Acoustical Tuning Of Car Audio System

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A Thesis Submitted to

Indian Institute of Technology Hyderabad

In Partial Fulfillment of the Requirements for

The Degree of **Master of Technology**

Department of Mechanical and Aerospace Engineering

June 2016

Declaration

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Approval Sheet

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Acknowledgements

I am grateful to my parents and teachers who taught me how to live a responsible life with patience and love.

I would like to express my gratitude to mu guide Dr.B.Venkatesham for allowing me relish freedom to work and at the same time to keep an eye to see that I do not get drifted from the main objective. I would especially like to thank him for his moral support and calmness, at the time when I have been running on low confidence.

I take this privilege to thank all the people who helped me in the completion of this project. I would like to extend thanks to Mr. Sreedhar Vanam who helped me in the development of the jury software in clearing syntax doubts, without which this project wouldn't have been completed. I would like to express our sincere gratitude to Mr.Jack Joseph, Mr. Praveen.A, Mr. Jagadeesh.M, Mr. Venkata Avinash, Mr. Sarath.K and every other member of the Hyundai Motors of India Enterprises (HMIE) team for continuous feedback on product and conducting jury test that were used by me in this project.

I would like to thank all Acoustic lab mates Sachin Birodhkar, Nagaraja Jade, Tapan Kumar Mohanta, Deepak Akiwate and Harshad Keskar for providing a good work environment and their continuous support. I would like to thank IIT Hyderabad for providing resources in carrying out our research work.

I am very much thankful to all my friends for their support and making my stay in IITH on eof the most memorable ones.

I am also thankful to each and every member of IIT Hyderabad who had been supportive throughout my stay here for the last two years.

Abstract

Today's automobile industry is doing a good amount of work in developing gadgets and applications which provide safety, comfort and experience to the users. Audio system is one among them and music has become a part of our day-to-day life.

The current research work aims at providing appropriate car audio tuning parameters that suits the requirements of Indian customers by taking into account Indian culture, quality of sound, psycho-acoustic and engineering parameters. As the music/audio will be mostly described in qualitative manner, the main challenge here is to translate this information into engineering parameters. A methodology is developed to study the car audio system. Objective and subjective tests have been conducted and the results were compared to get the appropriate tuning parameters.

This project also provides the acoustic simulations inside a car interior using a combined Finite Element (FE) and Geometric Acoustics (GA) approach. The simulations are conducted for the whole audible frequency range using the loudspeakers of the car audio system as the source. It also focuses on the acquisition of the boundary and source data required for the simulation. Straight forward determination of this data is not possible because of the complexity and inhomogeneity of the materials and loudspeaker configurations. Different methods were applied to determine required data for various materials and loudspeaker configurations inside a car compartment. In order to keep the complexity of FE simulations at a manageable level, all passive boundary conditions were considered. the outcome of these simulations are identification of the "Hotspots" which indicates about the audio quality in the low frequency region and various psycho-acoustic parameters like Initial Time Delay Gap (ITDG), Inter-aural cross correlation (IACC) and Binaural Quality Index (BQI) that describes about the audio quality in the high frequency region.

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Chapter 1

Introduction:

The way a listener determines the sound quality of a product is a very complex phenomenon. The perceived quality results from a comparison process in which the subject assesses what is actually presented to him with respect to a "desired" stimulus. For example, the best sound quality rating is reached when the system under test fulfil the listener's expectations. Moreover, for the quality assessment relies not only on the system under test itself but also on numerous external factors such as the stimulus type, background noise and the listening environment as these parameters could modify the way it is perceived and affect the sound quality.

1.1 Problem statement:

The tuning of car audio system was done taking the global music scenario into consideration. The challenge here is to find acceptance of global settings for local market. The current project aims to find evaluation procedure and develop target curves for Indian Market.

1.2 Literature survey:

Literature study aims at finding out the factors that effect the quality of sound inside the car. The main factors are -
1. Back ground noise
2. Cognitive effects
3. Listening environment
4. Sound generation by the audio system

1.2.1 Back ground noise:

Back ground noise plays an important role in determining the quality of sound inside the car. The back ground noise in a car is usually because of the noise from the engine, traffic outside, interaction between the wheel and road, etc., The car is in stationary condition (engine is off) while the sound pressure levels were measured inside the car. Hence, the effect of back ground noise on the sound inside the car can be minimized.

1.2.2 Cognitive effects:

As the sound is judged by normal people (here customers) in terms of quality, subjective tests are needed to be performed. But in the case of subjective tests the test environment plays an important role. So, it is important to maintain same type of test environment for all the subjects. In this way, the cognitive effects are also minimized.

1.2.3 Listening environment:

In sound quality of car audio problem, the listening environment is car interior. Compared to a normal listening room, a car compartment is a very special sound field. As the car compartment has only around $3m^3$ of space and is full of sound absorbing objects such as seats, carpet and roof treatment. So, reverberation of sound is very little [1]. On the other hand, as the reflective objects such as glass windows are close to speakers in a car compartment, naturally there are large early-reflected sounds close to the direct sound. Listeners feel the reflected sounds as the changed timbre of direct sounds, not as reverberation of sounds. The problem is worsened by the non-uniformity of sound pressure levels across the seating positions inside the car compartment.

To characterize the car interior [2] certain parameters are to be measured. They are Sound levels inside the car, Spaciousness and Localization, Impulse response (IR) function for various seating positions, Reverberation time and Transfer function between speaker and listener.

Spaciousness and Localization: Both are perceptual attributes. "Spaciousness" [3] refers to the perceived size of the sound source and how much you feel enveloped with sound. The main contributing factors for spaciousness are "early lateral reflections".

"Localization" [3] refers to the ability to detect the direction of a specific sound source. It is almost entirely determined by the direction of the first arrival of sound (almost always the direct sound from the speaker in this case) and thus is almost entirely immune from the effects of reflections.

Inter-aural Level Difference (ILD) and Inter-aural Time Difference (ITD) [1,4] are the parameters required to know the spaciousness and localization.

ITD is described as - sound from right side reaches the right ear earlier than the left ear. The human auditory system evaluates inter-aural time differences from Phase delays at low frequencies and Group delays at high frequencies.
ILD is described as - sound arrives from the right side has a higher level at the right ear than at the left ear, because the head shadows the left ear. These level differences are highly frequency dependent and they increase with increasing frequency.

For frequencies below 800Hz, mainly inter-aural time differences are evaluated (Phase delays), for frequencies above 1600Hz mainly inter-aural level differences are evaluated. Between 800Hz and 1600Hz there is a transition zone, where both mechanisms play a role.

Evaluation for low frequencies: For frequencies below 800Hz , the dimensions of the head (Approximate distance between two ears is 21.5 cm, corresponding to an inter-aural time delay of 625 μs), are smaller than the half wavelength of the sound waves. So, the auditory system can determine phase delays between both ears without confusion. Inter-aural level differences are very low in this frequency range, especially below about 200Hz [1, 4], so a precise evaluation of the input direction is nearly impossible on the basis of level differences alone. As the frequency drops below 80Hz, it becomes difficult or impossible to use either time difference to determine a sound's lateral source, because the phase difference between the ears becomes too small for a directional evaluation.

Evaluation for high frequencies: For frequencies above 1600Hz the dimensions of the head are greater than the length of the sound waves. An unambiguous determination of the input direction based on inter-aural phase alone is not possible at these frequencies. However, the inter-aural level differences become larger, and these level differences are evaluated by the auditory system. Also, group delays between the ears can be evaluated, and is more pronounced at higher frequencies; that is, if there is a sound onset, the delay of this onset between the ears can be used to determine the input direction of the corresponding sound source [1, 4]. This mechanism becomes especially important in reverberant environments. After a sound onset, there is a short time frame where the direct sound reaches the ears, but not yet the reflected sound. The auditory system uses this short time frame for evaluating the sound source direction, and keeps this detected direction as long as reflections and reverberaion prevent an unambiguous direction elimination.

The mechanisms described above cannot be used to differentiate between a sound source ahead of the hearer or behind; therefore additional cues have to be evaluated.

Sound localization in the median plane (front, above, back, below):
Monoaural cues: The human outer ear, i.e. the structure of the Pinna and the external ear canal, form direction-selective filters. Depending on the sound input direction in the median plane, different filter resonances become active. These resonances implant direction-specific patterns into the frequency responses of the ears, which can be evaluated by the auditory system (directional bands) for vertical sound localization [1]. Together with other direction-selective reflections at the head, shoulders and torso, they form the outer ear transfer functions.

Dynamic binaural cues: When the head is stationary, the binaural cues for lateral sound localization (ITD and ILD) [1] do not give information about the location of a sound in the median plane. Identical ITDs and ILDs can be produced by sounds at eye level or at any elevation, as long as the lateral direction is constant. However if the head is rotated, the ITD and ILD change dynamically, and those changes are different for sounds at different elevations. If the presentation of binaural cues to the two ears during head movement is reversed, the sound will be heard behind the listener.

Impulse Response(IR) function for various seating positions: In acoustic and audio applications, impulse responses are used to identify the acoustic characteristics of any closed environment.

Reverberation time: Reverberation is the persistence of sound after a sound is produced. Reverberation is frequency dependent: the length of the decay, or reverberation time [5], receives special consideration in the architectural design of spaces which need to have specific reverberation times to achieve optimum performance for their intended activity.

The time it takes for energy of a signal to drop by 60dB is the reverberation time, denoted by RT_{60}. The optimum reverberation time for a space in which music is played depends on the type of music that is to be played in the space. Rooms used for speech typically need a shorter reverberation time so that speech can be understood more clearly. If the reflected sound from one syllable is still heard when the next syllable is spoken, it may be difficult to understand to what was said. If on the other hand the reverberation time is too short, tonal balance and loudness may suffer. Reverberation changes the perceived spectral structure of a sound, but does not alter the pitch.

Basic factors that affect a room's reverberation time include the size and shape of the enclosure as well as the materials used in the construction of the room. Every object placed within the enclosure can also affect the reverberation time, including people and their belongings.

1.2.4 Car audio system:

Car audio system comprises of audio player and loudspeakers. The characteristics of these two have a good impact on the sound inside the car.

Audio Player: The audio player is characterized by it's tuning settings / equalizer settings [6] across different frequency bands. The tuning settings represent the gain in sound pressure levels given for various frequency bands.

Equalizers: Equalizers are an essential part of any sound reproducing system. They have many applications for various users. An equalizer is a filter that allows a person to control the tone (frequency response), of a sound system. The equalizer itself does not manipulate sound or sound waves. Knowing how an equalizer works is not absolutely necessary to operate one. However, it can help in making accurate judgements about them.

Loud Speakers: A loudspeaker is an electro-acoustic transducer, a device which converts an electrical audio signal into a corresponding sound. The most widely used type of speaker today is the Dynamic speaker. The dynamic speaker operates on the same basic principle as a dynamic microphone, but in reverse, to produce sound from an electrical signal. When an alternating current electrical audio signal is applied to it's voice coil (a coil of wire suspended in a circular gap between the poles of a permanent magnet), the coil is forced to move rapidly back and forth due to Faraday's law of induction. The one end of the voice coil attached to a diaphragm also moves back and forth pushing the air to create sound waves. The sound source (e.g., a sound recording or a microphone) must be amplified with an amplifier before the signal is sent to the speaker.

Speakers are typically housed in an enclosure which is often a rectangular or square box made of wood or sometimes plastic, and the enclosure plays an important role in the quality of the sound. Where high fidelity reproduction of sound is required, multiple loudspeakers are often mounted in the same enclosure, each reproducing a part of the audible frequency range. In this case the individual speakers are referred to as "drivers" and the entire unit is called a loudspeaker. Drivers made for reproducing high audio frequencies are called "Tweeters", those for middle frequencies are called "Mid-Range drivers", and those for low frequencies are called "Woofers".

Subwoofer: A subwoofer is a woofer driver used only for the lowest part of the audio spectrum: typically below 200Hz for consumer systems, below 100Hz for professional live sound. Since sound in this frequency range can easily bend around corners by diffraction, the speaker aperture does not have to face the audience, and subwoofers can be mounted in the bottom of the enclosure, facing the floor.

Woofer: A woofer is a driver that reproduces low frequencies. The driver combines with the enclosure design to produce suitable low frequencies. Some loudspeaker systems use a woofer for the lowest frequencies, sometimes well enough that a subwoofer is not needed. Additionally, some loudspeakers use the woofer to handle middle frequencies, eliminating the mid-range driver. This can be accomplished with the selection of a tweeter that can work low enough that, combined with a woofer that responds high enough, the two drivers add coherently in the middle frequencies.

Mid-Range driver: A mid-range speaker is a loudspeaker driver that reproduces middle frequencies. Mid-range driver diaphragms can be made of paper or composite materials, and can be direct radiation drivers (rather like smaller woofers) or they can be compression drivers (rather like some tweeter designs). If the mid-range driver is a direct radiator, it can be mounted on the front baffle of a loudspeaker enclosure, or, if a compression driver, mounted at the throat of a horn for added output level and control of radiation pattern.

Tweeter: A tweeter is a high-frequency driver that reproduces the highest frequencies in a speaker system. A major problem in tweeter design is achieving wide angular sound coverage (off-axis response), since high frequency sound tends to leave the speaker in narrow beams.

The terms for different speaker drivers differ, depending on the application. In two-way systems there is no mid-range driver, so the task of reproducing the mid-range sounds falls upon the woofer and tweeter. Home stereos use the designation "tweeter" for the high frequency driver, while professional concert systems may designate them as "HF" or "highs".

Crossover: When multiple drivers are used in a system, a "filter network", called a *Crossover*, separates the incoming signal into different frequency ranges and routes them to the appropriate driver. The drivers receive power only in their usable frequency range (the range they were designed for), thereby reducing distortion in the drivers and interference between them. A loudspeaker system with n separate frequency bands is described as ”n-way speakers”: a two-way system will have a woofer and a tweeter; a three-way system employs a woofer, a mid-range, and a tweeter.

Loudspeakers can be characterized by their SPL measurements, FRF measurements, Directivity and Total Harmonic Distortion(THD).

Directivity: Directivity of a loudspeaker [7] describes about the direction in which the loudspeaker is emitting maximum sound pressure level.

Frequency Response Function: Frequency response [7] is the quantitative measure of the output spectrum of a system or device in response to a stimulus, and is used to characterize the dynamics of the system. It is a measure of magnitude and phase of the output as a function of frequency with respect to the input. In simplest terms, for a given sine wave, a linear system will respond at that same frequency with a certain magnitude and a certain phase angle relative to the input.

Frequency response curves are often used to indicate the accuracy of electronic components or systems. When a system or component reproduces all desired input signals with no emphasis or attenuation of a particular frequency band, the system or component is said to be ”Flat”, or to have a flat frequency response curve.

Total Harmonic Distortion(THD): THD of a signal is a measurement of the harmonic distortion [7] present and is defined as the ratio of the sum of the powers of all harmonic components to the power of the fundamental frequency.

$$THD = \sqrt{(H_2)^2 + (H_3)^2 + ... + (H_n)^2)}/H_1 \tag{1.1}$$

$$THD = \sqrt{(H_2)^2 + (H_3)^2 + ... + (H_n)^2)}/H_1$$

$$H_2, ...H_n = \text{Power levels of the harmonics}$$

$$H_1 = \text{Power level of the fundamental tone}$$

THD is used to characterize the linearity of audio systems and the power quality of electric power systems.

In audio systems, lower distortion means the components in a loudspeaker, amplifier or microphone or the other equipment produce a more accurate reproduction of an audio recording.

1.3 Room acoustic simulations of car audio systems in car passenger compartments

Any kind of room acoustic simulation requires an appropriate geometric model and suitable mathematical models for sound radiation, reception and boundary conditions such as absorbing materials, sound reflections at room boundaries and sound reception. So does the car interior acoustic simulation. The quality and accuracy of the simulation is dependent on the accuracy of these models and their input data. Using the present day available designing and analysis tools a high quality geometric model of a car interior can be established and the boundary and source conditions can be accurately represented. However, it is important to note that the commonly applied source, boundary and reception models vary considerably depending on the underlying sound field model(e.g. FEM and GA). Moreover, while determining the acoustic boundary conditions using different methods for different types of materials, a lot of uncertainties arise. But the combination of all these methods and the best choice of a specific method for the boundaries and materials in a car compartment is to be well studied considering the applicability and uncertainty into the final sound pressure field and auralization. For both boundary and source conditions, the analysis is structured into two parts which deal with the following [8] -

(1) Which data is required in FE and GA domain and how does the different input data in both domains relate to each other?

(2) What are the conclusions from the results and their uncertainties in the case of the car materials and loudspeakers?

1.4 Establishment of a geometric model

Establishing a proper geometrical model is quite necessary for a simulation of sound field by the loudspeakers inside the car passenger compartment. Generally, it is possible to simulate in the low frequency range using FEM or BEM approach. Geometrical Acoustic Method based on ray tracing is used in the mid-to-high frequency range to simulate wave propagation in the car interior. However, the car passenger compartment [9] is a highly difficult simulation environment as compared to a normal listening room. This has many reasons. Firstly, the car interior is not designed with the primary intention of good acoustics. Relatively very few priori information exists on the acoustic characteristics of the boundary materials and sound sources in the car. The most important reason is that the car materials are mostly not accessible to a straight forward determination of their acoustic characteristics, due to curved shapes and highly inhomogeneous material structures.

The Geometrical acoustics simulation faces limitations caused by (a) various diffraction edges at the front seats and head rests, (b) high geometrical details and complex, curved shape of many boundary surfaces and (c) close vicinity of sound source and receiver to each other as well as to surrounding boundaries which is a limitation for the Image source method (ISM) and the Stochastic ray tracing method (SRT).

1.5 Boundary conditions

The boundary conditions in this project refers to the passive boundary surfaces that comprises of inner lining of the car interior [8]. This includes the car floor carpet, seat cushion material, glass windows etc., . The acoustic reflection characteristics can be described by giving suitable coupling conditions between the fluid and the structure domain. The fluid in the car interior is air. As the boundary conditions are passive, weak coupling condition is given to model acoustic reflection characteristics. Acoustic absorption characteristics can be described by modelling the absorption materials using various models as per the type of the material.

1.5.1 Required boundary data:

FE domain

The acoustic surface impedance can sufficiently define the physical behaviour of the boundary wall [8]. Following Mechel's empirical expressions [10], the general type of boundary condition at the interface of two inviscid fluids can be modelled by considering that the sound pressure and the surface normal velocity are continuous across the boundary. Value of surface impedance is given by $Z_s = \frac{p}{v_n}$. However, these equivalent formulations are more like "coupling conditions" rather than "boundary conditions" as they are not independent of the sound field in front of the boundary. However by assuming that the propagation of sound occurs only in the perpendicular direction to the boundary surface, the behaviour of an acoustic boundary can be described solely by its acoustic surface impedance Z_s, which is independent of the incident sound field. Such boundaries are called "locally reacting" as the surface normal velocity v_n at a point is dependent only on the sound pressure p at that point and is independent of the surrounding pressure distribution. This approximation is not true for elastic plates as the adjacent materials are coupled by their bending stiffness. However this approximation is admissible in case of porous absorber materials only if the absolute value of the their propagation constant is much higher than that of the air.

Hence the impedance boundary approach in the FE domain neglects possible coupling effects caused by the structural vibrations of the car. However, these effects are only considered to significantly effect the room sound field only in the low frequency range, as there will be a strong modal coupling between fluid and structural waves.

GA domain

In geometrical acoustics simulations the reflection characteristics of the room boundaries are generally modelled by assigning the diffuse-field absorption coefficient α_{diff} and scattering coefficient s to each boundary. Unlike the wave-based frequency domain models the geometrical models require these data as frequency band averages (octaves or third-octaves). With regard to the absorption characteristics at the boundary this means that both the phase shift at the boundary reflection as well as the angle dependence of the reflection coefficient are neglected in typical GA simulations. This simplification is generally admissible atleast for the late part of the impulse response.

Unlike the absorption coefficient he determination of scattering coefficient can be done with a huge simplification of the scattered process that it applies only to ideally diffuse reflecting surfaces. More accurate representation of scattering process can be done by considering all angles of incidence on a hemisphere for all considered frequency bands. For room acoustic simulations where a large number of surfaces are to be characterized, such a measurement effort is infeasible.

1.5.2 Boundary conditions for Car materials

In order to specify boundary conditions for the FE and GA domains [8], the acoustic impedances and absorption coefficients are required for the car interior materials. The car interior materials under consideration are seat cushions, floor carpet and windows. Each of these are a combination of different types of materials and have different boundary conditions.

Car seat cushions

Car seats are made up of complex and inhomogeneously layered material configurations, consisting of a leather or fabric cover, a metal frame, multiple porous layers, a plastic back lining and in some cases of high-class vehicles even heating, ventilation or massage units [8]. The car seat cushions considered in this project have a fabric seat cover, three types of porous materials of different thickness and a plastic back lining which provides rigid termination.

Car floor carpet in foot space

The leg room in the front and back of the car is lined with a porous floor carpet with a thin airtight backing which is glued to a heavy foam of inhomogeneous thickness which is formed to fit into the underlying sheet metal structure [8]. In the floor areas this carpet is mostly covered with additional car mats, which are similarly made of a porous fabric with a stiff porous backing.

Windows

The glass windows in the car [8] can be considered as almost acoustically rigid except for the lowest frequency bands. The windows were modelled as a surface mass m' with a free-field termination Z_0:

$$Z_s = j\omega m' + Z_0 \tag{1.2}$$

A low frequency approximation exists to correct the surface mass term m' for coincidence effects. The effective surface mass m'_{eff} is in this case given as a function of the angle of incidence θ_0):

$$m'_{eff} = m'[(1 - (\frac{f}{f_{cr}})^2 \sin^4\theta_0) - j(\eta(\frac{f}{f_{cr}})^2 \sin^4\theta_0))] \tag{1.3}$$

where f_{cr} is the critical frequency of the considered elastic plate. According to Mechel, these formulas are however only valid up to frequencies which lie at least half an octave below f_{cr}.

$$f_{cr} = 2000\sqrt{\frac{RT_{60}}{V}}Hz \tag{1.4}$$

where RT_{60} is the Reverberation time and V is the volume of the car compartment. For the considered glass windows f_{cr} is $3kHz$ the mass correction was only applied in the calculation of the surface impedance for the FE frequency range. The absorption characteristics for the GA frequency domain were calculated the surface mass $j\omega m'$ which gives an approximately rigid behaviour of the windows in GA frequency domain.

1.6 Source conditions

As this project focuses on the sound field in the car compartments which is excited by the car audio system, it is important to model the sound emission from the loudspeaker sound sources [8]. These sources can be modelled independent of the input signal by considering their free field transfer function in the main direction of radiation and their directivity pattern. This section deals with the method of obtaining necessary loudspeaker data for simulation purpose for both FE and GA domains.

1.6.1 Required source data in FE and GA domain

In the FE domain, loudspeaker sound source can be modelled by assigning the surface normal velocity v_n to the loudspeaker diaphragm membrane i.e., boundary surface τ_v where v_n is the surface normal membrane velocity. This source representation requires a full model of the loudspeaker in it's actual built-in situation. The directional characteristics of loudspeaker source are mostly caused by diffraction at the loudspeaker cabinet and in case of a multi-channel loudspeaker interference between the different driver membranes, such a model of the driver units (membranes) in their built-in situation could capture the directivity characteristics of the loudspeaker. Thus for FE domain, a spatial and frequency dependent scan of membrane velocity of each loudspeaker is required.

In GA domain, the loudspeakers are modelled as point sources from which sound rays are emitted into multiple directions in the room. In order to model the directional and frequency characteristics of a real sound source, it is thus necessary to suitably adjust the spherical energy distribution of the emitted sound rays in all considered frequency bands. This can be done by measuring the directivity function of the considered sound source, which gives the directional frequency characteristics of a source relative to it's principle direction.

1.6.2 Determination of source data

By taking the advantage of the radial symmetry of most loudspeakers drivers the directivity can be determined by measuring the pressure responses on a quarter circle. Laser Doppler Vibrometer is used in order to obtain the necessary source data for the room acoustic FE simulations.

As an alternative to the direct measurement of the loudspeaker membrane velocity [8], an electrical analogy network was used to model the low frequency performance of a loudspeaker and thus determine it's diaphragm velocity and radiated sound power as a function of the input voltage at the voice-coil terminals. The parameters used for this network model are widely known as "Thiele-Small" parameters [11–13]. It is important to note that electrical analogy network model is applicable in the "piston range" of a loudspeaker. This is because at higher frequencies it is no longer suitable to model the mechanical and acoustical part of the loudspeaker by the lumped components as used in the network diagram which was presented in the next section. This is because the loudspeaker and it's suspension can no longer be approximated as an ideal piston with a damped spring suspension, due to the formation of eigen modes on the loudspeaker diaphragm.

In the next step, the measured or calculated membrane velocities $v_m(f)$ at the voice cols were then used to predict the on-axis free field pressure transfer function at 1 m under the assumption that the loudspeakers work as ideal piston sources in infinite baffle.

On the other hand, in case of room acoustic GA simulations, this model can only be used with considerable precaution, that the prediction of the free field pressure transfer function for the mid- and high- frequency range contains non-negligible simplifications and that the model gives no indication about the directional characteristics of the loudspeaker system.

1.7 Structure of the report:

Chapter 1 is the project introduction which discusses the problem statement, literature survey for the target curves and acoustic simulations. Chapter 2 describes the modelling of a loudspeaker using an electrical analogy and thereby calculating the loudspeaker membrane velocities.Chapter 3 describes the modelling of porous materials using the laws of Delany and Bazely. These porous materials are used extensively inside the car. Chapter 5 describes the methodology adapted and implemented for the performing objective tests and subjective tests. It also discusses the simulation methods, tools and settings. Chapter 6 presents and discusses the results obtained by following the methodology discussed in chapter 5. It presents the Target curves which are outcome from subjective and objective tests. It also presents the simulation results from both Finite element (FE) and Geometric Acoustic (GA) domains. Chapter 7 summarizes the entire project and gives a scope of extending the work further.

Chapter 2

Loudspeaker modelling: Driver parameters and Equivalent diagrams

This section discusses about the modelling of the loudspeaker.

2.1 Loudspeaker Driver Construction

The idea of a loudspeaker driver is to move air by sending alternating electrical current through a coil positioned in a magnetic field and connected to a membrane. The schematic of the loudspeaker is shown in the Fig.2.1.

The magnet and the pole piece are used to create a magnetic field in the air gap. When an alternating current is sent through the voice coil, it will make the voice coil and the membrane attached to it move according to the frequency. The spider is used to keep the voice coil centred in the air gap, and keeping it from touching the magnet and the pole piece. The spider and the suspension is responsible for introducing mechanical resistance and compliance to pull the membrane back to it's resting position. The compliance together with the mass create a resonance frequency. The membrane, the dust cap and part of the suspension are the parts of the loudspeaker that moves the air. It is responsible for giving a better coupling to the air to more efficiently convert movements of the voice coil to movement of air. Furthermore the dust cap and the spider has to protect the air gap against dust.

2.2 Loudspeaker parameters

Voice coil resistance, R_e The voice coil resistance is the part of the voice coil impedance that is resistive. It is measured in Ω.

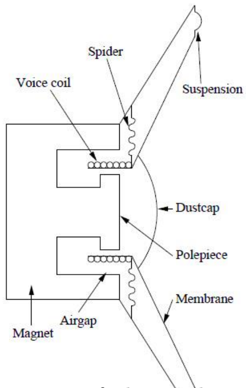


Figure 2.1: Cross-section of electrodynamic loudspeaker driver

Voice coil inductance, L_e The voice coil inductance is the part of the voice coil impedance that is reactive. It is measured in Henry.

Voice coil inductance correction factor, n The voice coil correction factor n is included to have a better model of how a lossy inductor behaves. The correction factor is used as shown in the equation-

$$j\omega L_e \rightarrow (j\omega)^n L_e \tag{2.1}$$

where n is a value between 0 and 1. When using the correction factor, the size of the inductance has to be adjusted.

Moving mass, M_m The moving mass is the weight of the membrane assembly. This includes the membrane, the dust cap, the voice coil and partly the suspension and the spider. This mass does not include the air that moves along the driver. The moving mass is measured in Kg.

Mechanical resistance, R_m The mechanical resistance is formed by the suspension and the spider of the driver. It is the part of the drivers mechanical impedance that is resistive. the mechanical resistance is measured in $\frac{Ns}{m}$.

Mechanical compliance, C_m Mechanical compliance is formed by the suspension and the spider. it is the part of the mechanical impedance that is reactive. it is responsible for pulling the membrane back to it's resting position after excitation. the mechanical compliance is measures in $\frac{m}{N}$.

Force factor, Bl The magnetic force is the product of the magnetic flux in the air gap, and the length of the wire in the voice coil. This describes the strength of the loudspeaker motor. The magnetic force factor is measured in $\frac{N}{A}$.

2.2.1 General equivalent diagram

The parameters can be used to make a model of a loudspeaker. The model consists of three parts describing the electrical, mechanical and acoustical part of the driver [14].

Electrical components The electrical part can be directly derived from knowledge of the construction of a driver. It consists of a coil and a resistor in series connection. It can be see in Fig.2.2

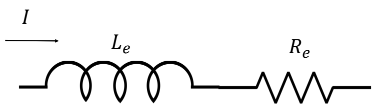


Figure 2.2: Equivalent diagram of the electrical part of the loudspeaker

Therefore, the electrical impedance is given by -

$$Z_e = R_e + (j\omega)^n L_e \tag{2.2}$$

Mechanical components The mechanical part of the model includes the moving parts of the system. That is the membrane assembly, the spider and the suspension. The weight of the moving parts M_m multiplied with the acceleration of the membrane dv_n/dt describes the force acting on the membrane. v_n is the surface normal velocity of the membrane. The spider and suspension act as a spring with a total compliance C_m. When the membrane is moved out of its resting position, this spring will pull the membrane to the resting position with a force of $(1/C_m)\int v_n dt$, where $\int v_n dt$ is the displacement of the membrane.

There is a mechanical loss R_m. This arises when movement is converted into heat in the suspension and spider of the driver. All the mechanical parts act as forces on the membrane, and they can be added together:

$$v_n.Z_{mech} = \sum external forces = M_m \frac{dv_n}{dt} + r_m v_n + \frac{1}{C_m}\int v_n dt \tag{2.3}$$

Laplace transformed:

$$\sum external forces = sM_m V_n + r_m V_n + \frac{1}{sC_m} V_n \tag{2.4}$$

The external forces is the magnet motor force,BlI. From the Laplace transformed equation it can directly be seen, that an electrical analogy should consist of a series connection of an inductor, a resistor and a capacitor. This can be seen in Fig.2.3.

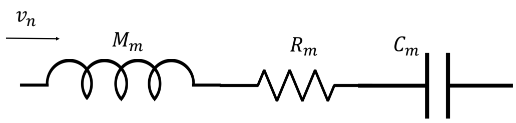


Figure 2.3: Equivalent diagram of the mechanical part of the loudspeaker

Therefore, the mechanical impedance is given by -

$$Z_m = j\omega M_m + R_m + \frac{1}{j\omega C_m} \tag{2.5}$$

Acoustical parameters The acoustical part of the model consists of two forces acting on the membrane. One on the front side of the membrane and one on the back. It is difference of these forces that has to be included in the diagram as the stationary pressure is the same on both sides of the membrane. The force acting on the membrane is defined as:

$$F = A(p_{back} - p_{front}) \tag{2.6}$$

where A is the area of the membrane. The acoustical equivalent diagram can be seen in Fig.2.4.

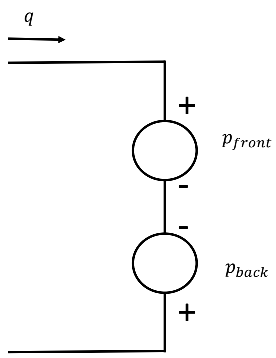


Figure 2.4: Equivalent diagram of the acoustical part of the loudspeaker

where q is volume velocity, which is defined as:

$$q = v_n.A \tag{2.7}$$

This is used to relate pressure to acoustic radiation impedance:

$$\frac{p_{front}}{q} = \frac{-p_{back}}{q} = Z_r \tag{2.8}$$

where Z_r is the acoustic radiation impedance.

The three individual equivalent diagrams can be combined to a complete equivalent diagram for the loudspeaker. The connection between the parts of the diagram is determined by the magnetic force factor Bl and the membrane area A. The connection from the electrical to the mechanical part is made by a gyrator, with a ratio of $Bl : 1$. The connection between the mechanical and the acoustical part is made by a transformer with a ratio of $A : 1$. The complete equivalent diagram is shown in Fig.2.5

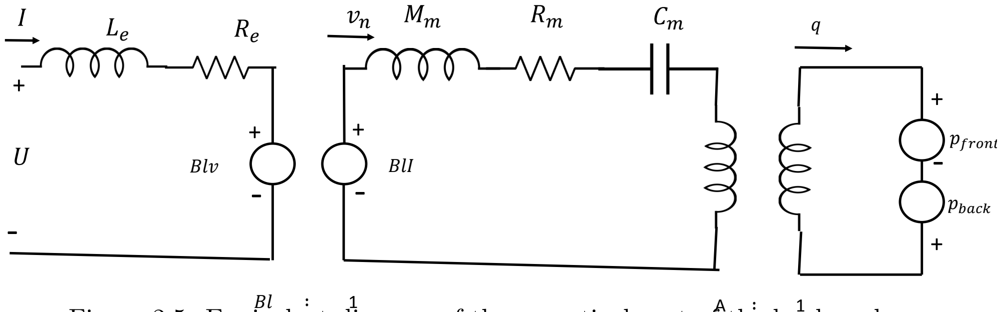


Figure 2.5: Equivalent diagram of the acoustical part of the loudspeaker

Using these parameters, the loudspeaker can be modelled by obtaining the surface normal velocities of the loudspeaker membrane. The methodology for calculating them was provided in the next section.

2.2.2 Acoustic response: Infinite Baffle case

A 5" loudspeaker mounted in an infinite baffle was considered, the acoustical radiation impedance, $Zr, acou$ [15] is given as -

$$Z_{r,acou} = \frac{\rho_0 c}{A}.(1 - \frac{2J_1\frac{2\omega a}{c}}{\frac{2\omega a}{c}} + j\frac{2H_1\frac{2\omega a}{c}}{\frac{2\omega a}{c}}) \tag{2.9}$$

where J_1 is first order Bessel function and H_1 is first order Struve function. This radiation impedance has to be included twice, since there is radiation from both the front and the back of the driver.

To simulate the acoustical response of a loudspeaker, it is most convenient to express electrical and acoustical impedances in terms of equivalent of mechanical impedance.

When impedances are moved across the transformer between the mechanical and acoustical side, the transformation factor is:

$$Z_{r,mech} = Z_{r,acou}.A^2 \tag{2.10}$$

where $Zr, acou$ is the radiation impedance of the driver and is dependent on how the driver is mounted. After shifting the acoustical parts to mechanical side, the mechanical impedance can be found as:

$$Z_{mech} = j\omega M_m + R_m + \frac{1}{j\omega C_m} + Z_{r,mech} \tag{2.11}$$

To convert everything to mechanical impedances, now all the parts on the electrical side have to be moved to the mechanical side. Since a gyrator is dividing the two sides, the process is [15]:

- Series connections of impedances change to parallel connections of admittances. This means inductors change to capacitors and vice versa.

- When moving impedances from the electrical to the mechanical side, first divide by $(Bl)^2$ and then transform to admittance.

- When moving impedances from the mechanical to the electrical side, first transform to admittance and then multiply by $(Bl)^2$.

- Voltage and current are transformed as if it was a normal transformer.

By moving all the components to mechanical side, the equivalent diagram changes as show in the Fig.2.6

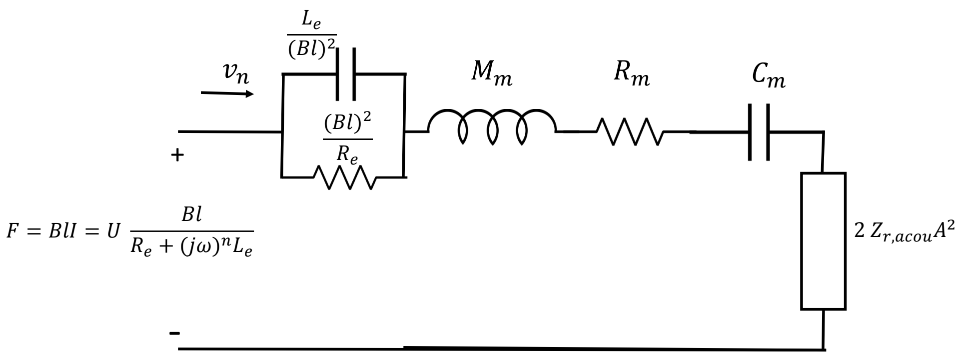


Figure 2.6: Equivalent diagram of loudspeaker driver mounted in an infinite baffle, with all component moved to mechanical side

The total mechanical impedance when the loud speaker is mounted in an infinite baffle is given by -

$$Z_{mech,tot} = j\omega M_m + R_m + \frac{1}{j\omega C_m} + 2Z_{r,acou}.A^2 \tag{2.12}$$

The membrane velocity v_n in mechanical system is in analogy to the current in the electrical system. When the membrane velocity is known, the volume velocity can be found as per the equation 2.6

$$q = v_n.A(m^3/s) \tag{2.13}$$

here q is the volume velocity, v_n is the membrane velocity and A is the effective membrane area. To calculate the pressure from the velocity v_n, the distance and the area has to be used -

$$p = |\frac{\rho_0 A v_n}{2\pi x} j\omega|(Pa) \tag{2.14}$$

where x is the distance to the source from the measurement position. The distance x is taken as $1m$. The 2π is used because of the infinite baffle, which causes the loudspeaker to radiate into a hemisphere. Other values could be 4π, describing radiation into free field or $\pi/2$ describing a speaker positioned in a corner.

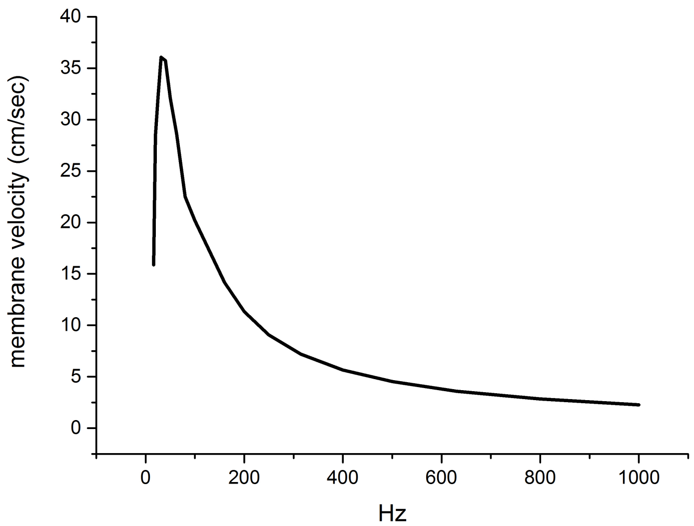


Figure 2.7: Variation of membrane velocity as a function of frequency

Chapter 3

Modelling of Porous materials

3.1 Porosity and flow resistivity in porous materials

Porosity: The porous materials [16] consists of an elastic frame which is surrounded by air. The porosity σ is the ratio of the air volume V_a to the total volume of the porous material V_T. Thus,

$$\sigma = V_a/V_T \tag{3.1}$$

Let V_b be the volume occupied by the frame in V_T. The quantities V_a,V_b and V_t are then related by

$$V_a + V_b = V_T \tag{3.2}$$

Only the volume of air which is not locked within in the frame is considered in V_a and thus in the calculation of the porosity. The latter is also know as the open porosity or connected porosity. For most of the fibrous materials and plastic foams, the porosity lies very close to 1.

Flow resistivity One of the most important parameters that influence the absorption of a porous material is its flow resistance. It is defined by the ratio of the pressure differential across a sample of the material to the normal flow velocity through the material. The flow resistivity Ξ is the specific (unit area) flow resistance per unit thickness.

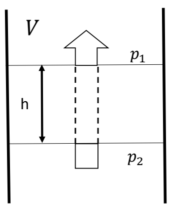


Figure 3.1: A portion of porous material inserted in a pipe. The pressure difference of $p_1 - p_2$ causes a steady flow V of air per unit of material

The schematic of the set-up for the measurement of flow resistivity,σ is shown in the Fig.3.1. The flow resistivity is given by -

$$\sigma = (p_2 - p_1)/Vh \tag{3.3}$$

In this equation, the quantities V and h are mean flow velocity of air per unit area of material and the thickness of the material respectively.

3.2 Microscopic ans macroscopic description of sound propagation in porous media

The parameters that are involved in sound propagation can be defined locally, on a microscopic scale, for example in a porous material with cylindrical pores [**?**]having a circular cross-section, as functions of the distance to the axis of the pores. On the microscopic scale because of the complicated geometries, it is difficult to study the propagation of sound in porous materials. Only the mean values of the parameters are of practical interest. The averaging must be performed on a macroscopic scale, on a homogenization volume with dimensions sufficiently larger for the average to be significant. But, it should be taken care that these dimensions should be smaller than the acoustic wavelength. The description of sound propagation in porous materials can be complicated as the sound excites and moves the frame of the material. If the frame is assumed to be motionless, the air inside the porous medium can be replaced on the macroscopic scale by an equivalent free fluid. This equivalent free fluid has a complex effective density ρ and a complex bulk modulus K. The wave number k and the Characteristic impedance Z_c of the equivalent fluid are also complex.

3.3 The laws of Delany and Bazley and flow resistivity

The laws of Delany and Bazley [16] are used to replace a layer of porous material with an equivalent layer of fluid material. Delany and Bazley have measured the complex wave number k and the characteristic impedance Z_c for a large range of frequencies in many fibrous materials with porosity close to 1. Expressions for the wave number k and the characteristic impedance Z_c given by Delany and Bazley shows that the values of k and Z_c are mainly dependent on angular frequency ω and on the flow resistivity Ξ of the material.

$$Z_c = \rho_0 c_0 [1 + 0.057X^{-0.754} - j0.087X^{-0.732}] \tag{3.4}$$

$$k = \frac{\omega}{c_0}[1 + 0.0978X^{-0.700} - j0.189X^{-0.595}] \tag{3.5}$$

where ρ_0 and c_0 are the density of air and the speed of sound in air and X is a dimensionless parameter -

$$X = \rho_0 f/\Xi \tag{3.6}$$

f being the frequency, where $\omega = 2\pi f$.

Delany and Bazley suggested the following boundary for the validity of their empirical expressions in terms of X, as follows:

$$0.001 < X < 1.0 \tag{3.7}$$

It may not be expected that these relations give a perfect prediction of acoustic behaviour of all the porous materials in the frequency range mentioned by the above equation. These expressions for k and Z_c were modified and improved by Mechel [17]. They are -

$$\begin{aligned} \frac{Z_c}{\rho_0 c_0} &= 1 + 0.0485A^{-0.754} - j0.087A^{-0.73}, A < 60 \\ &= \frac{0.5A/\pi + j1.4}{(-1.466 + j0.212A)^{\frac{1}{2}}}, A > 60 \end{aligned} \tag{3.8}$$

$$\begin{aligned} \frac{k}{k_0} &= -j0.189A^{0.6185} + 1 + 0.0978A^{0.6929}, A < 60 \\ &= (1.466 - j0.212A)^{\frac{1}{2}}, A > 60 \end{aligned} \tag{3.9}$$

where $k_0 = \omega/c_0$ and A is the normalized flow resistance of a λ-deep layer; that is

$$A = \frac{\Xi\lambda}{\rho_0 c_0} \tag{3.10}$$

Here λ is the wavelength of the air (at ambient temperature).

Chapter 4

Methodology

As mentioned in the Introduction chapter, two cars were considered for methodology development and they were named as MC1 and MC2. The developed methodology was validated on 18 cars which were named as VC1, VC2,...., VC18 . It was intended to know the following -

1. sound propagation inside these cars
2. settings of the head unit
3. development of the target curve which should provide the preliminary information about the car audio quality

The methodology was developed to know about these attributes based on -

1. Objective test
2. Subjective tests

The results from these objective test and subjective tests were together analysed to get the target curve.

4.1 Objective Test:

To know about these attributes we need to measure sound pressure levels inside the car at all the four seating positions by maintaining certain settings. To measure the sound pressure levels, we need to play standard signals through the head unit in the car.

4.1.1 Uniformity Test:

To know how uniformly the sound is being propagated inside the car across all the seating positions, a uniformity test was done.

Excitation signals:
Pink noise is used as a excitation signal in the test. Pink noise is a signal or process with a frequency

spectrum such that the power spectral density (energy or power per Hz) is inversely proportional to the frequency of the signal. In pink noise, each octave (halving / doubling in frequency) carries an equal amount of acoustic power.

Default settings:
A special care was taken such that the head unit does not induce any gains to the standard input signal. The chosen default settings were Bass = 0, Middle = 0, Treble = 0.

Procedure:
The measurement was done using a single microphone. The microphone was positioned to the level of driver's ear with the help of a tripod and holder. The volume level of the head unit is increased uniformly and sound pressure level data was stored in the form of 1/3 octave band data and FFT. This exercise was performed for all the seating positions in both the cars. The results were presented in the next chapter 5.1.1

Figure 4.1: A test setup for measuring sound pressure levels using single microphone

Fig. 4.1 shows the equipment setup used for measuring the Sound Pressure Levels to check for Uniformity inside the car.

4.1.2 Tuning conditions of the audio system :

Generally, any audio system introduce the distortion for a given input signal while reproducing. An ideal audio system will reproduce exactly the input signal. However, it is quite challenging to get an ideal system. So, each audio system is tuned for a particular car model. One of the objective for the current measurements is to find the tuning parameters of a given car. This data will help for comparing different car audio systems.

Excitation signal:

A standard Pink noise signal carries an equal amount of sound power for each octave band. Hence, when a Pink noise is played through an ideal audio system it should give a flat response (i.e., the audio system should not modify the input signal).

Head unit settings:

Default settings were maintained (Bass = 0, Middle = 0, Treble = 0). 80 dB sound pressure level was maintained at the driver's position.

Procedure:

The sound pressure level measurements were done using Earthworks M30 microphone. The measurements were taken only at the driver position maintaining 80dB. The sound pressure level information is obtained in the form of octave band data. This exercise was performed on 18 cars so that this data would be helpful in developing an algorithm to know how the audio system was tuned in a car.

The most popular genres in India according to the age is given in a SONY survey [18]. These popular genres are -

1. Rock
2. Pop
3. Electronic
4. Metallica
5. Classic
6. R&B
7. Jazz

The tuning settings of these standard genres were provided in Appendix A.

Taking these settings as a reference, we tried to find the default audio system settings in various cars using the proposed algorithms. These algorithms will provide the dominance of standard genres in a default tuning system.

Algorithm - 1: (Single Genre Identification Method)

Step-1: Audible range of frequency is considered, 32 - 16000Hz

Step-2: This frequency range is divided into octave bands

$$f_{n+1}/f_n = 2 \tag{4.1}$$

$$f_n = \text{central frequency of } (n)^{th} \text{ band}$$
$$f_{n+1} = \text{central frequency of } (n+1)^{th} \text{ band}$$
$$f_1 = 32\text{Hz}$$

Total 10 octave bands were obtained - 32, 64, 125, 250, 500, 1000, 2000, 4000, 8000, 16000Hz.

Step-3: Sound pressure levels inside the car were measured in the octave band

$$(L_p)_{band} = (L_p)_{32}, (L_p)_{64}, ..., (L_p)_{16000}$$

$$(L_p)_n = \text{Sound pressure level of } n^{th} \text{ band}$$

Step-4: The overall sound pressure level maintained at the driver's seat is given by -

$$L_{eq} = 10log_{10}(\Sigma 10^{0.1L_{pi}}) \tag{4.2}$$

The mean sound pressure level is given by -

$$L_{mean} = 10log_{10}(1/n(\Sigma 10^{0.1L_{pi}})) \tag{4.3}$$

$$\Rightarrow L_{mean} = L_{eq} - 10$$

This L_{mean} is the ideal level of L_{pi} that is to be maintained for all the octave bands for a given Pink noise. But the actual values of L_{pi} might be higher or lower than these ideal values. This gives us the variation in tuning of audio system.

Step-5: How much each octave band is deviating from the mean value was calculated.

$$Deviation, \Delta = L_{pi} - L_{mean} \tag{4.4}$$

The Δ values of all the octave bands indicate the default tuning condition of the given audio system.

Step-6: Now, the default tuning setting of a given audio system was compared with the standard genres (reference settings, S_E) and it shows the deviations from the reference. The difference was squared to apply Least Square approach.

$$(Error)_{band}, E_{band} = ((S_E)_{band} - \Delta)^2$$

Step-7: To have a good audio experience, gains should be given depending on whether it is a low frequency band or mid frequency band or high frequency band. So, the 10 octave bands were divided into 3 groups

(a) Low frequency group comprises of 32, 64, 125Hz octave bands

(b) Mid frequency group comprises of 250, 500, 1000, 2000Hz octave bands

(c) High frequency group comprises of 4000, 8000, 16000Hz octave bands

Step-8: The total squared error sum for each frequency group is calculated. This was done for all the reference settings

Sum of squared errors of low frequency group

$$E_{Low} = E_{32} + E_{64} + E_{125} \tag{4.5}$$

Sum of errors of mid frequency group

$$E_{Mid} = E_{250} + E_{500} + E_{1000} + E_{2000} \tag{4.6}$$

Sum of errors of high frequency group

$$E_{High} = E_{4000} + E_{8000} + E_{16000} \tag{4.7}$$

Step-9: E_{Low},E_{Mid} and E_{High} of each group was compared across all the reference settings and the least value was identified. It was considered that the given car audio was dominantly tuned to the reference setting corresponding to the least square error in a particular frequency group.

Step-10: A new tuning curve was formed by combing the selected reference setting values for each frequency group.

This algorithm was applied for audio systems of 18 cars. The proposed algorithm has a limitation in providing weighting to closely spaced genres. This algorithm is considering only a single genre which has least error value. It is not considering the genres whose values are very close to the least error value. These small variations in the error values might be because of some trivial reasons which may be unavoidable at the time of taking measurements. This is the limitation of this algorithm.

The limitation of algorithm 1 is reduced by proposing an alternative algorithm. The methodology is discussed as follows -

Algorithm - 2: (Muitiple Genre Identification Method)

Step-1: Audible range of frequency is considered, 32 - 16000Hz

Step-2: This frequency range is divided into octave bands

$$f_{n+1}/f_n = 2$$

$$f_n = \text{central frequency of } (n)^{th} \text{ band}$$
$$f_{n+1} = \text{central frequency of } (n+1)^{th} \text{ band}$$
$$f_1 = 32\text{Hz}$$

Total 10 octave bands were obtained - 32, 64, 125, 250, 500, 1000, 2000, 4000, 8000, 16000 Hz.

Step-3: Sound pressure levels inside the car were measured in the octave band

$$(L_p)_{band} = (L_p)_{32}, (L_p)_{64}, ..., (L_p)_{16000}$$

$$(L_p)_n = \text{Sound pressure level of } n^{th} \text{ band}$$

Step-4: The overall sound pressure level maintained at the driver's seat is given by -

$$L_{eq} = 10log_{10}(\Sigma 10^{0.1L_{pi}})$$

The mean sound pressure level is given by -

$$L_{mean} = 10log_{10}(1/n(\Sigma 10^{0.1L_{pi}}))$$

$$\Rightarrow L_{mean} = L_{eq} - 10$$

This L_{mean} is the ideal level of L_{pi} that is to be maintained for all the octave bands for a given Pink noise. But the actual values of L_{pi} might be higher or lower than these ideal values. This gives us the variation in tuning of audio system.

Step-5: How much each octave band is deviating from the mean value was calculated.

$$Deviation, \Delta = L_{pi} - L_{mean}$$

The Δ values of all the octave bands indicate the default tuning condition of the given audio system.

Step-6: Now, the default tuning setting of a given audio system was compared with the standard genres (reference settings, S_E) and it shows the deviations from the reference. The difference was squared to apply Least Square approach.

$$(Error)_{band}, E_{band} = ((S_E)_{band} - \Delta)^2$$

Step-7: To have a good audio experience, gains should be given depending on whether it is a low frequency band or mid frequency band or high frequency band. So, the 10 octave bands were divided into 3 groups

(a) Low frequency group comprises of 32, 64, 125Hz octave bands

(b) Mid frequency group comprises of 250, 500, 1000, 2000Hz octave bands

(c) High frequency group comprises of 4000, 8000, 16000Hz octave bands

Step-8: The total squared error sum for each frequency group is calculated. This was done for all the reference settings

Sum of squared errors of low frequency group

$$E_{Low} = E_{32} + E_{64} + E_{125}$$

Sum of errors of mid frequency group

$$E_{Mid} = E_{250} + E_{500} + E_{1000} + E_{2000}$$

Sum of errors of high frequency group

$$E_{High} = E_{4000} + E_{8000} + E_{16000}$$

Step-9: E_{Low},E_{Mid} and E_{High} of each group was compared across all the reference settings and the least value was identified. It was considered that the given car audio was dominantly tuned to the reference setting corresponding to the least square error in a particular frequency group. Up to this step, the procedure is same as in algorithm 1.

Step-10: Standard deviation (σ) of errors was identified across all the reference settings for all the three frequency groups.

Step-11: A Target limit was calculated for each frequency group for each car.

$$TargetLimit, TL_{freq.group} = (min.)_{freq.group} + 0.5 * (\sigma)_{freq.group} \tag{4.8}$$

Step-12: Error normalization was done based on the target limit (TL) value for all frequency groups of given cars.

Step-13: Reference settings with normalized values less than 1 are said to be with in the Target Limit (TL). Multiple genres may exist for each frequency group.

Step-14: The normalized error values conveys us how far the default audio settings of a given car was tuned from the standard genres.

This algorithm was applied for audio systems of 18 cars. The results of this algorithm were discussed in the next section.

4.2 Subjective tests:

As this problem is about the sound quality, the user perception is also to be taken into consideration. For this purpose three types jury tests were conducted [19]. Three sound samples were selected which have dominance of low, mid and high frequency ranges. These samples were modified according to the requirement of the jury tests and recorded using Head Measurement System (HMS) [20].

Figure 4.2: A test setup for recording the sound samples using HMS

Fig.4.2 shows the HMS equipment setup used for recording the samples for the jury tests.

4.2.1 Jury test - 1:

Test objective: Selection of a good car audio.

Samples: 30 second signals [21] recorded at default settings in both the methodology cars, MC1 and MC2.

Environment: Participants are not allowed to adjust the loudness and they have to take the test in a room instead of car.

4.2.2 Jury test - 2:

Test objective: To check the natural selection of car audio settings by the user, here the user is a music expert.

Samples: The selected sound samples for all three frequency ranges were played in the competitor cars, MC1 and MC2.

Environment: The user (i.e., the music expert) was allowed to adjust the loudness and settings. They have to take the test in the car itself.

4.2.3 Jury test - 3:

Test objective: Selection of a good equalizer setting for car audio.

Samples: 30 second signals recorded at default settings and the settings obtained from Jury test - 2 in both the methodology cars, MC1 and MC2.

Environment: Participants are not allowed to adjust the loudness and they have to take the test in a room instead of a car.

The jury tests were conducted on a customized software which was developed using VB.NET. Seperate modules were developed separately for jury test 1 and jury test 3.

The details of the jury tests conducted are presented in the next section.

The results from objective test and subjective tests are analyzed together to come up with an appropriate target curve.

After finding the Target curves, it was intended to know simulate the realistic sound field inside the car compartment to know the role of the car interior materials on the sound field inside the car compartment. For this purpose both FE and GA simulations were performed. The methodology followed to achieve this was mentioned in the next section.

4.3 Receiver model

For FE simulations, the sound pressure levels were calculated at the node points which are of our interest. In FE method, nodes are placed at the location of the ear of the listener. But in case of GA method, a field point mesh is generated in front of the head positions of the listeners inside the car compartment. The interested results from the GA method are calculated on these field point meshes only. The receiver model for both the methods are shown in Fig.4.3

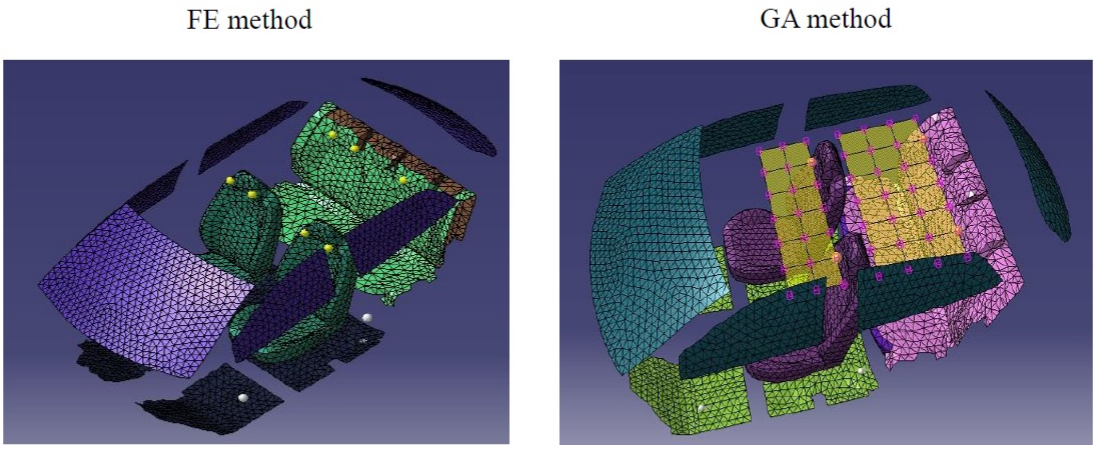


Figure 4.3: Nodes at desired locations in the FE method and Field point mesh in GA method

4.4 Simulation methods, tools and settings

The combined simulation approach uses a frequency domain FE method to calculate the room transfer function for low frequency and a time-domain GA-based algorithm for the high frequency room transfer function. The simulation results from both methods are finally combined in the frequency domain [**??**].

The source was modelled as an omnidirectional point source in the FE and GA domain. The sound power levels as a function of frequency were given as input to the sources in FE and GA domains.

With regard to the simulation set up in FE and GA domain, the following settings have been used for all simulations throughout this study.

- Geometrical model was meshed with parabolic tetrahedron elements using an average element size of approximately 80 mm. The simulations for 80mm mesh were run upto 1kHz. A minimum of three elements per wave length is required to represent the wave character of the sound field for parabolic elements, which yields the following meshing criteria:

$$l_{Element} < \frac{\lambda_{min}}{3f_{max}} = \frac{c}{3f_{max}} \tag{4.9}$$

  The frequency step width of 10 Hz was chosen. The expert settings regarding the direct FE solver was chosen i.e., MUMPS solver with a tolerance of 10^{-6} in *Virtual Lab*.

- In the GA domain the user has to specify a lot more parameters. The frequency range from 1000 - 8000 Hz was simulated in steps of 10 Hz. The simulations are done for the store levels of both "No Transient" and "Full STI" with time resolution of energy histograms = 5 ms; maximum ray path order = 5; no. of rays = 10000; reflection order = 10.

- In both FE and GA simulations, the car interior parts are provided with their respective boundary properties i.e., surface impedance Z_s. Except for the car interior materials the other part of the car compartment was provided with the property of air.

The materials of the car interior parts are porous, highly inhomogeneous in nature and have multiple layers. These porous layers were replaced by equivalent fluid materials using Delany and Bazley [16] laws. These materials should be given surface impedance as the boundary property in acoustic simulations. These simulations were performed in LMS Virtual Lab. In LMS Virtual lab, instead of modelling the car interior parts as layered structure and giving surface impedances to them individually, an equivalent surface impedance was calculated and was given as boundary condition on the outer surface of the part. This equivalent surface impedance was calculated using "Transfer matrix method".

4.5 Transfer matrix method

Transfer matrix relates the total sound pressure and volume velocity at two points in a porous element [16]. As mentioned in 1.5.1, the total sound pressure and volume velocity are assumed to be continuous across the boundary layer of the material. This assumption is useful in writing a transfer matrix for multiple layers and finding equivalent surface impedance.

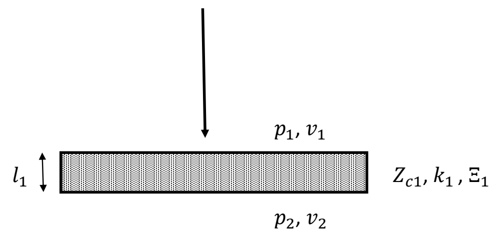


Figure 4.4: A layer of porous material with characteristic impedance Z_{c1}, wave number k_1, flow resistivity Ξ_1 and thickness l_1. p_1, v_1 and p_2, v_2 are the sound pressure and volume velocity on the face of incidence and the back face of the material

Fig.4.4 shows a layer of porous material with characteristic impedance Z_{c1}, wave number k_1, flow resistivity Ξ_1 and thickness l_1. The transfer matrix for this layer relating p_1, v_1 and p_2, v_2 can be written as -

$$\begin{bmatrix} T_1 \end{bmatrix} = \begin{bmatrix} \cos k_1 l_1 & j Z_{c1} \sin k_1 l_1 \\ \frac{j}{Z_{c1}} \sin k_1 l_1 & \cos k_1 l_1 \end{bmatrix} \tag{4.10}$$

$$\begin{bmatrix} p_1 \\ v_1 \end{bmatrix} = \begin{bmatrix} T_1 \end{bmatrix} \begin{bmatrix} p_2 \\ v_2 \end{bmatrix} \tag{4.11}$$

Similarly, transfer matrix can be written for a layered structure also. Consider a material with n layers of different thickness and different materials.

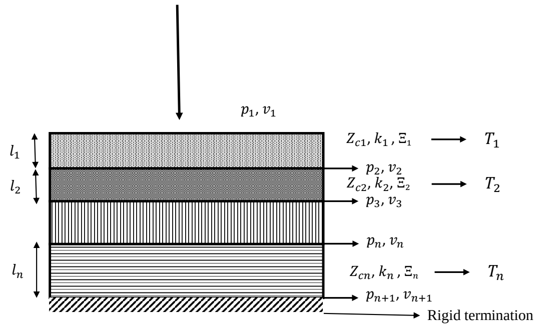


Figure 4.5: A structure with different layers of porous materials is shown. Each layer is provided with characteristic impedance Z_{cn}, wave number k_n, flow resistivity Ξ_n and thickness l_n

Fig.4.5 shows the labelled structure of a car interior material with respective properties for each layer. The overall transfer matrix foe this structure can be written as -

$$\begin{bmatrix} T \end{bmatrix} = \begin{bmatrix} \cos k_1 l_1 & jZ_{c1}\sin k_1 l_1 \\ \frac{j}{Z_{c1}}\sin k_1 l_1 & \cos k_1 l_1 \end{bmatrix} \begin{bmatrix} \cos k_2 l_2 & jZ_{c2}\sin k_2 l_2 \\ \frac{j}{Z_{c2}}\sin k_2 l_2 & \cos k_2 l_2 \end{bmatrix} \ldots\ldots\ldots\ldots$$

$$\ldots\ldots\ldots\ldots\ldots\ldots \begin{bmatrix} \cos k_n l_n & jZ_{cn}\sin k_n l_n \\ \frac{j}{Z_{cn}}\sin k_n l_n & \cos k_n l_n \end{bmatrix} \tag{4.12}$$

$$\begin{bmatrix} T \end{bmatrix} = \begin{bmatrix} T_1 \end{bmatrix} \begin{bmatrix} T_2 \end{bmatrix} \begin{bmatrix} T_3 \end{bmatrix} \ldots\ldots \begin{bmatrix} T_n \end{bmatrix} \tag{4.13}$$

$$\begin{bmatrix} p_1 \\ v_1 \end{bmatrix} = \begin{bmatrix} T \end{bmatrix} \begin{bmatrix} p_{n+1} \\ v_{n+1} \end{bmatrix} \tag{4.14}$$

Using the equations (3.3),(3.5) and (3.7), any layered structure can be replaced with an equivalent fluid and the sound pressure and volume velocity at any two different points across the layer can be related. The equivalent surface impedance for the layered structure is given by -

$$Z_s = \frac{p_1}{v_1} \tag{4.15}$$

From the above equations, it can be observed that the surface impedance is a complex number. The real part indicates resistance and the imaginary part indicates the reactance of the material. The reflection coefficient of the material is given by the expression -

$$R = \frac{\frac{Z_s}{Z_0} - 1}{\frac{Z_s}{Z_0} + 1} \tag{4.16}$$

where Z_0 is the characteristic impedance.$Z_0 = \rho_0 c_0$, $\rho = 1.225Kg/m^3$ which is he density of air and $c_0 = 344m/s$ which is the speed of sound in the air. The reflection coefficient (R) is also a complex number. The absorption coefficient (α) is given by the expression -

$$\alpha = 1 - |R|^2 \tag{4.17}$$

From the equations (3.8),(3.9) and (3.10), it can be observed that the surface impedance (Z_s), reflection coefficient (R) and absorption coefficient (α) are functions of the frequency f. From the Fig.4.3, it can be observed that the seats, floor carpet and glass windows were present in different colours in both FE and GA meshes. The materials with similar properties were grouped and were provided with equivalent properties. The configurations of the car interior parts considered and their corresponding material properties were given in Appendix F and G

The expected outputs from the Finite element simulations are the Acoustic modes and Acoustic forced response. These gives us an idea of the collection of resonances that exist in a car interior volume when it is excited by an acoustic source such as a loudspeaker. These resonances effect the low-frequency response of a sound system in the car interior volume and are one of the biggest obstacles to accurate sound production.

The expected outputs from the Geometrical Acoustic simulations are Initial Time Delay Gap (ITDG), Binaural Quality Index (BQI) and Inter Aural Cross-Correlation (IACC).

ITDG : The Initial Time Delay Gap describes the time difference between arrival of the direct wave and first strong reflection at the listener. This is shown in the Fig.4.6. Nearby sources causes a long ITDG. The ITDG is one of the most important cues for estimating the distance regarding the sound source. The ITDG isn't a constant for a room; it is dependent on source and listener position. Best values for ITDG in concert halls are somewhere between 12 and 25 milliseconds. For car interior, the ITDG value should be less than 80 milliseconds to have a good listening experience.

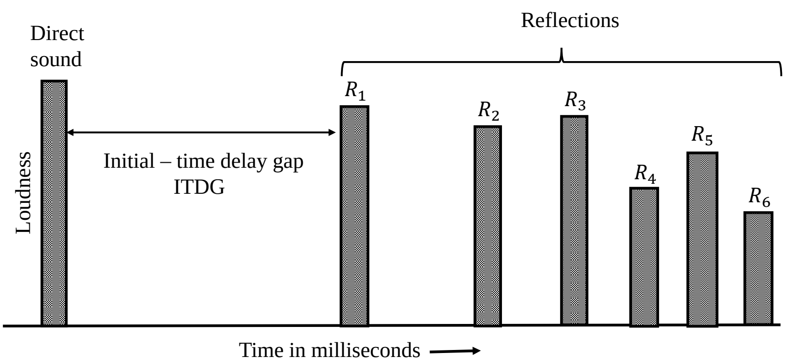


Figure 4.6: Schematic of the sound signals comprising of direct sound and reflected sounds on the time scale

IACC : To explain IACC, when a lateral reflection reaches a listener, it impinges on the closest ear without change. After it travels around the head, it will be decreased in amplitude and will be delayed in time. The IACC takes into account all of the lateral reflections impinging on both sides of the head within a stated time, including the differences in their amplitudes and time differences at the two ears. The value of IACC range between -1 and +1. If IACC = -1, it means that both the ears are getting same signal but they are opposite in phase. If IACC = +1, it means both the ears are getting the signals which are equal in amplitude and phase. IACC = 0 means the signals received by both the ears are completely different.

$$\gamma(\tau) = \frac{\int_{t_1}^{t_2} p_L(t)p_R(t+\tau)\,dt}{\sqrt{\int_{t_1}^{t_2} {p_L}^2(t){p_R}^2(t+\tau)\,dt}} \tag{4.18}$$

$$IACC = mac(|\gamma(\tau)|) \tag{4.19}$$

BQI : The Binaural quality index equals the quantity $[1 - IACC_{E3}]$ [22], where 'E' refers to the early time and '3' refers to the average of the three octave bands 500, 1 K and 2 K Hz. BQI speaks about the quality of the signal reached to the receiver when the receiver's position is not stable with respect to the source.

Chapter 5

Results and Discussion

The current chapter discuss the results of the objective test and subjective tests obtained for two car models. A detailed inference will also be presented for all the results.

5.1 Objective test:

Objective test was performed to check the uniformity of sound pressure levels inside the car and to identify the settings to which the car audio was tuned for default conditions.

5.1.1 Uniformity:

The graphs representing the uniformity of sound pressure levels inside the car are represented in Fig.5.1

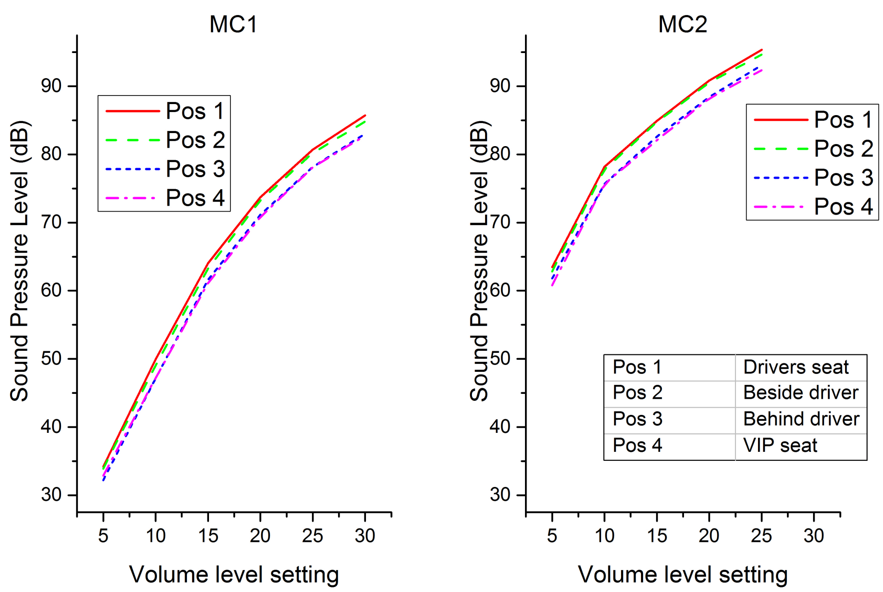


Figure 5.1: Uniformity curves of MC1 and MC2

In the uniformity graphs, the x - axis represents the volume level setting of the head unit of the car and y - axis represents the sound pressure level inside the car.

In the Fig.5.1, taking the position - 1 i.e., driver seat as the reference, from the two graphs it can be observed that the front two seating positions are having almost same amount of sound pressure levels and the rear two seating positions are having almost same amount of sound pressure levels . But there is difference between the sound pressure levels of front and rear positions of the car. This difference increases with the volume level of the audio system in the car and the difference was in the order of 3 dB.

Similarly, the uniformity condition in other two cars that are considered for uniformity test, UC1 and UC2 is shown in the Fig.5.2

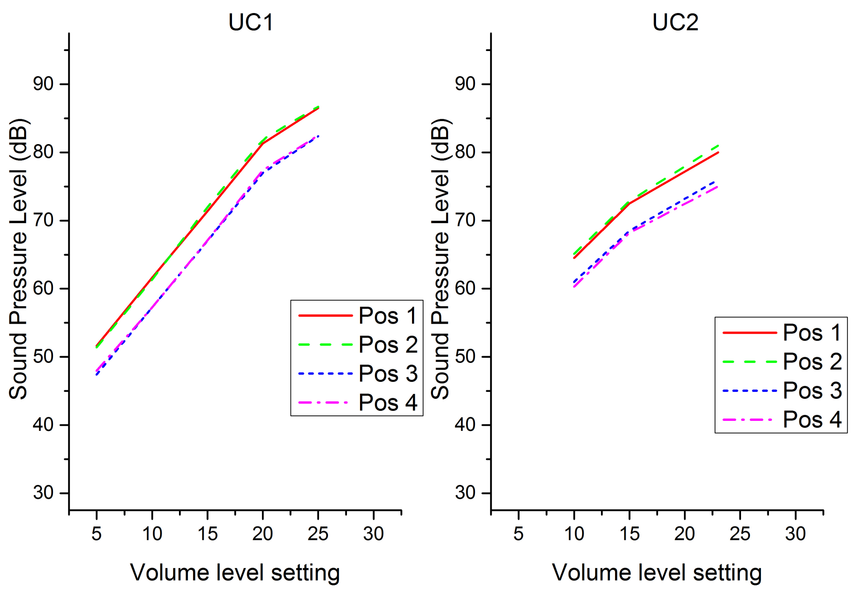


Figure 5.2: Uniformity curves of UC1 and UC2

From the Fig.5.2, it can be observed that in these two cars UC1 and UC2, the sound pressure levels between the front and rear part of the car are having more difference when compared to that of the two competitive cars MC1 and MC2. Particularly in UC2, the difference in sound pressure levels between the front and rear part of the car is as high as 5dB which is not desirable.

5.1.2 Genre identification:

This section discuss the results obtained from Algorithm - 1 and Algorithm - 2 for all 18 cars.

The methodologies of Algorithm - 1 and 2 were discussed in the previous chapter.

Algorithm - 1: (Single genre Identification Method)

This Algorithm is intended to identify which reference setting is predominant in the default tuning settings of the car in three frequency groups.

Algorithm - 1 has been validated on 18 different cars. For example let us discuss the validation car VC1 results -

Table 5.1: Algorithm - 1 results for VC1

Ref. setting	ΣE_{low}	ΣE_{mid}	ΣE_{high}
Rock	115.3	34.4	1108.1
Pop	59.2	72.5	670.7
Classical	112.8	41.8	1133.4
Metal	316.9	112.2	1784.9
Electronic	114	6.5	1141
R&B	79.9	40.8	1113.9
Jazz	104.4	34.8	1026

From Table 5.1, it can be observed that for VC1 in low frequency group "Pop" is having the minimum ΣE_{low} value. In mid frequency group "Electronic" is having the minimum ΣE_{mid} value. In high frequency group again "Pop" is having the minimum ΣE_{high} value. From this the Algorithm - 1 says that the audio system of Renault Duster is tuned to Pop for low frequency range, Electronic for mid frequency range and Pop for high frequency range in default settings.

Similarly, let us discuss another validation car VC15 results -

Table 5.2: Algorithm - 1 for VC15

Ref. setting	low	mid	high
Rock	93.1	59.8	367.5
Pop	109.0	187.0	141.8
Classical	101.1	74.4	380.0
Metal	132.5	219.3	778.1
Electronic	76.6	60.6	382.0
R&B	109.5	80.3	373.1
Jazz	78.2	62.0	318.2

Table 5.2 shows results of VC15. It can be observed that in low frequency group "Electronic" is having the minimum ΣE_{low} value. But, it can also be observed that the difference between Electronic and Jazz values is very small. This difference might be because of many trivial aspects. This is the same case between Rock, Electronic and Jazz in the mid frequency range. In high frequency group again "Pop" is having the minimum ΣE_{high} value.

From this we can say that Algorithm - 1 gives us only the standard genres that is very close to the tuning condition of the audio system of a given car. It is not considering the other standard genres which also have a influence in the tuning setting of the audio system. This has motivated us to develop Algorithm - 2.

Results of all the 18 cars by this algorithm were provided in Appendix B.

Algorithm - 2: (Multiple Genre Identification Method)

Algorithm - 2 gives multiple standard genres contribution in the tuning of the audio system of a given car for a particular frequency range.

Following the methodology of Algorithm - 2, we can say that for a given car in a particular frequency range the reference settings whose normalized $\Sigma E_{freq.range}$ values are less than or equal to 1 have their influence in the default tuning settings.

Let us discuss into the previous example of VC15 -

Table 5.3: Algorithm - 2 results for VC15

Ref. setting	low	mid	high
Rock	93.1	59.8	367.5
Pop	109.0	187.0	141.8
Classical	101.1	74.4	380.0
Metal	132.5	219.3	778.1
Electronic	76.6	60.6	382.0
R&B	109.5	80.3	373.1
Jazz	78.2	62.0	318.2

From Table 5.3, it can be observed that for low frequency range

$$\text{Minimum } E_{Low} \text{ value} = 76.6$$
$$\text{Standard deviation } (\sigma) = 19.5$$
$$\text{From Eq.2.6, Target Limit for low frequency range } (TL_{Low}) = 86.4$$

A similar exercise is followed for mid frequency and high frequency bands and their respective target limit values were calculated. Using these target limit values normalization of squared error values was done for each frequency range.

Table 5.4: Normalized values for VC15

Ref. setting	Low	Mid	High
Rock	1.0	0.6	1.5
Pop	1.2	2.0	0.6
Classical	1.1	0.8	1.6
Metal	1.5	2.3	3.3
Electronic	0.9	0.6	1.6
R&B	1.2	0.8	1.5
Jazz	0.9	0.6	1.3

From Table 5.4 it can be observed that the normalized ΣE_{low} values of Electronic and Jazz are less than 1. For mid frequency range, the normalized ΣE_{mid} values of Rock, Classical, Electronic , R&B and Jazz are less than 1. For high frequency range, the normalized ΣE_{high} values of Pop is less than 1.

The normalized sum of errors in each frequency range conveys the dominance of standard genres in a default tuned conditions or each frequency range. If the normalized value is "0", it means the audio system was tuned exactly to that reference setting. The normalized value of the target limit is "1".

Results of all the 18 cars by this algorithm were provided in Appendix C.

A further research is required to find an appropriate scaling factor for finding the Target Limit (TL) and also the contribution of single genre vs multiple genre on preferred sound quality.

These two algorithms helps us in comparing different car audio quality without subjective tests.

5.2 Subjective tests:

The purpose of the subjective test is to capture the user perception on the product quality. Based on the project objective, three types of jury tests were conducted. The results have been discussed in the following section.

5.2.1 Jury test - 1: (Pair comparison test)

For Jury test 1, as it was intended to find out which car has the better audio quality, the software was developed in a way to compare the same sound sample that was recorded at default settings in the two competitive cars MC1 and MC2. Hence, samples were presented in pairs to the subject. In this pair comparison test, the subject can listen to the samples as many times as he can and has to rate the samples on a verbal scale of 1 - 5. One pair of samples will be presented at a time to the subjects and they have to compare and rate six such pairs of samples. To maintain the consistency in the responses, the first three sample pairs were presented in a reverse order as the last three sample pairs. The responses given by the subject were recorded time to time and the software was designed in a manner such that the subject can't exit the test in the middle.

Verbal Scale:

1 - sample 1 is strongly preferred

2 - sample 1 is preferred

3 - sample 1 and sample 2 are equally preferred

4 - sample 2 is preferred

5 - sample 2 is strongly preferred

Settings: (Default settings)

Bass = 0, Middle = 0 and Treble = 0.

Sample nomenclature:

Table 5.5: Sample nomenclature for Jury test - 1

Sample type	MC1	MC2
Low Frequency	S1	W1
Medium Frequency	S2	W2
High Frequency	S3	W3

Table 5.5 gives the names of the samples that were used in Jury Test 1.

Results:

Total 21 participants took jury test 1. The responses of all the participants were collected and averaged. If the average response value is less than 3 then it can be interpreted as sample 1 is preferred than sample 2 and if the average response value is more than 3 then it can be interpreted as sample 2 is preferred than sample 1.

Table 5.6: Results of Jury test - 1

Jury Test 1		
	Avg Rating	Prefered Car
MC1 (S1) Vs MC2 (W1)	3.2	MC2
MC1 (S2) Vs MC2 (W2)	3.3	MC2
MC1 (S3) Vs MC2 (W3)	3.3	MC2
MC2 (W1) Vs MC1 (S1)	2.2	MC2
MC2 (W2) Vs MC1 (S2)	2.6	MC2
MC2 (W3) Vs MC1 (S3)	2.6	MC2

From the results provided in Table 5.6, it can be observed that MC2 car audio system at default settings was preferred to that of MC1 car audio.

The snapshot of the GUI for Jury test - 1 was presented in Appendix D.

5.2.2 Jury test - 2:

In this the user was allowed to adjust the loudness and settings of the car audio as per his requirement to attain a proper sound quality.

A music expert was made to participate and allowed him to adjust the audio settings in both the cars.

Results:

The settings adjusted by the music expert were the outcome of this test.

Table 5.7: Expert settings

MC1	MC2
Bass - +5	Bass - +5
Mid - +7	Mid - -3
Loudness - ON	Treble - +7

Table 5.7 provides the expert settings in both the competitor cars.

5.2.3 Jury test - 3: (Preference rating)

In this test, the samples recorded in MC1 & MC2 at both default and expert settings were rated on a verbal scale of 1 - 10 by jury members. This test is done only for Low frequency and Mid frequency samples because from the Algorithm - 1 and 2 results shows that all the cars were tuned to "Pop" genre for high frequencies.

Jury test 3, was intended to find out which equalizer setting was preferred by the jury. Two different settings were considered for the jury test. So, the subject has to compare four samples at once which makes the Pair Comparison Test not possible in this case. So, we opted for the "Preference Rating Test". In this test the subject has to listen to all the four samples and should give a rating to it between 1 - 10. The following settings were designed in the software for listening test -

1. the subject can't rate the sample without listening to it
2. the subject can't rate the samples without listening to all the samples atleast once
3. the subject can't exit the test in the middle.

The snapshot of the GUI for Jury test - 3 was presented in Appendix E.

Sample nomenclature:

Table 5.8: Sample nomenclature for Jury test - 3:

	Low frequency samples			Mid frequency samples	
Setting	MC1	MC2		MC1	MC2
Default	pd1	fd1		pd2	fd2
Expert	pe1	fe1		pe2	fe2

In the nomenclature of the samples provided in Table 5.8, "pd" stands for default settings in MC2 and "pe" stands for expert settings in MC1. The nomenclature "fd" and "fe" stands same for MC2 as in the case of MC1.

Results:

Total 51 participants took jury test 3. The responses of all the participants were collected and averaged. The purpose of this analysis is to know how the responses were given among the four samples in each test. For this purpose the standard deviation of the responses was calculated as -

$$\sigma = \sqrt{(x - x_i)^2/n} \tag{5.1}$$

$$x = \text{mean of the responses}$$
$$x_i = \text{individual response value}$$
$$n = \text{total no. of responses}$$

Now, the range of responses was found.

$$Range = max.response - min.response \tag{5.2}$$

No. of Standard deviations (σ) that can be accommodated in the range

$$= Range/\sigma \tag{5.3}$$

Table 5.9: Results for Jury test - 3

	Low frequency samples					Mid frequency samples			
	fd1	fe1	pd1	pe1		fd2	fe2	pd2	pe2
Average	7.1	5.8	6.2	6.9		6.9	6.7	6.7	7.5
Min	3.0	2.0	3.0	3.0		2.0	3.0	3.0	3.0
Max	10.0	10.0	9.0	9.0		10.0	10.0	10.0	10.0
Stdev	1.6	1.8	1.3	1.4		1.7	1.5	1.5	1.7
no of stdev	4.5	4.6	4.5	4.3		4.7	4.7	4.6	4.2

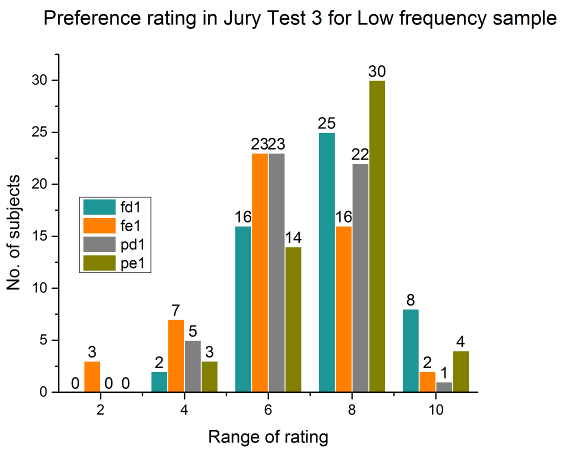


Figure 5.3: Distribution of responses among the samples for Low frequency samples in Jury test - 3

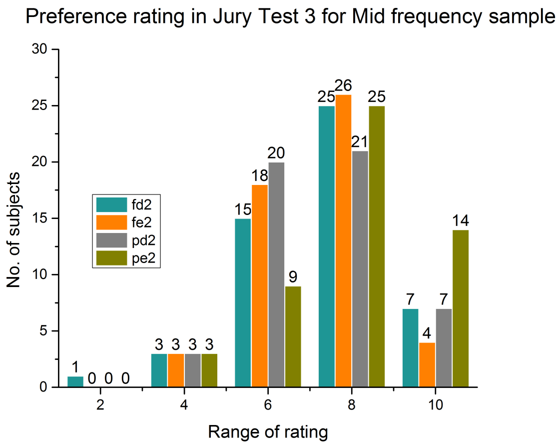


Figure 5.4: Distribution of responses among the samples for Mid frequency samples in Jury test - 3

From Table 5.9 , it can be observed that the no. of standard deviation values is between 4 and 5. So, the value was approximated to the nearest integer 5. For the given example, the size of response range is $10/5 = 2$.

From the Table 5.9, it can be observed that for low frequency range, jury response was towards the default settings in MC2 and for mid frequency range, jury response was towards the expert settings in MC1.

Fig.5.3 and Fig.5.4 shows the distribution of response values among the subjects.

5.3 Target Curves:

The target curves were developed based on subjective and objective test results. The gains of these target curves were obtained from the Algorithms.

Pink noise was played through the audio systems of MC1 & MC2 by applying default settings and expert settings. SPL data in each octave band was obtained, then Algorithm - 1 gives the Δ values. These Δ values are considered as the gains of the respective octave bands for that particular settings in both the cars.

From the subjective tests it was known that Jury preferred "MC2 default settings" for low frequencies and "MC1 expert settings" for mid frequencies. So for Target curve - 1, we considered MC2 default settings for all the frequency ranges and for Target curve - 2, we considered MC2 default settings for low and high frequency ranges and MC1 expert settings for mid frequency range. These proposed curves need to be validated in actual testing conditions.

Table 5.10: Target Curves

	MC2	MC2_Exp	MC1	MC1_Exp	Target Curve 1	Target Curve 2
32	-3.5	-2.9	-2.7	0.8	-3.5	-3.5
64	-0.9	3.0	4.8	6.1	-1.0	-1.0
125	4.7	4.8	6.3	5.3	5.0	5.0
250	3.1	3.7	0.4	-2.3	3.0	-2.0
500	0.9	-2.2	-4.4	-8.7	1.0	-9.0
1000	0.6	-3.5	-7.3	-8.3	0.5	-8.0
2000	-2.5	-3.7	-5.4	-9.4	-2.5	-9.0
4000	-2.8	-5.2	-8.2	-7.6	-2.8	-2.8
8000	-8.0	-6.1	-12.6	-9.7	-8.0	-8.0
16000	-11.9	-9.5	-20.2	-15.7	-12.0	-12.0

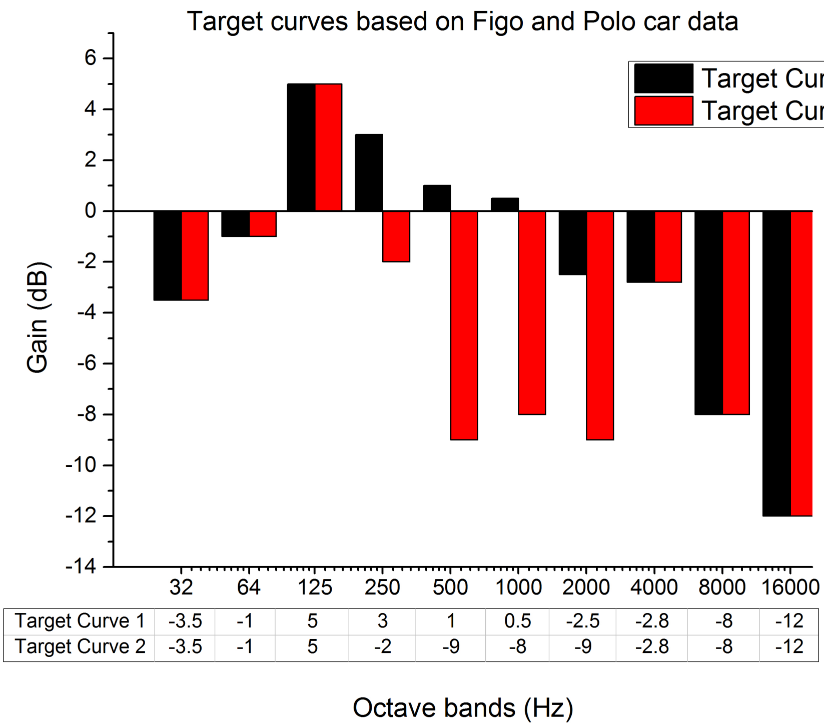


Figure 5.5: Target curves based on MC1 and MC2 data

Table 5.10 and Fig.5.5 shows the gain values that were obtained by applying the algorithms.

5.4 Boundary properties of the car interior parts using Transfer Matrix Method

This section provides the absorption coefficients and surface impedance values of the car materials which are obtained using Transfer matrix method.

Seat cushions

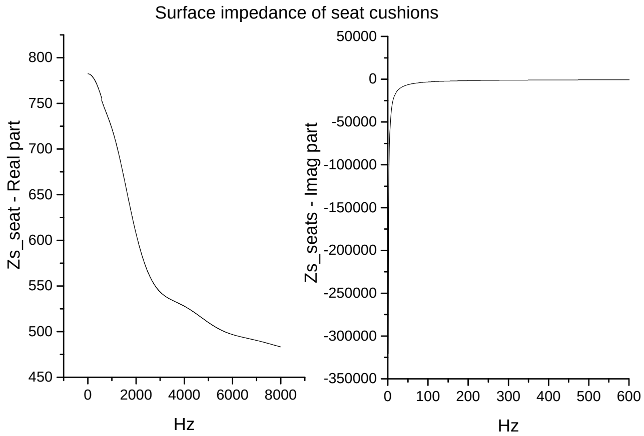


Figure 5.6: Variation of real and imaginary parts of the surface impedance of seat cushions with frequency

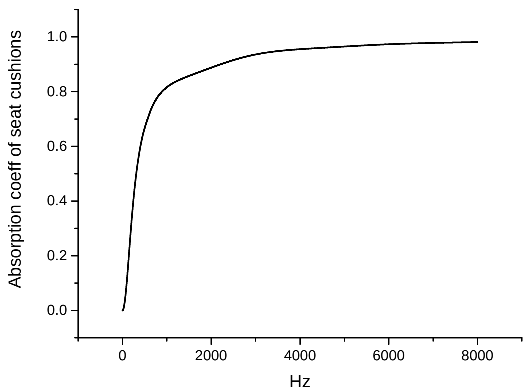


Figure 5.7: Variation of absorption coeff. of seat cushions with frequency

Glass windows

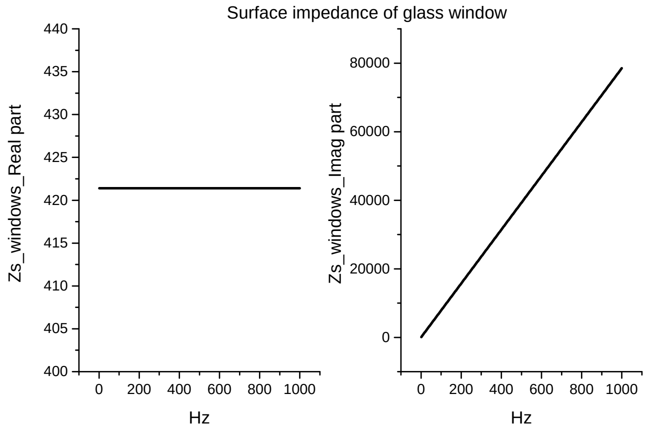


Figure 5.8: Variation of real and imaginary parts of the surface impedance of windows with frequency

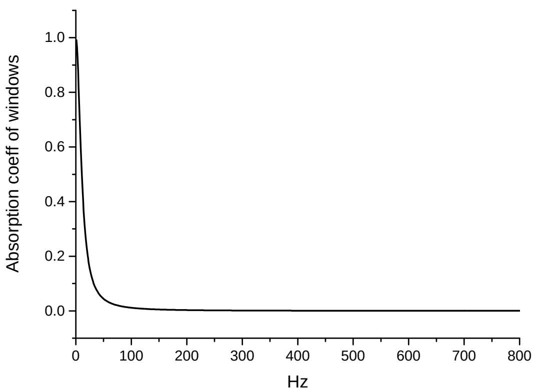


Figure 5.9: Variation of absorption coeff. of windows with frequency

Carpets

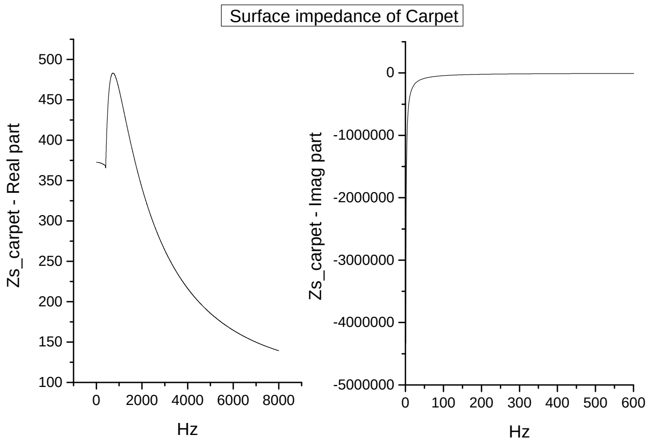


Figure 5.10: Variation of real and imaginary parts of the surface impedance of Carpet with frequency

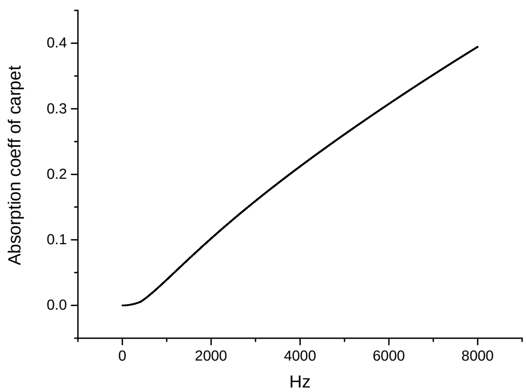


Figure 5.11: Variation of absorption coeff. of carpet with frequency

Among the three car interior materials considered, only seat cushions and carpet have comparitively good absorption characteristics. This can be observed from Fig.5.7, 5.9 and 5.11. For both seat cushions and carpet, the absorption coeff. is increasing with the frequency. But as the glass window is not a sound absorbing material, it is noticed that the absorption coeff. value of glass window rapidly falls to a value close to zero. Glass windows show higher absorption values at very low frequencies.

From the fig.5.6, 5.8 and 5.10, it can be observed that the surface impedance is a strong function of frequency. It can be observed that the materials which have better absorbing characteristics show a considerable variation of the resistance (i.e., the real part of the surface impedance). But the glass windows which doesn't have good sound absorption has it's surface impedance value almost equal to that of the Characteristic impedance. The reactance values(i.e., the imaginary part of the surface impedance) of seat cushions and carpets which have good sound absorption were almost close to zero from the low frequency range itself. But in case of glass windows the reactance values were varying linearly with the frequency.

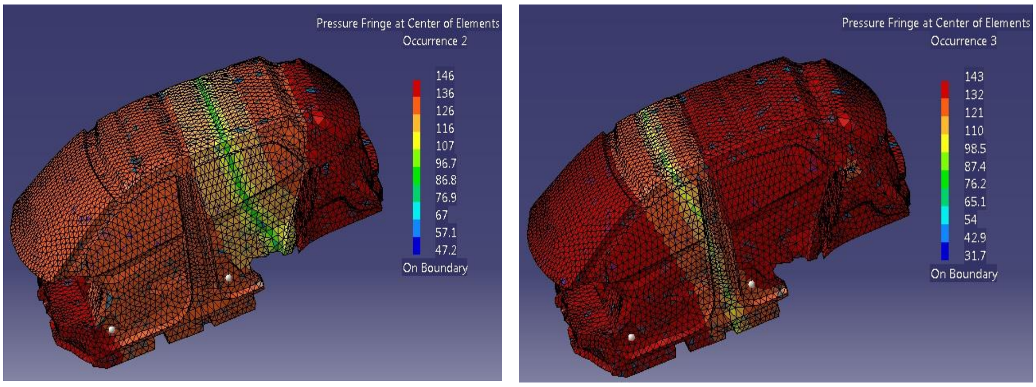
5.5 Results from FE and GA domain simulations

The results from the acoustic simulations were intended to give us an idea about the audio quality condition of the car interior. The expected results from the Finite element (FE) simulations are acoustic modes, acoustic forced response and the Inter-aural level difference (ILD). The expected results from Geometric acoustics (GA) simulations are Initial Time Delay Gap (ITDG), Inter-Aural Cross Correlation (IACC) and Binaural Quality Index (BQI).

5.5.1 Finite element simulations:

The acoustic modes are calculated. Room modes are the collection of resonances that exist in a room when the room is excited by an acoustic source such as a loudspeaker. These resonances affect the low-mid-frequency response of a sound system in the room and are one of the biggest obstacles to accurate sound reproduction. The first ten modes of the car cabin are given in the Fig.5.12

Mode no.	Hz
1	$2.81e^{-4}$
2	51.1
3	92.0
4	117.0
5	123.0
6	147.2
7	150.5
8	162.0
9	175.4
10	176.2



Fundamental mode @ 51.1Hz

Second acoustic mode @ 92.0Hz

Figure 5.12: First ten acoustic modes and mode shapes for fundamental and second acoustic modes

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From the acoustic forced response, the sound pressure levels at the nodes of interest can be obtained. The nodes are located at the positions of left and right ears of the all four listeners inside the car. The ILD for the four seating positions was given in the Fig.5.13

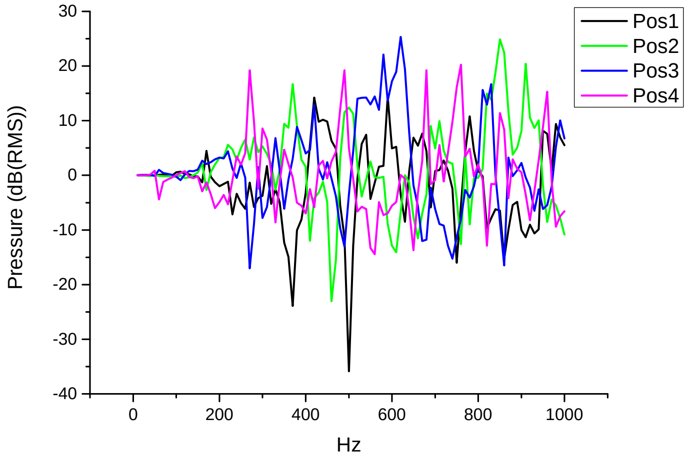

Figure 5.13: Inter-Aural Level Difference at all the four seating positions in car compartment

It can be observed from the Fig.5.13 that after around 250 Hz, at all the positions the ILD is more than 3 dB which is not desirable for a good audio experience. Hence, all the four seating positions don't have the property of a "Hotspot".

5.5.2 Geometrical Acoustics simulations:

The results from GA simulations are - ***Initial Time Delay Gap(ITDG)***

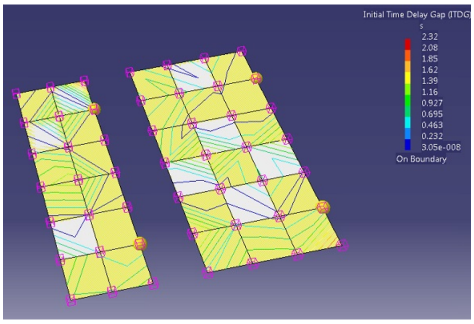

Figure 5.14: Initial Time Delay Gap at all the four seating positions in car compartment

From the Fig.5.14 it can be observed that the ITDG values went as high as 2.32 seconds which is not even in close proximity to the optimum value of 80 milliseconds. This might be due to the presence of more amount of acoustic material which causes absorption of the sound and delays the reflections.

Inter Aural cross Correlation(IACC)

It can be observed from the Fig.54, that the maximum value of IACC on the field mesh is 0.395 which says that the signals reaches to the two ears of the listener are not exactly identical. This might be because of the inappropriate orientation of the speakers in the door cavity.

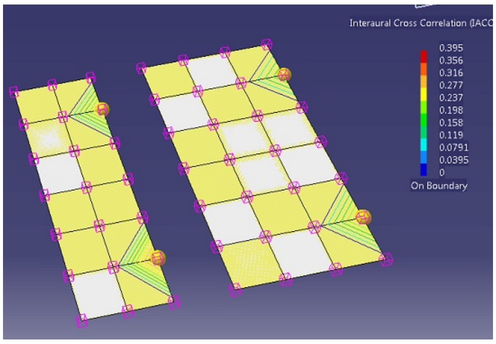

Figure 5.15: Inter Aural cross Correlation at all the four seating positions in car compartment

Binaural Quality Index(BQI)

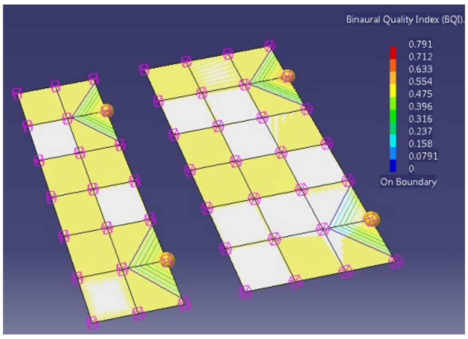

Figure 5.16: Binaural Quality Index at all the four seating positions in car compartment

The optimum value of BQI to have a good audio experience is around 0.65 [22]. In the Fig.5.16, it can be observed from the contour plots that the region around the passenger's head with optimum values of BQI is very small.

Chapter 6

Summary and Future work :

6.1 Summary:

An extensive literature study has been done to find out the factors affecting the car audio quality. Car audio measurement methodology has been established. Interior Sound Pressure Level distribution were compared in the competitor cars (Uniformity check). Indian customer's preferences in music/audio were obtained from SONY survey. It provides information on the preferred genre according to age group.

A new algorithm is proposed to find how the audio system is tuned in a given car with respect to the standard genre and measured data. It was found that almost all the cars are tuned in the same manner for High frequencies. Three sound samples were selected for jury test that represents Low, Mid and High frequency ranges. Subjective test procedure has been proposed in which three types of jury tests have been conducted to establish a target curve. In Jury Test – 1, samples recorded in MC1 and MC2 at default audio settings are used. MC2 audio is preferred in Jury Test – 1 for all the frequency ranges. Jury test 2 provides expert settings for preferred audio. In Jury Test – 3, samples recorded at default and expert settings in both the cars for low and mid frequency ranges is used. MC2 default settings are preferred for Low frequency range and MC1 expert settings are preferred for Mid frequency range. Based on the objective and subjective data analysis, two target curves have been proposed.

Simulation methodology of acoustic sound field inside the car was developed for low to mid frequency region using FE method and from mid to high frequency region using GA methods. Appropriate models for geometry, source, receiver and boundary conditions were provided for both FE and GA domains. Identification of "Hotspots" is an important outcome from the FE simulations. These Hotspots gives an idea about the quality of sound received by the passenger at low frequencies. It was observed that none of the seating positions inside the car has the property of a "Hotspot". Using the GA simulations, psycho-acoustic parameters like ITDG, IACC and BQI which describes about the quality of sound in high frequency region inside the car are verified. ITDG has a value much higher than it's desirable value, which might be due to the presence of acoustic material near the seating positions. The IACC and BQI values from GA simulations indicate that there is no proper correlation between the signals reached to the two ears of the listener. This might be because of the inappropriate orientation of the speakers in the door cavity.

6.2 Future work:

Although the methodologies developed here are very useful to understand the sound quality of audio system inside the car yet the following research topics or endeavours are worthy of further studies:

1. Validation of Target curves.
2. Characterization of Loud Speakers and Car interior.
3. Correlate the data measured in HMIE and IITH.
4. There is a good amount of research scope for fine tuning the Target curve by eliminating the current assumptions.
5. The established methodology of the simulations should be validated with the objective tests i.e., sound pressure measurements in the car compartment.
6. The subjective tests have to be conducted and the results are to be correlated with the psycho-acoustic parameters obtained from the simulations.
7. A parametric study can be done by taking different configurations of loudspeakers and by changing their location. This gives design guidelines to acoustically model the car compartment to get a good audio experience.

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Appendix

Appendix A:

Table 1: Tuning settings of Standard equalizers

Octave band(Hz)	Rock	Pop	Classical	Metal	Electronic	R&B	Jazz
32	4.5	-1.5	4.5	10.5	4	2	3.5
64	4	-1	4	7.5	3	7	2.5
125	2.5	0	3	0	1	5	1
250	1	1.5	2.5	5	-0.5	1.5	1.5
500	-1	4	-1.5	0	-2.5	-2	-1.5
1000	-1.5	4	-1.5	0	1.5	-1.5	-1.5
2000	0	1.5	0	6	0.5	2	0
4000	2	0	1.5	0	1	2.5	1
8000	3	-1.5	3	9	3.5	3	2.5
16000	4	-2	4.5	11.5	4.5	4	3

Appendix B:

Table 2: Algorithm - 1 results for all test cars

	32	64	125	250	500	1000	2000	4000	8000	16000		Low freq.	Mid freq.	High freq.
VC1	-1.5	-1	0	-0.5	-2.5	1.5	0.5	0	-1.5	-2		pop	electronic	pop
VC2	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop
VC3	-1.5	-1	0	-0.5	-2.5	1.5	0.5	0	-1.5	-2		pop	electronic	pop
VC4	2	7	5	-0.5	-2.5	1.5	0.5	0	-1.5	-2		R&B	electronic	pop
VC5	-1.5	-1	0	2.5	-1.5	-1.5	0	0	-1.5	-2		pop	classical	pop
VC6	-1.5	-1	0	2.5	-1.5	-1.5	0	0	-1.5	-2		pop	classical	pop
VC7	2	7	5	-0.5	-2.5	1.5	0.5	0	-1.5	-2		R&B	electronic	pop
VC8	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop
VC9	-1.5	-1	0	1	-1	-1.5	0	1	2.5	3		pop	rock	jazz
VC10	2	7	5	1.5	-1.5	-1.5	0	1	2.5	3		R&B	jazz	pop
VC11	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop
VC12	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop
VC13	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop
VC14	3.5	2.5	1	1	-1	-1.5	0	0	-1.5	-2		jazz	rock	pop
VC15	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		electronic	rock	pop
VC16	2	7	5	1.5	-1.5	-1.5	0	0	-1.5	-2		R&B	jazz	pop
VC17	3.5	2.5	1	1	-1	-1.5	0	0	-1.5	-2		Jazz	rock	pop
VC18	-1.5	-1	0	1	-1	-1.5	0	0	-1.5	-2		pop	rock	pop

Appendix C:

Table 3: Algorithm - 2 results for all test cars

	Rock	Pop	Classical	Metal	Electronic	R&B	Jazz	Total
VC1(L)	1.1	0.6	1.1	3.1	1.1	0.8	1.0	2.0
VC1 (M)	1.5	3.1	1.8	4.8	0.3	1.7	1.5	1.0
VC1 (H)	1.3	0.8	1.4	2.1	1.4	1.3	1.2	1.0
VC2(L)	1.3	0.6	1.3	3.1	1.1	1.0	1.1	2.0
VC2(M)	0.5	2.6	0.8	2.7	0.9	0.5	0.6	5.0
VC2(H)	1.6	0.5	1.7	3.7	1.7	1.7	1.4	1.0
VC3(L)	1.2	0.9	1.2	1.4	1.1	1.3	1.1	1.0
VC3(M)	1.0	1.1	1.0	1.2	1.0	1.0	1.0	1.0
VC3(H)	1.1	1.0	1.1	1.2	1.1	1.1	1.1	1.0
VC4(L)	0.9	0.9	0.9	2.4	0.8	0.7	0.8	6.0
VC4(M)	0.5	2.2	0.9	2.8	0.5	0.8	0.7	5.0
VC4(H)	1.6	0.4	1.7	4.1	1.8	1.7	1.4	1.0
VC5(L)	1.4	0.4	1.3	4.2	1.2	1.4	1.1	1.0
VC5(M)	0.7	1.4	0.6	2.8	1.3	1.3	0.7	3.0
VC5(H)	1.6	0.6	1.6	3.4	1.6	1.6	1.4	1.0
VC6(L)	1.3	0.6	1.4	3.2	1.2	1.0	1.1	1.0
VC6(M)	0.8	1.0	0.7	2.2	0.9	1.2	0.8	5.0
VC6(H)	1.5	0.7	1.5	2.8	1.5	1.5	1.3	1.0

	Rock	Pop	Classical	Metal	Electronic	R&B	Jazz	Total
VC7(L)	0.9	0.6	0.9	3.2	0.9	0.5	0.8	6.0
VC7(M)	0.5	2.5	0.5	3.3	0.4	0.9	0.4	5.0
VC7(H)	1.6	0.6	1.6	3.4	1.7	1.6	1.4	1.0
VC8(L)	1.5	0.6	1.5	3.0	1.4	1.3	1.3	1.0
VC8(M)	0.6	1.4	0.7	3.0	1.2	1.2	0.7	3.0
VC8(H)	1.5	0.5	1.6	3.9	1.6	1.5	1.3	1.0
VC9(L)	1.5	0.7	1.5	2.6	1.4	1.2	1.3	1.0
VC9(M)	0.6	2.4	0.7	2.8	1.3	1.0	0.6	4.0
VC9(H)	0.6	0.6	0.7	3.9	0.6	0.6	0.4	6.0
VC10(L)	1.1	0.9	1.1	2.9	1.2	0.6	1.1	2.0
VC10(M)	0.5	2.8	0.5	3.1	1.1	0.9	0.4	4.0
VC10(H)	1.3	0.8	1.3	1.9	1.3	1.3	1.3	1.0
VC11(L)	1.3	0.5	1.3	3.5	1.1	1.0	1.0	1.0
VC11(M)	0.6	2.3	0.7	2.6	1.9	1.1	0.6	3.0
VC11(H)	1.7	0.3	1.7	4.6	1.8	1.7	1.4	1.0
VC12(L)	1.3	0.6	1.3	3.0	1.2	1.0	1.1	2.0
VC12(M)	0.2	3.3	0.4	4.2	1.6	1.0	0.3	4.0
VC12(H)	1.7	0.3	1.7	4.7	1.8	1.7	1.4	1.0
VC13(L)	1.4	0.5	1.5	3.6	1.3	1.2	1.1	1.0
VC13(M)	0.5	2.5	0.6	2.6	2.0	1.0	0.6	3.0
VC13(H)	1.6	0.3	1.7	4.7	1.8	1.6	1.3	1.0
VC14(L)	0.9	1.1	1.0	2.0	0.8	0.8	0.8	5.0
VC14(M)	0.6	2.0	0.7	2.5	1.1	1.0	0.6	3.0
VC14(H)	1.6	0.6	1.7	3.5	1.7	1.7	1.4	1.0

	Rock	Pop	Classical	Metal	Electronic	R&B	Jazz	Total
VC15(L)	1.1	1.3	1.2	1.5	0.9	1.3	0.9	2.0
VC15(M)	0.6	2.0	0.8	2.3	0.6	0.9	0.7	5.0
VC15(H)	1.5	0.6	1.6	3.3	1.6	1.6	1.3	1.0
VC16(L)	1.1	1.3	1.1	3.7	1.3	0.5	1.2	1.0
VC16(M)	0.7	2.3	0.7	2.0	1.1	0.9	0.7	4.0
VC16(H)	1.4	0.8	1.4	2.3	1.4	1.4	1.3	1.0
VC17(L)	1.1	0.9	1.2	1.8	0.8	1.4	0.8	3.0
VC17(M)	0.8	1.4	1.2	2.1	1.0	1.0	0.9	3.0
VC17(H)	1.6	0.5	1.7	3.7	1.7	1.6	1.4	1.0
VC18(L)	1.2	0.7	1.2	2.7	1.1	0.8	1.1	2.0
VC18(M)	1.0	2.2	1.3	3.5	0.5	1.3	1.1	1.0
VC18(H)	1.7	0.3	1.7	5.1	1.8	1.7	1.3	1.0

Appendix D:

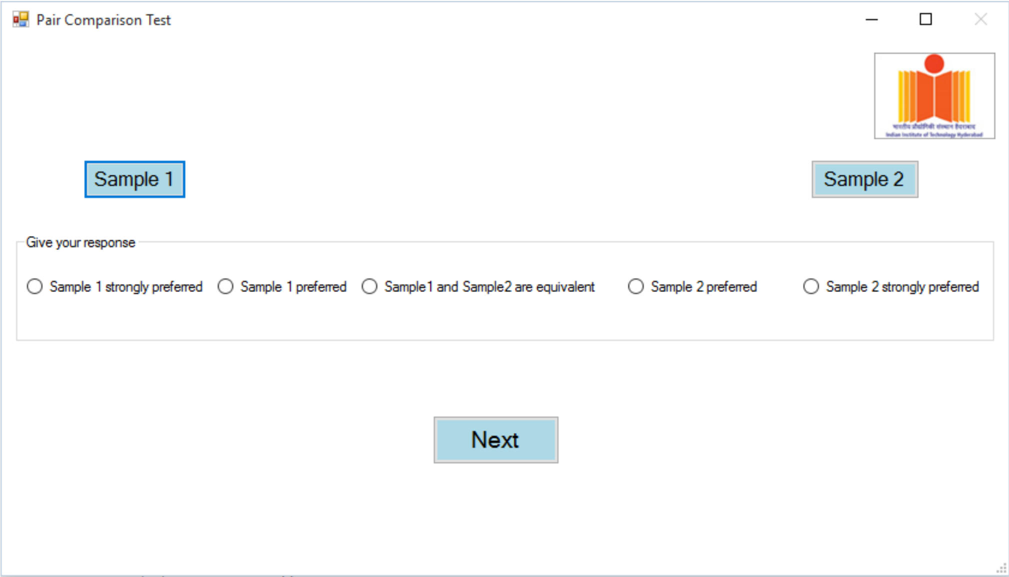


Figure 1: Snapshot of GUI for Jury test - 1 (Pair Comparison Test)

Appendix E:

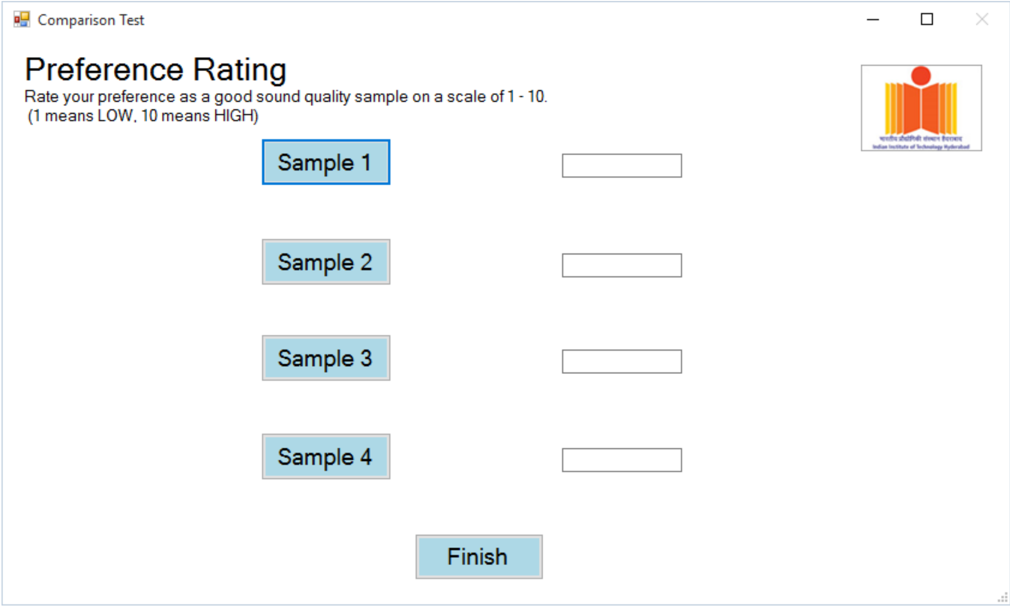


Figure 2: Snapshot of GUI for Jury test - 3 (Preference Rating Test)

Appendix F:

The layered absorber configurations of the car interior materials are:

seat cushion: A $\rightarrow$ B $\rightarrow$ C $\rightarrow$ D $\rightarrow$ Y
floor carpet: E $\rightarrow$ Y
windows: F $\rightarrow$ Z

Appendix G:

Description of the layers according to their properties:

Table 4: Thickness and Flow resistivity of the absorber materials that were considered for the seat cushions and car carpet

Letter indicator	**Material**	**Thickness,** mm	**Flow resistivity,** $KPas/m^2$
A	Fabric cover for seats	1.3	800
B	White porous material	8.5	11.9
C	Grey porous foam	19.2	21.8
D	Beige porous foam	54	42.5
E	Black carpet (floor carpet)	4.0	30

Properties of glass plate are given as -

. Thickness = 5 mm

. Density = 2500 Kg/m^3

. Young's modulus = 70,000 N/mm^2

. Poissons ration = 0.3

The termination conditions for the car interior materials are -
Y : Rigid termination ($Z_{term} \rightarrow \infty$)
Z : Free field termination ($Zterm = Z_0$)

Appendix H:

Low frequency analysis of the acoustic simulation of sound field inside the car passenger compartment

This chapter explains the step-by-step procedure for the simulation of sound field inside the car compartment.

1. Open LMS Virtual.Lab. Start the Acoustic Harmonic FEM module. This will create a new analysis window with a pre-defined tree.

Start → Acoustics → Acoustic Harmonic FEM

2. Import the volume mesh of the desired car model with a compatible mesh file format as shown in Fig.3.

Links Manager → Right Mouse Button → Import

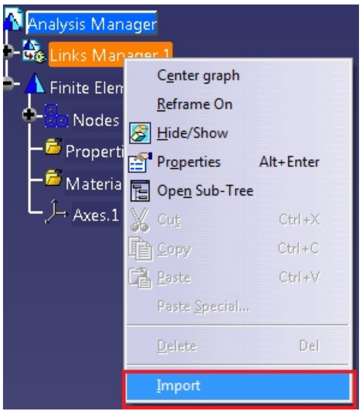


Figure 3: Importing the mesh (1)

3. After selecting the desired file, a window will open as shown in Fig.4. Select the working units for *Length* and *Mass* as required. The rest of the units are preferably kept as defaults.

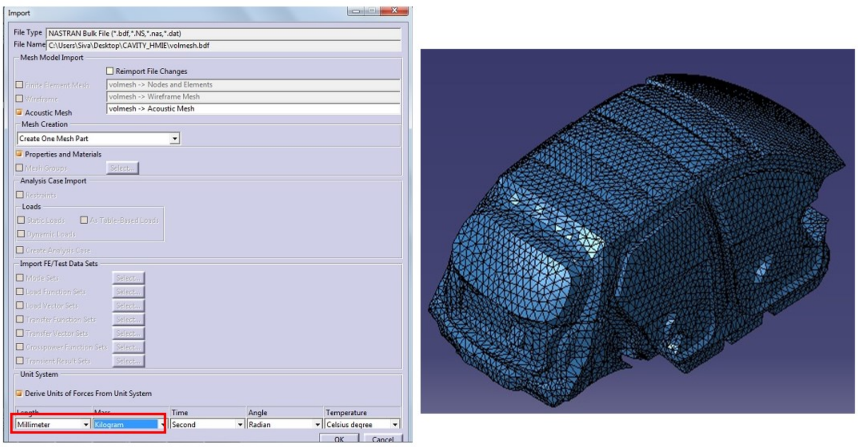


Figure 4: Importing the mesh (2)

53

4. The imported mesh needs to be set as an acoustic mesh to carry out the acoustic analysis.

Set as acoustic part as shown in Fig.5
Nodes and Elements → Expand → (mesh-file-name) → RMB → Set Mesh part Type → Set as Acoustical Mesh Part

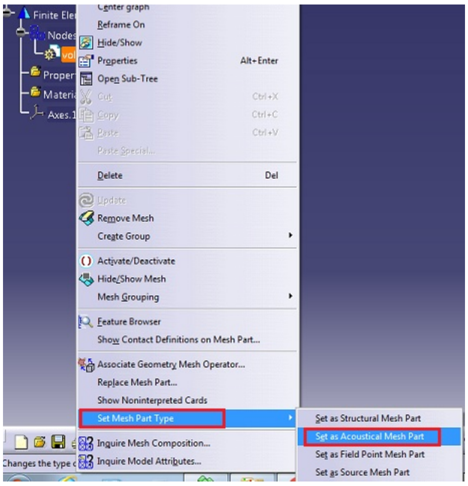


Figure 5: Setting as Acoustic mesh

5. Define acoustic material with properties *Sound Velocity* and *Mass Density*
Materials → RMB → New Materials → New Fluid Material
A new window shows up to input the valus of sound velocity and mass density as shown in Fig.6

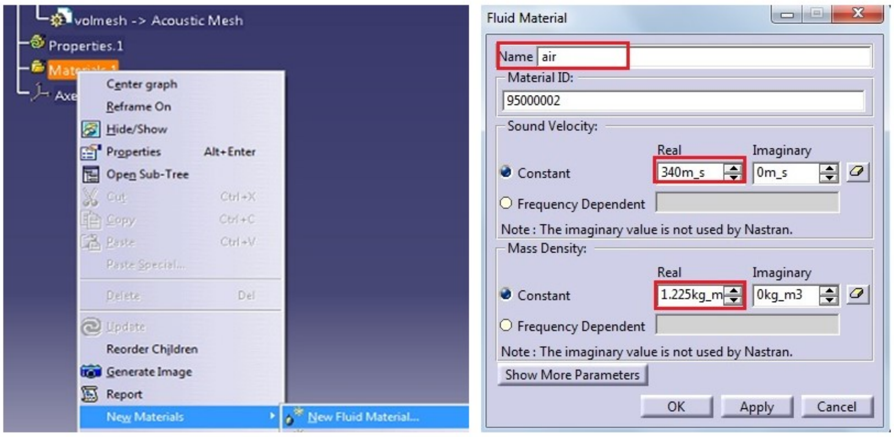


Figure 6: Creating an acoustic fluid material

6. The above created acoustic material needs to be assigned to the acoustic mesh. For this, a new acoustic fluid property has to be defined. 54
Properties → RMB → New Acoustic Properties → New Acoustic Fluid Property
As shown in Fig.7, a small window pops up. The *Application Region* is the acoustic mesh and the created material is applied in the *Fluid Material* selection.

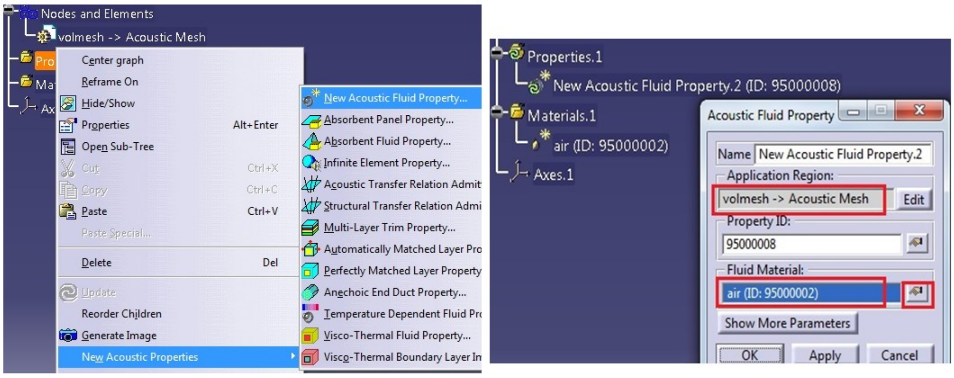


Figure 7: Assigning acoustic fluid material to acoustic mesh 55

7. Now Modal analysis has to be performed to know the contribution of the structure on the sound field inside the car compartment. To perform modal analysis, we have to insert the *Acoustic Modes case.*
Insert → Modal Analysis cases → Acoustic Modes case
As shown in Fig.8, a small window pops up for the specification of the boundary conditions.

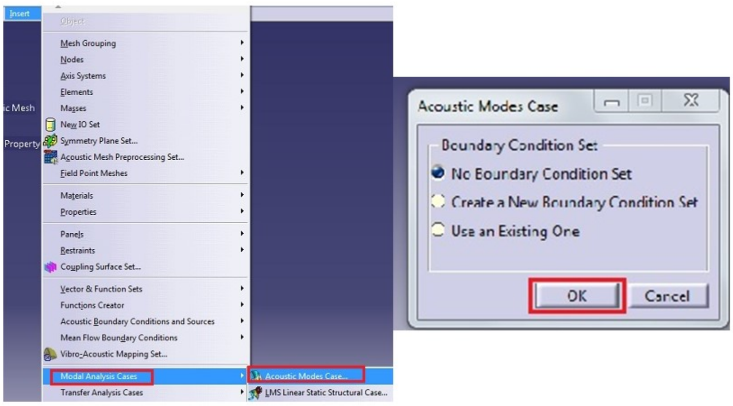


Figure 8: Acoustic modal analysis (1) 56

8. After the *Acoustic modes case* was inserted, *Acoustic modes case* will present in the *Analysis Manager* tree as shown in the Fig.9.
Acoustic modes case → RMB → Acoustic Modes Solution Set.1

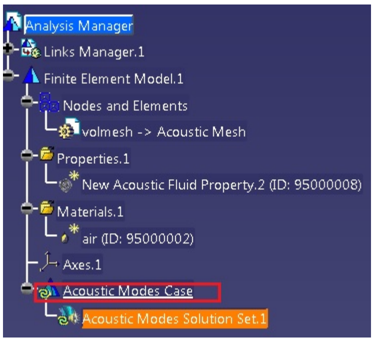


Figure 9: Acoustic modal analysis (2)

9. To check the maximum frequency level to which all the elements in the acoustic mesh are valid, the following sequence has to be followed -
Acoustic Modes Solution Set.1 $\rightarrow$ *RMB* $\rightarrow$ *Maximum Frequency Report*
Then a window pops up as in Fig.10 showing the validity of various percent of elements for the respective maximum frequencies.

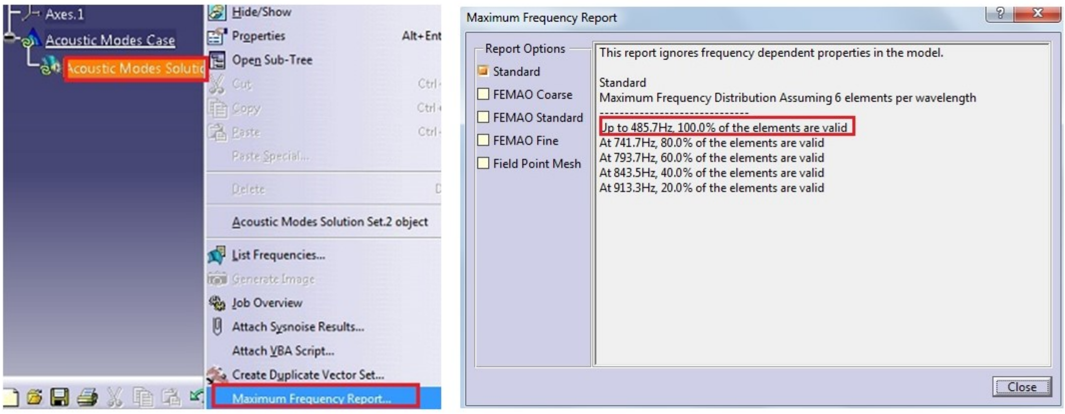


Figure 10: maximum frequency report

10. To perform the modal analysis, solution parameters have to be selected as shown in Fig.11. 57
Acoustic Modes Solution Set.1 $\rightarrow$ *RMB* $\rightarrow$ *Acoustic Modes Solution Set.1 object rightarrow Definition*
11. After the selection of desired no. of modes or desired frequency range, the Acoustic Modes Solution Set.1 has to be updated for the calculation of acoustic modes.
Acoustic Modes Solution Set.1 $\rightarrow$ *RMB* $\rightarrow$ *Update*
A job details window pops up as shown in Fig.12

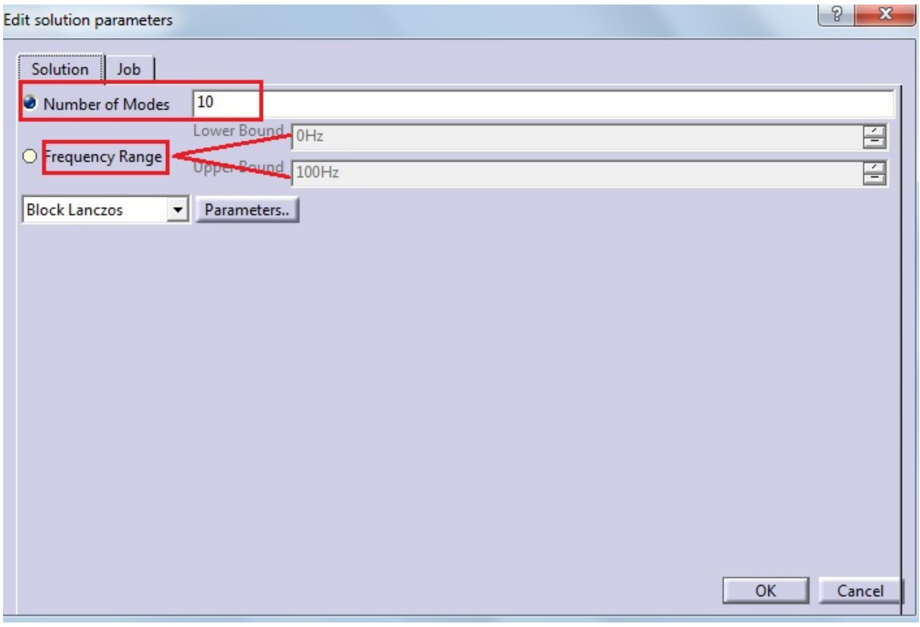


Figure 11: Job parameters for Acoustic modal analysis

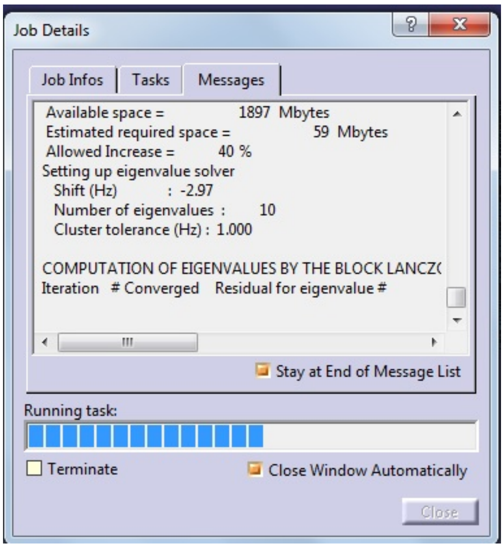


Figure 12: Calculation of acoustic modes

12. To visualize the acoustic modes -
Acoustic Modes Solution Set.1 → *RMB* → *Generate Image*
A window will show up as in Fig.13 in which we have to specify the parameter to be displayed and the mode to be visualized. 13. An *Acoustic Mesh preprocessing set* has to be included in the

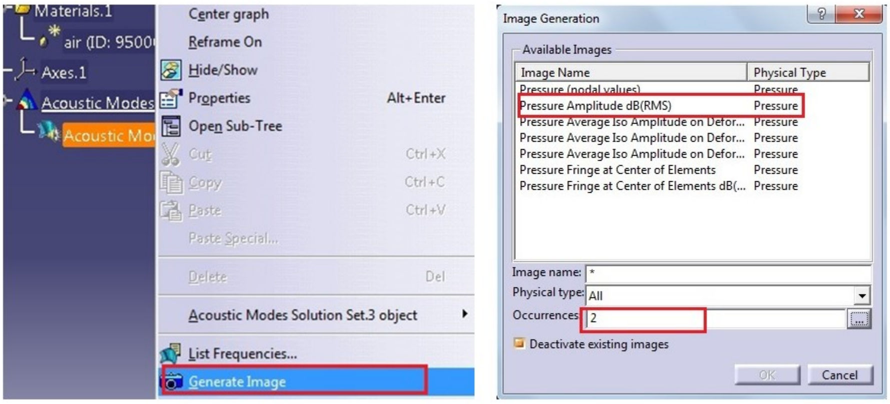


Figure 13: Visualization of acoustic modes

Analysis manager tree as shown in Fig.14 to perform *Acoustic Response case* which is required to 58
simulate the sound field due to the excitation from loud speakers.
Insert $\rightarrow$ *Mesh Preprocessing set*
After this an *Acoustic Envelope* will be created under the *Nodes and Elements* section and *Acoustic Mesh Preprocessing Set* section will be created in the *Analysis Manager* tree. After this, all the sections have to be updated.

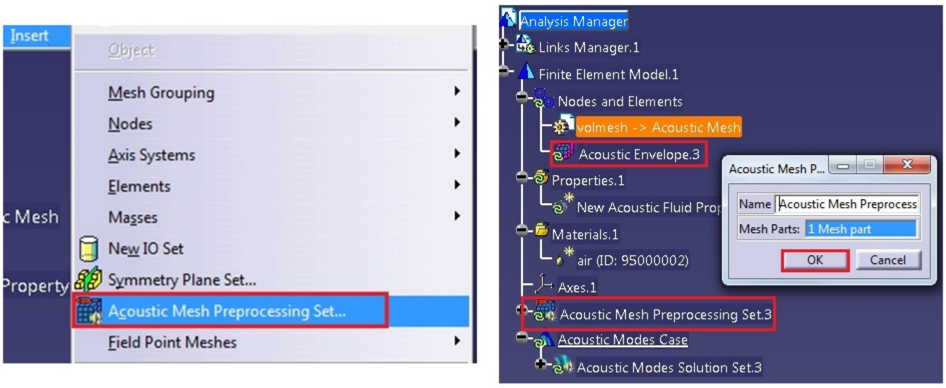


Figure 14: Acoustic mesh preprocessing

14. The materials inside the car compartment should be given their impedance values as function 59
of frequency. These act as the boundary condition in the simulation of sound field. For this initially, the materials of similar properties have to be grouped.

Insert → Mesh grouping → Auto Update group

A window will show up asking on which basis the grouping should be done. *Feature Angle* was selected as shown in the Fig.15

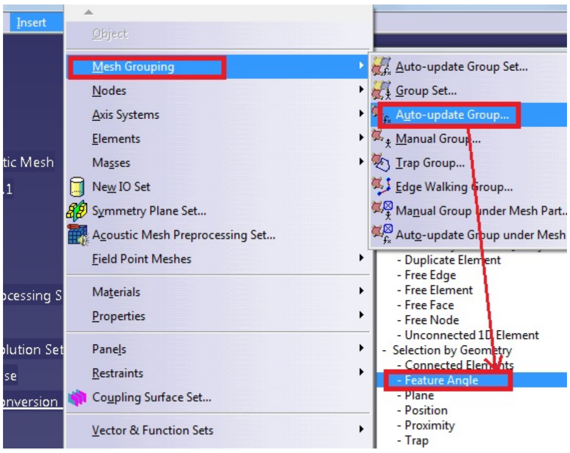


Figure 15: Mesh grouping (1)

15. *Acoustic Envelope* was selected as the input mesh part. An appropriate feature angle was selected and the starting element was selected as shown in the Fig.16. In this manner all the required parts should be grouped.

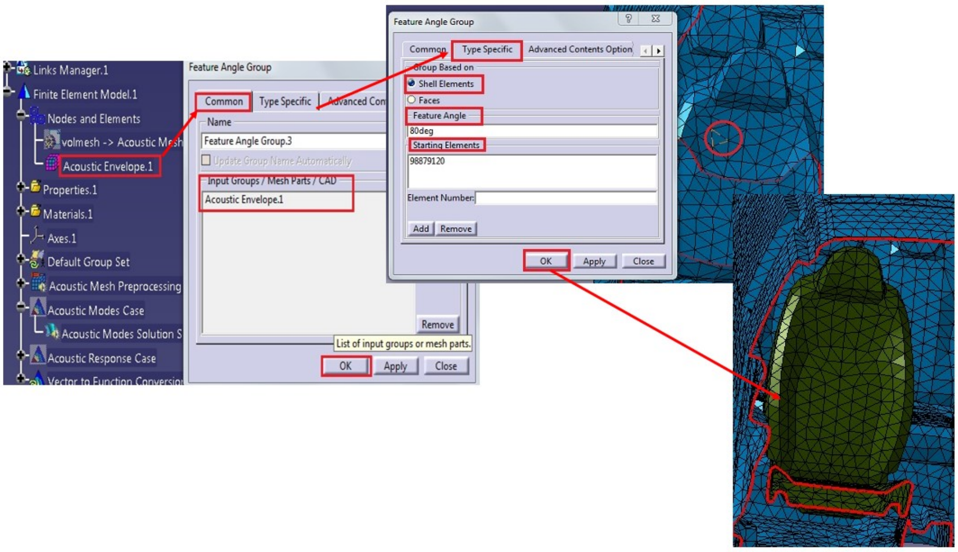


Figure 16: Mesh grouping (2)

55

16. Now Impedance as a function of frequency has to be applied to the grouped parts as shown in the Fig.17

Property → RMB → New Acoustic properties → Absorbent panel property

17. To simulate the sound field due to the excitation from loud speakers *Acoustic Response case* has to be included in the *Analysis Manager* tree as shown in Fig.18

Insert → Acoustic Response → Acoustic Response case

A window will show up as shown in Fig.18 asking the required information for the *Acoustic Response*

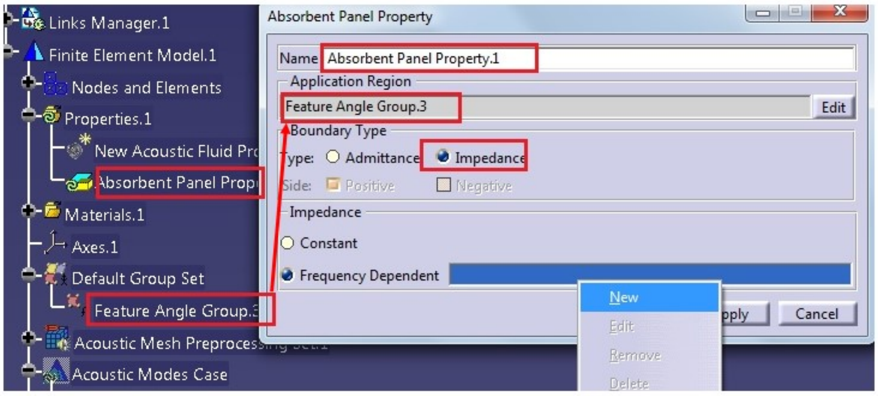


Figure 17: Defining properties to mesh groups

case. Then *Acoustic Response Case* section will be added to *Analysis Manager* as shown in Fig.19

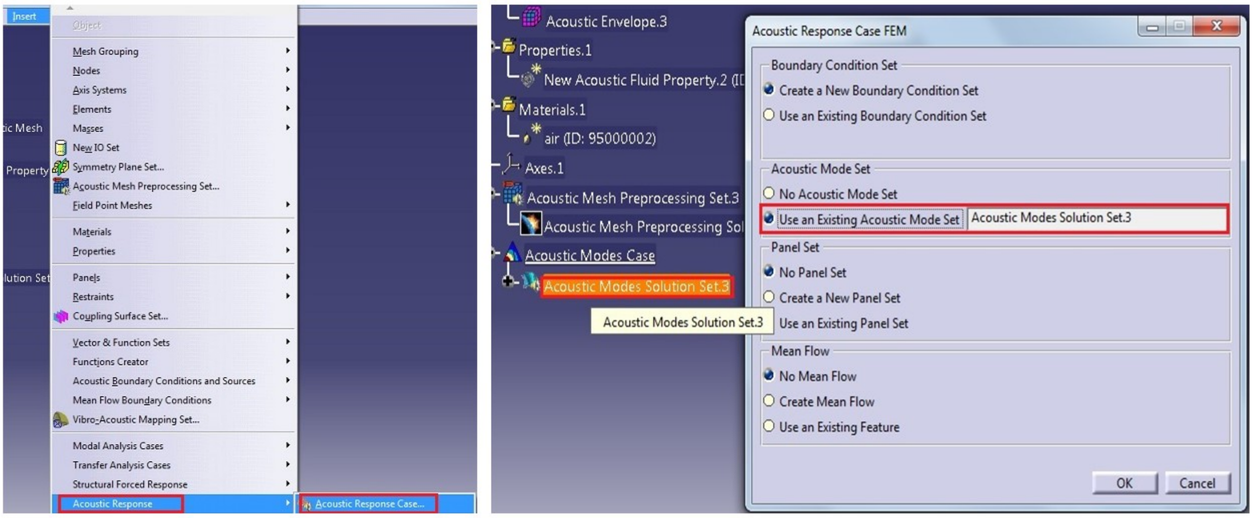


Figure 18: Inserting Acoustic response case (1)

60

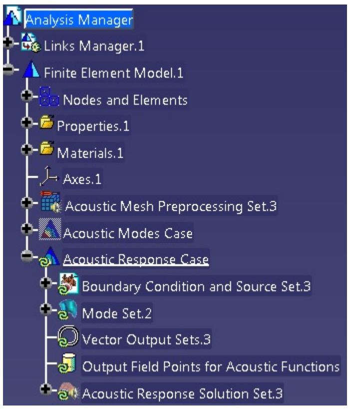


Figure 19: Inserting Acoustic response case (2)

18. Double click on *Acoustic Response Case* to expand the section. To add the acoustic sources- *Boundary Condition and Source Set* → *RMB* → *Acoustic sources* → *Monopole*
Then a pop up will come asking the location and amplitude of the source as shown in Fig.20

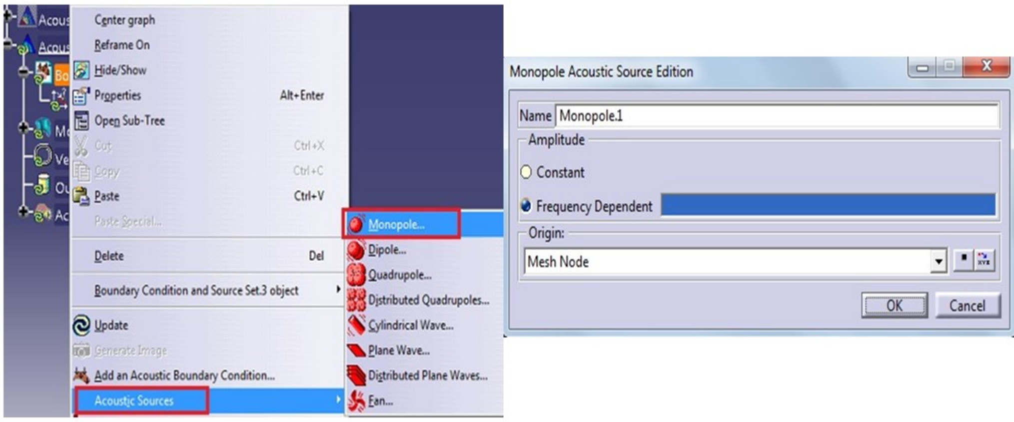


Figure 20: Inserting acoustic source (1)

19. The amplitude value of the monopole source can be given either as a constant or a function 61
of frequency. To give amplitude as a function of frequency -
Check the Frequency dependent option → *RMB* → *New*
Then a window will be shown up as shown in Fig.21 in which frequency dependent input to the source can be given.

20. To give the location of the speaker, mesh node or CAD point or co-ordinates of the desired location of source are to be given as shown in the Fig.22

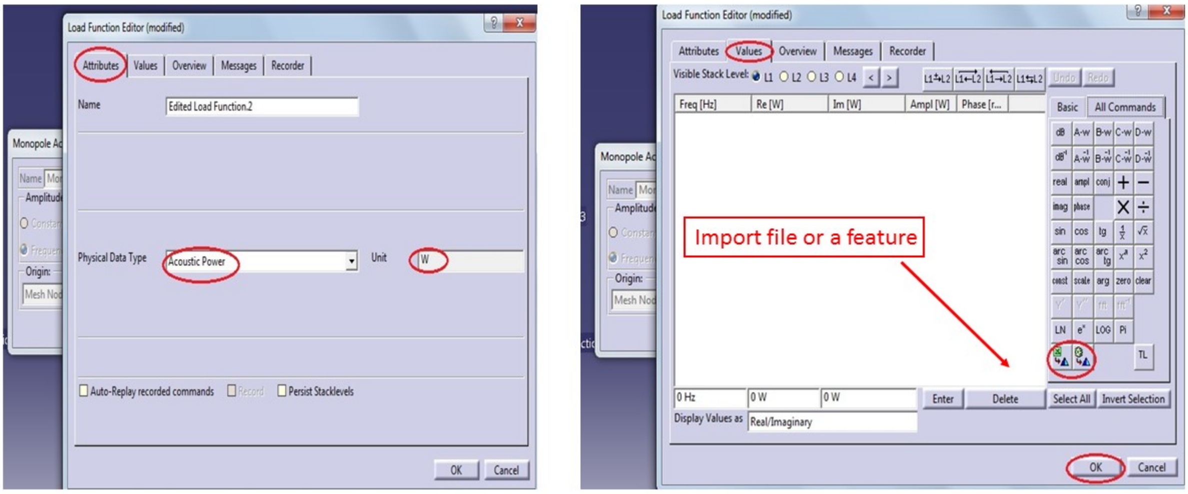


Figure 21: Giving frequency dependent input to acoustic source 62

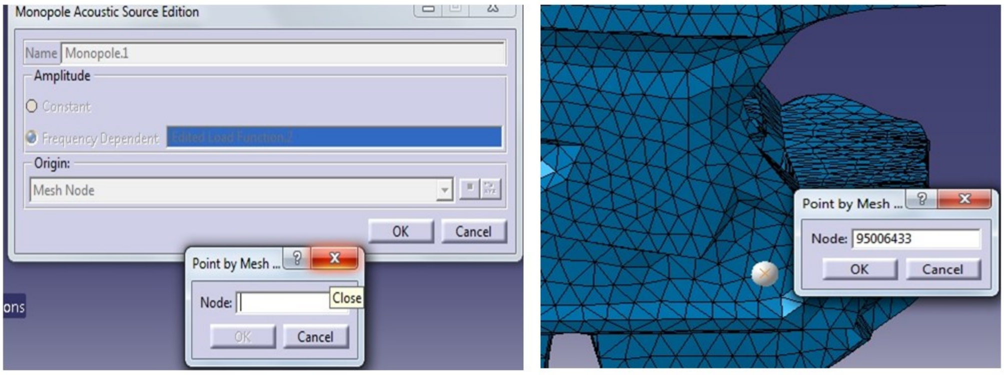
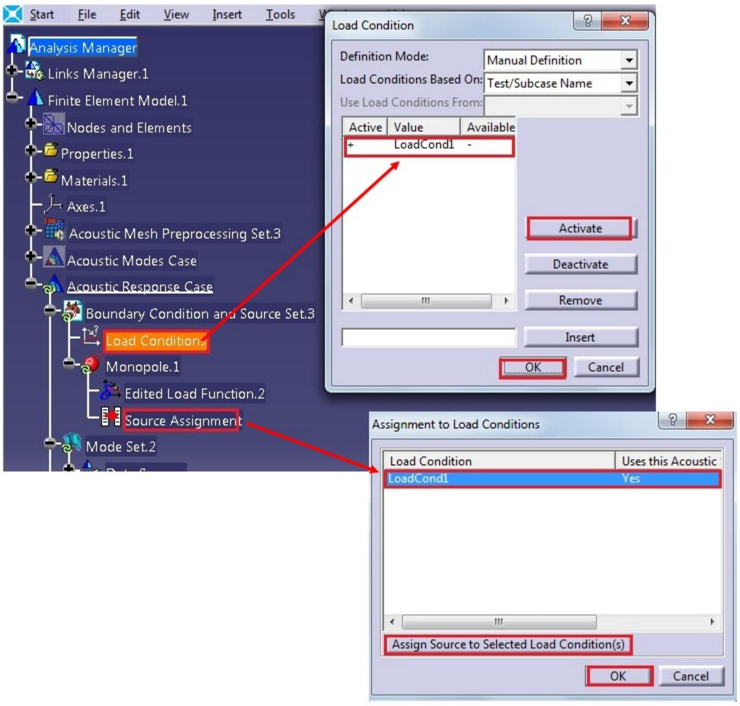


Figure 22: Giving location to acoustic source

21. The *Load condition* should be activated and the *Source assignment* should be done. These options should be double clicked and options has to be followed as shown in the Fig.23 63

22. The modal analysis data was provided to the *Mode Set* in *Acoustic Response case* as shown in the Fig.24



64

Figure 23: Load condition and Source assignment

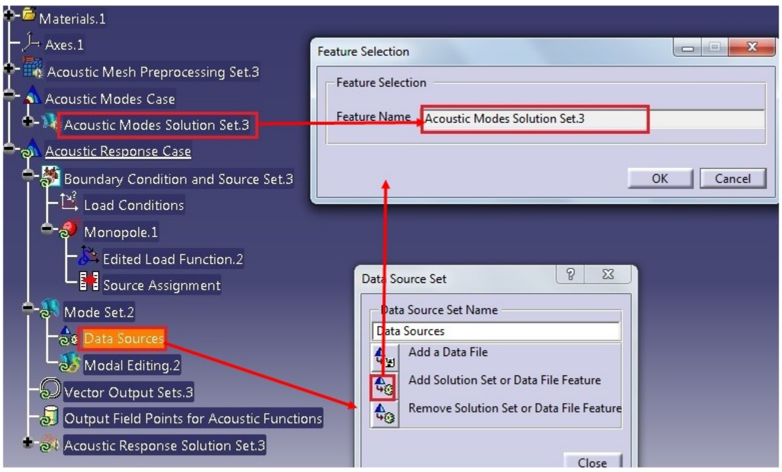


Figure 24: Providing Modal analysis data to Acoustic response case

65

23. *Vector Output Set* was defined on the entire acoustic mesh region as shown in Fig.25

24. Update all the subsections of *Acoustic Response case*. At last double click on *Acoustic Response Solution Set* and edit the solution parameters as shown in Fig.26

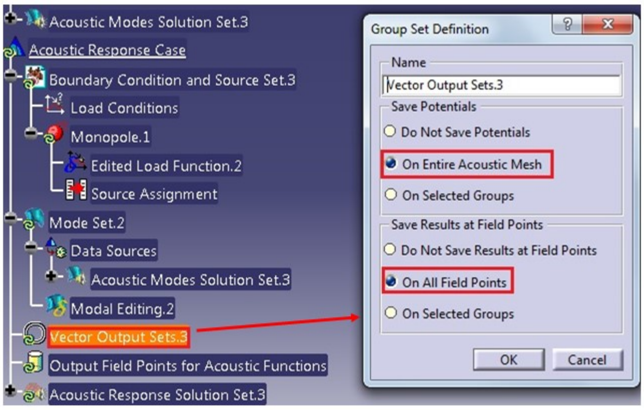


Figure 25: Defining Vector output set

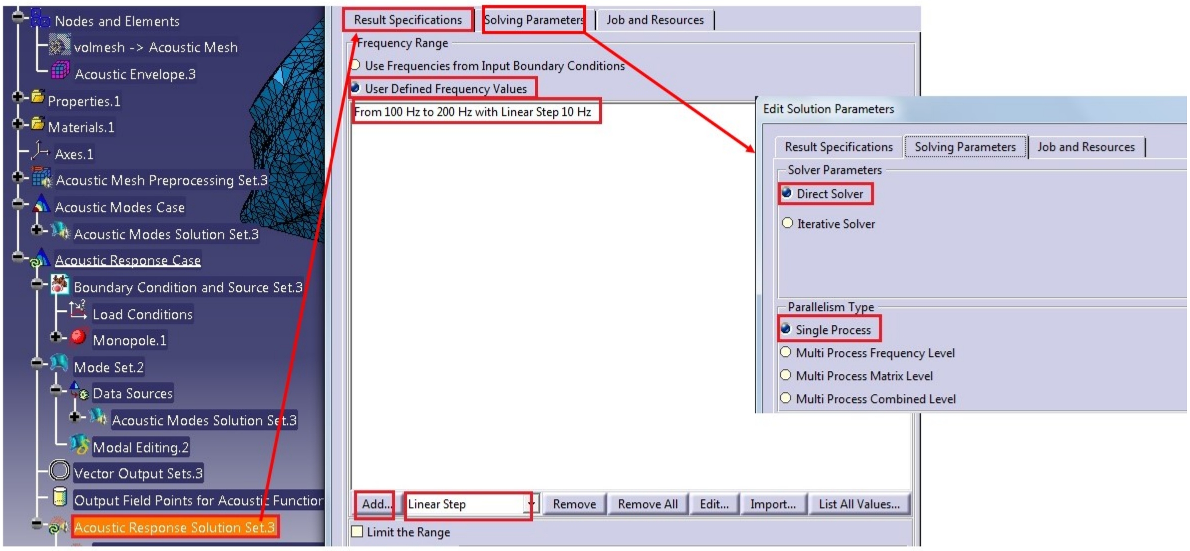


Figure 26: Editing solution parameters of Acoustic Response case

68

25. Now update the *Acoustic Modal participation Factors.* A job details window will show up as shown in the Fig.27

26. A *Vector to Function Conversion case* is inserted by - *Insert* → *Pre- and Post- processing* → *Vector to Function Conversion case.*

A window will show up asking the input details for the *Vector to Function Conversion case* as show in the Fig.28

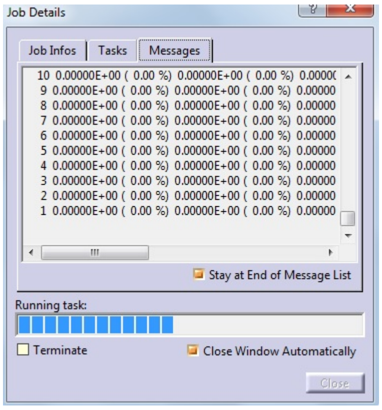


Figure 27: Job details of Acoustic response case

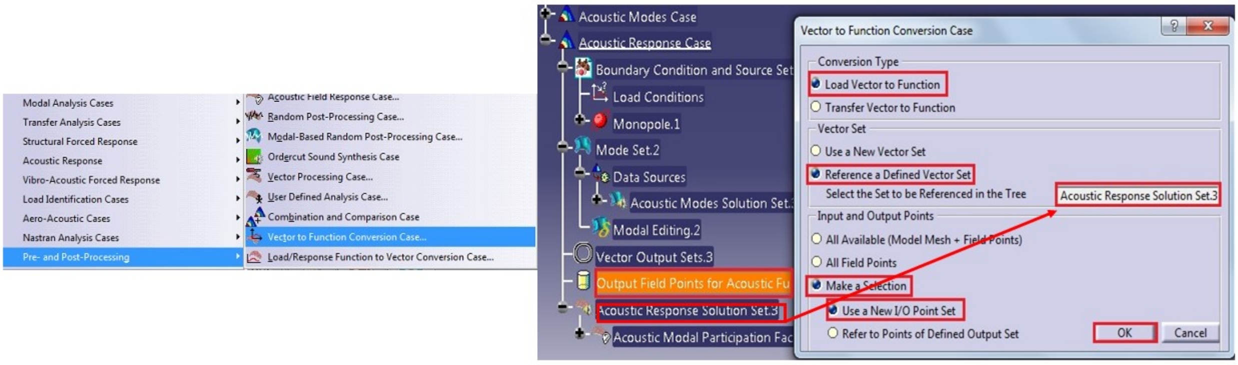


Figure 28: Job details of Acoustic response case

27. The *Load Condition* has to be updated as mentioned in the *Step 18*. Now *Output Field Points for Acoustic Functions* have to be selected. These points have to be selected from the 54
Acoustic Envelope that is created in the *Step 13*. For feasibility of selecting the desired points, hide the Acoustic mesh and make only Acoustic Envelope visible as shown in Fig.29
Nodes and Elements → Double click → Acoustic mesh → RMB → Hide/Show

28. Now nodes has to be created at the desired points inside the car interior where we want the results. Here, the nodes has to be created at the left and right ears of the passengers as shown in the Fig.30
Output Points → Double click → Single Point Selection → Select the desired node from the mesh

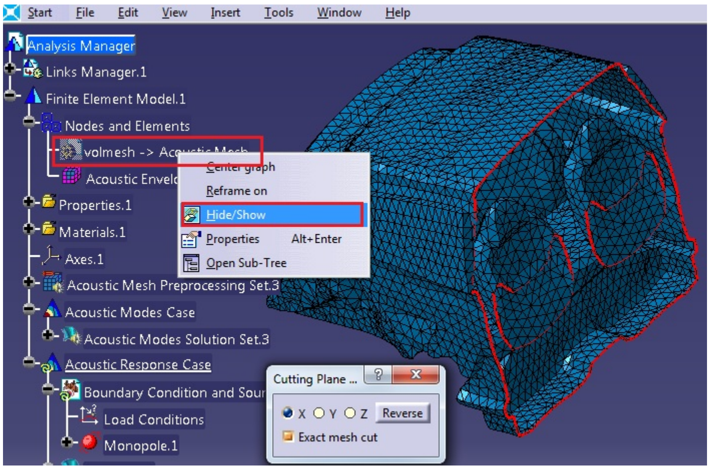


Figure 29: Defining Vector output set

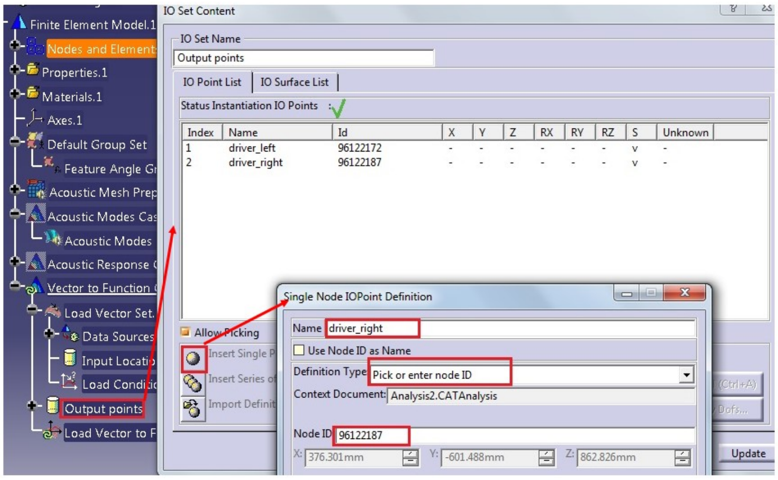


Figure 30: Selection of Output points

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29. All the subsections in *Vector to Function Conversion* case have to be updated and at last the *Load vector to Function Conversion case* has to be updated.
Load vector to Function Conversion case → *RMB* → *New Function Display*
Pop ups will appear as shown in the Fig.31

30. A new window appears asking to select the point of interest to display the curve. The format of data can be modified by right clicking on the axis name of Y- axis (here, dB(RMS)) and the range of frequency can be modified by right clicking on the axis name of the X-axis.

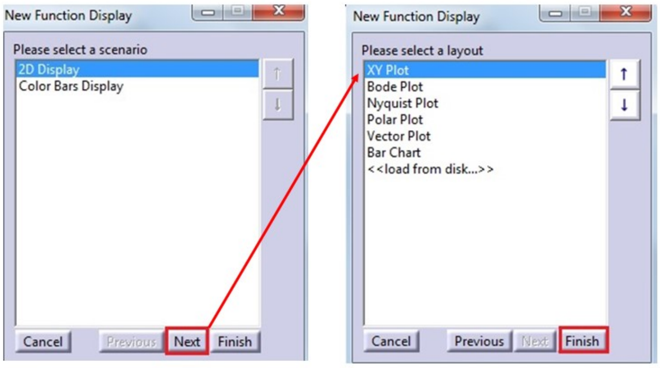

Figure 31: New function display

58

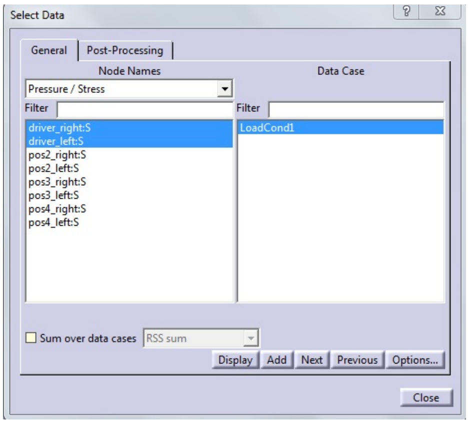

Figure 32: Selection of data for displaying

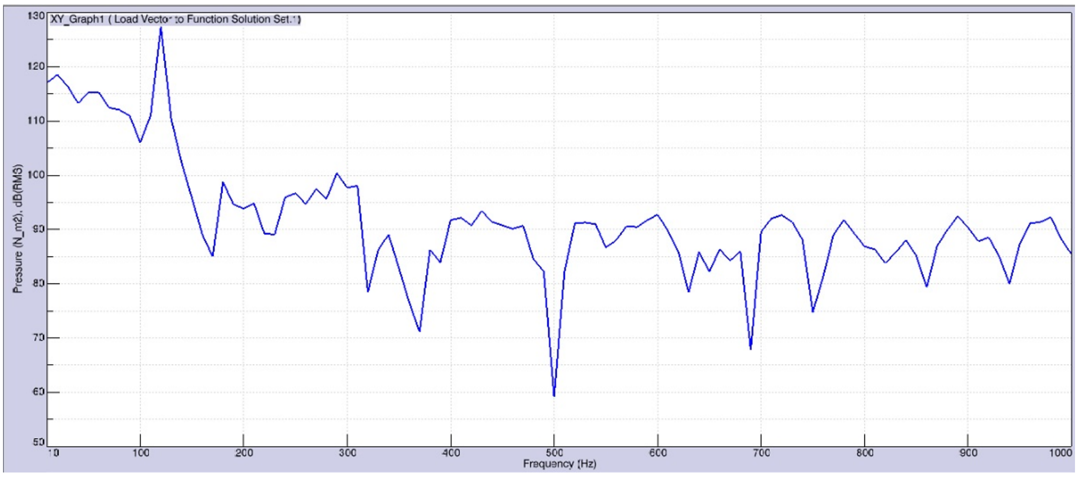

Figure 33: Pressure distribution as a function of frequency

Appendix I:

High frequency analysis of the acoustic simulation of sound field inside the car passenger compartment

1. Open LMS Virtual.Lab. Select the Ray Tracing Acoustics License as shown in the Fig.34 *Tools → Options → General → LMS Licensing → Ray Tracing Licence*

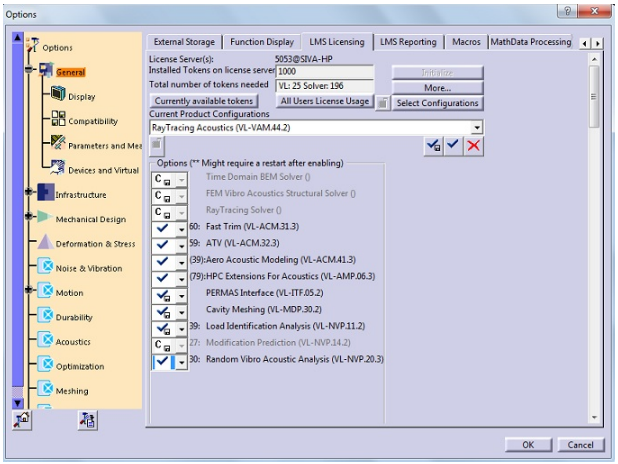

Figure 34: Selecting Ray Tracing Acoustics license

2. Start the Ray Acoustics module. This will create a new analysis window with a pre-defined tree.

Start → Acoustics → Acoustic Ray Analysis

3. Import the surface mesh of the desired car model with a compatible mesh file format.

Links Manager → Right Mouse Button → Import

4. Group the materials of similar properties inside the car compartment and give them impedance boundary condition as a function of frequency as mentioned in the *Steps 14, 15 and 16* as mentioned in the Appendix H.

5. Create Field Point Meshes in front of the passenger seats as shown in Fig.35

Insert → Field Point Meshes → Plane Field Point Meshes

A window will pop up asking the dimensions of the mesh and no. of divisions of the mesh. The mesh should be divided in such a way that a node should come at the location of the passenger's head. Two field meshes should be created: one for front passengers and the other for rear passengers as shown in Fig.36

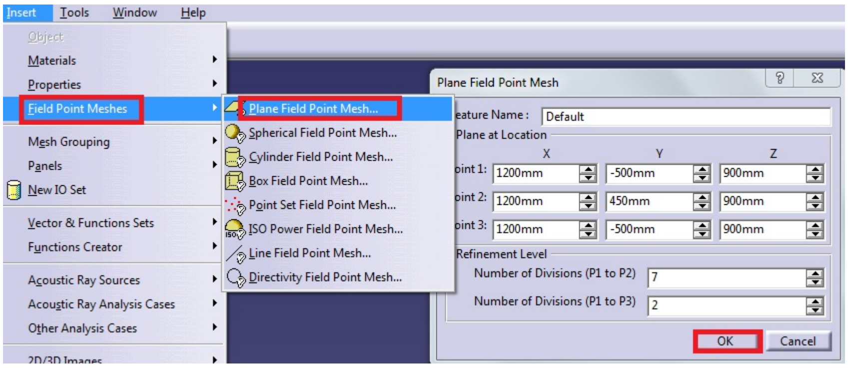

Figure 35: Creating field point meshes

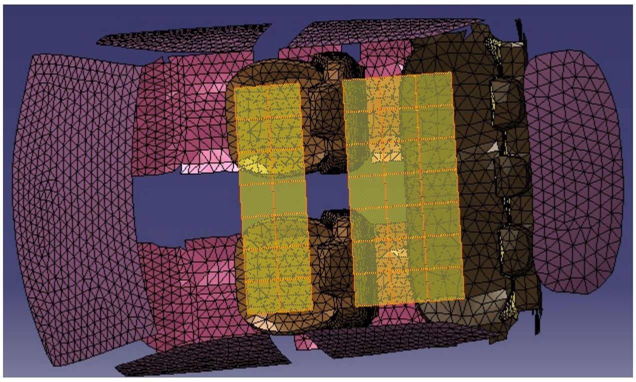

Figure 36: Field point meshes

6. Acoustic source should be provided for the simulation of sound field. The acoustic sources are inserted into the *Analysis Manager* as shown in Fig.37
Insert → Acoustic ray Sources → Acoustic Ray Source Set

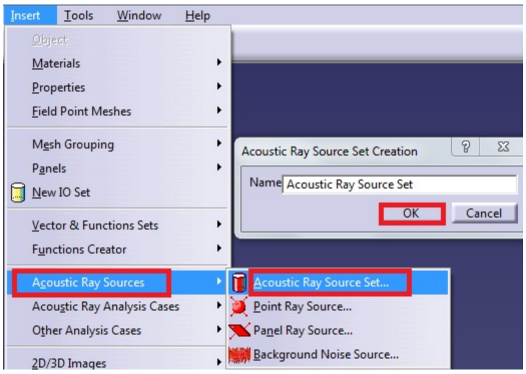

Figure 37: Inserting Acoustic ray source set

7. Point sources were provided at the loudspeaker locations as shown in the Fig.38
Acoustic Ray Source Set → Acoustic Ray Sources → Point Ray Source

8. A window will be shown up as shown in the Fig.39 asking the input conditions of the Point source. Amplitude to the Point source can be given as a function of frequency as mentioned in Step 19 of Appendix H. A mesh node at the point of location of the Point source has to be selected as mentioned in the Step 20 as mentioned in Appendix H.

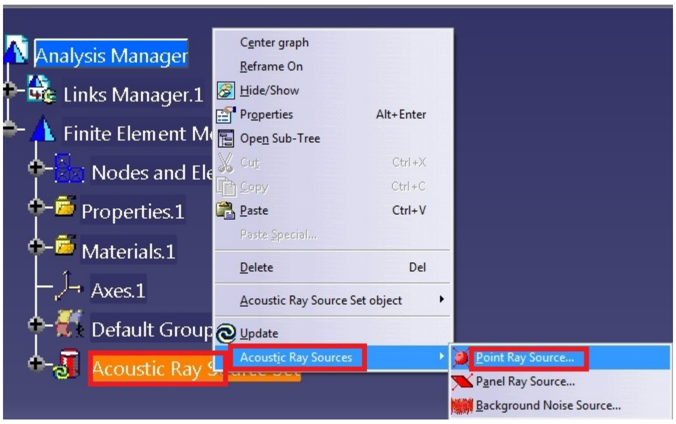

Figure 38: Inserting Point sources

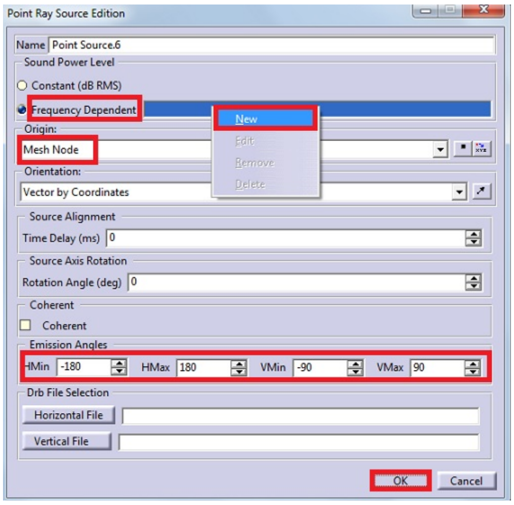

Figure 39: Point ray source addition

9. New I/O set has to be inserted and input points have to be selected on the field mesh at the location of the passenger's head as shown in the Fig.40
Insert → *New I/O Set* Now the input points at the locations of the heads of four passengers can be selected as mentioned in the Step 28 of Appendix H.

10. Create an Acoustic Field Response Analysis Case.
Insert → *Acoustic Ray Analysis cases* → *Acoustic Ray Field Response Case* and choose *Acoustic Ray Source Set* as shown in Fig.41

11. Double click the *Acoustic Ray Field Response Solution Set.* In the *Result Specification* tab, specify the frequency range as 1000Hz to 8000Hz in steps of 20Hz as shown in Fig.42. The Store level specified as *No Transient.*

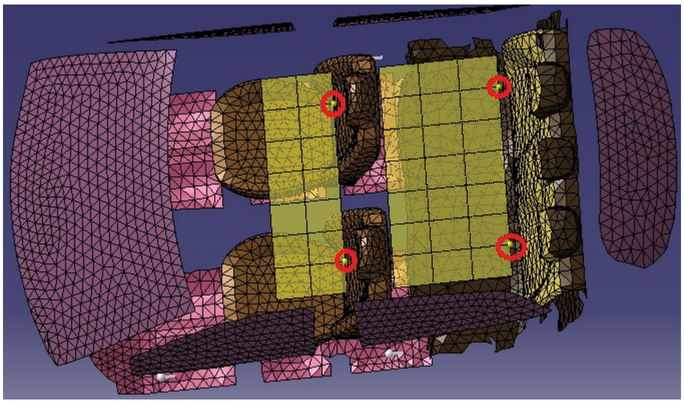

Figure 40: Input points at passenger heads on the field mesh

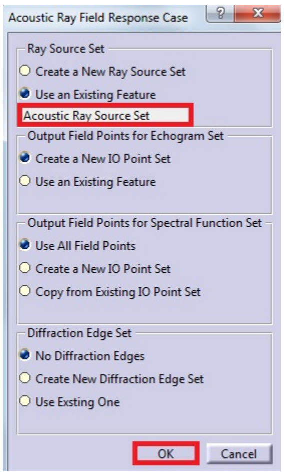

Figure 41: Insert Acoustic ray field response case

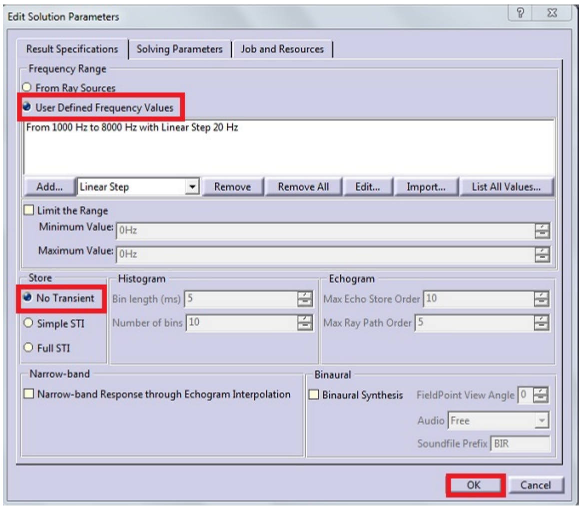

Figure 42: Result specifications

12. In the *Solving Parameters* tab, the parameters are defined as no. of rays = 10000, reflection order = 10, time window (ms) = 50 and dynamic range (dB) = 90 as shown in the Fig.43

13. To launch the computation the *Acoustic Ray Field Solution Set* has to be updated. Then

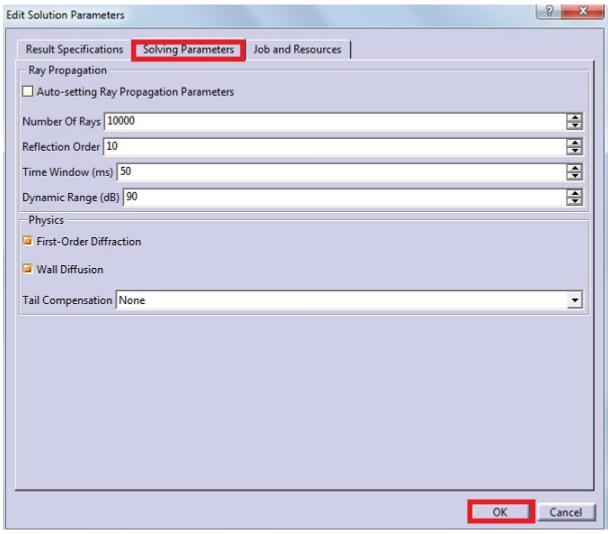

Figure 43: Solving parameters

the job details window pops up as shown in the Fig.44
Acoustic Ray Field Solution Set → RMB → Update

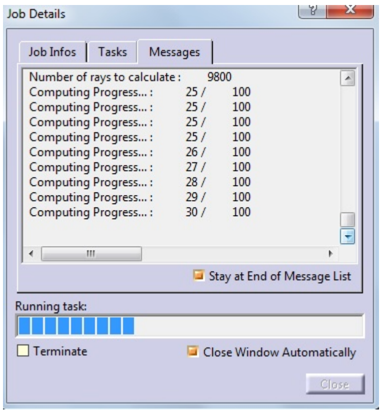

Figure 44: Job details

14. Update all the subsections in the *Acoustic Ray Field Analysis Case*. Check for the existence of *Siva-505647-Acoustic.sdb* file or a similar one under the *Links Manager* as shown in the Fig.45

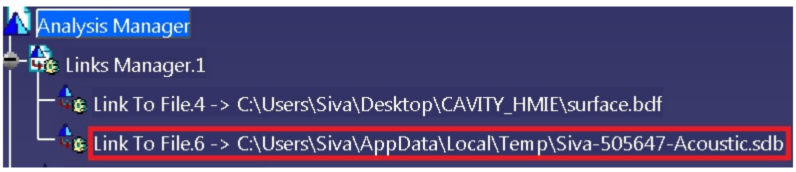

Figure 45: Existence of .sdb file under Links Manager

15. Now check the result of the performed simulation.
Acoustic Ray Field Response Solution → RMB → Generate Image → SPL(dB) → OK
16. To plot the *Spectral Function* in one field point -
Spectral Function Set → New Function Display → Select 2D Display/Bode Plot → Select required point for display

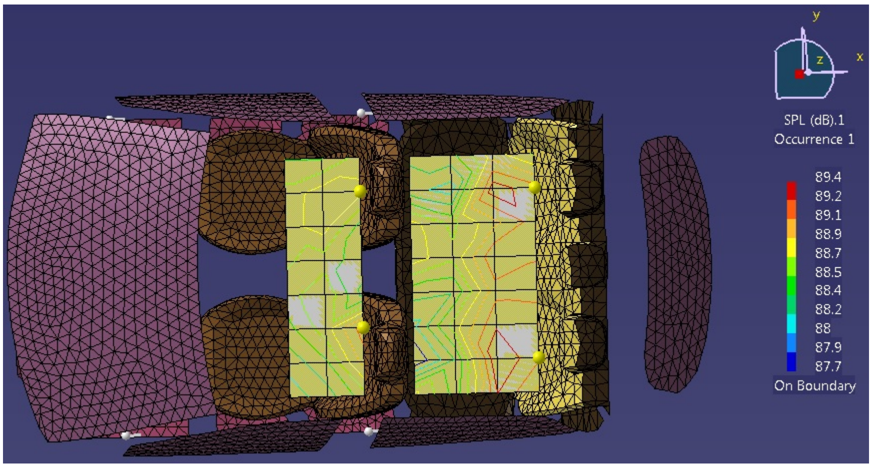

Figure 46: Contour plot of SPL(dB) in the field meshes

Figure 47: Spectral function in two field points

17. Use Post-processing tab of the Select Data dialog to transform to time domain as shown in the Fig.48

18. Now insert another *Acoustic Ray Field Response Analysis Case* as mentioned in the *Step 10.*

19. Double click the *Acoustic Ray Field Response Solution Set* and in the Result Specifications tab, specify the frequency range as 1000Hz to 2000Hz in steps of 20Hz. Define the Store Level to *Full STI*. Switch to *Binaural Synthesis* option. Use audio file in *Free*-format The *FieldPoint View Angle* indicates the viewing direction of the binaural heads at each field point location. It is always measured from the positive X-axis. If we put *FieldPoint View Angle* equal to -180 as shown in the

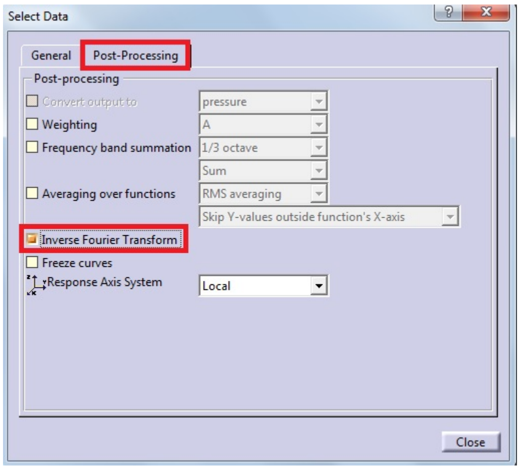

Figure 48: Inverse Fourier Transform

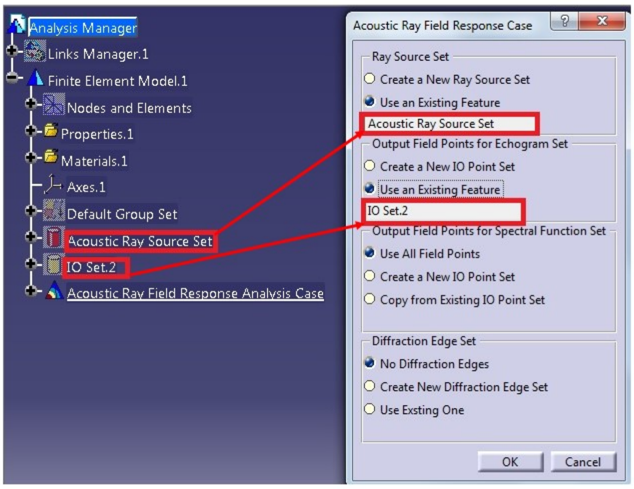

Figure 49: Insert Acoustic ray field response case

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Fig.50. This means that the binaural heads are looking in a direction opposite to the global X-axis. The Result Specification were made as shown in the Fig.51

19. The Solving Parameters were maintained as mentioned in the *Step 12*.

20. To launch the computation update the *Acoustic Ray Field Solution Set*. After the computation was done check for the existence of *Siva-505671-Acoustic.sdb* or similar file attached to the *Links Manager* as shown in the Fig.52

21. Also look for the existence of *BIR11537.txt*, *BIR11506.txt*, *BIR11517.txt* and *BIR11491.txt* files in the working directory of the server. These contain the BIRs of the response field points. Update all the sections under *Acoustic Ray Field Response Solution Set*.

22. Plot the *Interaural Cross Correlation* (**IACC**) by following the sequence as shown in the

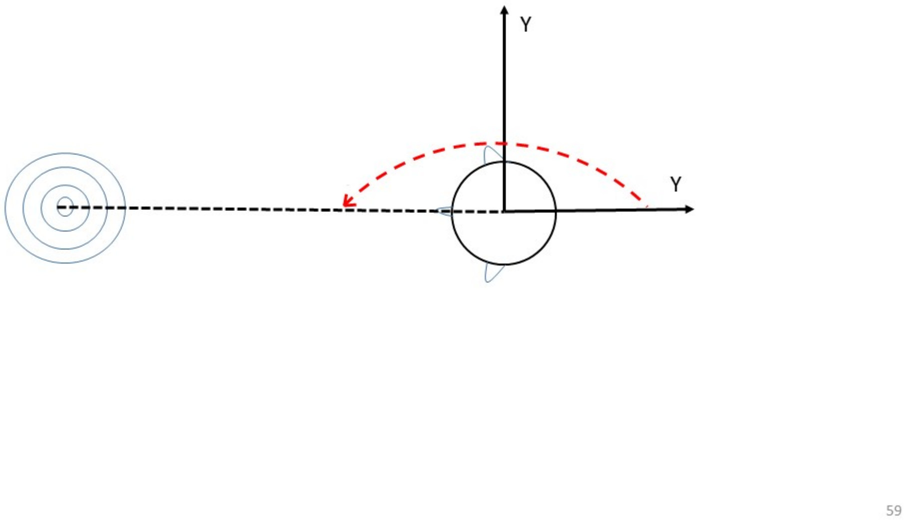

Figure 50: Field point view angle equals -180^0

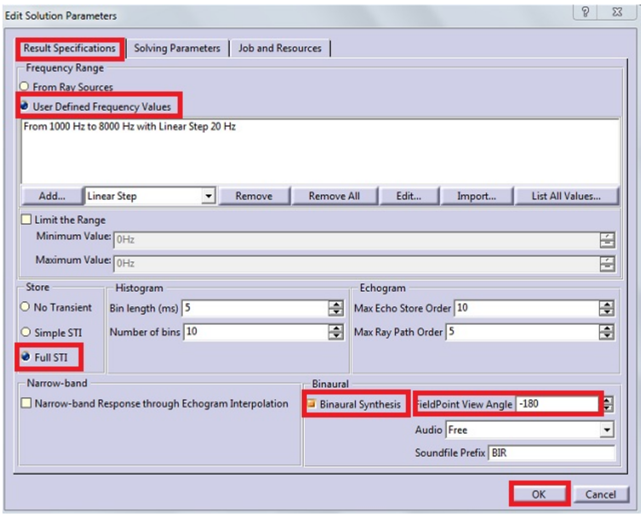

Figure 51: Result Specifications

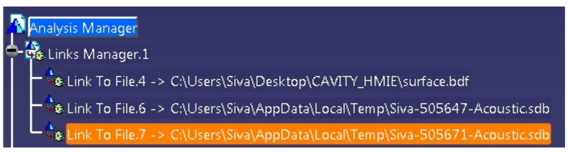

Figure 52: Checking for *.sdb* file under Links Manager

Fig.53

Overall vector Set $\rightarrow$ *RMB* $\rightarrow$ *Generate Image* $\rightarrow$ *Interaural Cross Correlation* $\rightarrow$ *Ok*

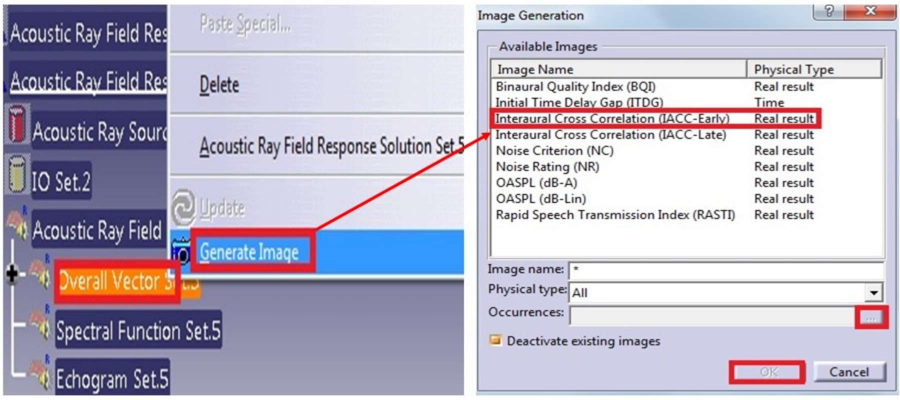

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Figure 53: Generating IACC plot

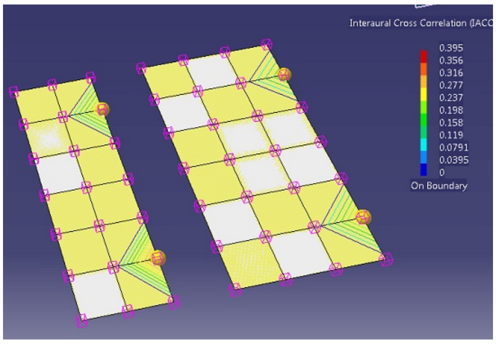

Figure 54: Inter Aural cross Correlation at all the four seating positions in car compartment

23. In the similar way, Initial Time Delay Gap (**ITDG**) and Binaural Quality index (**BQI**) can also be plotted.