N.V.H Analysis of a Vehicle using Statistical Energy Analysis

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Abstract:

Modern vehicle development heavily requires noise and vibration refinement to deliver the proper level of customer satisfaction and methods are improved daily. Statistical Energy Analysis (SEA) presented in 70’s and in the recent decade it has been highly applied in the automotive industry and different commercial software have been introduced. LeanNova Engineering Co. has decided to study the method and apply it in the design and engineering services. In this thesis the concept of the method was studied. Some simple examples were solved (by using MATLAB and FEM software), and the results were compared to a SEA commercial software named VA One results. Measurement related to acoustic properties of Porous Material (in order to use as input in the VA One software) by Impedance Tube and based on a method presented by University of Sherbrooke and software of FOAMX have been done. Finally a whole car body model was generated and the effect of sound pressure as an input at different locations of body was evaluated.

Keywords:

Statistical Energy Analysis (SEA), VA One, Impedance Tube, Porous Material, Transmission Loss
Acknowledgements

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Trollhättan
Ilia Dokhani
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1. Notation

\[ A \quad \text{Matrix of damping} \]
\[ C \quad \text{Damping coefficient} \]
\[ \text{dB} \quad \text{Decibel} \]
\[ E \quad \text{Energy of vibration} \]
\[ f \quad \text{Frequency} \]
\[ F \quad \text{Force} \]
\[ \text{Hz} \quad \text{Hertz} \]
\[ L_I \quad \text{Sound intensity level} \]
\[ L_P \quad \text{Sound pressure level} \]
\[ M, m \quad \text{Mass} \]
\[ N \quad \text{Number of modes} \]
\[ \text{Pa} \quad \text{Pascal} \]
\[ Q \quad \text{Q factor} \]
\[ T \quad \text{Kinetic Energy} \]
\[ U \quad \text{Potential Energy} \]
\[ v \quad \text{Velocity} \]
\[ x \quad \text{Displacement} \]
\[ \beta_{ij} \quad \text{Coupling factor from object } i \text{ to } j \]
\[ \varepsilon \quad \text{Average modal energy} \]
\[ \zeta \quad \text{Damping ratio} \]
\[ \eta \quad \text{Loss factor} \]
\[ \Pi \quad \text{Power dissipated from a system} \]
\[ \omega_n \quad \text{Radian natural frequency} \]
\[ \overline{\delta f} \quad \text{Average frequency} \]
**Abbreviation**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
</tr>
<tr>
<td>CAM</td>
<td>Computer Aided Manufacturing</td>
</tr>
<tr>
<td>CAE</td>
<td>Computer Aided Engineering</td>
</tr>
<tr>
<td>FEM</td>
<td>Finite Element Method</td>
</tr>
<tr>
<td>MATLAB®</td>
<td>Language for numerical computation designed by MathWork</td>
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<tr>
<td>SEA</td>
<td>Statistical Energy Analysis</td>
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<tr>
<td>STL</td>
<td>Sound Transmission Loss</td>
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<tr>
<td>VA One</td>
<td>Vibro-Acoustic One, software generated by ESI group</td>
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</tbody>
</table>
2. Introduction

2.1 Importance of vehicle industry
According to estimates, global auto sales in 2013 hit more than 80 million units and some forecasts say it will steadily increase up to 100 million in 2018. Over the last years, technological changes in automotive engineering with a focus on safety and performance have been significant [1]. Modern vehicle development requires noise and vibration refinement to deliver the proper level of customer satisfaction and acceptance. For the customer’s perception of vehicle quality, the noise and vibration characteristics of the vehicle need to be considered [2]. Therefore finding and applying new, accurate and economical methods is an advantage for any company to be able to be successful in this industry.

2.2 Importance of CAD/CAM/CAE
Development in computational and analytical methods and using computers for running the complicated algorithms in the last couple of decades made Computer Aided Engineering and Design become very popular in different field of industries because of being economic, less time consuming and flexible. Advances in computer aided design and engineering (CAD/CAE) and mechanical system dynamic software already have dramatically changed the way some components and systems are being designed and engineered. Still experts suggest that as much as 40% of development time for a complete vehicle can be further slashed with wider application of CAD/CAE.

Combining the capabilities of surface and solid elements modelling, animation software and powerful computers, the industry can develop vehicles much more cost-effectively than in the past. Technology designers can style and immediately view a concept in three dimensions. Also engineers can test every aspect of a vehicle's performance, and manufacturing types can run pre-production assembly and fit tests without building even a single prototype or clay model. One of the major benefits of CAE is giving engineers the opportunity to try different design options without the cost of a full-size physical prototype to see how they look or if they will work.
Historically, it was the aeronautical that for the first time took advantage of computer-aided design and engineering in the 1950s and 1960s and then two-dimensional computer-generated wire frame or mesh renderings replacing drafting boards as CAD crept into the auto industry in the late 60’s and early 70’s. Between 1970 and 1980 surface and solid modelling were introduced, and 3-D CAD wire frames came on the scene in the early '80s. As an example, while in 1988 GM had no CAD/CAM/CAE workstations in 1995 it had 3,500 seats and a total of 6,500 workstations in 1995 [3]

The Global CAE market is expected to post revenue of US$3,402.30 million by 2016 according to TechNavio's report, the Global CAE Market 2012-2016. The Global CAE market is driven by many growth factors. One of the major drivers in the market is the increasing need to reduce time-to-market. There are, however, certain factors that may affect the growth of the market. One of the major challenges in the market is the increasing threat of open-source CAE software that could affect the growth of the market during the forecast period [4].

### 2.3 Sound, vibration and noise control in vehicle industry

Noise and vibration performance in a vehicle is an important criterion, since it directly affects the overall perception of a vehicle. Some customers pay more attention to style while other on safety or reliability. One of the most important factors considered by every customer is the sound. Of course this matter is a subjective one and people’s views differ. For example for some drivers a “sporty sound” can be attractive while for others it is annoying. Noises affect the driver’s and passengers’ comfort and can induce feelings of insecurity. Modern vehicles require noise and vibration refinement to deliver the proper level of customer satisfaction and acceptance. It is common for a customer’s perception of quality to relate closely to the noise and vibration characteristics of the vehicle.

Based on [1], [2], it is possible to categorise a vehicle sound in 3 subjects:

1. Interior Noise
2. Exterior Noise and
3. Passenger Noise
In addition it is possible to divide the concept of interior noise (as the most important one for vehicle manufacturer) to 4 subcategories:

1. Engine
2. Tire road
3. Wind
4. Miscellaneous

If it is required to consider only the sound and noise issue, a good vehicle is required to have the following characteristics:

1. Low wind noise
2. Low road noise
3. Low engine noise
4. Clarity of Speech (for passengers)
   1. Low level of exterior noise
   2. Low level of interior noise

On the other hand, noise can be beneficial for example in terms of alarming pedestrians for upcoming vehicle.

### 2.4 Traditional method evaluation

Finite Element Analysis (FEA) for more than 60 years has been applied in many different fields such as Stress Analysis, Fluids and Thermodynamics even in Vibration and Acoustics. It started in the 1950's in aerospace industry and since these early days it has been widely used in structural mechanics [5]. In NVH analysis, however, due to various reasons such as effect of damping provided by the vehicle parts, computation cost is very high and usually the frequency domain limited to below 200 Hz (except for research and development projects). In this regard usually for higher frequencies running a test on the object in the advanced acoustic laboratories is applicable [6].

Statistical Energy Analysis (SEA) is another new method that after being applied in the aviation and space industry (just like the FEA!), recently have been used in the automotive industry. In the beginning of the 1960’s, aerospace engineers needed to predict the connection between vibrational
responses of satellite launch vehicles to rocket noise. This was the birth of SEA. Because of the size of the models and the speed of computation, engineers were able to predict only a few of the lowest order modes by ordinary methods. It posed a serious difficulty because for example in the Saturn launch vehicle, it is estimated that 500000 natural frequencies in the range of 0 to 2000 Hz [7] exists. In order to have a better understanding two simple examples will be used. According to [8] that it shows for 18 points in the steel plate in a room where a harmonic load “Force” is acting. All measured transfer functions between the harmonic load and the sound pressure (on a logarithmic scale) are presented in the figure 2-1. The first peak at about 11 Hz shows the first resonance frequency of the plate.

The points are very close to each other. Consequently, for low frequencies the air pressure at all microphones is nearly equal. However, it can clearly be seen that at higher frequencies the transfer functions differ considerably. As it mentioned this was an example of simple plate in a room with known parameters so just imagine how much results will be varies for a complex object with a lot of other uncertainties: for example a slight variation in temperature, air pressure etc.

![Figure 2-1 Measured transfer functions of 18 point of a steel plate](image)
Another famous example is presented by Fahy in [9]. Structure-born sound generated by tyre-road interaction, engine and exhaust tend to be exceeded by contributions to interior noise below about 400 Hz. He presented the result of a set of 41 nominally identically beer cans that were subjected to the same acoustic excitation. In figure 2-2 obvious variations can be seen:

![Figure 2-2, 41 nominally identically beer can](image)

By considering the uncertainties in these two simple examples, the necessity of probabilistic methods and models such as Statistical Energy Analysis that deals with energetic quantities is understood. This method called “Statistical” because the system that is considered from a statistical point of view, “Energy” denotes that the variable that is studied is energy. And “Analysis” emphasize that it is not one particular technique but it is a framework of study [10].

### 2.5 Aim and scope
LeanNOVA Engineering as an engineering service company decided to study and apply Statistical Energy Method (SEA) for modelling and add it to its current services for the clients. For this reason in cooperation with National Electric Vehicle Sweden in Trollhättan (NEVS), they defined a project and the main deliverable of this is a methodology for how to use SEA to predict high frequency noise in vehicles.
Identifying its possibilities and limitations is an important task in the thesis project. As the traditional noise and vibration analysis of vehicles are based on Finite Element theory and hence have an upper limit of high frequencies that can be studied, statistical or other methods need to be added to the toolset that CAE engineers use to predict and improve the noise and vibration performance of a vehicle.

Such model enables the user to evaluate the effect of changes in the input variables to the whole car body for high frequencies. As an example, by having the model, suppose user wants to evaluate the effect of a new pump in the overall internal noise of the car and compare it with the old pump. Also it will help to choose between different alternatives based on their efficiencies while it is possible to see and trace effects at the same time and decide for the required insulation and trim.
3. Statistical Energy Analysis

3.1 History
The first developments of this method were independently made in 1959 by the two researchers R.H. Lyon and P. W. Smith. In 1960 they started to work together. After years of cooperation with other scientists and contributions from some parallel studies by others, the basic theory and procedure was published by Richard H. Lyon in 1975 [10]. Since then, lots of refinement and analytical (for calculation of different variables) have been made [11].

3.2 Energy of Vibration
The Energy of Vibration \( E \) of a system depends upon the particular system and also on the type of the motion that is considered. Here it is taken to mean the long-time averaged sum of the kinetic and potential energies under stationary and random excitation:

\[
\langle E \rangle = \langle T \rangle + \langle U \rangle \tag{3-1}
\]

Here \( \langle \cdot \rangle \) means that the parameter is averaged over time, \( T \) is the kinetic. Considering a simple, linear, damped oscillator, the kinetic and potential energies will be equally larger [11], [12]:

\[
\langle T \rangle = \langle U \rangle \tag{3-2}
\]

And the total energy is:

\[
E = M \langle v^2 \rangle \tag{3-3}
\]

Where \( M \) is mass and \( \langle v^2 \rangle \) is the averaged mean square of vibration velocity.

3.3 Definition and formulation
Power is force multiplied by velocity, and the power dissipated from a system with viscous damping is [13]:

\[
\]
\[ F_d = cv \quad (3-4) \]

\( F_d \) is the damping force

\[ \Pi = Fv\dot{x} = cv\dot{x}^2 \quad (3-5) \]

By adding the definition of the damping ratio \( \zeta \), and the radian natural angular frequency \( \omega_n \) and the oscillator mass \( m \) we got:

\[ cv = 2\zeta \omega_n m \quad (3-6) \]

So by considering \( E \) from the formulas (3-3), (3-5) and (3-6) is the energy there will be:

\[ \Pi = 2\zeta \omega_n mx'^2 = 2\zeta \omega_n E \quad (3-7) \]

And finally by adding the quality factor \( Q \):

\[ Q = \frac{1}{2\zeta} \quad (3-8) \]

We got:

\[ \Pi = \frac{\omega_n E}{Q} \quad (3-9) \]

Finally by introducing \( \eta \), the loss factor at resonance:

\[ \eta = 2\zeta = \frac{1}{Q} \quad (3-10) \]

\[ \Pi = \eta \omega_n E \quad (3-11) \]

It is possible to extend this concept for a single oscillator to a number of oscillators in specified frequency bands (usually one third octave bands) so by this modification \( \omega_n \) is substituted by, \( \omega \), the geometric mean centre frequency of the band and consequently \( \eta \) will be the mean loss factor of all modes within the band. We now start to formulate the procedure of SEA.

In the first introduction of Statistical Energy Analysis by Richard H. Lyon, he proposed a two subsystems model and suggested the following concepts (figure 3-1) and formulation:
Where $E_1, E_2$ are total dynamical energy of the subsystem 1, to subsystem 2 at frequency $f$ and $\eta_1, \eta_2$ are damping loss factor (3-10). Recalling formula (3-3) Lyon [10] suggested the following formula as the net transmitted power:

$$\Pi_{12} = 2\pi f \beta_{12}(\varepsilon_1 - \varepsilon_2)$$  \hspace{1cm} (3-14)

For this formulation Lyon has introduced $\varepsilon_1, \varepsilon_2$ as the averaged modal energies and $\beta_{12}$ is a coupling factor that depends only on physical properties.

By considering the averaged modal energy as:

$$\varepsilon_1 = \frac{E}{N} = \overline{E} \frac{\Delta f}{\Delta f}$$  \hspace{1cm} (3-15)

Where $N$ is the number of modes and $\Delta f$ is the average frequency spacing between the modes of the subsystem in the frequency band $\Delta f$ and introducing the new quantity $\eta_{12}, \eta_{21}$ called coupling loss factors the formula will become:

$$\Pi_{12} = 2\pi f (\eta_{12}E_1 - \eta_{21}E_2)$$  \hspace{1cm} (3-16)
Where
\[ \eta_{12} = \beta_{12} \frac{\delta f}{\Delta f} \]  
(3-17)

And there exist reciprocity property as below:
\[ \eta_{21} = \eta_{12} \frac{N_1}{N_2} \]  
(3-18)

The damping loss factor \( \eta_i \) is a measure of rate of energy dissipation in a subsystem \( i \), the coupling loss factor \( \eta_{ij} \) is a measure of the rate of transition energy from subsystem \( i \) to subsystem \( j \). The final formulation for the mentioned example is:

\[
2 \pi f \eta_1 E_1 + 2 \pi f \eta_{12} E_1 - 2 \pi f \eta_{21} E_2 = \Pi_{1,in}
\]  
(3-19)

\[
2 \pi f \eta_2 E_2 + 2 \pi f \eta_{21} E_2 - 2 \pi f \eta_{12} E_1 = \Pi_{2,in}
\]  
(3-20)

It can be easily formulated as:

\[
2 \pi f [A]\{\mathcal{E}\} = \{\Pi_{in}\}
\]  
(3-21)

By supposing (3-15), and considering the reciprocity property, (3-14) it is possible to introduce a symmetric damping matrix \([A]\):

\[
[A] = \begin{bmatrix}
(\eta_1 + \sum_{i \neq 1}^{n} \eta_{1i})N_1 & -\eta_{12}N_1 & \ldots & -\eta_{1n}N_1 \\
-\eta_{21}N_2 & (\eta_2 + \sum_{i \neq 2}^{n} \eta_{2i})N_2 & \ldots & -\eta_{2n}N_2 \\
\ldots & \ldots & \ldots & \ldots \\
-\eta_{n1}N_n & \ldots & \ldots & (\eta_n + \sum_{i \neq n}^{n} \eta_{ni})N_n
\end{bmatrix}
\]  
(3-22)

The reciprocity property makes it possible to have a symmetric matrix. Here \( n \) is the number of oscillators, for the current case 2, and \( N \) is the number of modes.
The formulation will become [10], [14], [7]:

\[
\begin{bmatrix}
(\eta_1 + \sum_{i \neq 1}^{n} \eta_{1i})N_1 & -\eta_{12}N_1 & \ldots & -\eta_{1n}N_1 \\
-\eta_{21}N_2 & (\eta_2 + \sum_{i \neq 2}^{n} \eta_{2i})N_2 & \ldots & -\eta_{2n}N_2 \\
\vdots & \vdots & \ddots & \vdots \\
-\eta_{n1}N_n & \ldots & (\eta_n + \sum_{i \neq n}^{n} \eta_{ni})N_n \\
\end{bmatrix}
\begin{bmatrix}
\frac{E_1}{N_1} \\
\frac{E_2}{N_2} \\
\vdots \\
\frac{E_n}{N_n}
\end{bmatrix}
\]

\[
= \begin{bmatrix}
\Pi_1^i \\
\Pi_2^i \\
\vdots \\
\Pi_n^i
\end{bmatrix}
\]

(3-23)

When the SEA model is established, for each subsystem i the following parameters will be necessary [10], [13], [15]:

1. The mode count (N_i),
2. The damping loss factor (\eta_i),
3. The coupling loss factor (\eta_{ij}), and
4. The input power (\Pi_i) from the external sources of excitation.

The steps that should be followed will be:

1. Define subsystems containing groups of natural modes and frequencies for each components,
2. Define the physical coupling between subsystems, and
3. Define the form of external excitations to the subsystems.
By knowing the loss factors (coupling loss factor $\eta_{ij}$ and internal loss factor $\eta_i$) and assessing them theoretically, based on formula (3-21) one is able to predict the structural behaviour to the proposed vibration even at the designing stage. Of course these values of loss factors affect the result. In addition the effect of number of modes (N) is very considerable in this approach. The reader is invited to study the chapter 3-7 of current paper. As the method’s name (Statistical) implies, the larger the number of modes per frequency band, the closer average is to the correct value (consider formula 3-1 to 3-3).

3.4 Mode and Modal Analysis

One of the main assumptions in SEA is that the response (amplitude of vibration) of a system is due to the resonances and that the other motions can be ignored. It means that the response of a subsystem is directly related to damping. Modes occur when the path travelled by a wave is such that after travelling it arrives back at its starting-place. Although existence of resonances in each subsystem is important, the actual frequencies of any resonance and the mode shapes have less importance. Because it is a statistical property and it is just needed to have enough resonances in the frequency band. The knowledge of the number of modes in a frequency band and the estimated values of the first modes is necessary for calculation of coupling as well as selection of subsystems. It also gives the overview of validity of results.

Recalling the concept of “Wavenumber” and “Wavelength” from [10], [13], [14] we will have the wavenumber $k$ defined as the spatial frequency of a wave. These “Spatial” variations in a wave motion are represented by the phase change per unit increase of distance:

$$k = \frac{\omega}{c} \quad (3-24)$$

where $c$ is the wave velocity. The spatial period of a harmonic wave motion (the distance over which the wave's shape is repeated) is described by its wavelength, $\lambda$:

$$\lambda = \frac{2\pi}{k} \quad \text{or} \quad k = \frac{2\pi}{\lambda} \quad (3-25)$$
For the two-dimensional wave number of a mode of vibration it is:

$$k_{n,m} = \frac{2\pi}{\lambda_{n,m}} = \left\{ k_x^2 + k_y^2 \right\}^{\frac{1}{2}} = \left\{ \left( \frac{n\pi}{L_x} \right)^2 + \left( \frac{m\pi}{L_y} \right)^2 \right\}^{\frac{1}{2}}$$  \hspace{1cm} (3-26)

With

$$L_x = \frac{n\lambda_x}{2}, \; L_y = \frac{m\lambda_y}{2}$$  \hspace{1cm} (3-27)

Now the simple example of a two-dimensional, homogeneous isotropic simple supported flat plate with the dimension of D_x and D_y will be considered. The mode shapes will be:

$$\psi_{n,m} = 2\sin \frac{n\pi x}{D_x} \sin \frac{m\pi y}{D_y}$$  \hspace{1cm} (3-28)

, and the resonance frequencies are:

$$\omega^2_{n,m} = \left[ \left( \frac{n\pi}{L_x} \right)^2 + \left( \frac{m\pi}{L_y} \right)^2 \right]^2 K^2 Cl^2$$  \hspace{1cm} (3-29)

where n, m are integers, K is radius of gyration (here for the plate cross section) and Cl is the longitudinal wave speed.

As it appears in figure 3-2, each point, in this graph belongs to a mode. The distance from the centre of coordinate, shows the resonance frequency of the mode \(\omega_n\). This procedure enables us to count the number of modes that will resonate in any particular frequency interval.
3.5 Analogy to: Heat Transfer

In order to have a better understanding of this method, we can look at the model as a thermal diffusion problem [16]: Suppose two identical thermally conducting objects figure 3-3, are connected and each one loses heat by radiation to the surrounding area as indicated below:

Figure 3-3 Presentation of Heat Transfer problem
Suppose that object 1 is heated by a source. The parameters of the SEA system can be defined as:

1. The Modal Density can be considered as Thermal Capacity of each object
2. Damping of a vibration mode as Radioactive Loss
3. Coupling Loss Factor as Conductivity (loss by coupling)
4. Flow of vibration energy can be seen as flow of heat

### 3.6 Analogy to: Hydrodynamic

Another simple example in order to have a better understanding of SEA is to consider it as two connected tanks [8].

Suppose that there are two water tanks that are connected by a valve and both of tanks and the valve have leakage and we can them from above by water as indicated below (figure 3-4):

![Figure 3-4 Presentation of a Hydrodynamic problem](image)

Here each subsystem is described as a tank of water. The energy E can be raised by filling in some water from above (A and B). Tapping the water is the effect of dissipating energy (C and D). Because of different width, they have different energies: Here we can see them as two objects with two different modal densities. They are connected to each other so water can flow in between. Because of their connection, vibration can expand from
one object to another. By adding water to one or both tanks: Introducing sound or vibration power \( \Pi \) the water (vibration energy \( E \)) flows from one to another and at the same time because of leakage some water is dissipated from each one. This water dissipation can be seen as each objects damping \( \eta_i \). In addition the leakage at tanks connection can be seen as coupling loss factor \( \eta_{ij} \). So in summary:

1. Modal Density can be considered as width or hydraulic energy of each tank
2. Leakage in each tank as Damping in each object
3. Leakage at connecting pipe between two tanks as Coupling Loss Factor
4. Flow of vibration energy can be seen as flow of water

### 3.7 Advantages and disadvantages

This method is easy and fast in comparison with FEM. There is a matrix to be solved and the elements will be directly related to the number of the sub-systems. So the number of matrix elements is very small in comparison to FEM.

As the method’s name indicates, SEA provides a statistical framework and does not give the exact deterministic solution. Also any prediction for an object (for example sound transmission through a wall) is not specific for that object and it can be considered for any other similar cases (walls) that have the same general properties (such as area, mass and stiffness or in general have the same Modal Density, Loss Factor and …).

Again, in this method, the number of modes per frequency band (or modal density) acts the major role and the accuracy of obtained results largely depends on it. Regarding the minimum required number of resonant frequencies in a special frequency band, different authors suggested different values. For example Craik [14] suggests from 2 to 30 while Cremer [17] suggests at least 6 frequencies per band for each element. However Renji [18] indicates that even if there is only one mode in a particular subsystem in a specific frequency band, if the interacting
subsystem has several modes in the same frequency band, SEA is applicable.

One of the main advantages of SEA is that it can help to identify the major contributor in the overall energy of system. Also it is easy to monitor the effects of changes in the design [1]. In addition Serradj [19] indicates that the power - energy relation is not sensitive to small parameter changes, Energy quantities can be averaged more easily, Most often, the actual goal of a computation are energy quantities (e.g. the sound pressure level that is an energy quantity).
4. Examples

4.1 Two cavity and a wall
Here a simple example tries to show how the SEA method can be used for two connected objects:

Suppose three objects are connected (A, B and C), while the power P is acting on A (This example usually defines the problem of two side by side rooms - here A and C - separated by a wall - here B - in between) figure 4-1:

![Diagram of two side by side rooms](image)

Figure 4-1 Problem of transmitted sound in two side by side rooms

For every $f$ we will have:
\( \Pi = E_1 \eta_1 2 \pi f + (E_1 \eta_{12} 2 \pi f - E_2 \eta_{21} 2 \pi f) + (E_1 \eta_{13} 2 \pi f - E_3 \eta_{31} 2 \pi f) \)
\( 0 = E_2 \eta_2 2 \pi f + (E_2 \eta_{21} 2 \pi f - E_1 \eta_{12} 2 \pi f) + (E_2 \eta_{23} 2 \pi f - E_3 \eta_{32} 2 \pi f) \)
\( 0 = E_3 \eta_3 2 \pi f + (E_3 \eta_{31} 2 \pi f - E_1 \eta_{13} 2 \pi f) + (E_3 \eta_{32} 2 \pi f - E_2 \eta_{23} 2 \pi f) \)

According to what have been previously mentioned, we will have:

\[
[A] = \begin{bmatrix}
(\eta_1 + \sum_{i \neq 1}^{3} \eta_{1i}) N_1 & -\eta_{21} N_1 & -\eta_{31} N_1 \\
-\eta_{12} N_2 & (\eta_2 + \sum_{i \neq 2}^{3} \eta_{2i}) N_1 & -\eta_{32} N_2 \\
-\eta_{13} N_3 & -\eta_{23} N_1 & (\eta_1 + \sum_{i \neq 3}^{3} \eta_{3i}) N_3
\end{bmatrix}
\]

\( \{\Pi_{in}\} = \begin{bmatrix}
\Pi_1 \\
0 \\
0
\end{bmatrix} \)

\( \{\varepsilon\} = \begin{bmatrix}
\frac{E_1}{N_1} \\
\frac{E_2}{N_2} \\
\frac{E_3}{N_3}
\end{bmatrix} \)

As you can see, there is a 3 by 3 matrices that should be generated (compare it to the same example in FEM) and by having according to the parameter that we are interested to calculate the equation can be solved. These parameters can be Internal loss, Energy or etc.

### 4.2 Some general formulas

Since the introduction of SEA in 70’s, many efforts have been done in order to calculation of mentioned variables analytically.
For example Cremer [17] suggests a table of different formulas for calculating Modal Density for different objects (Beam, Plate, Cavity etc.) For a plate he suggests:

\[ Cl = \left[ \frac{E}{\rho(1-v^2)} \right]^2 \]  \hspace{1cm} (4-1)

\[ n(f) = \frac{s\sqrt{12}}{2Clt} \]  \hspace{1cm} (4-2)

Where S is the surface area, t is its thickness, v is Poisson’s ratio, \( \rho \) is density.

Similarly for a three dimensional volume it will be:

\[ n(f) = \frac{4\pi f^2V}{c^3} + \frac{\pi fA}{2c^2} + \frac{P}{8c} \]  \hspace{1cm} (4-3)

V is the volume, A is the total area and P is the total edge length (perimeter).

Other formulas are available in White and Walker and Craik [12], [14].
5. Introducing the software

The software ESI-VA One 2014, is a solution for simulating the response of vibro–acoustic systems across the full frequency range and it consists of different modules. For the current thesis I have used the Statistical Energy Analysis (SEA) module. This is the evolution of the industry standard software for mid and high frequency noise and vibration design, AutoSEA2[20].

5.1 Simple example with the software VA One

After presenting the concept of SEA and it related calculations, now it is the best time to follow some simple examples and evaluate the results of the software. For this reason the following object is considered:

Steel plate $1\text{m} \times 1\text{m} \times 0.001\text{m}$ with the following properties:

- $E=2.1\text{e}11$ Young’s modulus
- $\nu =0.3$ Poisson’s ratio
- $\rho =7800$ Density

The modal density in the third octave band is calculated as the figure 5-1:
In another example, for a plate that usually used in the vehicle body:
Steel with the diameter 0.7 mm and the rest of properties same as the previous object, the results can be seen in figure 5-2:
Finally in a cubic cavity with the following properties:

Dimension: 1m × 1m × 1m

C=343 m/s  Sound speed in air

The modal density is presented in figure 5-3.

Figure 5-2 Modal density of flat plate t=0.0007 m calculated by VA One
In all 3 mentioned cases, the frequency is the variable of the horizontal axis and the number of resonant modes per radian is on the vertical axis.

5.2 Validation and comparing to analytical formula
It is possible to validate the presented results from the mentioned examples. For this reason, the formulas that have been presented before in 4.2 have been used for the same modelled object. By using MATLAB as a solver and final results are presented in the third octave band frequency. The results from VA One software and the analytical formula are shown in the same graph. The figures 5-4 and 5-5 indicate the result for the 1 mm thick
and 0.7 mm thick plates, while figure 5-6 shows modal density results for the one cubic meter cavity:

*Figure 5-4 Comparing results for 1 mm steel plate*
In the first two examples the formula 4-1 and 4-2 were applied, and for comparing the result for the one cubic meter air cavity, formula 4-3 was applied.

*Figure 5-5 Comparing results between for 0.7 mm steel plate*
The results have the same trend in all three cases especially in higher frequencies.

It is possible to compare the number of resonant frequencies per frequency band from the FEM software to the current model. The same models established by the FEM software (here HyperMesh 2012) the resulted resonant frequencies were used and finally the number of mode per frequency band in 3rd octave frequency were calculated. Here it should be announced that due to the problem of memory, the modelled frequency range was modified to 8000 Hz.

The final results are indicated below in figure 5-7 for 1 mm and figure 5-8 for 0.7 mm thick plates and again the same matching can be seen.

*Figure 5-6 Comparing results for $1^3$ m air cavity*
5.3 More complicated part from the vehicle

For this part of thesis, one of the most important parts of a car, the Firewall, is subjected for more evaluation and analysis. Firewall is the separator of cabin from engine space and has a very important role in the noise reduction of the cabin. In fact, if one wants to categorize the possible noises inside
the cabin of vehicle, the most important noises belongs to the “Engine bay” which means the different noises from engine, pumps, cooling system etc. Therefore using sound treatment of body structure, especially for the dash (fire wall) and floor, is crucial. The acoustic treatments for these parts are usually similar and both of them focused on Sound Insulation so having a good model is crucial.

As said before, the number of objects that can be evaluated by the method and software in order to calculate the number of mode per frequency bands analytically has so far been limited to: Flat Plate, Singly Curved Plate, Doubly Curved Plate, Beam and Acoustic Cavities. Therefore to be able to model a complex model such as vehicle, that is has many different complex shapes, it is suggested that one try to define the most similar shape with almost the same number of frequencies per band.

Another point that needs to be considered is that when the geometry imported in the SEA software, it is possible to use. Then define a flat plate based on that. In this case, although you see a complex shape object, all the calculations will be done based on a flat plate object then in another model a singly curved plate and at last a doubly curved object.

At the first step the FEM software (again HyperMesh figure 5-9) is used to give the resonant frequencies until up to a limited frequency (6300 Hz). And then two models in VA One based on that geometry is generated and they appear in figures 5-10, 5-11. The results are plotted in figure 5-12 by using MATLAB and these values can be compared.
Figure 5-9 Model in FEM software (HyperMesh)

Figure 5-10 Using the same geometry in VA One and defining properties as a Flat Plate
Figure 5-11 Define in VA One as a Doubly Curved Plate
Figure 5-12 Comparing result between FEM and different model by VA One

The red line shows the acquired results from FEM software for the trim with the defined geometry dimension. It is not a flat line and has a curvature which means that it is not acting like a simple plate (black line). However in the lower frequency (below 3500 Hz) the result seems close. On the other hand at frequencies more than 4000 it appears that the behaviour of the dash is more close to the doubly curved plate. So maybe we can conclude that depends on the frequency that we plan to evaluate, if we can model the dash based on a flat plate or doubly curved plate.
6. The porous material

6.1 Acoustic properties

The amount of noise inside a vehicle cabin is one of the major indexes of quality in this industry. One of the very first decisions is to use some noise absorbing material such as foam, fibre or multi-layered noise treatment. For porous materials at high frequencies, an adiabatic process takes place that produces heat loss due to friction when the sound wave crosses the irregular pores (in fact the sound energy converts to heat) [21]. At low frequencies, porous materials absorb sound by energy loss caused by heat exchange. Today by advancing in every branch of science and technology, new materials are being introduced by industrial companies every day and the need of having an analytical method for evaluation of acoustical properties of such materials is growing. In fact by having such information, managers are able to choose between the different materials in the shortest time with the lowest cost.

Recently researchers in the University of Sherbrooke (Université de Sherbrooke) and E.N.T.P.E. (École nationale des travaux publics de l’État) in Lyon proposed two methods for determination of these parameters based on the results from standardized impedance tube measurements ASTM E1050, ISO 10534-2, ASTM E2611 [22]–[24]. The software FOAM X uses the same theory for the model to be fitted. In fact the software uses results from an impedance tube to estimate Porosity, the static airflow resistivity ($\sigma$), tortuosity ($\alpha\infty$), the characteristic lengths ($\lambda$ and $\lambda'$), and the static thermal permeability ($k'0$) of air-saturated porous media.

6.2 Impedance tube measurement

Based on what have been presented in the previous chapter, now the acoustic properties derived from impedance tube must be measured. By using these values as inputs in the FOAMX software, it calculates and gives the values that mentioned before (porosity, tortuosity, characteristic lengths and static thermal permeability and static thermal permeability) and finally it will be possible to use them for defining new foam materials for modelling. This measurement follows the standards ISO 10534-1:1996, ISO 10534-2:1998, (For the software it is also possible to follow the
standard ASTM E 1050). And the following data should be extracted from it:

1. Frequency in Hertz
2. Sound absorption coefficient ($\alpha$)
3. Real part of the complex reflection coefficient ($Re(R)$)
4. Imaginary part of the complex reflection coefficient ($Im(R)$)
5. Real part of the normalized complex surface impedance ($Re(Z)$)
6. Imaginary part of the normalized complex surface impedance in ($Im(Z)$)

For this reason the current definition and adjustment was done to the experimental setup based on the [25]:

The Brüel & Kjær impedance tube 4206 with $\frac{1}{4}''$ microphones and the software: PULSE has been applied. In this experiment 2 different size of cylinders, 29 mm (cylinder for frequency range of 500~6300) and 100 mm (cylinder for frequency range of 50~1600) have been used. Consequently for every material (here two subjects), two different sizes (29 mm and 100 mm) were cut by using a cutter blade. Here it should be mentioned that in order to get the best result it is recommended that (to avoid mechanical deformation from cutting process) samples are cut by water jet. However, it is still possible to use the classical device as shown below in figure 6-1.
The following precautions should be taken:

- The drill press should be aligned and rotation of the cutter centralized.
- Test several samples and require their absorption to be similar.
- Move the rotating cutter slowly down into the material.
After connecting the equipment and putting the sampled foam in the related size cylinder (the experimental set shows in figure 6-2), it is time to start to run the measurement. So the following adjustment based on the sample size is required:

*Figure 6-2 Measurement set, Thermometer and Barometer*
<table>
<thead>
<tr>
<th>Tube</th>
<th>Large</th>
<th>Small</th>
</tr>
</thead>
<tbody>
<tr>
<td>Microphone Spacing:</td>
<td>0,05 m</td>
<td>0,02 m</td>
</tr>
<tr>
<td>Distance to Sample from Mic. B, Pos. 3:</td>
<td>0,1 m</td>
<td>0,035 m</td>
</tr>
<tr>
<td>Distance to Source from Mic. A, Pos. 2:</td>
<td>0,15 m</td>
<td>0,37 m</td>
</tr>
<tr>
<td>Diameter:</td>
<td>0,100 m</td>
<td>0,029 m</td>
</tr>
<tr>
<td>Lower Frequency Limit:</td>
<td>50 Hz</td>
<td>500 Hz</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Lines</th>
<th>1600</th>
<th>6400</th>
</tr>
</thead>
<tbody>
<tr>
<td>Span</td>
<td>1.6 kHz</td>
<td>6.4 kHz</td>
<td></td>
</tr>
<tr>
<td>Averages</td>
<td>100</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Centre Frequency (Hz):</td>
<td>800,0</td>
<td>3200,0</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Generator</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Generator Active</td>
<td>TRUE</td>
<td>TRUE</td>
</tr>
<tr>
<td>Waveform:</td>
<td>Random</td>
<td>Random</td>
</tr>
<tr>
<td>Signal Level:</td>
<td>1,414Vrms</td>
<td>1,414Vrms</td>
</tr>
<tr>
<td>Pink Filter:</td>
<td>Off</td>
<td>Off</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Environment</th>
<th>Atmospheric Pressure:</th>
<th>&quot;Based on experiment moment&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative Humidity:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Velocity of Sound:</td>
<td>344,41 m/s</td>
<td></td>
</tr>
<tr>
<td>Density of Air:</td>
<td>1,195 kg/m³</td>
<td></td>
</tr>
<tr>
<td>Characteristic Impedance of Air:</td>
<td>411,5 Pa/(m/s)</td>
<td></td>
</tr>
</tbody>
</table>
Defining the environmental indexes such as temperature, air pressure and humidity should be done by a proper device and choosing the values in the related boxes the software.

6.3 Using the results from Impedance tube into FOAM X

After the samples have been cut and examined in the impedance tube, finally the results are extracted in text format and applied by FOAMX. The related adjustments (such as frequency range based on sample size) should be defined and room and tube condition properties such as temperature, air Pressure and humidity should be measured by valid instruments and proposes to the software.

The results of the Foam with the thickness of 10 mm and diameter of 29 mm can be seen in figure 6-3. The values for lower frequencies (<500) varies dramatically and they do not seem reliable, so here the frequency range of 500~6400 is applicable. For example for one of the chosen foams, 3 samples were evaluated with the diameter of 29 mm and the adjustments (frequency range 500~6400 as well as temperature, air pressure) and absorption coefficient results over frequency are indicated and other settings are adjusted as indicated in the figure 6-5.
Figure 6-3 Measured result for the sample, diameter: 29 mm
Figure 6-4 Other adjustments related to the FOAM X software

The final results for the mentioned factors are presented in the following table, Table 6-2:

Table 6-2 Acquired results from FOAMX for the two selected materials

<table>
<thead>
<tr>
<th>Measured Variance</th>
<th>Measured Variance</th>
<th>Measured Variance</th>
<th>Measured Variance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Porosity</td>
<td>0.859</td>
<td>0.037</td>
<td>0.885</td>
</tr>
<tr>
<td>Resistivity</td>
<td>102587</td>
<td>21198.8</td>
<td>117857</td>
</tr>
<tr>
<td>Tortuosity</td>
<td>1.102</td>
<td>0.332</td>
<td>1</td>
</tr>
<tr>
<td>Viscous Length</td>
<td>18</td>
<td>32.6</td>
<td>326.2</td>
</tr>
<tr>
<td>Thermal length</td>
<td>336.1</td>
<td>138.8</td>
<td>326.2</td>
</tr>
</tbody>
</table>

Comparing different values from the two different FOAMs: 2cm and 1cm Thick

Now by having these parameter values it is possible to define the required parameters for the software VA One in order to define new foam materials (treatments) in the car body. By choosing the related menu in the VA One,
you can see by defining these value, it is possible to see the effects of such material on a body part:

Figure 6-5 Window for defining the acoustic properties of foam materials
7. Transmission Loss experiment

One of the major experiments that is very common in the automotive industry in the field of sound reduction is measurement of “Sound Transmission Loss and Insertion Loss”. A good STL is critical in many car applications, especially when it comes to attenuate the engine noise.

For the current thesis this experiment also planned and in order to do that required plate material, steel with the thickness of 0.85 * 0.85 * 0.007 m ordered and in the related place installed, the window dimension was 0.5m by 0.5m . Regarding this experiment there are two standards as ISO 9614-1:1993, ISO 9614-2:1996 for that and Fahy in the [26] explained the whole theory behind it.

Regarding running this experiment certain facilities and equipment are required and for this reason the mentioned sheet mounted in the window (with 0.5m by 0.5m dimensions) between two connected room so called the Reverberation room and Anechoic chamber.

The overall steps of running the experiment are to mount an object between reverberation and anechoic room. Then generate a noise in the reverberation room while it measured by rotating microphone set and on the other side in the anechoic room the sound pressure is measured by using a probe in the proper way [26]–[29]. The Sound Transmission Loss (STL) will be:

\[
STL = L_p - L_I - 6dB
\]  

(7-1)

Where the Sound Pressure Level (L_p) is

\[
L_p = 10log_{10}\left(\frac{p^2}{p_{ref}^2}\right)
\]  

(7-2)

And Sound Intensity Level (L_I) is:

\[
L_I = 10log_{10}\left(\frac{l}{l_{ref}}\right)
\]  

(7-3)

So according these formulas the test is run and the result acquired then a modification on the object for example covering the object with a sound absorbent material is executed and the final important measure that is interested to be measured is calculated:

\[
\text{Insertion Loss} = STL_{\text{with refinement}} - STL_{\text{without refinement}}
\]  

(7-4)
That means usually for running the test the object is measured and the related STL is calculated then again the same test is executed while now the added part (usually foam as a sound insulation).

*Figure 7-1 Reverberation Room*

*Figure 7-2 Anechoic Chamber*
There is no doubt that this measurement is very sensitive to the circumstances and execution of the procedure should accurately be followed.

For the thesis the experimental set and the object installed and the measurement run once and the results generated. After processing the results it was founded that they are not accurate and after investigations it was founded that the calibration of the probe was not correct. Unfortunately due to some economical situation at the NEVS company there was no time for redoing the test.

The learned lesson here is that Sound Intensity is a quantity that can ‘see’ the direction in which the sound energy is flowing. In a perfect world or setup, the energy comes from the source room and passes through the test specimen. If it is measured correctly, it will be only positive intensity values in this case. When it gets a negative intensity value for some frequencies or octave bands, it means that there is more sound energy flowing from outside towards the source room. For this case, I have no idea how much energy is actually coming from the source room. It can be other sound source around probe when the measurement was running, or bad calibration. These are causing the negative intensities or the sound from the source room is also leaking to the outside through other walls/leaks/holes and to your intensity measurement area.
8. Whole Body Model by the software VA One

The final step is to make the whole body model. Similar to most of the modelling software, in order to establish a model the first step is defining the geometry for the software. Here the data that imported from CAD software to Finite Element software has been applied. It should be mentioned again that it is very important for this method that the value from modal density and damping are close to reality as said in Chapter 3 there are four important variables:

1. The mode count,
2. The damping loss factor,
3. The coupling loss factor, and
4. The input power from the external sources of excitation.

For this reason the data from the geometry of body has been used and the different parts defined for the model such as Doors (Front, back, left and right), Front window, Back window, Side windows, Fire wall (Left, right) and many more materials were defined [30].

It is also necessary to create and define the air cavities inside and outside of the cabin and defining internals such as the locations of passengers (head and body), and externals (in order to define the energy dissipation in the environment). In figure 8-1 the body can be seen and in figure 8-2 the internal cavities appear. For each passenger two cavities are defined one around the head and one around the rest of body. Finally in figure 8-3 the external cavities are presented.
Figure 8-1 The modeled car body

Figure 8-2 The body with the air cavities inside
In this phase other important factors must be considered as well. For instance: the covering layer of the body such as dash, the seats material, and covering layer of ceiling. Also as the input, two pressure defined with the spectrum defined as figure 8-4 and they are equal to 1 Pa sound pressure for the frequency in the 1/3 octave band from 16 to 8000 Hz. These sound pressers defined on the firewall (on the face connected to the engine bay) and the reason is to have an overall view in order to understand the sound propagation from engine to the different locations in the cabin. The frequency range can be adjusted easily and the current frequency range can be changed whenever it is required.
Figure 8-4 The power input
9. Results and discussion

As discussed so far, the aim of this thesis to study SEA method, its strength and weakness and trying to find a primary model for a vehicle. Following that, as can be seen in chapter 4 some general formula based on the references were introduced (formula 4-2 and 4-3). Then some examples were shown where the number of modes per frequency band has been calculated analytically and by using the software VA One. These results were compared in the figures 5-4, 5-5 and 5-6. In all examples lower frequencies especially at the beginning as it indicated in the following a fluctuation and divergence can be seen and as frequency increases. But the results converge more and more for high frequencies. In another example, the comparison between FEM and SEA is subjected (figure 5-7, 5-8) shows a good conformity. In figure 9-1, 9-2 the zoomed graphs presented:

![Modal Density Comparing 1 mm Steel Plate between VA One and Analytical Formula](image)

*Figure 9-1 Comparing results for 1 mm steel plate*
In Chapter 6, different steps to apply the results from impedance tube in order to estimate the acoustic properties of foam materials in the FOAMX software were explained and all steps were followed for two types of foams. As it said in the related chapter by increasing the number of samples and improving the accuracy of cutting (mostly considering the speed of cutting), more accurate result can be extracted specially it can be seen that by having accurate samples, the final results will have narrower boundaries (figure 9-3).
Figure 9-3 Results from FOAMX and the tolerance

Considering the whole body modelling, it is now required to understand the body parts behaviour. Also it is important to try to have the same modal density in our model compared to real part. By running the established model, the following results have been acquired figure 9-4, 9-5, 9-6:
This model with two pressures on the dash as input the major effects will be on the front seats. As it can be seen in below for the air cavity defined as the space around the driver head the energy will be:
The power input to the space located in the driver head is:
Figure 9-6 Power inputs in drivers head from different objects

The major role in this result is due to the cavity attached to the Firewall (here named as “Front Down Left” cavity) and also the powers from the Front Window, Front Side Window have effects on that.

With a very simple sound insulation material defined as a cover on the Firewall and the response is indicated and compared to without the insulation in figure 9-7, 9-8:
Figure 9-7 Comparing the energy around the driver head before (hashed) and after (solid) insulation
Figure 9-8 Comparing the Power Inputs around the driver head before (hashed) and after (solid) insulation

It should be considered that these modifications can be easily done on the SEA model (in comparison with FEM) and the results can be extracted in less than a minute.
10. Conclusion

Noise, vibration, and harshness analysis is a wide area that consists of many different activities, especially for a vehicle in the automotive area. It is a search for the source of a noise, shake, or vibration, and it refers to the entire range of vibration perception, from hearing to feeling.

In this thesis a complete cycle of the sound environment is presented. Overall study over Statistical Energy Analysis basics, evaluation of advantages and disadvantages were done. In addition some examples were presented and studied. Then it brought to action and execution, this was done through modelling the car cabin by using the geometry data and establishing them in SEA software VA One. Also a study over the concepts of acoustical properties of porous material was done and after that, the measurement regarding acoustical determination of new material through impedance tube and related establishments was executed.

As it mentioned in chapter 3 there are some analytical formula for calculation of required parameters to establish SEA model so prediction and validation of numerical formulas using two software VA One and MATLAB were executed and the result also are compared to the results from the FEM model established in Hypermesh.
11. Future Works

Regarding measuring of acoustic properties of a foam material, it is possible to run an exact measurement to compare the result from FOAMX. Another improvement to the existing approach is to measure the noise produced by different parts located in the engine bay and then define them as the input for our model and monitor the results. Finally, measuring the amount of sound inside the vehicle by installing microphones at different locations, and then comparing them to the previous results seems very useful. Also following chapter 7, it is highly recommended to run again the measurement STL considering the lessons learned in the current work.
12. References
