



**Faculty of Mechanical Engineering**

**A CORRECTED MODEL OF  
STATISTICAL ENERGY ANALYSIS (SEA)  
IN A NON-REVERBERANT ACOUSTIC SPACE**

**Al Munawir**

**Master of Science in Mechanical Engineering**

**2014**

**A CORRECTED MODEL OF  
STATISTICAL ENERGY ANALYSIS (SEA)  
IN A NON-REVERBERANT ACOUSTIC SPACE**

**AL MUNAWIR**

**A thesis submitted  
in fulfillment of the requirements for the degree of Master of Science  
in Mechanical Engineering**

**Faculty of Mechanical Engineering**

**UNIVERSITI TEKNIKAL MALAYSIA MELAKA**

**2014**

## DECLARATION

I declare that this thesis entitled "A Corrected Model of Statistical Energy Analysis (SEA) In A Non-Reverberant Acoustic Space" is the result of my own research except as cited in the references. The Thesis has not been accepted for any degree and is not currently submitted in candidate of any other degree.

Signature : .....

Name : .....

Date : .....

## APPROVAL

I hereby declare that I have read this thesis and in my opinion this thesis is sufficient in terms of scope and quality for the award of Master of Science in Mechanical Engineering

Signature : .....

Supervisor Name : .....

Date : .....

## **DEDICATION**

*"To my beloved parents and wife"*

## ABSTRACT

Statistical Energy Analysis (SEA) is a well-known method to analyze the flow of acoustic and vibration energy in a complex structure. The method is based on the power balance equation where energy in the divided subsystems must be reverberant. This study investigates the application of SEA model in a non-reverberant acoustic space where the direct field component dominates the total sound field rather than a diffuse field in a reverberant space. Here, a corrected SEA model is proposed where the direct field component in the energy is removed and the power injected in the subsystem considers only the remaining power after the loss at first reflection. To validate the model, a measurement was first conducted in a box divided into two rooms where the condition of reverberant and non-reverberant can conveniently be controlled. In the case of a non-reverberant space where acoustic material was installed inside the wall of the experimental box, the signals are corrected by eliminating the direct field component in the measured impulse response. Using the corrected SEA model, comparison of the coupling loss factor (CLF) with the theory shows good agreement. Secondly, a test was conducted in a car cabin where the front and rear cabins act as two separate subsystems. A loudspeaker was first used to inject the sound energy into the subsystems and several microphones were located to measure the transfer function. The CLF and the damping loss factor (DLF) were obtained using the classical SEA model. The corrected CLF and DLF are then calculated using corrected SEA model after eliminating the direct field components. The engine was then turned on to provide the input energy into the cabin. The sound power transmitted into the cabin was measured and from here the sound pressure level (SPL) can be obtained, either using the uncorrected CLF and DLF or using the corrected CLF and DLF. The results were compared with the directly measured SPL showing that good agreement is obtained from those using the corrected SEA model.

## ABSTRAK

*Analisis Statistik Tenaga (SEA) merupakan satu kaedah yang terkenal untuk menganalisis aliran akustik dan getaran tenaga dalam struktur yang kompleks. Kaedah ini adalah berdasarkan persamaan keseimbangan tenaga di mana tenaga dalam subsistem dibahagikan mestilah yg bergema. Kajian ini menyiasat penggunaan model SEA di dalam ruang akustik tanpa gema di mana komponen 'direct field' dari sumber bunyi mendominasi jumlah medan bunyi berbanding dengan medan resapan di dalam ruang yang bergema. Di sini, pembetulan terhadap model SEA dicadangkan di mana komponen 'direct field' dalam tenaga dikeluarkan dan kuasa disalurkan ke dalam subsistem hanya mempertimbangkan kuasa yang tinggal selepas kehilangan pada pantulan pertama. Untuk megesahkan model tersebut, ukuran pertama dilaksanakan di dalam kotak yang mempunyai dua bilik di mana keadaan yg bergema dan bukan yg bergema supaya mudah dikawal. Di dalam kes ruang yang tidak bergema di mana bahan akustik telah dipasang di dalam dinding kotak eksperimen, isyarat diperbetulkan dengan menghapuskan komponen 'direct field' sebagai tindak balas impuls yang diukur. Dengan menggunakan model SEA yang telah diperbetulkan, perbandingan terhadap faktor kehilangan gandingan (CLF) menunjukkan persamaan yang sepadan dengan teori. Kedua, pengukuran dijalankan di dalam kabin kereta dimana kabin depan dan kabin belakang bertindak sebagai dua subsistem yang terpisah. Pembesar suara digunakan untuk menyalurkan tenaga bunyi ke dalam subsistem dan beberapa mikrofon diletakkan untuk mengukur fungsi pindah. Kemudian CLF dan faktor redaman kehilangan (DLF) diperolehi daripada model SEA terdahulu. Seterusnya model SEA yang telah diperbetulkan digunakan untuk memperbaharui CLF dan DLF yang diperolehi setelah komponen 'direct field' dihapuskan. Enjin dihidupkan untuk memberi input tenaga ke dalam kabin. Kuasa bunyi yang dihantar ke dalam kabin diukur dan tahap tekanan bunyi (SPL) boleh diperolehi, sama ada menggunakan kesemua CLF dan DLF yang diperbetulkan ataupun tidak. Perbandingan dibuat dengan SPL yang diukur secara lansung menunjukkan bahawa keputusan tersebut sepadan dengan model SEA yang diperbetulkan. Ini menunjukkan model SEA yang diperbetulkan memberi ramalan yang lebih baik terhadap tahan tekanan bunyi (SPL) di dalam kabin kereta.*

## ACKNOWLEDGEMENTS

*In the name of Allah, The Beneficent, The Merciful*

First, I would like to give my sincere to my supervisor, Dr. Azma Putra for his perfect guidance, supervision and motivation in this research since the very beginning. I would like to also acknowledge UTeM for waiving the tuition fee during my study and Ministry of Higher Education Malaysia (MOHE) for the Fundamental Research Grant Scheme (FRGS) under which this study is funded as well as to support my monthly allowance.

My gratitude is also addressed to my beloved parents, Abdullah Usman and Wardiah. Thank you so much for your affection, advices, guidance, instruction and help in all my life. Also for my parents in law, Darmia and Faridah for their numerous kindness.

For all my Indonesian and Malaysian friends, PPI and KEPS-UTeM. I would like to thank for the beautiful friendship. I also pray for their successful life in the future. For my colleagues in Acoustics and Vibration Group, thank you for the brilliant discussion which gives much input to my work.

Most important recognition and appreciation are dedicated to my family who shared in the joys and frustrations of this study. My greatest praise is reserved for my wife, Zuchra Ulfa. I appreciate her admirable patience, understanding and encouragement, which helped immensely in sustaining my efforts especially during the writing of this thesis.

## TABLE OF CONTENTS

	<b>PAGE</b>
<b>DECLARATION</b>	<b>i</b>
<b>APPROVAL</b>	<b>ii</b>
<b>DEDICATION</b>	<b>iii</b>
<b>ABSTRACT</b>	<b>i</b>
<b>ABSTRAK</b>	<b>ii</b>
<b>ACKNOWLEDGEMENT</b>	<b>iii</b>
<b>LIST OF TABLES</b>	<b>vii</b>
<b>LIST OF FIGURES</b>	<b>viii</b>
<b>LIST OF APPENDICES</b>	<b>xi</b>
<b>LIST OF ABBREVIATIONS</b>	<b>xii</b>
<b>LIST OF SYMBOLS</b>	<b>xiii</b>
<b>LIST OF PUBLICATIONS</b>	<b>xiv</b>
<b>CHAPTER</b>	
<b>1 INTRODUCTION</b>	<b>1</b>
1.1 Background	1
1.2 Past researches on the SEA model development and its application	5
1.3 Problem statement	10
1.4 Objective	10
1.5 Scope of the study	10
1.6 Methodology	11
1.7 Thesis outline	13
1.8 Thesis contributions	14
<b>2 STATISTICAL ENERGY ANALYSIS</b>	<b>15</b>
2.1 Introduction	15
2.2 Coupling powers and coupling power proportionality	15
2.3 Basic equation of SEA	17
2.4 Weak and strong coupling	20

2.5	Theoretical SEA Parameters	22
2.5.1	Input power	22
2.5.2	Damping Loss Factor (DLF)	24
2.5.3	Coupling Loss Factor (CLF)	25
2.6	Experimental SEA	26
2.6.1	A simple approach	26
2.6.2	Power injection method (PIM)	27
<b>3</b>	<b>CORRECTED SEA MODEL</b>	<b>30</b>
3.1	Introduction	30
3.2	Direct field and reverberant field in an acoustic space	30
3.3	Determination and elimination of direct field component	33
3.3.1	Inverse square law technique	33
3.3.2	Impulse response technique	35
3.4	Corrected SEA model in a non-reverberant field	38
<b>4</b>	<b>VALIDATION OF THE CORRECTED SEA MODEL</b>	<b>39</b>
4.1	Arrangement	39
4.2	Measurement setup	41
4.3	Measurement of sound power	42
4.4	Estimation CLF and DLF from the reverberant condition	44
4.5	Estimation of CLF and DLF from non-reverberant condition	49
4.5.1	Removing the direct field component	51
4.5.2	Applying the modified SEA model	54
<b>5</b>	<b>EXPERIMENTAL SEA IN A CAR CABIN</b>	<b>59</b>
5.1	Division of subsystems	59
5.2	Experimental methodology	63
5.3	Measurement of CLFs and DLFs	64
5.4	Validation of the corrected SEA model in the car cabin	73
5.4.1	Measurement of sound power from the engine	73
5.4.2	Determination of the sound pressure level (SPL)	75
<b>6</b>	<b>CONCLUSION AND RECOMMENDATION</b>	<b>79</b>
6.1	Conclusion	79
6.1.1	The experimental SEA in a coupled-box	79
6.1.2	The experimental SEA in the car interior	80
6.2	Recommendation	81
	<b>REFERENCES</b>	<b>86</b>
	<b>APPENDICES</b>	<b>87</b>
<b>A</b>	<b>EQUIPMENT</b>	<b>87</b>
A.1	Sound intensity probe	87

A.2	dBSolo Analyzer	87
<b>B</b>	<b>MEASURED IMPULSE RESPONSE</b>	<b>89</b>
B.1	Impulse response in a coupled-box	89
B.2	Measurement impulse response in the car cabin	91

## LIST OF TABLES

<b>TABLE</b>	<b>TITLE</b>	<b>PAGE</b>
2.1	Types of coupling loss factor (CLF).	26
4.1	List of equipment used in the experiment.	42

## LIST OF FIGURES

FIGURE	TITLE	PAGE
1.1	Illustration of subsystem division in the SEA model.	3
1.2	Comparison between FRF and SEA predictions.	4
1.3	Frequency range response of FEA and SEA (ESI Group, 2010).	4
1.4	General research methodology flowchart.	12
2.1	Analogy of power flow in SEA model: fluid flows from 'high head' to 'low head'.	17
2.2	SEA model of two subsystems.	17
2.3	Illustration of PIM for a system consisting of two subsystems.	28
3.1	Illustration of direct and reverberant field in an acoustic space.	31
3.2	Graph of direct and reverberant field as a function of distance.	31
3.3	Direct and reverberant field represented as impulse response and time.	32
3.4	A spherical propagation of sound wave in a free-field.	33
3.5	Diagram of methodology to remove the direct field component from the frequency transfer function.	37
4.1	(a) The experimental coupled-box with two subsystems and (b) the schematic diagram of the box.	40
4.2	Measurement setup for the experimental SEA (for non-reverberant condition).	41
4.3	Experimental setup for the sound power measurement inside a semi-anechoic chamber.	43
4.4	Measured normalised sound power from the loudspeaker.	44
4.5	Experimental setup for the reverberant condition.	45
4.6	Measured impulse response in the reverberant condition.	45
4.7	Measured normalised energy in the subsystems with the sound power injected in: (a) subsystem-1 and (b) subsystem-2.	46
4.8	Coupling loss factor obtained from experimental SEA in a reverberant condition.	47
4.9	Damping loss factor from experimental SEA in a reverberant condition.	47
4.10	Transmission loss of a single perforated plate from experimental SEA in a reverberant condition.	49
4.11	Arrangement experimental SEA in the non-reverberant condition with sponge absorbent attached on the wall.	50
4.12	Measured sound absorption coefficient of the sponge material used in the experiment.	50

4.13	Measured normalised energy in the subsystem before (—) and after (--) installing the absorbent material.	51
4.14	Measured impulse response in the non-reverberant (with absorber material).	52
4.15	Measured impulse response in the non-reverberant after the direct field component removed.	52
4.16	Measured normalised energy in the subsystems with power injected in: (a) subsystem-1 and (b) subsystem-2 before (—) and after (--) removing direct field component.	53
4.17	Coupling loss factor from experimental SEA in non-reverberant condition before correcting the direct field component (classical SEA model).	54
4.18	Coupling loss factor from experimental SEA in non-reverberant condition after correcting the direct field component (corrected SEA model).	55
4.19	Percentage error of $\eta_{12}$ with theory in non-reverberant condition.	55
4.20	Percentage error of $\eta_{21}$ with theory in non-reverberant condition.	56
4.21	Damping loss factor from experimental SEA in non-reverberant condition before correcting the direct field component (classical SEA model).	57
4.22	Damping loss factor from experimental SEA in non-reverberant condition after correcting the direct field component (corrected SEA model).	57
4.23	Percentage error of $\eta_1$ with theory in non-reverberant condition.	58
4.24	Percentage error of $\eta_2$ with theory in non-reverberant condition.	58
5.1	The test car used for the experimental SEA in the car cabin.	60
5.2	Position of reference microphones in the car cabin during measurement: (a) side view and (b) top view.	61
5.3	Measured normalised energy in the car cabin with power injected in: (a) the front cavity and (b) the rear cavity.	62
5.4	Diagram of methodology for experimental SEA in the car cabin.	64
5.5	Measurement setup for experimental SEA in the car cabin.	65
5.6	Positions of the microphone in: (a) the front cavity (b) the rear cavity of the car cabin.	65
5.7	Location of the sound source and the reference microphone in: (a) the front cavity and (b) the rear cavity.	66
5.8	Measured sound absorption coefficient of the car cabin.	66
5.9	Measured impulse response in the subsystem before the direct field component removed: (a) and (b) indicates point location in the subsystem.	67
5.10	Measured impulse response in the subsystem after removing the direct field component: (a) and (b) indicates point location in the subsystem.	67
5.11	Measured normalised energy in the car cabin with power injected in: (a) the front cavity (b) the rear cavity before (—) and after (--) removing direct field component.	68
5.12	Measurement damping loss factor in the front cavity of car cabin before and after removing direct field component.	69
5.13	Measurement damping loss factor in the rear cavity of car cabin before and after removing direct field component.	70

5.14	Percentage error of measured $\eta_1$ in car cabin with theory.	71
5.15	Percentage error of measured $\eta_2$ in car cabin with theory.	71
5.16	Measured CLFs from front cabin to rear cabin before and after removing the direct field component.	72
5.17	Measured CLFs from rear cabin to front cabin before and after removing the direct field component.	72
5.18	Scanning area for measurement of sound power using sound intensity.	74
5.19	Measured sound power in car interior.	74
5.20	Measured SPL in car interior from classical SEA and corrected SEA: (a) subsystem-1; 1000 rpm (b) subsystem-2; 1000 rpm (c) subsystem-1; 2000 rpm (d) subsystem-2; 2000 rpm (e) subsystem-1; 3000 rpm (f) subsystem-2; 3000 rpm.	76
5.21	Percentage error of SPL in car interior from classical SEA and corrected SEA with directly measured SPL: (a) subsystem-1; 1000 rpm, (b) subsystem-2; 1000 rpm, (c) subsystem-1; 2000 rpm, (d) subsystem-2; 2000 rpm, (e) subsystem-1; 3000 rpm and (f) subsystem-2; 3000 rpm.	78
A.1	A typical sound intensity probe.	87
A.2	Location of the dBSolo analyzer to measure reverberation time in the car cabin: (a) in the front cavity (b) in the rear cavity.	88
B.1	Measured impulse response in the subsystem-1: (a), (b), (c), (d), and (e) indicate the location of the response microphone in the subsystem.	90
B.2	Measured impulse response in the subsystem-1 after removing the direct field component: (a), (b), (c), (d), (e) indicate the location of the response microphone in the subsystem.	91
B.3	Measured impulse response in the subsystem-2: (a), (b), (c), (d), and (e) indicate the location of the response microphone in the subsystem.	92
B.4	Measured impulse response in the subsystem-2 after removing the direct field component: (a), (b), (c), (d), (e) indicate the location of the response microphone in the subsystem.	93
B.5	Measured impulse response in the subsystem-1 (front cavity) before removing the direct field component: (a), (b), (c), (d), (e), and (f) indicate the location of the response microphone in the subsystem.	94
B.6	Measured impulse response in the subsystem-1 (front cavity) after removing the direct field component: (a), (b), (c), (d), (e) and (f) indicate the location of the response microphone in the subsystem.	95
B.7	Measured impulse response in the subsystem-2 (rear cavity) before removing the direct field component: (a), (b), (c), (d), (e), and (f) indicate the location of the response microphone in the subsystem.	96
B.8	Measured impulse response in the subsystem-2 (rear cavity) after removing the direct field component: (a), (b), (c), (d), (e) and (f) indicate the location of the response microphone in the subsystem.	97



## LIST OF ABBREVIATIONS

<b>CLF</b>	<b>Coupling Loss Factor</b>
<b>DLF</b>	<b>Damping Loss Factor</b>
<b>FFT</b>	<b>Fast Fourier Transform</b>
<b>FRF</b>	<b>Frequency Response Function</b>
<b>PIM</b>	<b>Power Injection Method</b>
<b>SEA</b>	<b>Statistical Energy Analysis</b>
<b>SPL</b>	<b>Sound Pressure Level</b>
<b>TL</b>	<b>Transmission Loss</b>

## LIST OF SYMBOLS

$E_i$	Energy in the subsystem- $i$
$E_j$	Energy in the subsystem- $j$
$E_{dir}$	Energy travelling directly from the sound source
$E_{rev}$	Energy reflected from the surface
$f$	Frequency
$F$	Force
$I$	Sound intensity
$j = \sqrt{-1}$	Imaginary unit
$k$	Acoustic wavenumber
$n$	Modal density of the subsystem
$p$	Sound pressure
$P_{in}$	Input power
$S_{xx}$	Auto-spectra from the reference microphone and the response microphone
$S_{xy}$	Cross-spectra between the reference microphone and the response microphone
$T_{60}$	Reverberation time
$V$	Volume
$W$	Sound power
$Y_p$	Point mobility of the structure
$Z$	Impedence
$\alpha_i$	Absorption coefficient in subsystem- $i$
$\sigma$	Perforation ratio
$\tau$	Transmission coefficient
$\omega$	Angular frequency
$\eta_i$	Damping loss factor in the subsystem- $i$
$\eta_j$	Damping loss factor in the subsystem- $j$
$\eta_{ij}$	Coupling loss factor from subsystem- $i$ to subsystem- $j$
$\eta_{ji}$	Coupling loss factor from subsystem- $j$ to subsystem- $i$

## LIST OF PUBLICATIONS

A. Putra, Al Munawir and W.M.F.W. Mohamad (2014). The effect of the direct field component on a statistical energy analysis (SEA) model, *Applied Mechanics and Materials Journal*, Vol. 471, pp 279-284.

Al Munawir, A. Putra and W.M.F.W. Mohamad. A corrected model of Statistical Energy Analysis (SEA) in a non-reverberant acoustic space, *Engineering Noise Control Journal* (under review).

A. Putra, Al Munawir and W.M.F.W. Mohamad . Prediction of sound pressure level in a motor vehicle cabin using corrected SEA model, *Advances in Acoustics and Vibration* (under review).

# CHAPTER 1

## INTRODUCTION

### 1.1 Background

In recent years, noise and vibration performance is always a major issue for manufacturers around the world. Vibration is a phenomenon existing in a machine, such as vibrating pumps, motors and washing machines as well as in a flexible structure of a car or an airplane body. The level of noise and vibration has become one of the subjective performance indicators of a system. The problem of noise and vibration in vehicles for example, requires more attention due to competitive market and increasing customer awareness on low noise emission either for quality comfort inside the vehicle or noise pollution to the environment. Therefore, reduction of noise and vibration becomes important.

There are many methods to analyze the noise and vibration problem. The choice of the right method is important to find a required result with minimum attempt and cost. Energy-based methods can be considered as an effective method due to their techniques of using energy quantities that is energy and power rather than quantities such as force and displacement used by the classical analysis of vibration (Sarradj, 2004). Energy-based methods have some advantages:

- (a) The power-energy relation is not so sensitive to small parameter changes.
- (b) Energy quantities can be averaged more easily.

Apart from the energy-based methods, Finite Element Analysis (FEA) and Boundary Element Analysis (BEA) are very well known and are usually implemented for analysis of

noise and vibration at low frequency range. However, at high frequency range these methods are not efficient for several reasons:

- (a) Finer discretization in the model to cope with very small wavelength requires long computational time.
- (b) Require more details of structural pattern for a more complex structure.

One possible solution to solve the noise and vibration problems at mid to high frequency is Statistical Energy Analysis (SEA). It describes a complex system in terms of a network of connected subsystem, each of which has a resonant multi-modal response or, equivalently, reverberant wave field. In SEA, no attempt is made to recover the detail displacement pattern of the structure like FEA or BEA, but rather the structure is modelled as an assembly of subsystems. The aim is to predict the 'average' vibrational energy level of each subsystem. This is done by establishing a set of power balance equations which are based on the key assumption that the energy flow between two connected subsystems is proportional to the difference in the subsystem modal energies (Fahy, 1994). Such 'average' can be achieved when the structural or acoustic wavelength is much smaller than the dimensions of the corresponding structure or cavity.

Subsystems in SEA is defined as part of a system separated by boundaries across which distinct discontinuities in physical properties exist, e.g thickness, mass density or volume. Figure 1.1 shows division of car structure into three subsystems to predict the sound pressure level in the cabin due to force applied on the bulkhead. The subsystems can be divided into car cavity, windshield, and bulkhead. The noise in the cabin is expected to directly radiate from the vibration of the bulkhead. Meanwhile, there is also propagating vibration wave

from the bulkhead to the windscreen causing the screen to vibrate and eventually radiate noise into the cabin.

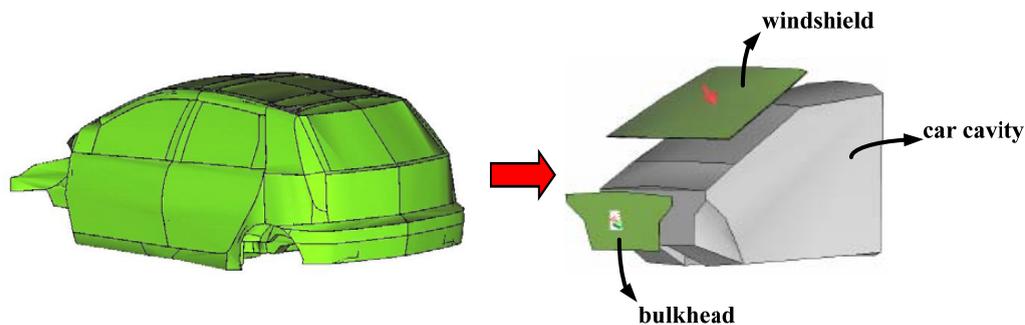


Figure 1.1 Illustration of subsystem division in the SEA model.

Because SEA is based on the statistical behaviour of the responses, therefore several assumptions apply:

- (a) Behaviour of the subsystem is dominated by resonances

A larger number of resonant modes in a frequency band smoothes the response spectra and the spatial variation in the responses (provided that the damping in the subsystems is not too large).

- (b) Weak coupling between subsystems

This is to allow 'control' of energy flow among the subsystems. In this case, the damping in both subsystems should be much larger than the coupling between subsystems. This means that almost all power dissipates in the excited subsystem and one subsystem does not affect the other subsystems.

Figure 1.2 shows example of a frequency response function (FRF) compared with the SEA prediction. At low frequency, the SEA can be seen to have large discrepancy with the FRF. This is because at low frequency, the FRF has very low modal density noted by the

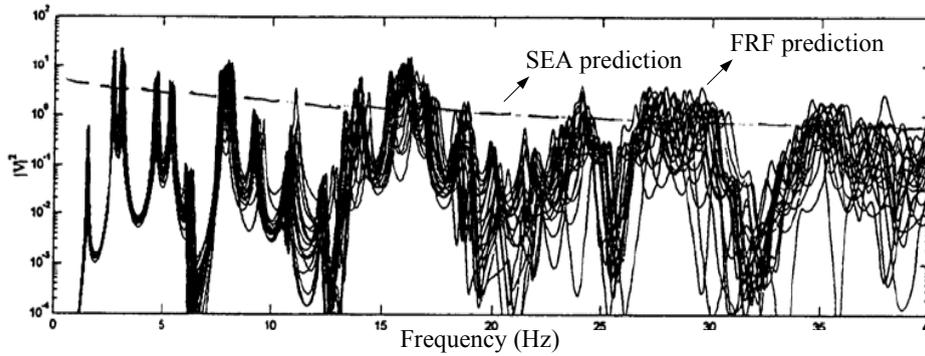


Figure 1.2 Comparison between FRF and SEA predictions.

distinct peaks in the graph (in this example below 15 Hz). The FRF can be seen to approach the SEA prediction at mid to high frequency as the modal density grows rapidly at higher frequency.

Figure 1.3 shows the typical frequency response of a structural-acoustic system, indicating frequency range of applicability for Finite Element Analysis (FEA), Boundary Element Method (BEM) and Statistical Energy Analysis (SEA). The SEA is best to predict the ensemble average at mid to high frequencies (Kenny, 2002).

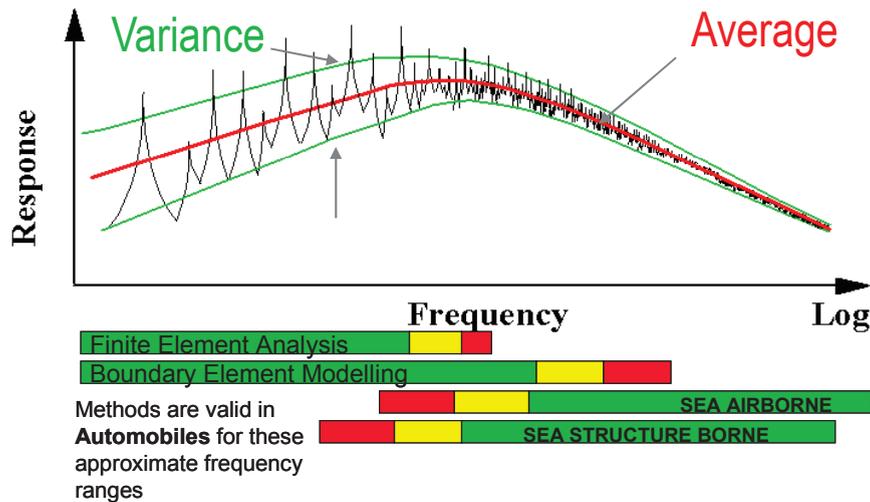


Figure 1.3 Frequency range response of FEA and SEA (ESI Group, 2010).

## 1.2 Past researches on the SEA model development and its application

Statistical Energy Analysis (SEA) was developed in 1960 by Lyon (1967) to predict the response of launch vehicles to rocket noise and to overcome the limitations of computational methods. He found that the power flow was proportional to the difference in uncoupled energies of the resonators and that it always flow from the resonator of higher to lower resonator energy. A complex structure was breached into subsystems and then stored and exchanged vibrational energy between these subsystems were analyzed.

The coupling loss factor (CLF) is the most important SEA parameter to be obtained from the SEA method. It represents how the energy flows from one subsystem to other subsystems. Good predictions of CLF is therefore critically important to obtain good estimation of the noise and vibration in a system. Price and Crocker (1969) formulated the CLF between room and cavity, assuming that transmission from room to cavity is the same as transmission from room to room. Cushieri and Sun (1994) presented a method for determining the dissipation and CLF from SEA model of a fully assembled machinery structure. The method is based on the experimental measurements of the total loss factors and the energy ratios between the subsystems of the machine structure when they are fully assembled. Yap and Woodhouse (1996) investigated the effects of damping on energy sharing in coupled systems. The approach taken is to compute the forced response patterns of various idealised systems, and from these the parameters of the SEA model for the systems are calculated.

Mace (1998) predicted the coupling loss factor using SEA theory for systems consisting of rectangular plates. It is found that if the damping is large enough (weak coupling) the response is independent of the shape of the plates and for lighter damping (strong coupling) the response depends significantly on the specific geometry of each plate. Skeen and