.

5.2.1.4 Vibration of auxiliary components

If the mounting brackets for alternator, power steering pump, or air conditioner compressor are made of material that is not sufficiently rigid, they will resonate with engine vibration. The vibration will then be transmitted through the engine mounts to the body, creating body booming noise [20].

6. EXPERIMENTAL CASE STUDY

In the sixth part of this thesis, several tests and measurements are made to identify the noise and vibration problems of a commercial vehicle. There is an unpleasant noise in the commercial vehicle cabin which gives clues of booming noise. The interior noise reaches to peak values at most commonly used engine speed ranges like 2500-3000 RPM. A number of projects for the determination of critical speeds, resonances and instabilities have been performed.

The measurements for identifying the interior noise problem are:

- Run-up acoustic performance test; for performing cabin noise tests using octave analysis and orders versus engine RPM.
- Run-up vibration diagnostics test; identification of structural noise and noise generated by vibrations, determination of critical speeds and resonances.
- Hammer test
- Sound intensity
- Engine mounts performance test
- Shaker test

Experiments have been done in Istanbul Technical University Automotive Department Laboratory Semi-Anechoic Room.

6.1 Run-up Acoustic Performance Test

The aim of the run-up acoustic performance test is to identify the interior noise problem. In the cabin noise test, order, octave and frequency analyses are performed according to the engine speeds in RPM.

6.1.1 Test Information

In the run-up acoustic performance test, engine speed is increased to 3500 RPM from 1000 RPM and the gearbox is in idle position. Several experiments have been made in different gear positions in which similar characteristics are obtained.

Because of this situation, the main measurements have been made in the idle position of gearbox while windows have been closed.

Test equipments are:

- B&K PULSE Type 3560 E Multichannel Data Acquisition Unit
- B&K 4189 Microphones
- B&K Tachometer
- Laptop (with Pulse 9.0)

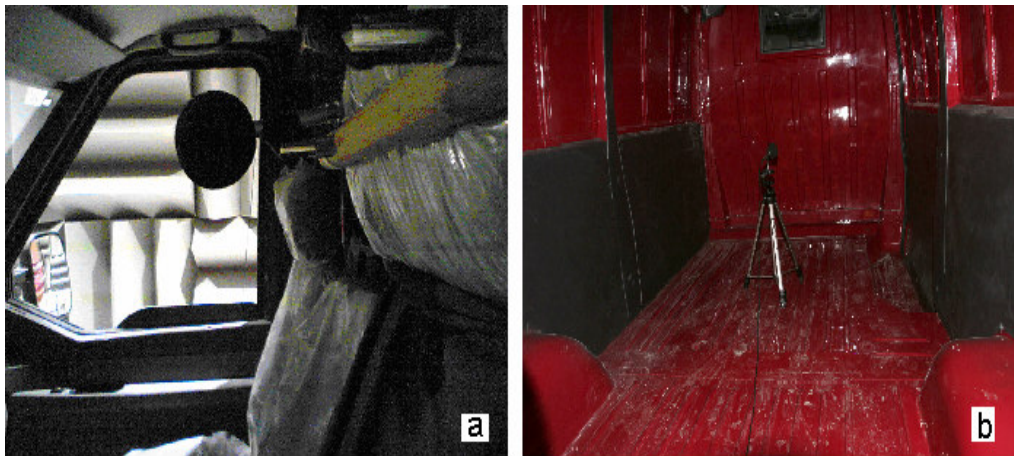


Figure 6.1 : Microphone Position (a) Driver Ear Position (b) Backside of the Vehicle

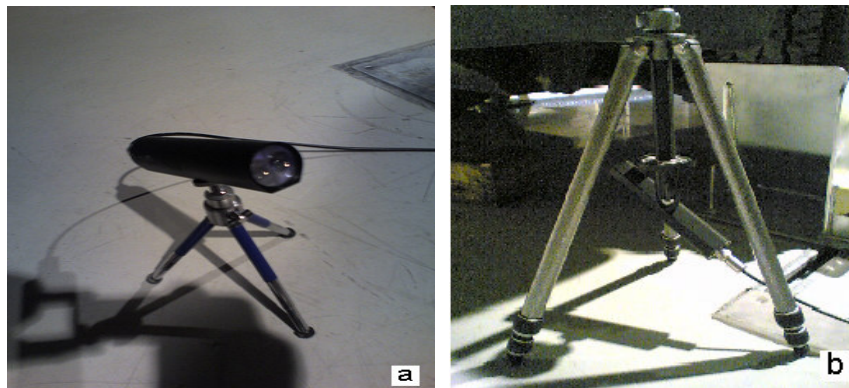


Figure 6.2 : Tachometer to Gather Engine Speed

Microphones have been located at driver right ear position and the backside of the vehicle as seen in Figure 6.1. The engine speed is obtained by the tachometer placed under the engine belt which is shown in Figure 6.2.

6.1.2 SPL at Driver Ear Position and the Backside of The Vehicle



Figure 6.3 : SPL at Driver Ear Position

As shown in Figure 6.3, while the horizontal axis represents the engine speed, the vertical axis shows the sound pressure level. On the chassis dynamometer engine is speed up to 3500 RPM and sound pressure level is measured. The peak value is 81.9 dB(A) which is obtained at 2891 RPM.

The analyzer is Constant Percentage Bandwidth Analyzer (1/3 octave band) and signal is the microphone at the driver ear position. Contour type graphical representation can be seen in Figure 6.4. This type of graphic contains three axes which show SPL, engine speed and the frequency. The horizontal axis represents the frequency, the vertical axis shows the engine speed and the third axis shows the sound pressure level with the colored scale. In this scale, sound pressure level is increasing when the colors turn dark to light. This means that the maximum SPL is shown with the lightest color which is yellow in this graphic. The horizontal axis is in the logarithmic scale which helps us to demonstrate wide range of frequencies especially we are concentrated on. Because of this logarithmic scale, the orders are parabolic curves.

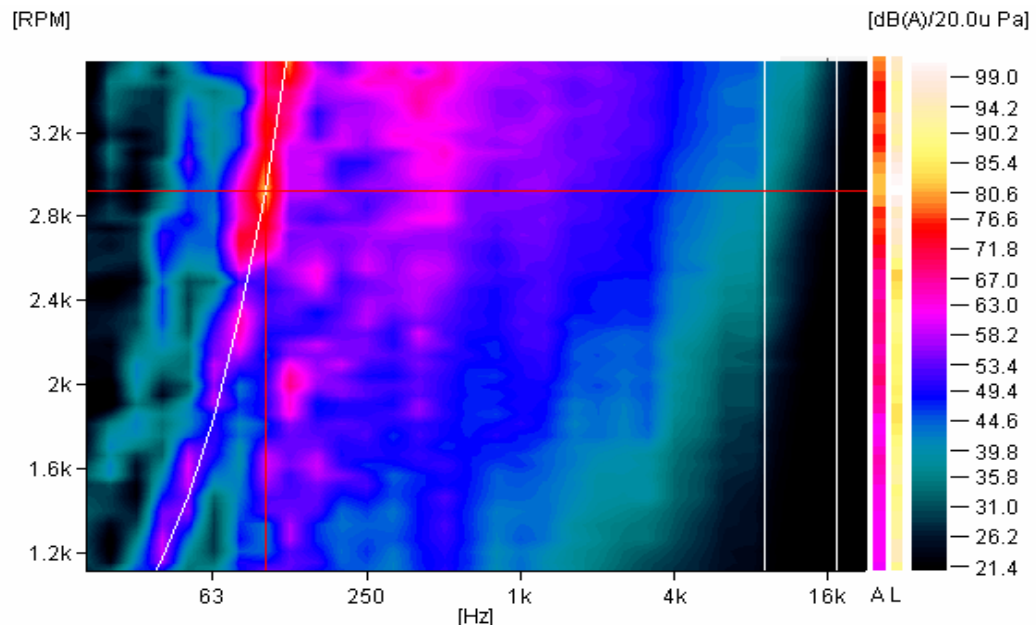


Figure 6.4 : CPB Buffer (1/3 octave band) - Driver Ear Position

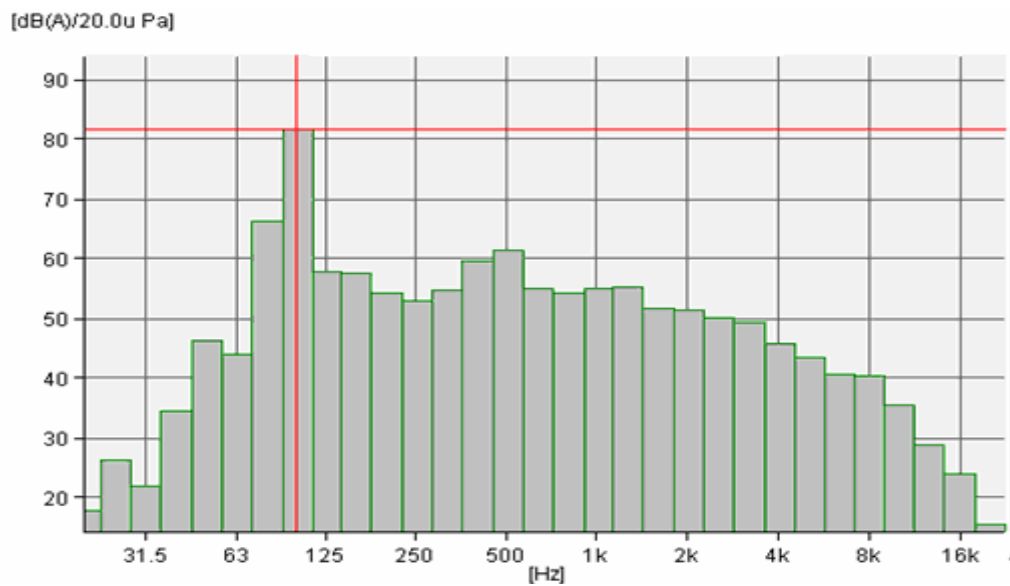


Figure 6.5 : CPB Autospectrum - Driver Ear Position

When the graphic is interpreted, the maximum sound pressure level is represented by the lightest yellow color in the second order around 100 Hz and at the 2900 RPM. In addition to this, maximum sound pressure level can be seen in Figure 6.5, the CPB Autospectrum, having the same cursor values.

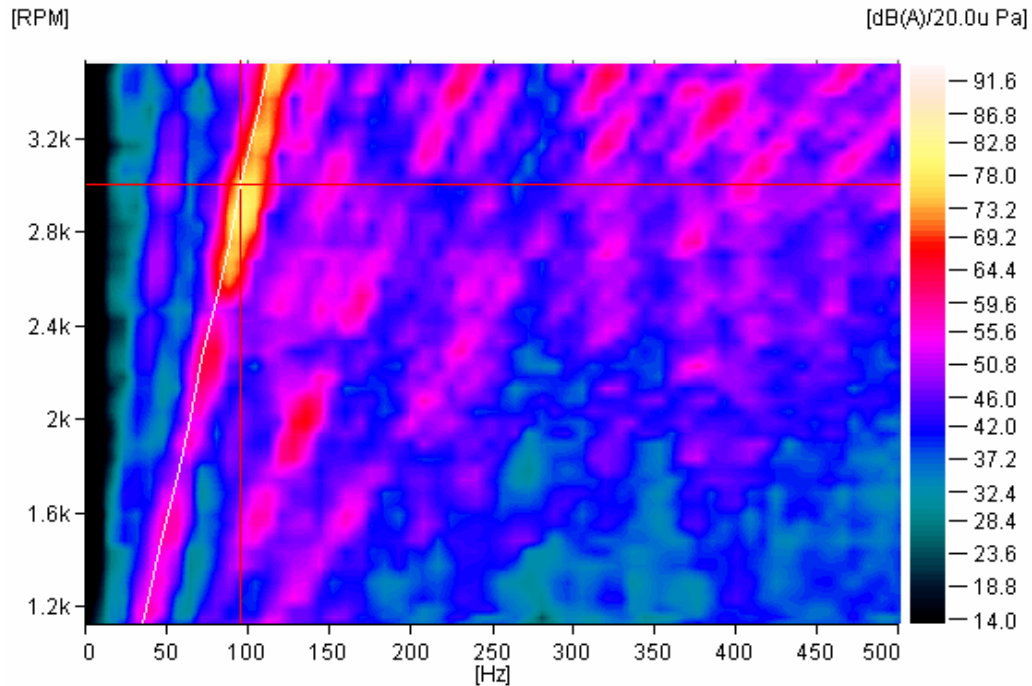


Figure 6.6 : FFT Buffer Triaxial Graph - Driver Ear Position

In the FFT analyzer, the signal is the microphone at the driver ear position. In this contour graphic, the horizontal axis which shows the frequency values is in the linear scale. Because of this linear scale, the orders are straight lines which can be seen in Figure 6.6. The dominant order, the engine excitation, is the second order. Besides, the advantage of using linear scale is to get the measurement data in a narrow frequency range. At 2900 RPM, the maximum SPL is 81.9 dB(A). The frequency can exactly be detected at 96 Hz.

Figure 6.7 shows the FFT Autospectrum at the driver ear position. In this graphic the vertical axis is SPL, the horizontal axis is frequency. On the other hand, engine speed is 2900 RPM which SPL is maximum value. In other words, the frequency spectrum belongs to the lightest yellow colored point in the FFT buffer triaxial graphic.

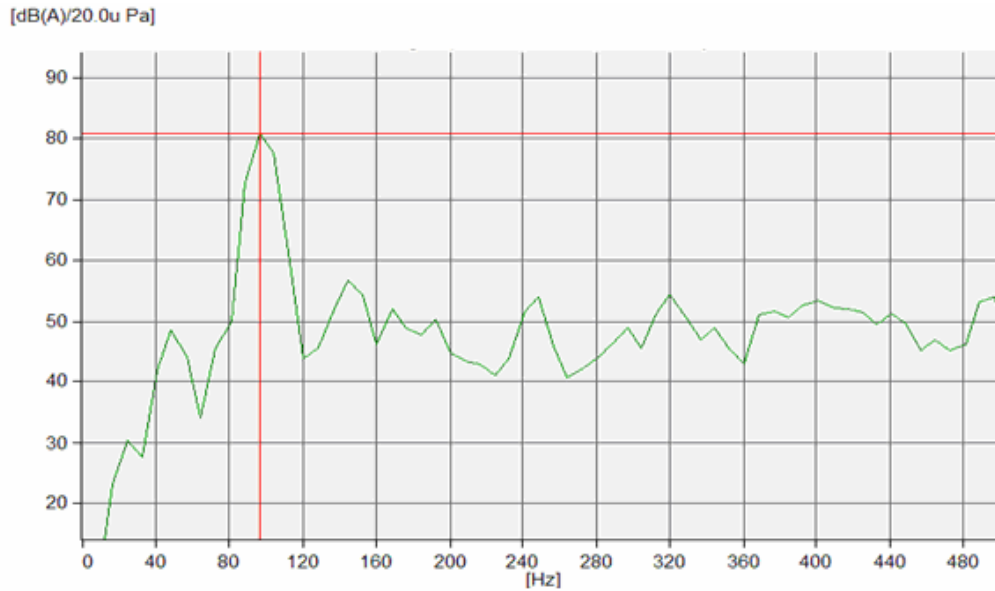


Figure 6.7 : FFT Autospectrum - Driver Ear Position

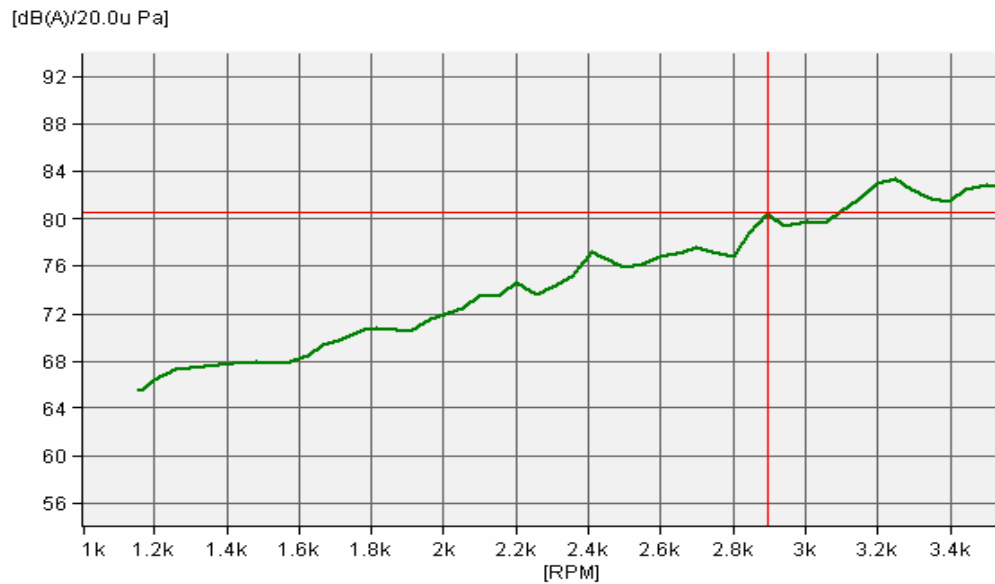


Figure 6.8 : Backside Sound Pressure Level versus Engine Speed

The increase of the sound pressure level in the backside of the vehicle is regular as shown in Figure 6.8. In other words, there are no unexpected peak values during the measurement. The SPL at the 2900 RPM is 80.4 dB(A).

The existing problem is that, when the engine speed is around 2900 RPM, 81.9 dB(A) sound pressure level is measured at the driver ear position while the frequency is 96 Hz. In this condition the dominant order is 2.021 which points out the engine excitation.

Under normal circumstances a regular increase in sound pressure level is expected. However, it can be easily understood from the Figure 6.9 that there is an 8 dB(A) difference between the expected SPL increase, shown with red dashed line, and the measured data with bold green curve. This 8 dB(A) difference occurs because of the booming noise in the cabin.

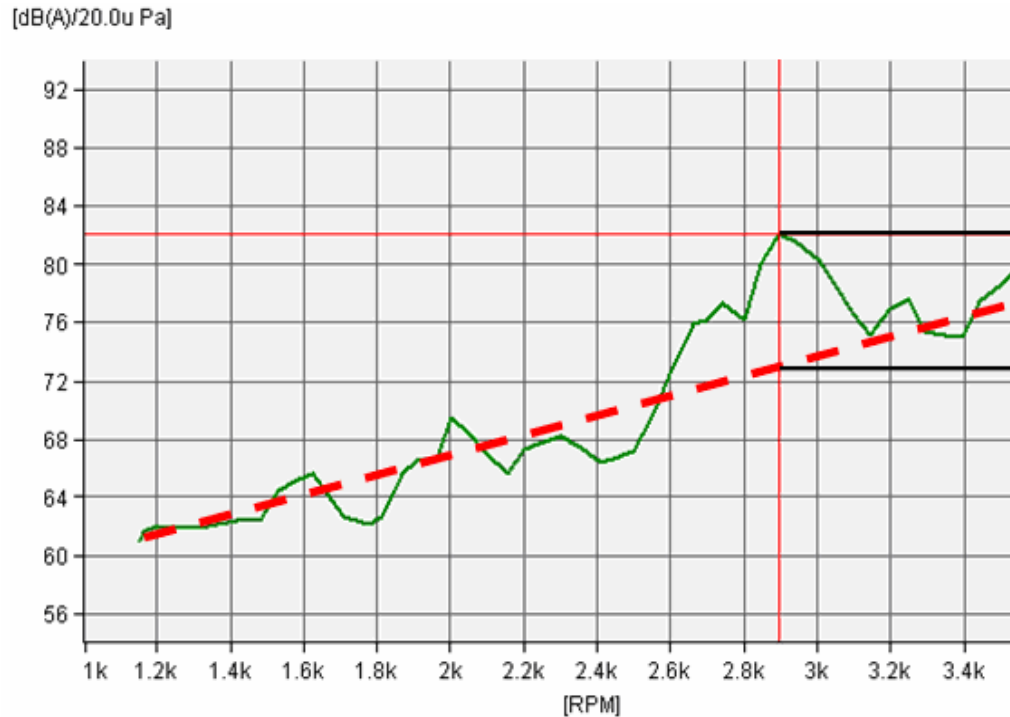


Figure 6.9 : The Existing Problem is the Difference 8 dB(A) at the Driver Ear Position

6.2 Run-up Vibration Diagnostics Test

The aim of this test is to identify the structural noise and noise generated by vibrations, and to determine the critical speeds and resonances. For this purpose, several accelerometers are placed at rear panel, roof, exhaust muffler, right and left doors of the vehicle. The vehicle is again speeded up to 3500 RPM from idle and vibration response data are collected and processed via the PULSE software.

6.2.1 Left & Right Door and Rear Panel Vibrations

In the Figure 6.10, pink line represents the vibrations of the left door while the green shows the right door. The third curve in blue color belongs to the vibrations of the rear panel.

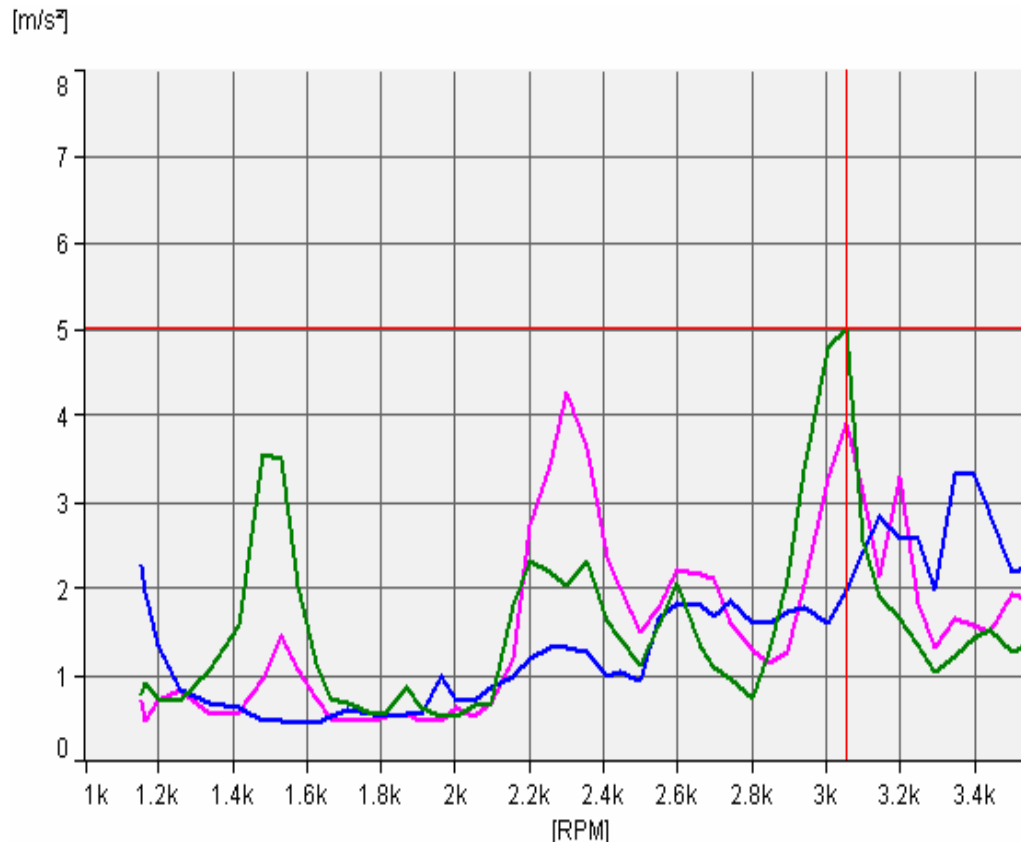


Figure 6.10 : Left Door, Right Door and Rear Panel Vibrations

For all collected data shown in the Figure 6.10, the maximum vibrations of each measurement points are observed. The left door vibrations shows peak values at 2300 RPM and 3050 RPM consequently. The vibrations of the right door reached to a maximum value at 3050 RPM while the rear panel vibrations show no specific peak value. Maximum sound pressure level is at 2850~2900RPM.

6.2.2 Rear Panel Vibration

Four accelerometers were located on the rear panel, as shown in Figure 6.11 in order to investigate the structural resonances.

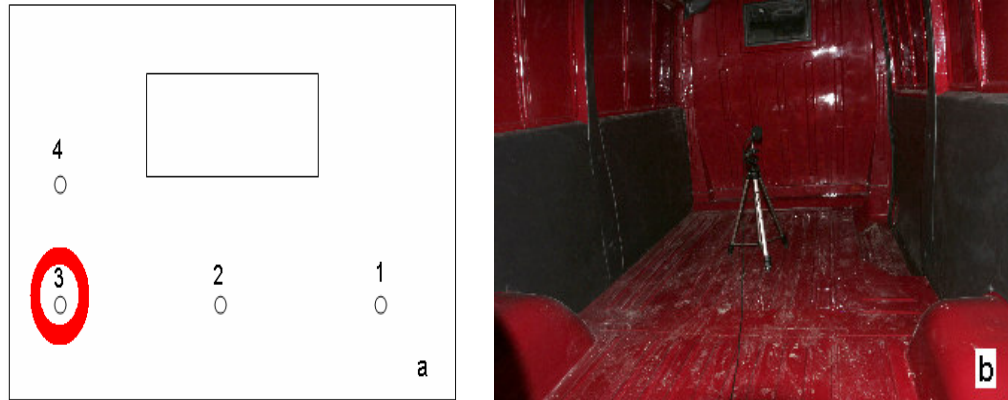


Figure 6.11 : Rear Panel (a) Drawing (b) Vehicle

In the measurement, the gearbox is in the first gear. At the end of the run-up test, there are no peak points and increases at the 2900 RPM. The other side of the rear panel was covered with an absorbent material to maintain a reduction in vibration of the panel. The third accelerometer detected a little increase but it is not too obvious. Figure 6.12 shows the rear panel vibrations.

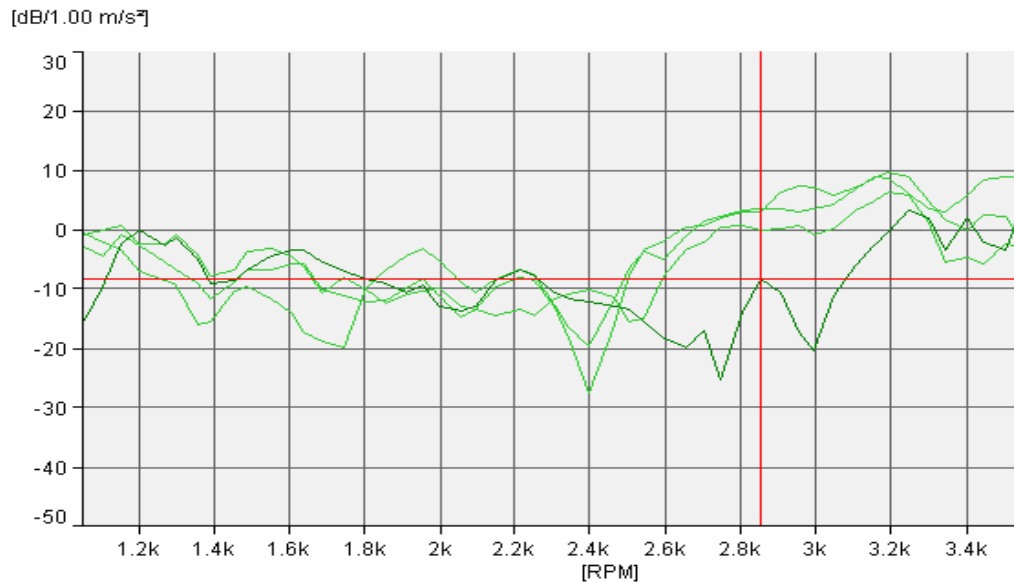


Figure 6.12 : Rear Panel Vibration

6.3 The Hammer Test

In the hammer test, the panels are excited by the impact hammer and the response is gathered by the accelerometers. The structural resonances are investigated. Frequency Response Function represents the ratio between output and input, as a

function of frequency. By the help of this application the frequencies in which the body reaches to its structural resonances can be detected.

6.3.1 The Hammer Test with Accelerometers at Rear Panel

Two accelerometers were located on rear panel surface. Gentle hits between the two accelerometers were made by the using the impact hammer. The impact hammer and the accelerometers used in this test are shown in Figure 6.13.



Figure 6.13 : Equipments (a) Impact Hammer (b) Accelerometer

When the Figure 6.14 is investigated, it can be seen that at 104 Hz there is a peak in FRF, which is pointed in dashed red circle. Coherence function, described in Chapter 3, shows the probability of the relationship between output and input data. In other words when the coherence function is 1, it means that the output data is caused by only the input. Vice versa when the coherence function is zero, there is no relationship between the input and the output data. In the measurement, the coherence function is 998m (0.998).

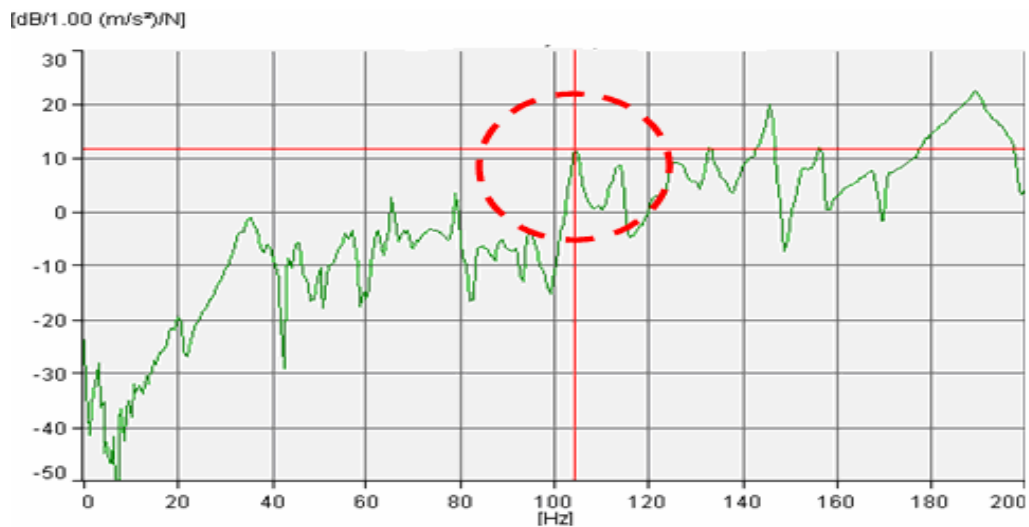


Figure 6.14 : FRF versus Frequency for the First Accelerometer

The Figure 6.15 shows the measurements made with the second accelerometer. Although gentle hits are made between the two identical accelerometers, different responses were obtained. The thing which has to be underlined is having a peak at 104 Hz similar to the first accelerometer, but the peak value is much more than the first one.

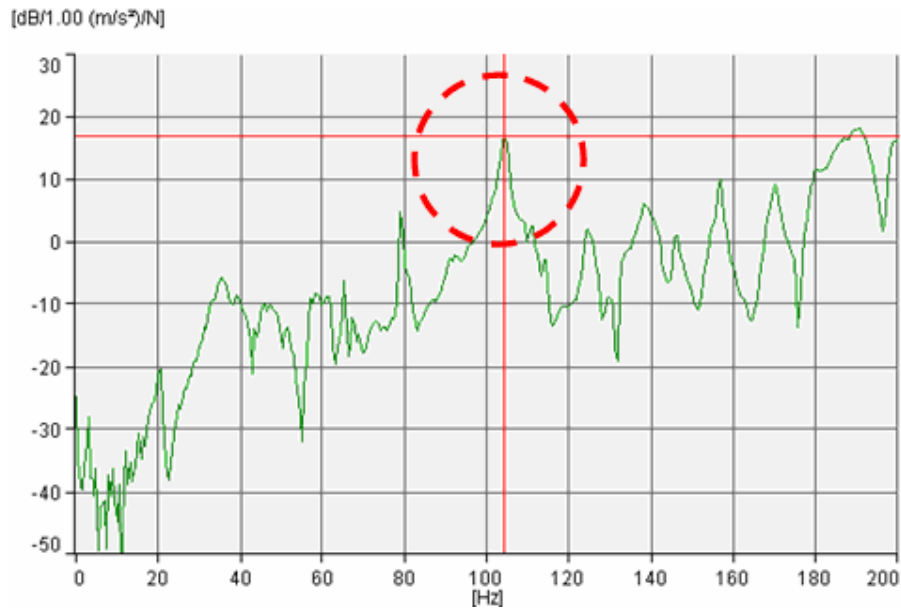


Figure 6.15 : FRF versus Frequency for the Second Accelerometer

6.3.2 The Hammer Test with Accelerometers at the Roof

Two accelerometers were located on the roof surface. Gentle hits between the two accelerometers were made by the using the impact hammer. The Figure 6.16 shows the measurements made with the accelerometers. Same response values were obtained in two accelerometers. There is a peak in FRF at 100 Hz in both of this graphics which is pointed in dashed red circle. For the first accelerometer, the coherence function is 996m. For the second accelerometer, the coherence function is 994m.

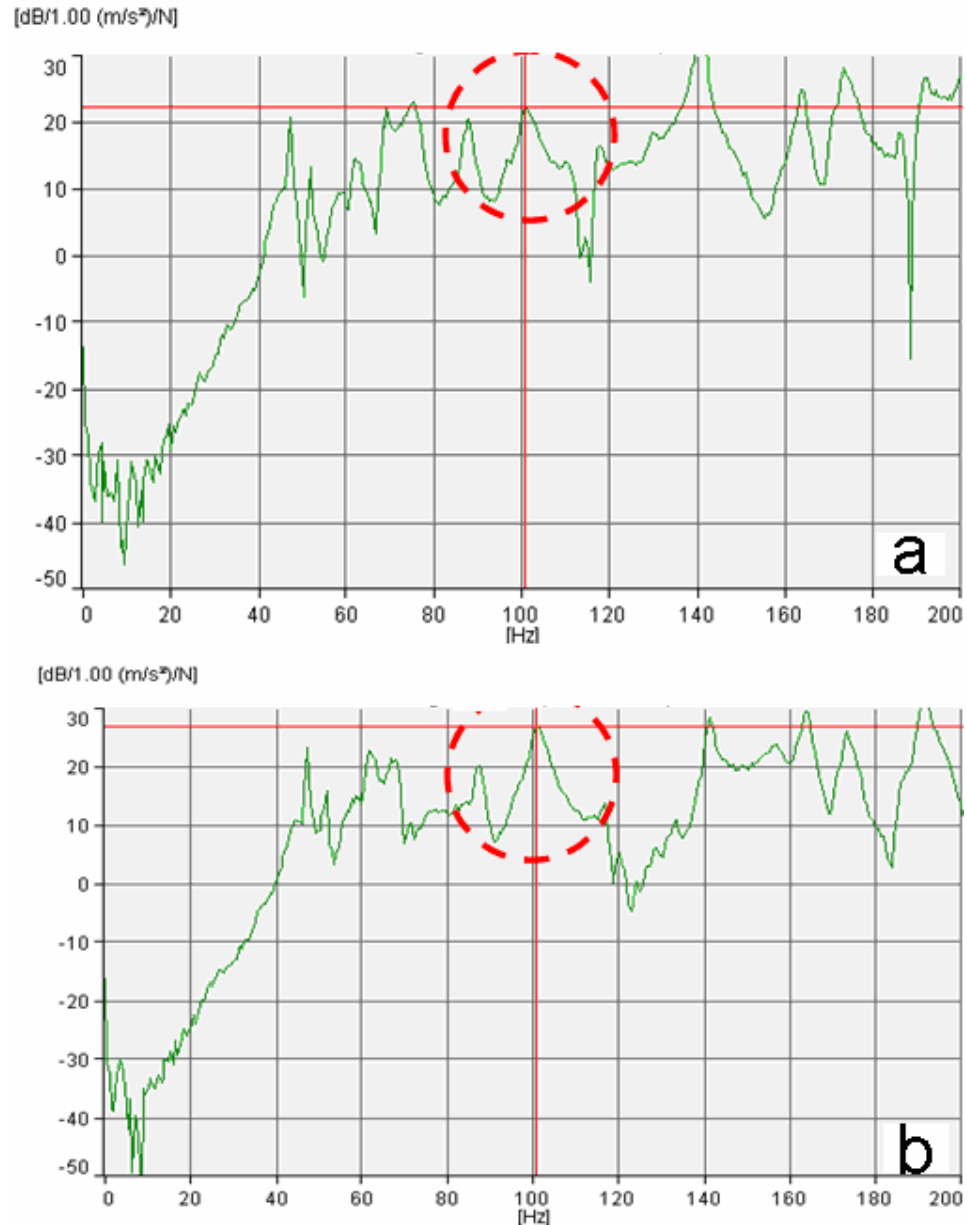


Figure 6.16 : FRF versus Frequency **(a)** The First Accelerometer **(b)** The Second Accelerometer

6.3.3 The Hammer Test with Microphone at the Rear Panel

In the hammer test with microphone, the rear panel is excited by the impact hammer and the response is gathered by the microphone. Frequency Response Function represents the ratio between output and input, as a function of frequency. In Figure 6.17 shows the test equipments.



Figure 6.17 : Hammer Test Equipments **(a)** Impact Hammer **(b)** Microphone

When the Figure 6.18 is investigated, it can be seen that at 94 Hz there is a peak in FRF. In the measurement, the coherence function is 1. This means that the response at the 94 Hz is caused by the impact excitation.

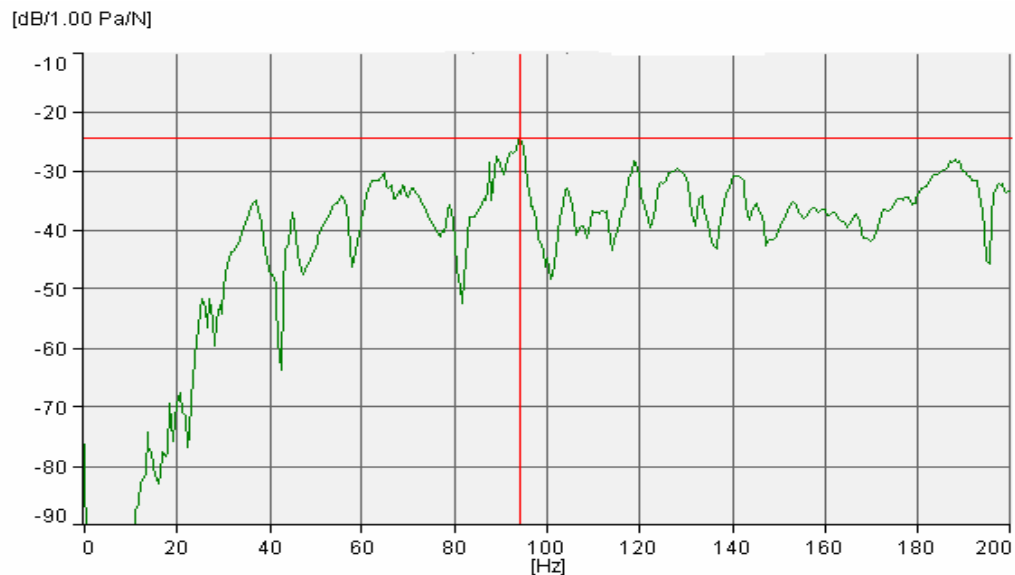


Figure 6.18 : FRF versus Frequency for Microphone

6.4 The Sound Intensity Measurements

The aim of sound intensity measurement is to determine the location where the sound intensity is higher in the cabin. In the experiment, the intensity probe has been located in some points of the cabin. The engine speed has been fixed at 2850 RPM in first gear.

Test equipments which are used in sound intensity measurement are sound intensity probe kit, sound intensity microphone pair and the dual preamplifier. In Figure 6.19 shows the sound intensity probe.

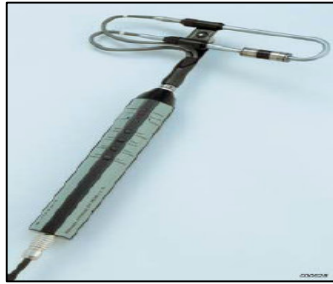


Figure 6.19 : Sound Intensity Probe

The maximum sound intensity is determined at the left door (near the loudspeaker), the rear panel (right side – driver ear position), the roof (driver right ear position) and the right side (floor). These locations are shown in Figure 6.20 and Figure 6.21.

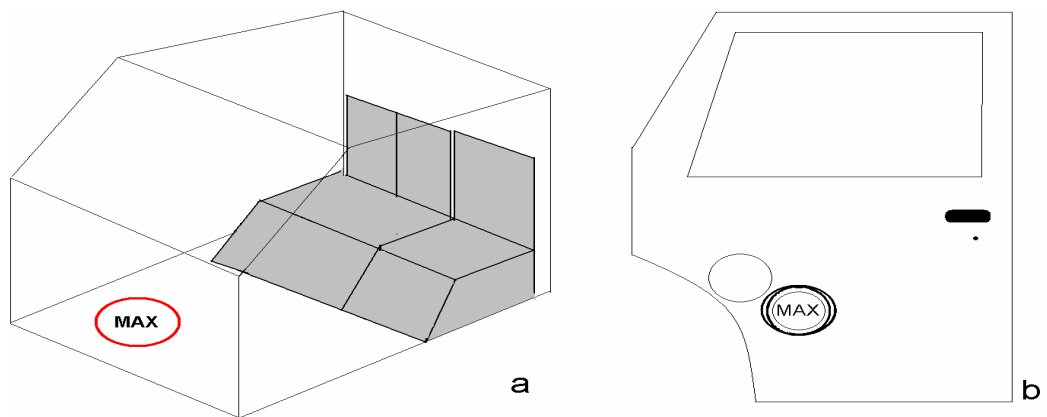


Figure 6.20 : The Maximum Sound Intensity Locations **(a)** The Floor **(b)** The Left Door

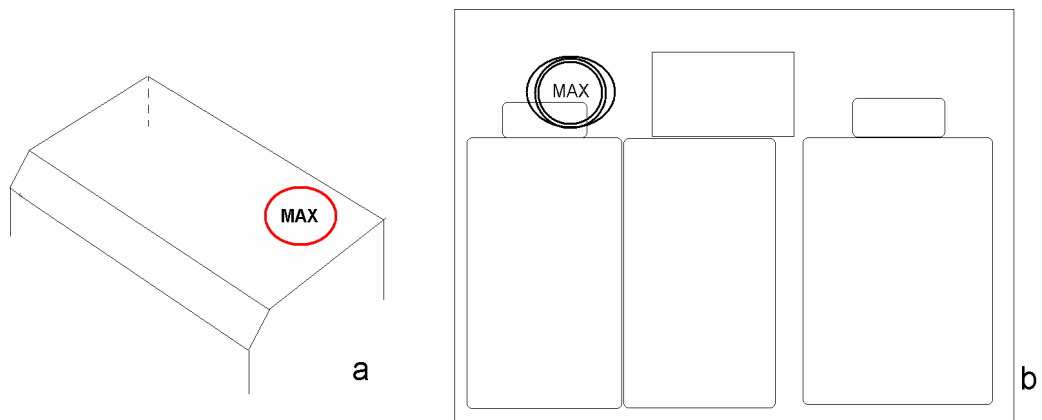


Figure 6.21 : The Maximum Sound Intensity Locations **(a)** The Roof **(b)** The Rear Panel

6.5 Performance Analysis of the Mounts

6.5.1 General Information about the Mounts Used in Vehicle

Although engine and gearbox mounts are relatively small, they are important in controlling NVH. They link the vehicle body to the engine, the largest source of vibration. It is through these mounts that the engine vibration is transmitted to the body. The vibrations from the tires are also transmitted to the engine, causing the engine to resonate with them.

The three characteristics required for the engine and gearbox mounts are:

- A great ability to dampen vibrations and reduces the vibrations from the engine
- A small spring constant reduces the engine vibration transmitted to the body
- A suitable natural frequency is a frequency which falls outside of normal operating vibration range, determined by the engine weight and the mount spring constant.

The active side of the mount can be described as the side which is exposed to direct vibrations caused by the engine. On the other hand, as the vibrations are damped through a mount, the passive side is expected to have less vibration levels when compared to the active side.

6.5.2 Run-up Tests for Analyzing Mount Performances

In this section the performance of the engine and gearbox mounts are investigated by using triaxial accelerometers. Using triaxial accelerometers leads to a detailed measurement as it is possible to have the vibration data in the entire longitudinal, vertical and the lateral directions at the same time. Thus, more precise information can be gathered and it gives the opportunity to compare the vibrations at each directions. Figure 6.22 shows the triaxial accelerometers on the passive sides of the gearbox and the right engine mount.

Triaxial accelerometers are located at:

- Engine left mount passive side
- Engine left mount active side
- Engine right mount passive side

- Engine right mount active side
- Gearbox mount passive side
- Gearbox mount active side

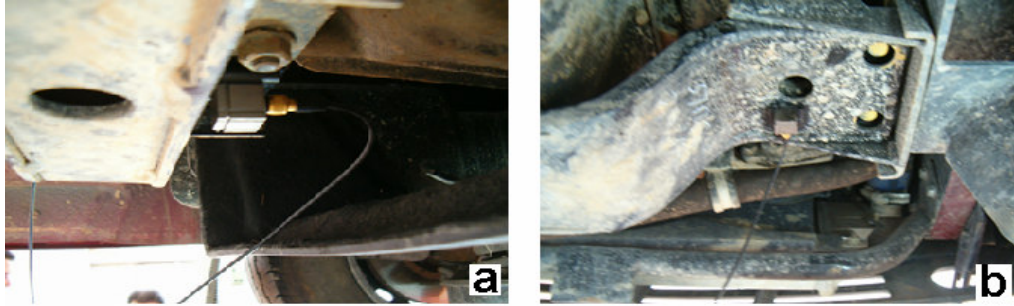


Figure 6.22 : Accelerometers on the Passive Sides of the **(a)** Gearbox **(b)** Right Engine Mounts

The Figure 6.24, Figure 6.25, Figure 6.26 show the performances of left engine mount, right engine mount, gearbox mount, left traverse mount at the critical engine speed (2850 RPM). Both the active and passive sides have been investigated. Bold curves are the representation of the vibrations at the active side while the thin curves show the vibrations at passive side.

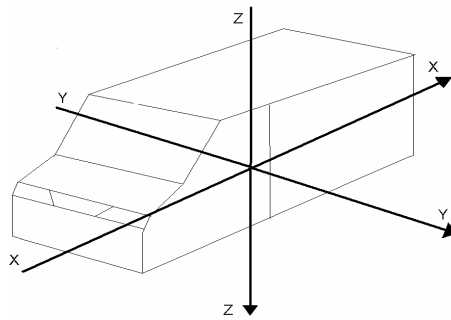


Figure 6.23 : The Representation of Longitudinal, Vertical and the Lateral Directions

In figures, three different colors are used to express the directions which are longitudinal, vertical and the lateral. Blue color is longitudinal axis of the mount, green color is the vertical axis of the mount and red color is the lateral axis of the mount. Figure 6.23 is the representation of longitudinal, vertical and the lateral directions of the vehicle.

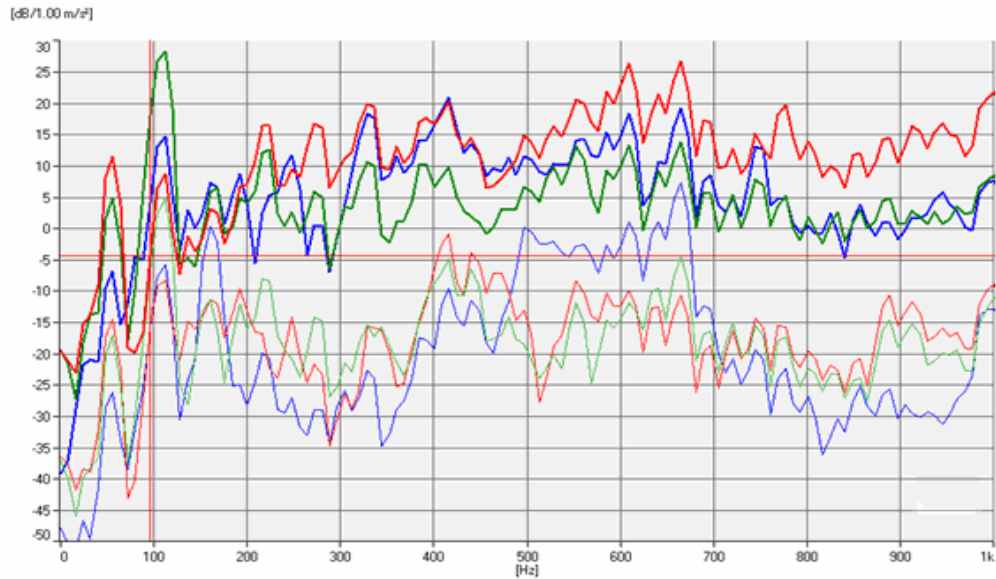


Figure 6.24 : The Active and Passive Sides of Left Engine Mount

The Figure 6.24 and Figure 6.25 show the performances of the left and right engine mounts. As expected, the vibrations of the active side are at the upper side of the graphic. There should be an adequate difference between the vibrations of the active and passive sides. When the graphic is examined comprehensively it can be seen that the mount transmission performance is very low around 100 Hz.

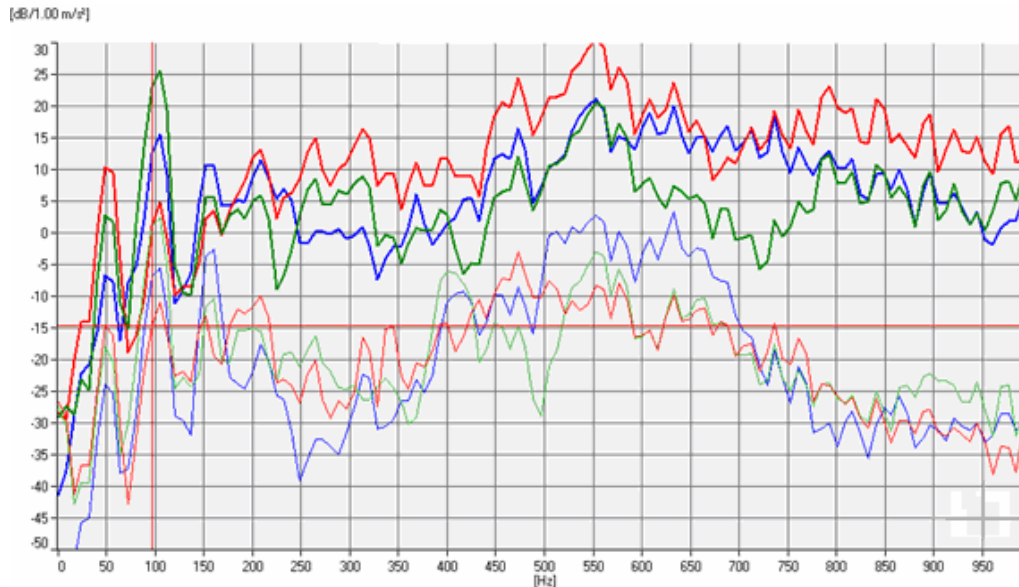


Figure 6.25 : The Active and Passive Sides of Right Engine Mount

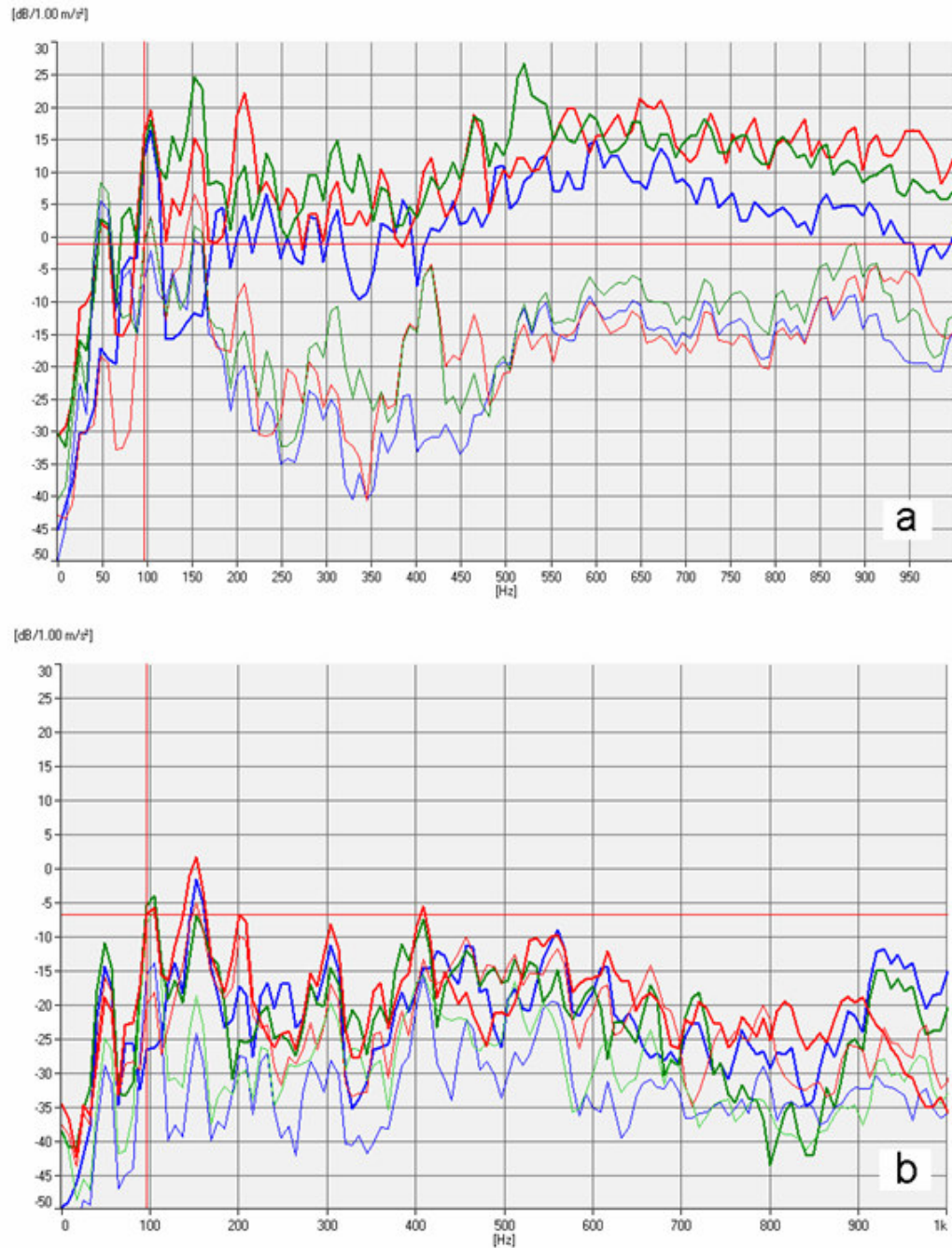


Figure 6.26 : Active and Passive Sides of **(a)** the Gearbox **(b)** the Left Traverse Mount

The Figure 6.26 consists of two graphics; one of them is the vibration data collected from the active and passive sides of the gearbox. While the performance of the mounts are said to be satisfactory in 250-350 Hz range, at low frequencies like 100 Hz, which is especially important for the study, the amount of reduction in the vibrations is negatively far beyond the expectations. Thus, this low performance in the gearbox mount leads to an unpleasant cabin noise. The other graphic

represented with the letter b, shows the performance of the left traverse mount. The performance of the mount seems to be unsatisfactory as there are no obvious differences in vibration levels between the active and passive sides. But as the vibrations which affect the left traverse mount are relatively small when it is compared with the other mounts, it is not fair to expect adequate performance, obvious reduction in vibration levels, from this mount.

6.5.2.1 The investigation of mount performances by using triaxial graphs

Contour type triaxial graphical representation can be seen in figures. This type of graphic contains three axes which show acceleration, engine speed and the frequency. The horizontal axis represents the frequency, the vertical axis shows the engine speed and the third axis shows the acceleration with the colored scale. In this scale, acceleration is increasing when the colors turn dark to light. This means that the maximum acceleration is shown with the lightest color which is yellow in this graphic. The horizontal axis is in the linear scale. Because of this linear scale, the orders are straight lines. The frequency ranges of graphics are 0-1000 Hz. The active and passive sides of the mounts are investigated in vertical direction.

When the passive side of the engine left mount graphic, shown in Figure 6.27, is interpreted, the maximum acceleration is represented by the lightest yellow color in the second order around 100 Hz and at the 2900 RPM. In addition to this, there are structural resonances at the frequency of 410 and 630 Hz. In Figure 6.28, active and passive sides of left engine mount are shown.



Figure 6.27 : Accelerometer on the Left Active Side

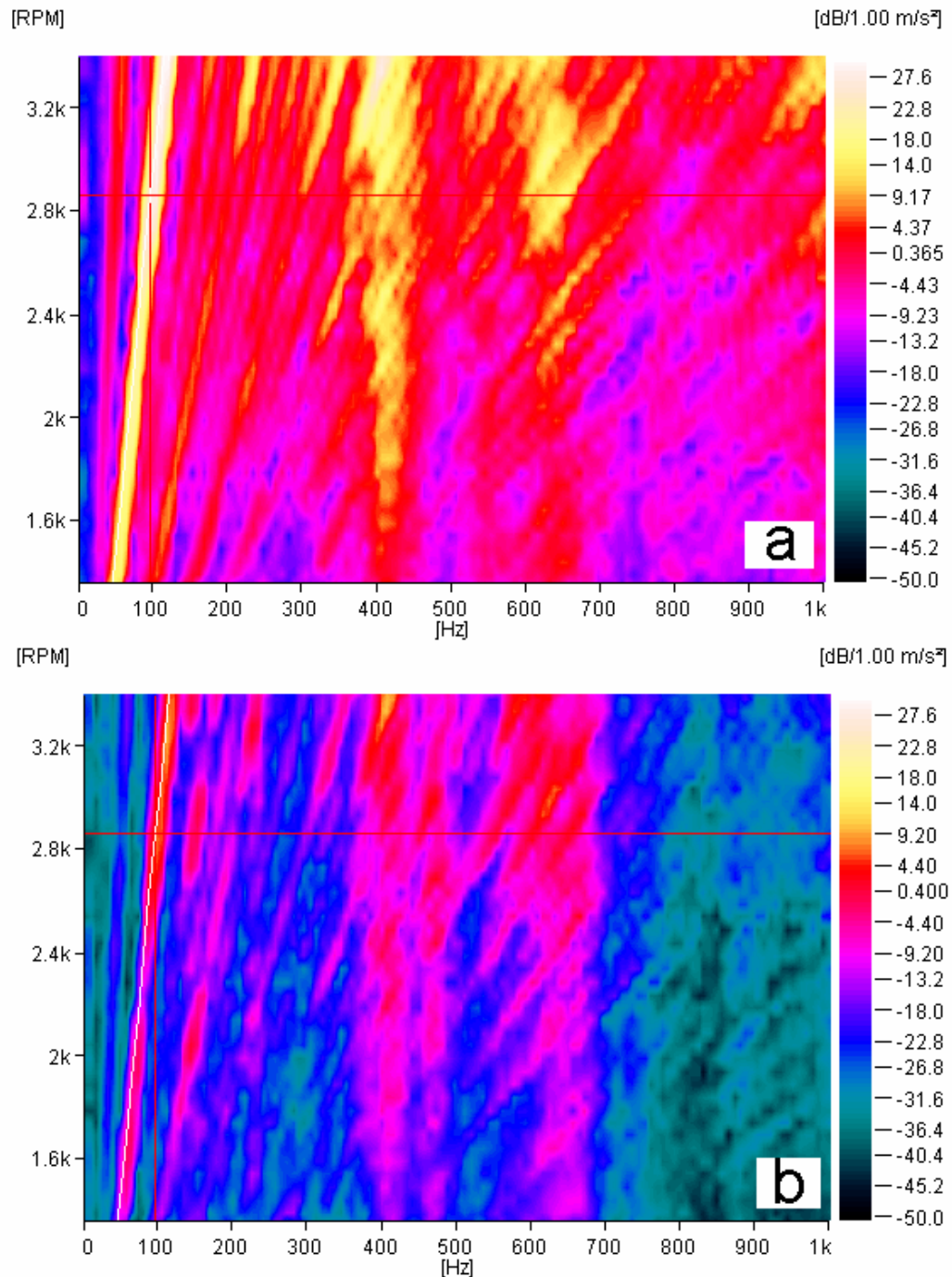


Figure 6.28 : Engine Left Mount **(a)** Active Side **(b)** Passive Side

When the passive side of the right engine mount graphic is interpreted, the maximum acceleration is represented by the lightest yellow color in the second order around 100 Hz and at the 2900 RPM. In addition to this, there are structural resonances at the frequency of 650 Hz. In Figure 6.29, engine right mount active and passive sides are shown.

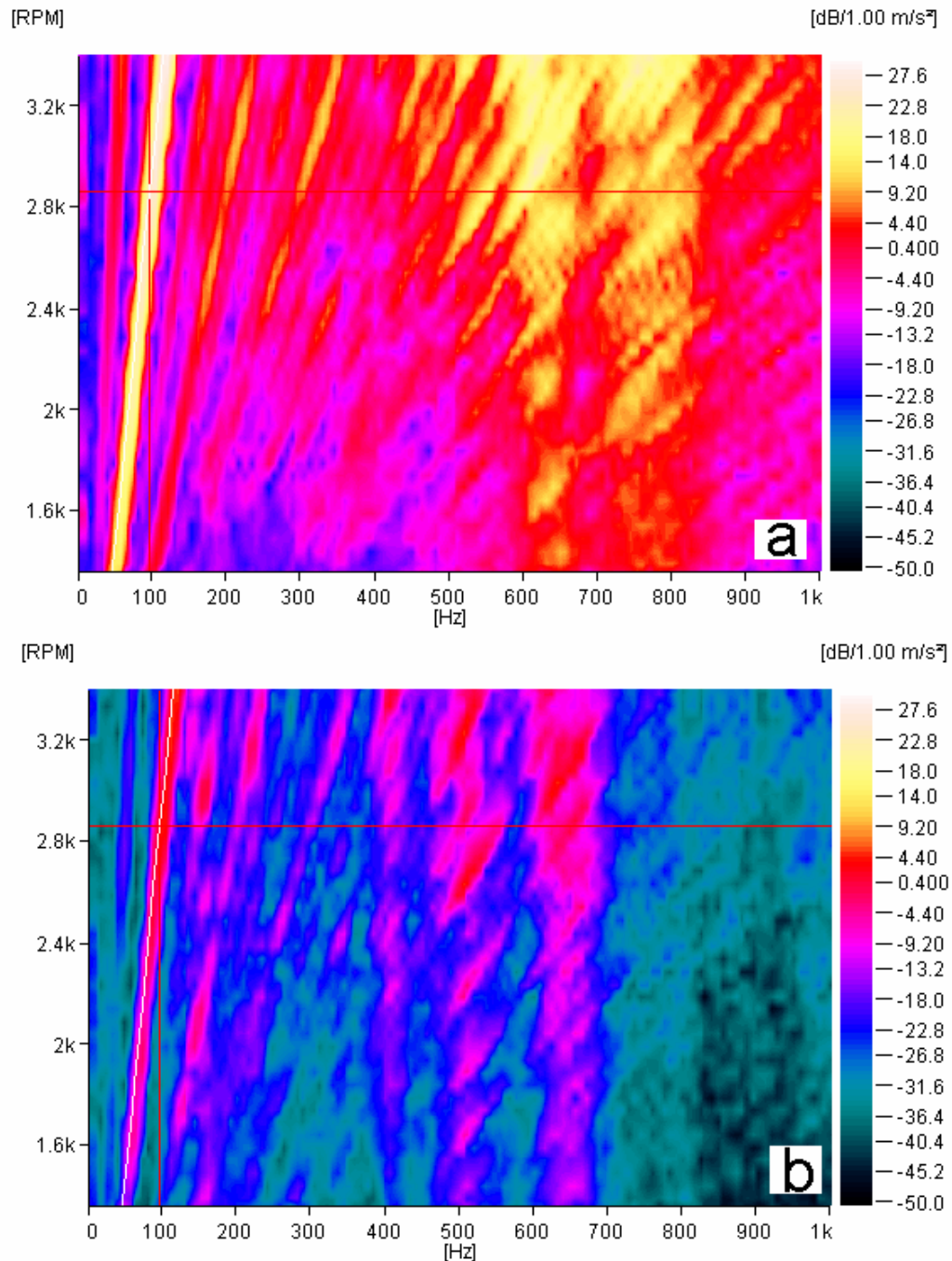


Figure 6.29 : Engine Right Mount (a) Active Side (b) Passive Side

When the passive side of the gearbox mount graphic is interpreted, the maximum acceleration is represented by the lightest yellow color in the second order around 160 Hz and at the 2900 RPM. In addition to this, there are structural resonances at the frequency of 160 Hz. The performance of the mount seems to be unsatisfactory as there are no obvious differences in vibration levels between the active and passive sides at 160 Hz. But, this low performance in the gearbox mount does not

lead to an unpleasant cabin noise. In Figure 6.31, engine right mount active and passive sides are shown. The accelerometer, shown in Figure 6.30, is on the active side of the gearbox mount where the vibration is higher than the other side.

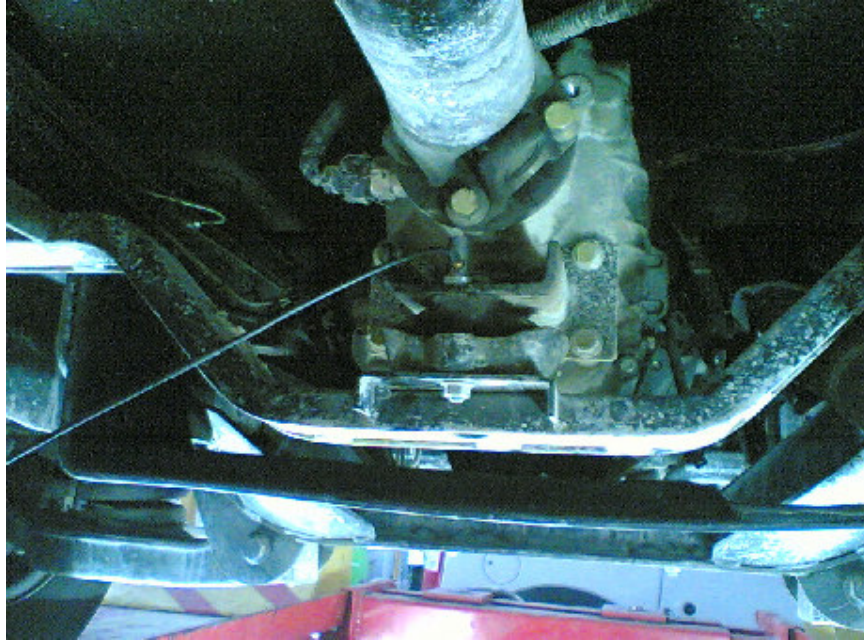


Figure 6.30 : Accelerometer on Gearbox Mount Active Side

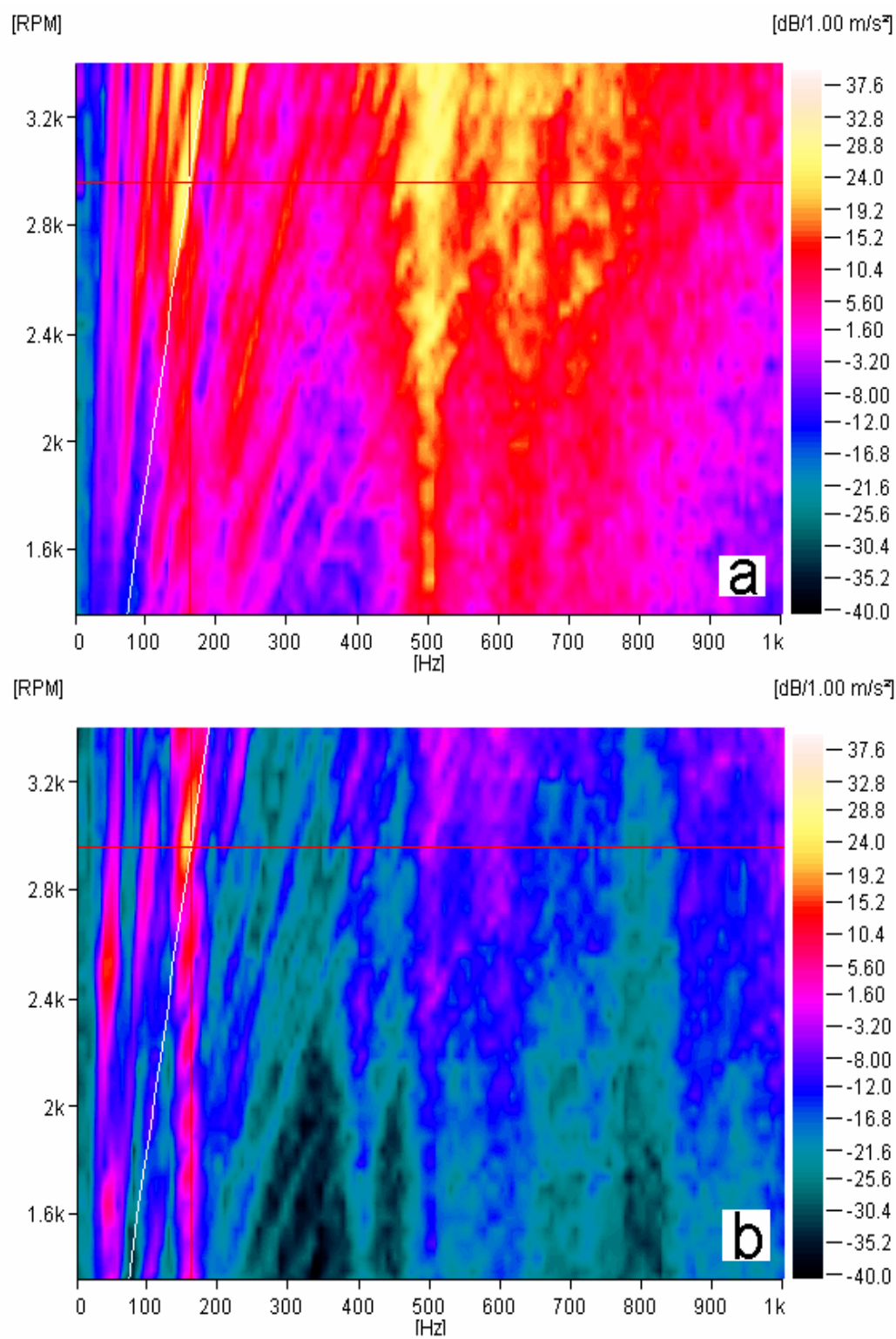


Figure 6.31 : Gearbox Mount **(a)** Active Side **(b)** Passive Side

6.5.3 Run-up Tests for Analyzing Traverse Mounts Performances

As shown in Figure 6.32, the traverse carries the gearbox and its arms are connected to the chassis. Two types of mounts are working on it. The first one which is between the gearbox and the traverse dampens and reduces the vibrations from gearbox. The other mounts which are between the traverse arms and the chassis dampen and reduce the vibration from the traverse to the chassis.

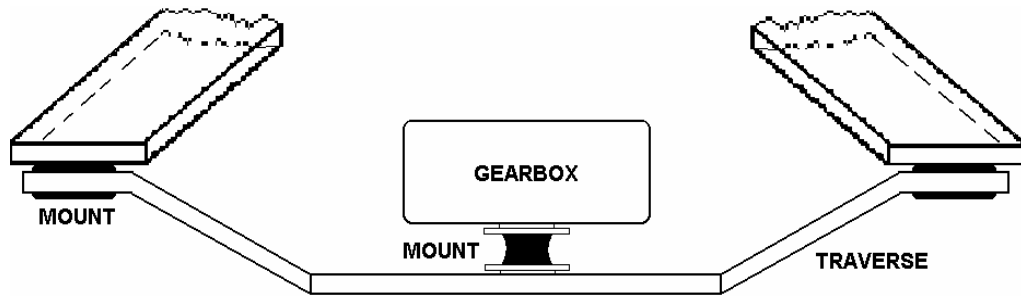


Figure 6.32 : Traverse and Gearbox System

Run-up tests have been made to identify the performance of mounts. Accelerometers have been located on the traverse. The accelerometers have been located at:

- Traverse left mount passive side
- Traverse left mount active side
- Traverse right mount passive side
- Traverse right mount active side
- Gearbox mount passive side
- Gearbox mount active side

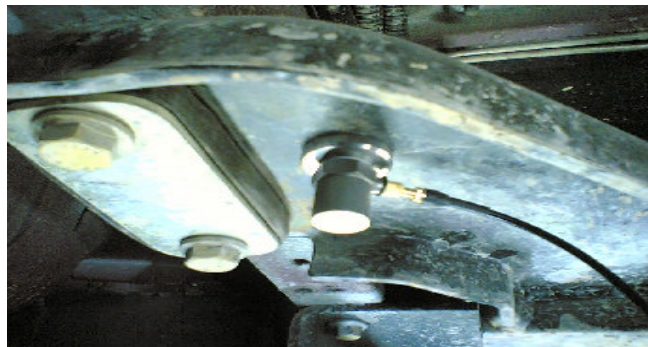


Figure 6.33 : Accelerometer on the Active Side of the Left Traverse Mount

The passive side of the gearbox mount is the active side of the traverse mounts. In Figure 6.33, the accelerometer is at the active side of the left traverse mount. Measurements have been made in vertical direction.

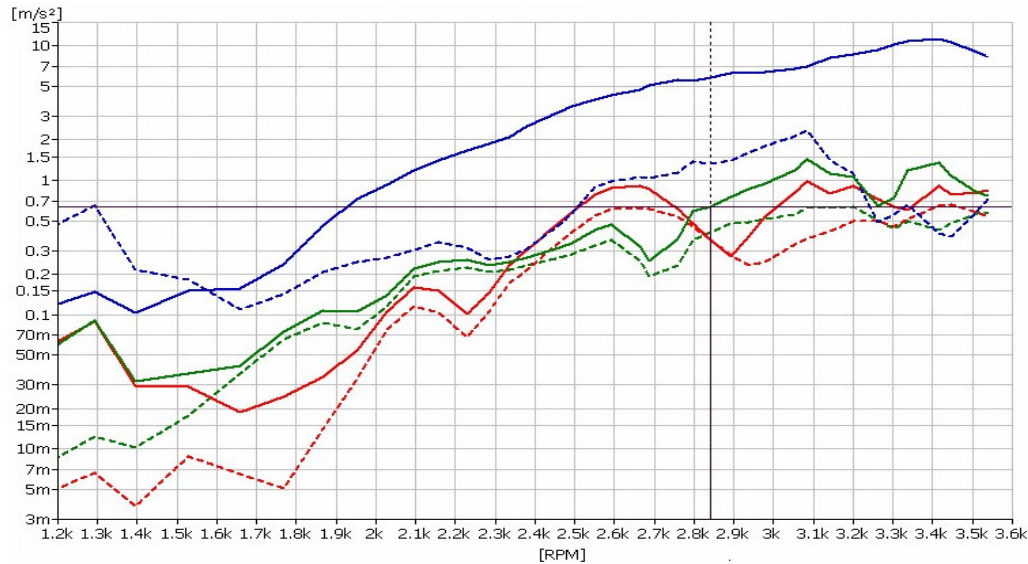


Figure 6.34 : Accelerations on Gearbox and Traverse Mounts

In Figure 6.34, three different colors are used to express the mounts which are the right traverse, the left traverse and the gearbox mount. Blue color is the gearbox mount, green color is the left traverse mount and red color is the right traverse mount. Continuous lines are active sides of the mounts, dashed lines are passive side of the mounts. At the critical engine speed, there are no peak values for the second order.

Contour type triaxial graphical representation can be seen in figures. This type of graphic contains three axes which show acceleration, engine speed and the frequency.

When the passive side of the engine right mount graphic, shown in Figure 6.35, is interpreted, the maximum acceleration is represented by the lightest yellow color in the second order around 92 Hz and at the 2200 RPM. In addition to this, there is a structural resonance at the frequency of 412 Hz. This resonance is also shown in the passive side of the gearbox because of the accelerometers mounting surfaces are on the same part. When comparing the acceleration data, the values at the traverse mounts' are lower than the gearbox mounts'.

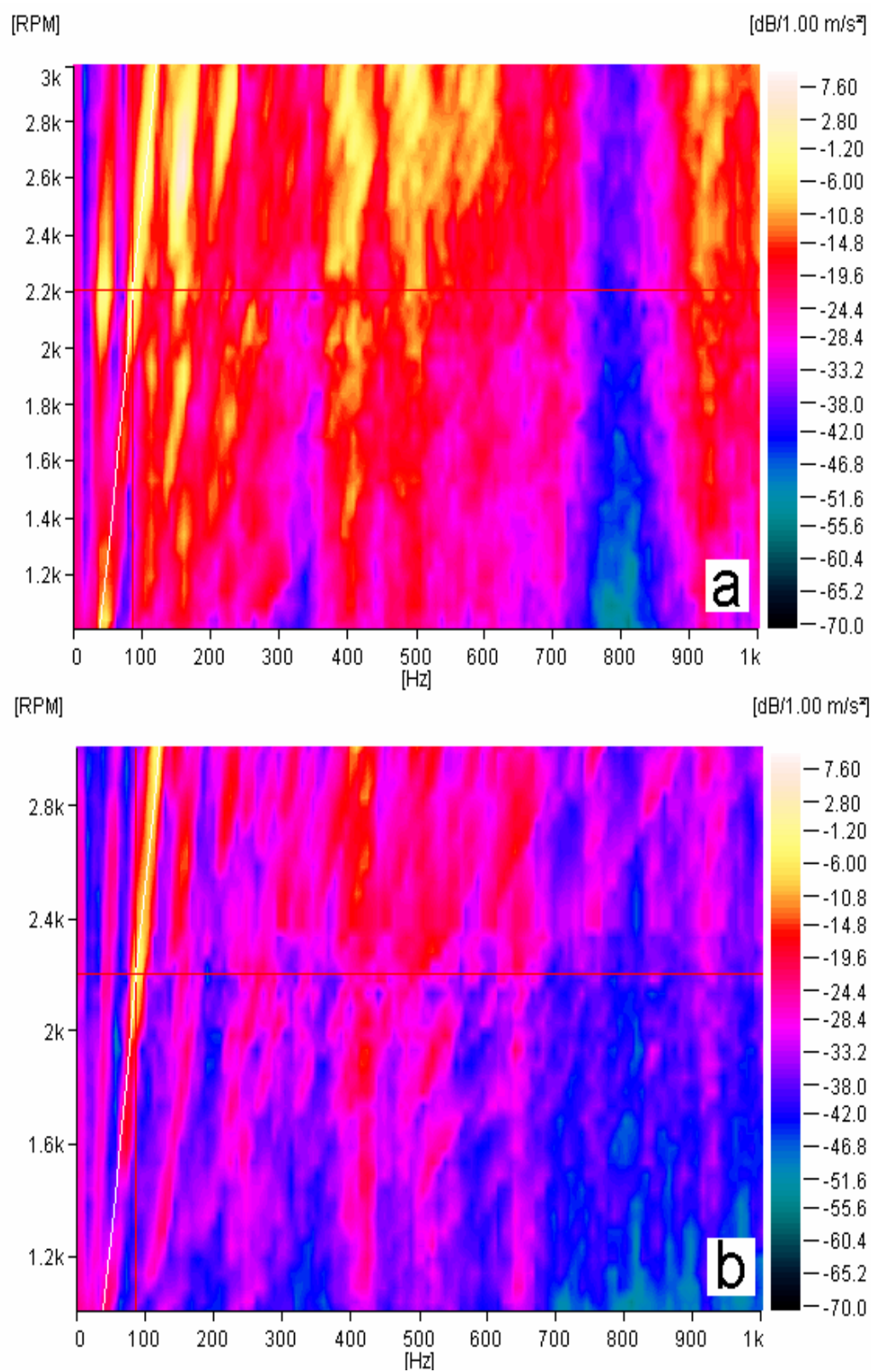


Figure 6.35 : Right Traverse Mount (a) Active Side (b) Passive Side

The triaxial accelerometers have been located at the active and passive sides of left traverse. Figure 6.36 shows the acceleration values for vertical, longitudinal and horizontal axes versus engine speed. Three different colors are used to express the directions which are longitudinal, vertical and the lateral. Red color is longitudinal axis of the mount, black color is the vertical axis of the mount and blue color is the lateral axis of the mount. Bold curves are active sides of the mounts, thin curves are passive side of the mounts. At 2850 RPM and 2900 RPM, there are no peak values.

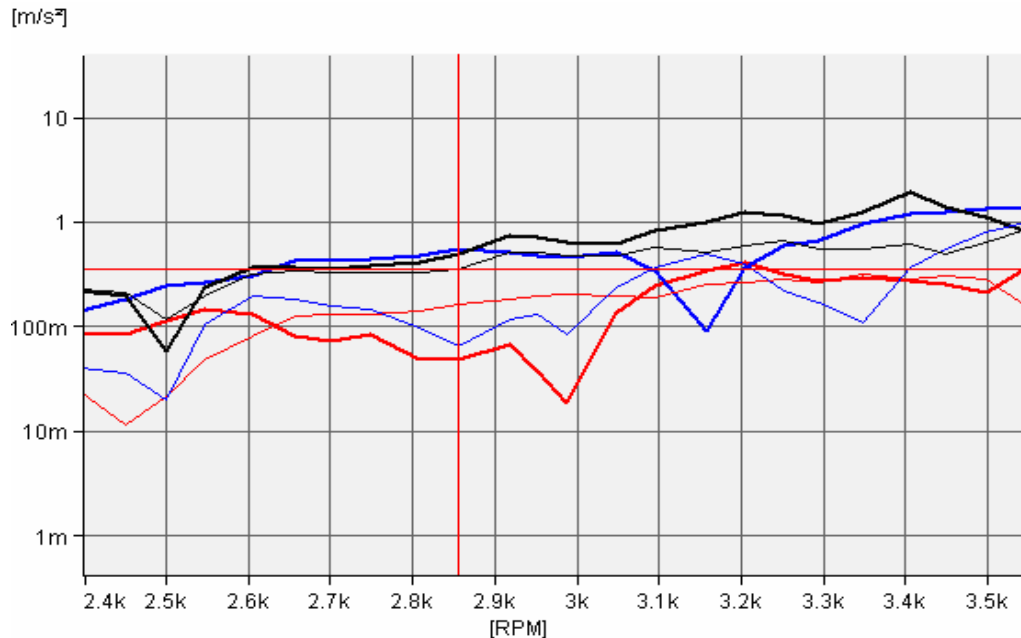


Figure 6.36 : Left Traverse Mount

When the passive side of the engine left mount graphic, shown in Figure 6.37, is interpreted, the maximum acceleration is represented by the lightest yellow color in the second order around 92 Hz and at the 2200 RPM. In addition to this, there is structural resonance at the frequency of 412 Hz. The performance of the mount seems to be unsatisfactory as there are no obvious differences in vibration levels between the active and passive sides at this frequency. This resonance is also shown in the passive side of the gearbox because of the accelerometers mounting surfaces are on the same part.

In both of the left and the right traverse mounts, the second order which is the excitation of the engine is always dominant. These mounts can not dampen the vibration in the second order.

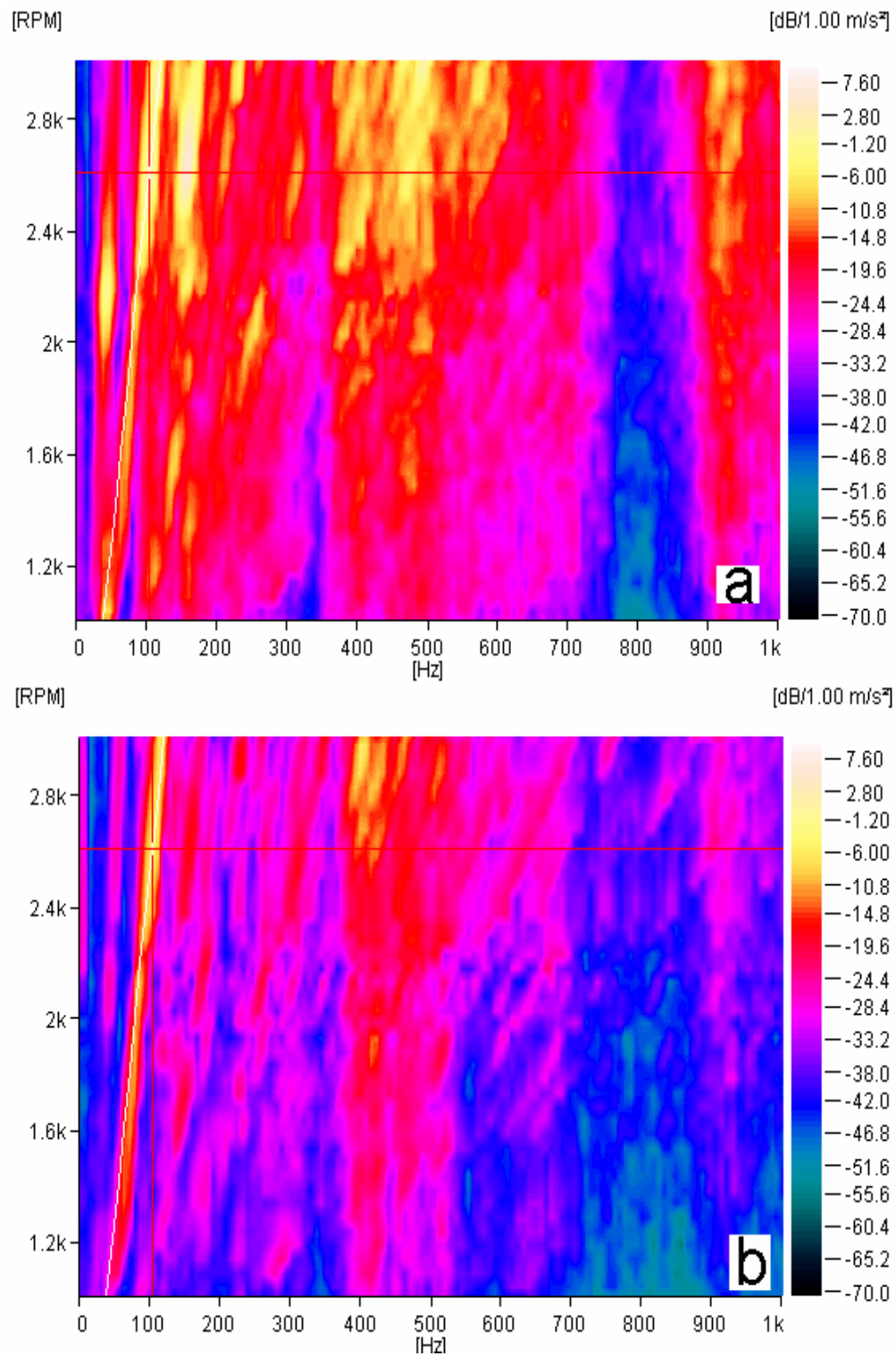


Figure 6.37 : Left Traverse Mount **(a)** Active Side **(b)** Passive Side

6.6 The Shaker Test

In the shaker test, which is made to determine the natural frequencies and modes, the engine of the vehicle is excited by the modal exciter. The connection between the exciter and the engine is supplied by a stringer, shown in Figure 6.40. Because of its plastic material, the excitation is only in one direction. The stringer is attached to the engine with the force transducer. The force transducer measures the excitation forces on the engine block. Figure 6.38 shows the system of the shaker test. In the test, the excitation signal, modified in the power amplifier, is swept sine wave form. In this wave form for every frequency in range, the excitation force can be at the maximum level. Triaxial accelerometers, shown in Figure 6.41, are located active and passive sides of the right engine mount, left engine mount and gearbox mount.

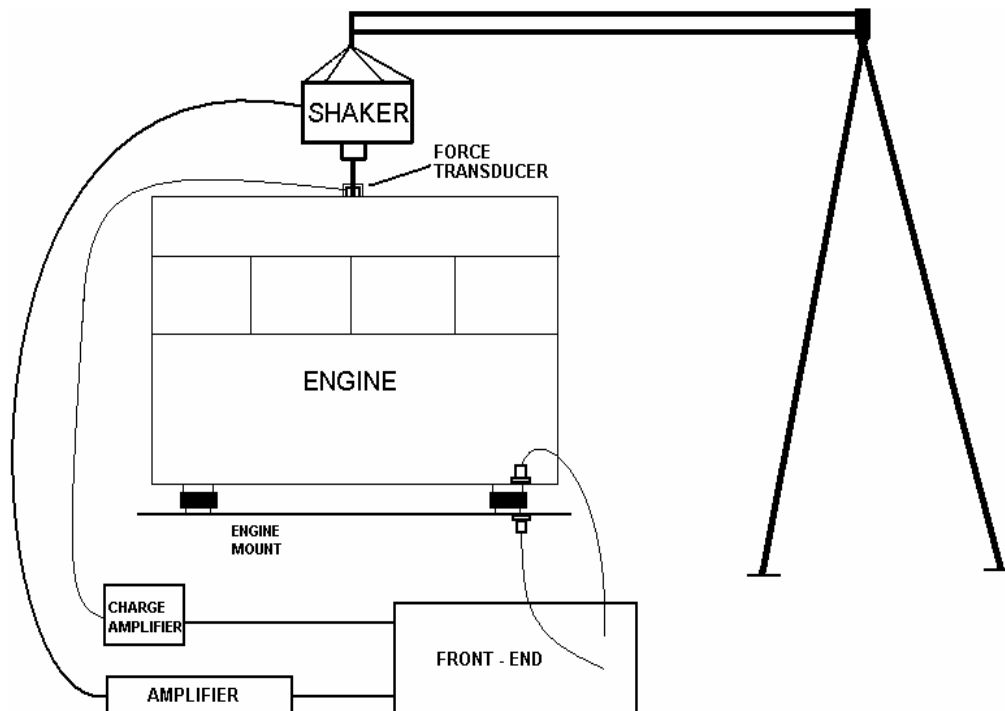


Figure 6.38 : The Shaker Test System

In this test, the modal exciter, stringer, force transducer, power amplifier and the triaxial accelerometers are used. Some of them are shown in shown Figure 6.39.

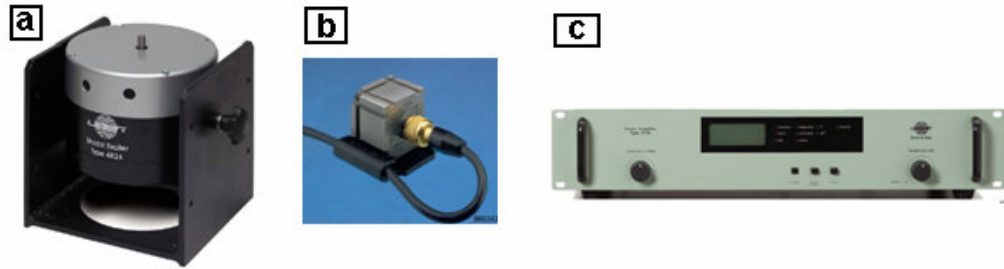


Figure 6.39 : Shaker Test Equipments **(a)** Modal Exciter **(b)** Triaxial Accelerometer **(c)**Power Amplifier

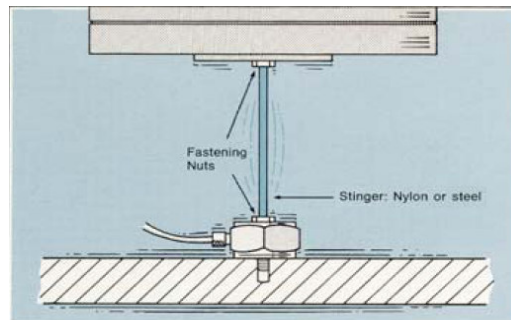


Figure 6.40 : Stringer Attachment

When the input data is the excitation force from the modal exciter which is measured by the force transducer, the output data is the acceleration at the active and passive sides of the mount. In the test, the frequency response functions have been investigated for the mounts. As seen in the acoustic and the vibration tests, the problem is found around the frequency of 100 Hz. So that, the frequency ranges are 0-200 Hz in the test.



Figure 6.41 : The Triaxial Accelerometer on the Mount

Triaxial accelerometers are located at:

- Engine right mount active side
- Engine right mount passive side
- Engine left mount active side
- Engine left mount passive side
- Gearbox mount active side
- Gearbox mount passive side

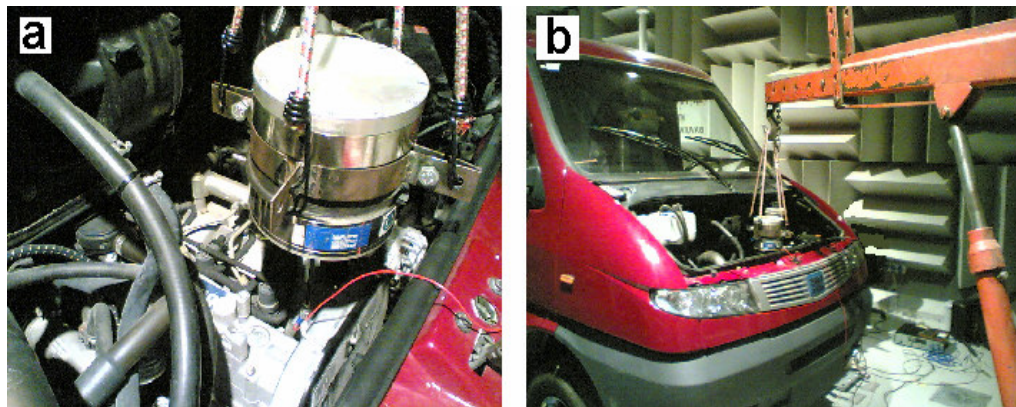


Figure 6.42 : The Modal Exciter Attached to the Engine Block

The coordinates of triaxial accelerometers, shown in Figure 6.43, are represented in letter x, y and z. The longitudinal axis is shown by the letter x and blue color for the graphics. The vertical axis is shown by the letter z and green color for the graphics. The lateral axis is shown by the letter y and red color for the graphics.

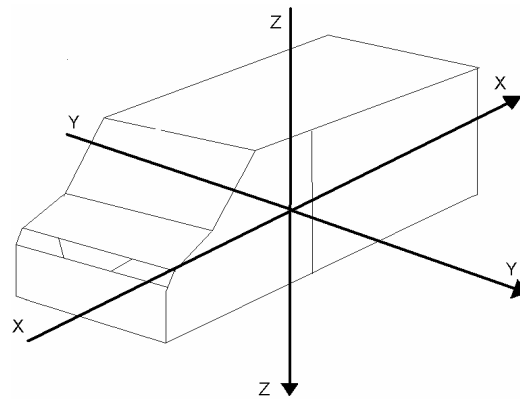


Figure 6.43 : The Representation of Longitudinal, Vertical and the Lateral Directions

6.6.1 The Shaker Test at the Right Mount

When the Figure 6.44 is interpreted, there are peaks at the active side of the right mount for each axis. There should be an adequate difference between the vibrations of the active and passive sides. The increases in the active side of the mount at 98 Hz are not in the passive side.

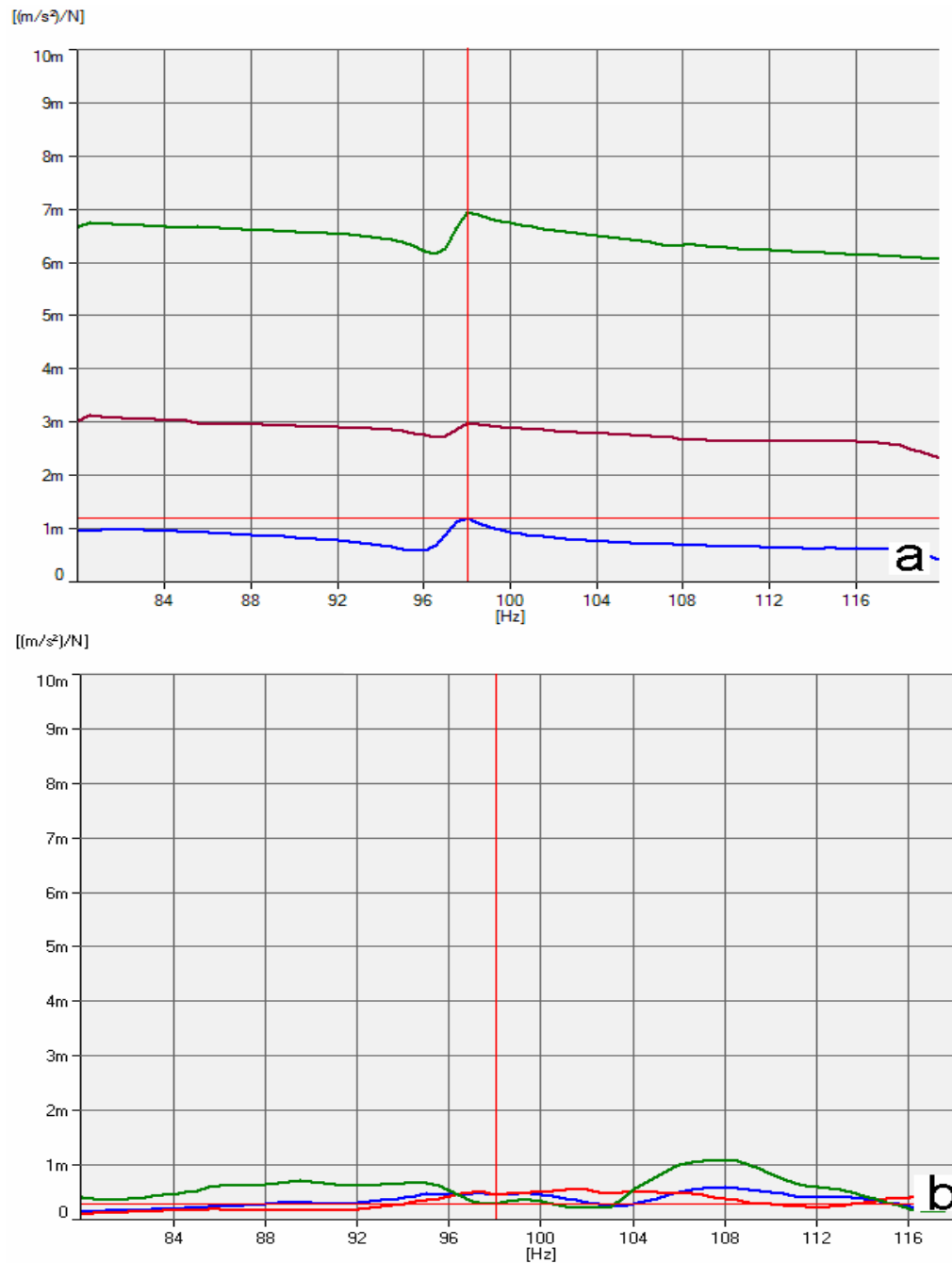


Figure 6.44 : Right Mount (a) Active Side (b) Passive Side

6.6.2 The Shaker Test at the Left Mount

When the Figure 6.45 is interpreted, there are peaks at the active side of the left mount for each axis. There should be an adequate difference between the vibrations of the active and passive sides. The increases in the active side of the mount at 98 Hz are not in the passive side.

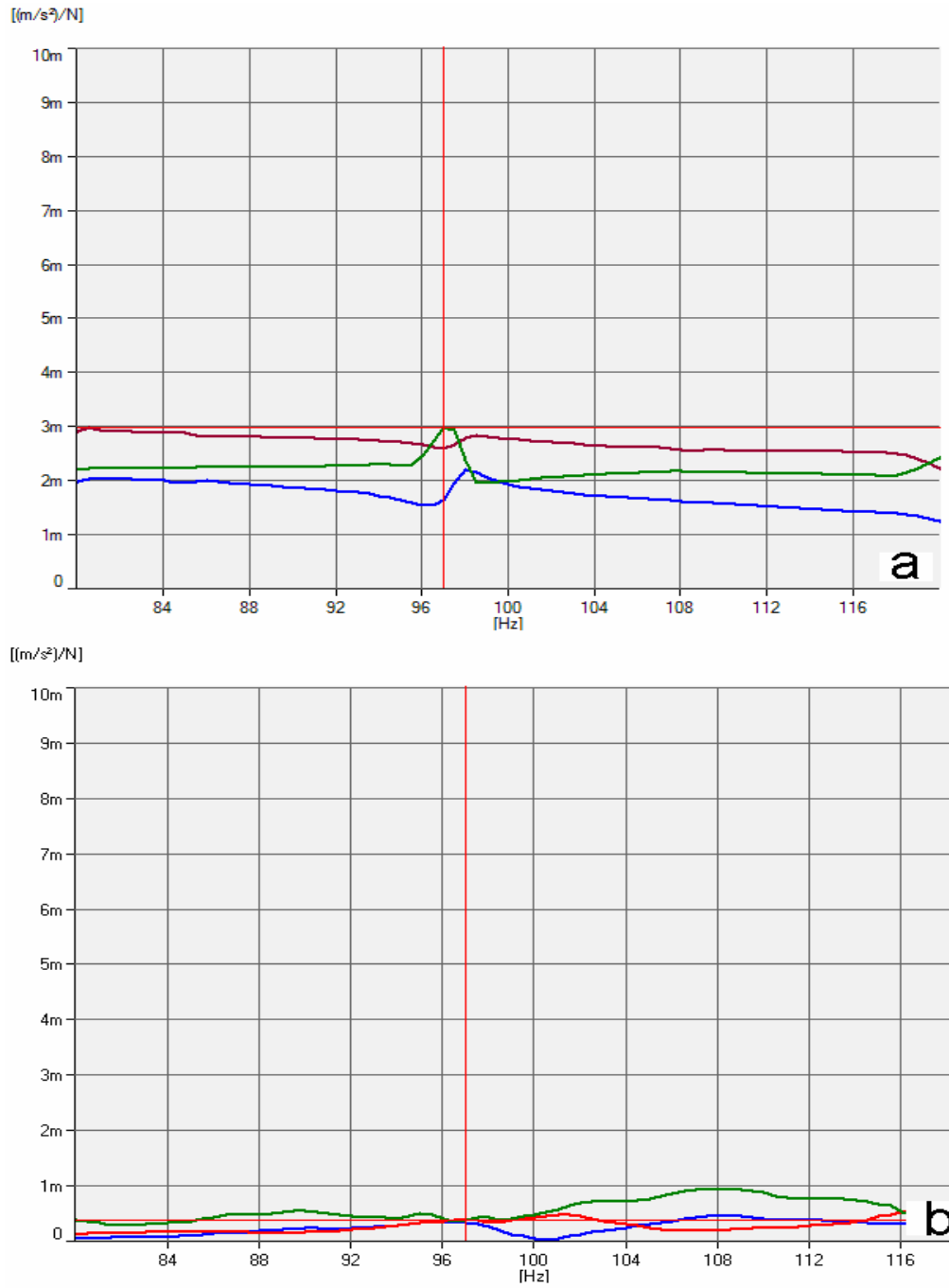


Figure 6.45 : Left Mount (a) Active Side (b) Passive Side

6.6.3 The Shaker Test at the Gearbox Mount

The Figure 6.46 shows the performances of both active (a) and passive side (b) of the gearbox mount. When the graphic is examined comprehensively it can be seen that the mount transmission performance is very low around 97 Hz in longitudinal axis.

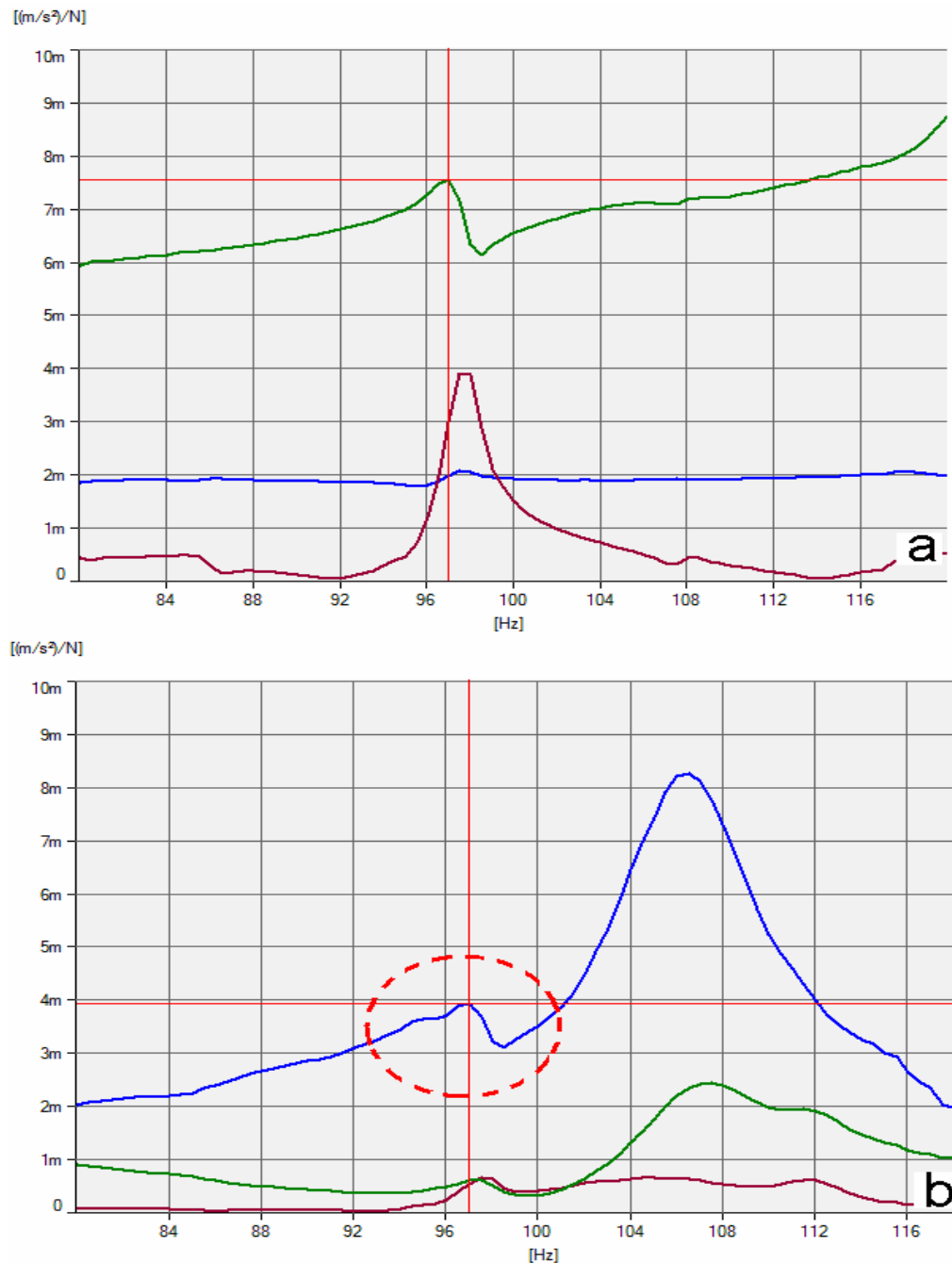


Figure 6.46 : Gearbox Mount (a) Active Side (b) Passive Side

There is a peak in FRF at 97 Hz in passive side of the mount which is pointed in dashed red circle. The thing which has to be underlined is having this peak. The excitation in the longitudinal axis causes vibrations at the rear panel. This is the main source of the interior noise.

6.6.4 Ear Position Response in the Shaker Test

In this measurement, when the input data is excitation force from the modal exciter, the output data is the sound pressure level at the driver ear position. The Figure 6.47 shows the sound pressure level at the driver ear position by the FFT Autospectrum graphic. There is a peak which is pointed in dashed red circle. The maximum SPL is 81.9 dB(A) at 96-98 Hz. An increase in frequency response function at the 98 Hz can because of this interior noise.

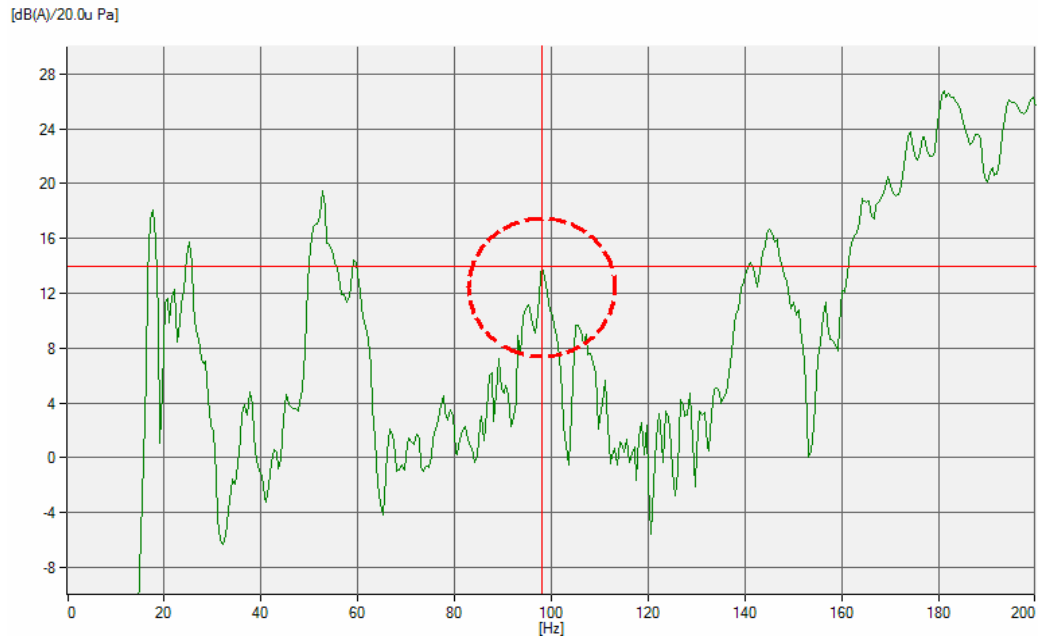


Figure 6.47 : Sound Pressure Level at the Driver Ear Position

When the frequency response function is investigated between the input data and the output data, the peak at 98 Hz is clearly seen. The coherence function is equal to 1, this means that the cause of the increase at 98 Hz is excitation of shaker. Figure 6.48 shows the frequency response function of SPL at the driver ear position and the excitation.

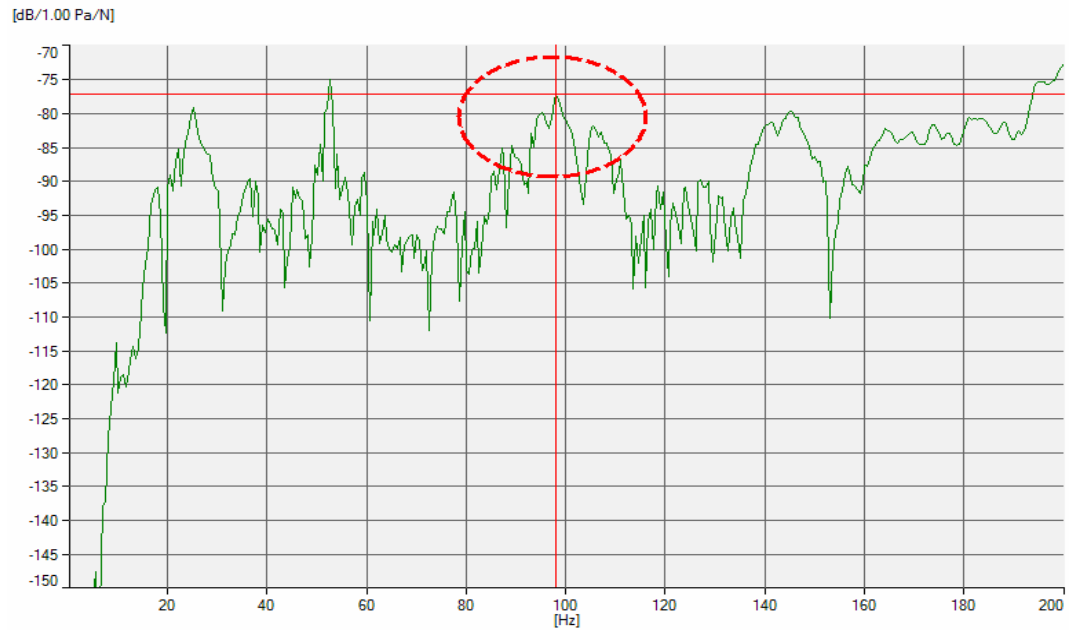


Figure 6.48 : The Frequency Response Function of SPL and the Excitation

7. CONCLUSION

Several tests and measurements were made to identify the noise and vibration problems of a commercial vehicle. There is an unpleasant noise in the vehicle cabin which gives clues of booming noise. The interior noise reaches to peak value at the most commonly used engine speed ranges like 2500-3000 RPM. Several tests were performed for the determination of critical speeds, structural resonances and instabilities.

The run-up acoustic performances, run-up vibration diagnostics, sound intensity, hammer, engine mounts performance and the shaker tests have been made for identifying the interior noise problem.

In conclusion of these experiments, the existing problem is that, when the engine speed is around 2900 RPM, an unexpected 81.9 dB(A) sound pressure level is measured at the driver right ear position.

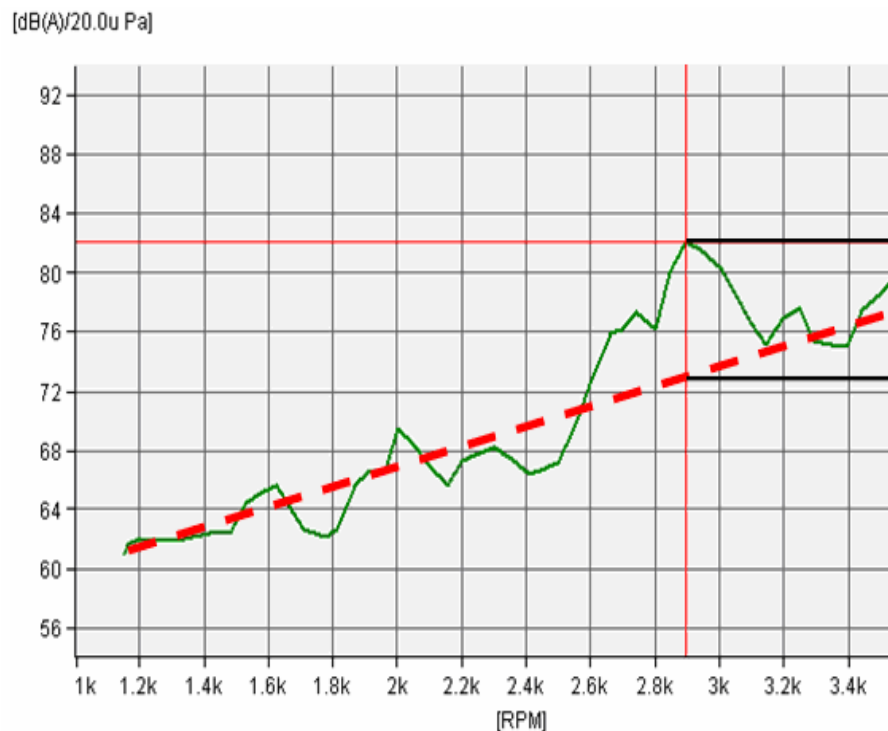


Figure 7.1 : The Target Line and the Measured Curve

Under normal circumstances, a gradual increase in sound pressure level is expected. However, it can be easily understood from the Figure 7.1 that there is an 8 dB(A) difference between the expected SPL increase, shown with red dashed line, and the measured data with bold green curve. This 8 dB(A) difference occurs because of the booming noise in the cabin.

Table 7.1: SPL Values at Different Measurement Locations

2900 RPM	Right Ear Position	Backside of Cabin	In Front of The Engine
SPL at 96 Hz	81.9 dB(A)	80.5 dB(A)	86.2 dB(A)

As it can be seen from the Figure 7.2, while the sound pressure level at the driver ear position is 81.9 dB(A), the frequency is 96 Hz. The colored scale lets us understand the peak value in the given frequency range with the lightest yellow color. In this condition, the dominant order is 2.021 which points out the engine excitation. The triaxial graphic contains three axes which show acceleration, engine speed and the frequency. By this information, the frequency and the order can be detected easily. The dominant order, marked with white line, is the second order which belongs to the engine excitation. Other colored inclined lines represent the orders of several sources in the vehicle.

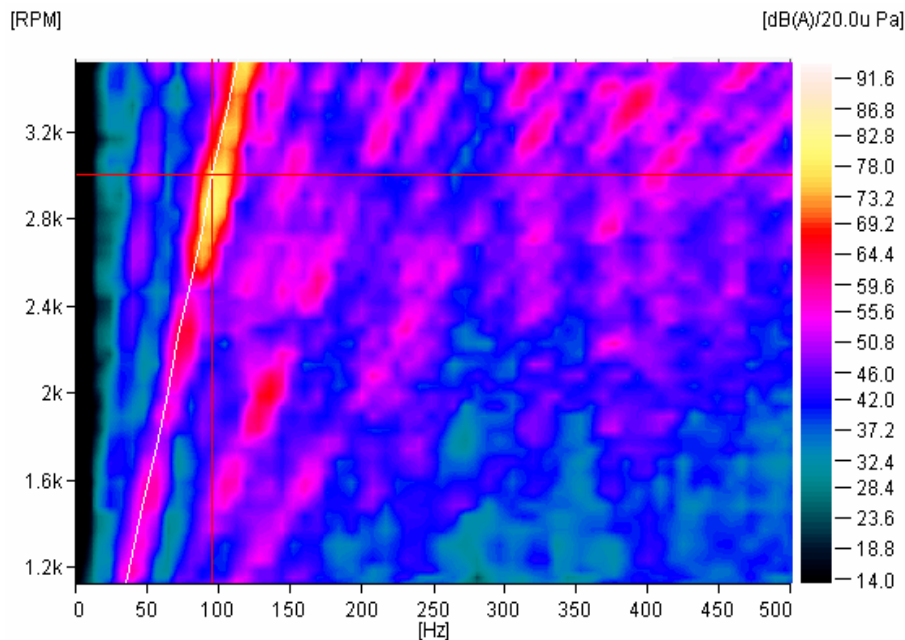


Figure 7.2 : FFT Buffer Triaxial Graph - Driver Ear Position

As mentioned in chapter 5 one feature of four-cylinder internal combustion engines is that they vibrate markedly at frequencies corresponding to the second and fourth harmonics of the engine rotation speed. From the figure above, this phenomenon can be seen easily and this resulting structural excitation is transferred to the chassis of the car via the engine mounts to the driver right ear position in cabin. However, the sound pressure level at a certain engine speed range is higher than expected because of the booming noise mechanism. In the passenger compartment, passengers will perceive an unpleasant dull booming sound in most of the compartment.

In order to overcome this problem detailed tests and measurements were made. Run-up vibration diagnostics tests, with using triaxial accelerometers leads to a detailed measurement as it is possible to have the vibration data in the entire longitudinal, vertical and the lateral directions at the same time, to identify the structural resonances and noise generated by vibrations, determination of critical speeds and resonances. By these tests, it is understood that the performance of the engine and gearbox mounts especially around 100 Hz are low.

In the hammer tests with accelerometers at the rear panel and the roof, the frequency response functions have peak values around 100 Hz. In the hammer test with microphone, the response, caused by the impact excitation, has a higher value around 100 Hz.

The aim of sound intensity measurement is to determine the location where the sound intensity is higher in the cabin. The maximum sound intensity is determined at the left door, near the loudspeaker, the rear panel, right side driver ear position, the roof at the driver right ear position and the right side of the floor.

When the input data is the excitation force from the modal exciter which is measured by the force transducer, the output data is the acceleration at the active and passive sides of the mounts. In the test, the frequency response functions have been investigated both for engine and gearbox mounts. There is a peak in FRF at the passive side of the mount around 100 Hz as seen in Figure 7.3. The longitudinal axis is represented in blue color while the vertical axis is shown with green and the lateral axis is shown by red color in the figure.

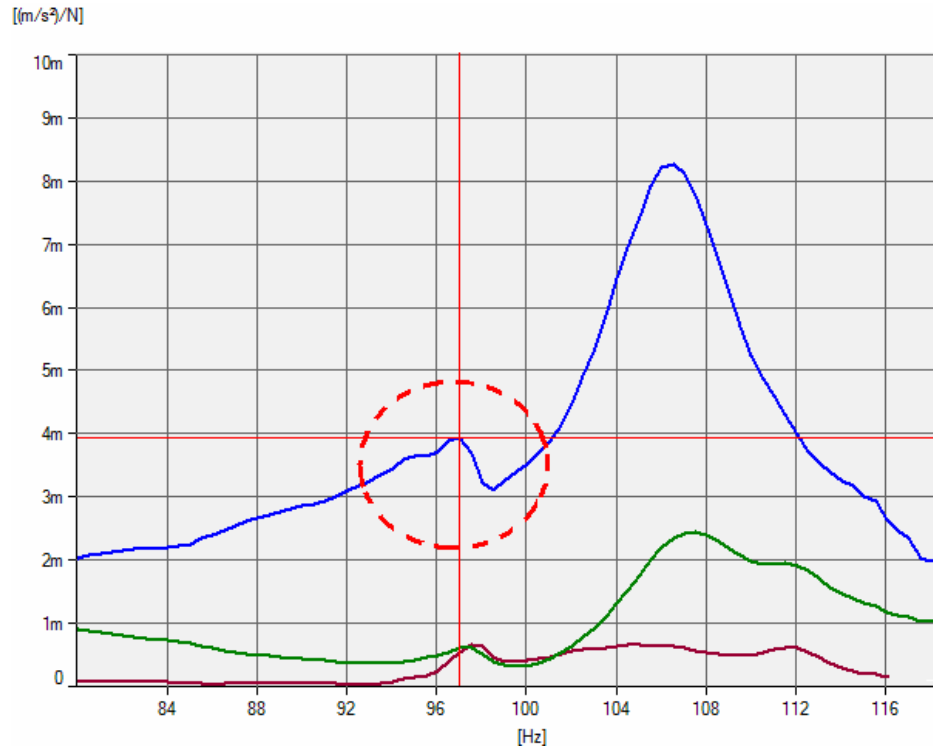


Figure 7.3 : Passive Side of the Gearbox Mount

The thing which has to be underlined is to have this peak in the frequency range where the booming noise is formed. This is one of the main sources of the unpleasant interior noise. The excitation peak in the longitudinal axis causes vibrations at the rear panel. Besides right-left door and roof vibrations originated from the engine via the transfer path along the vertical and lateral axes lead to the booming noise at the stated frequency value in most commonly used engine speeds 2500-3000 rpm.

Improvements in the performance of both the engine and gearbox mounts around 100 Hz can be suggested in the first place. Other solutions can be given as changes in the design of the cabin and rear panel to prevent the cavity resonance and avoid the excitations which cause structural resonances.

REFERENCES

- [1] **Brüel & Kjær**, 1980. Acoustic Noise Measurements, (BT 0010-12)
- [2] **Brüel & Kjær**, 1982. Noise Control - Principles and Practice, (188-81)
- [3] **Harris and Crede, McGraw-Hill**, 1976. Shock and Vibration Handbook
- [4] **Brüel & Kjær**, 1988. Technical Review No.1 - (BV 0033-11)
- [5] **National and International Standards and Recommendations**, 1981. Acoustics, Vibration & Shock, Luminance & Contrast, Brüel & Kjær.
- [6] **Brüel & Kjær**, 1991. Frequency Analysis (Brüel & Kjær Handbook BT 0007-11)
- [7] **Rosendahls Bogtrykkeri**, 2003. Brüel & Kjær BP1634 – 15
- [8] **Brüel & Kjær**, 2005. System Data, IDAe Hardware Configurations for PULSE X Types 3560 B and 3560E.
- [9] **Gade. S.**, 1982. "Sound Intensity (Part 1 Theory)". Brüel & Kjær Technical Review 3, 3-39.
- [10] **Ewins, D. J.**, 1998. "Modal Testing: Theory and Practice" Research Studies Press Ltd., Letchworth, Herts, England.
- [11] **Fahy, F, and Walker, J.**, 1998. Fundamentals of sound and vibration, E&FN Spon, London
- [12] **Brüel & Kjær Theory and Application Handbook**, 1986. Piezoelectric Accelerometers and Vibration Preamplifiers (BB 0694-12).
- [13] **Dixon J. D and Lalor N.**, 1998. Short course on vehicle noise via University of Southampton Institute of Sound and Vibration Research and Automotive Division Faculty of Mechanical Engineering, Istanbul Technical University, Istanbul, November 20-21.

- [14] **Brüel & Kjær**, 2001. Product Data, Modal Exciter - Type 4824
- [15] **Güney, A.** “Taşıtlarda titreşim ve gürültü” ders notları, İTÜ Makina Fakültesi Otomotiv Ana Bilim Dalı, İstanbul
- [16] **Brüel & Kjær**, 2004. Product Data, Miniature Triaxial DeltaTron Accelerometer Type 4506
- [17] **Ghesquiere H., Friot E., Albarrazin A.**, 1994. Active noise control for reducing booming noise in cars. Renault Research Department, Rueil-Malmaison, France
- [18] **Brüel & Kjær**, 2005. Product Data, “Source Path Contribution” (BP 2086 – 11)
- [19] **Ryu Y.**, 2005. Characteristics of NVH in Automobile and Automotive NVH Workshop via Brüel & Kjær University, ITU - OTAM, İstanbul
- [20] **Toyota NVH (Noise, Vibration & Harshness) Service Training Book**, 1996. Vol. 8. Step 3. Pub. No: TTM308E
- [21] **Brüel & Kjær**, 2004. Product Data, Sound Intensity Information, Sound Intensity Probe Kit — Type 3599
- [22] **Brüel & Kjær**, 2005. Product Data, Impact Hammers - Types 8206, 8206-001, 8206-002 and 8206-003.
- [23] **LMS International, Application Case**, 2003. “Vehicle Interior Noise Reduced by 6 dB in 5 Weeks” LMS Engineering Innovation, Leuven, Belgium.
- [24] **Brüel & Kjær**, 2005. Product Data, Power Amplifier - Type 2732
- [25] **Brüel & Kjær**, 1996. Technical Review No.2 (BV 0049-11)

AUTOBIOGRAPHY

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