Operational Modal Analysis Studies on an Automotive Frame
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Abstract

Conventional Experimental Modal Analysis (EMA) methods have utilized Frequency Response Functions (FRFs) obtained by measuring both output measurements and input forces in a system. In recent years, there has been development of output-only Operational Modal Analysis (OMA) methods that do not require the measurement of input forces under strict assumptions in terms of the nature of excitation forces. These techniques find extensive applications in study of bridges and other structures where the assumptions are satisfied, and where it is difficult to measure input forces. The aim of this thesis work is to explore the use of this methodology for automotive applications. It is important to note that the OMA assumptions might not be necessarily met in this study, and this becomes part of the objective.

The real operational condition of a vehicle is at most times very different from its static one. While EMA techniques have been successfully employed on automotive structures to study their modal behavior, it is to be noted that these are not real operating conditions for the automobile. Doing a test on a vehicle in real excitation conditions such as running on a test track also poses several logistical challenges in terms of instrumentation and data acquisition. This thesis work attempts to study automotive structures using output measurements alone, while exciting the structure using means which are closer to real conditions. Results from these tests are compared with well established experimental methods using standard validation tools.
1 Introduction

One of the earliest studies of structures probably began with Galileo’s book, “Two New Sciences”. From hand calculations of the 17th century to Fast Fourier Transforms of present day, this field of engineering has seen growth to encompass several aspects of its widespread applications. With tremendous growth in computing power in the last several decades, the field of structural dynamics has stormed into the 21st century with previously unimaginable capabilities. The need for solving complex problems in real time has also led to demand in accurate techniques that help in reducing costs and increasing safety.

Modal analysis is the branch of structures that deals with the study of dynamic characteristics of a system in terms of its natural frequencies, damping, mode shapes, and modal scaling. Modal analysis finds extensive applications in present day engineering such as design, Finite Element (FE) model updating, structural health monitoring (SHM), etc.

1.1 Experimental and Operational Modal Analysis

Experimental modal analysis methods most often measure both output responses and input forces applied to the system to construct frequency response functions (FRFs), subsequently used for obtaining modal parameters [Allemang, 1999; Maia, Silva, 1997; Ewins, 2000]. This is the conventional approach and it has been well established over several years, forming the basis of EMA for most applications.

An alternative experimental approach has emerged over the last few years in the form of Operational Modal Analysis [Zhang et al., 2005], where modal parameters can be estimated purely on the basis of response data, eliminating the need for measurement of input forces in
certain scenarios. OMA techniques have been successfully implemented by researchers in civil structures [James et al., 1996; Peeters, Ventura, 2003; Chauhan et al., 2008], aerospace [Goursat et al., 2001; Goursat et al., 2010] and other industrial applications [Hermans et al., 1999].

The OMA method has gained significance in recent years as it has certain compelling advantages over the conventional approach. The operational technique is extremely suitable in applications such as modal analysis of large civil structures and bridges which are subjected to ambient vibrations [James et al., 1996]. These structures can be excited using artificial means such as drop hammers, but this will generally increase the cost involved in testing. It is also nearly impossible to excite all the modes of huge structures using such equipment. The use of ambient vibrations to excite the structure reduces the effort involved in test setup and instrumentation, while reducing the cost involved in excitation too.

The OMA method comes with its own share of issues. The reduced effort in test setup and instrumentation is somewhat negated by the increased amount of steps in data acquisition and processing. To begin with, relatively longer time histories are required while recording data. This is necessary to get accurate estimates of the output Auto and Cross power spectra [Chauhan, 2008]. There are also some special tools required for parameter estimation. For using conventional frequency based parameter estimation algorithms, the power spectra obtained needs to be processed in order to obtain positive power spectra. Employing these algorithms on output-only power spectra data have also been known to have certain issues such as overestimation of damping, etc. [Chauhan et al., 2008]. It is also to be noted that ambient conditions may not excite some modes, thereby having an incomplete modal model. On the other hand, lack of control in terms of excitation forces may also lead to excitation of modes that are not in focus, thereby complicating the parameter estimation process. Another important aspect of the OMA technique
that limits its usage in applications such as FE model updating is the unavailability of the forcing function which is required to estimate modal scaling. Additional steps are required to extract scaled mode shapes [Aenlee et al., 2005].

The last condition leads to the two major assumptions under which OMA works efficiently:

1. The nature of the input force is random, broadband and smooth. This implies that the input power spectra is relatively constant or smooth and has no poles or zeroes in the frequency range of interest.

2. The excitation is spatially distributed throughout the structure being tested. (That is, the number of inputs \( N_i \) approaches the number of outputs \( N_o \), where the response is being measured all over the structure).

### 1.2 Modal Analysis in Automotive Applications

Modal analysis is used in the automotive industry for FE model validation and updating in the design stage. The modal estimates are used to validate the FE models. Based on the results from the validation, the FE model is updated to satisfy the design requirements. EMA methods have been traditionally used to obtain the modal parameters. Excitations are induced using impact hammers or electrodynamic shakers. Transducers are used to measure both output responses from the structure and input forces from the shaker or the impact hammer. This method of modal testing has been well established over several years.

Attempts to utilize operational modal techniques for automotive applications have yielded satisfactory results for a few cases [Peeters et al., 2008]. However, some approaches suffer from shortcomings when attempting to use OMA in its original form to validate FE
models. The presence of subcomponents such as the suspension system, which have vital roles in the functioning of an automobile, actually render the application of OMA methods in its original form to be ineffective. Having said that, the use of operational modal analysis methods on automotive structures is still worth investigating, given the potential advantages of OMA techniques over EMA techniques.

1.3 Motivation and Problem Definition

While studies in the past have utilized EMA methods for modal analysis of automotive structures, the EMA tests have required measurement of both the response and the reference (input) signals. Measuring naturally occurring excitation forces such as road induced vibrations, wind excitations, etc., is not practically possible when the vehicle is running on the road or on a test track. On the other hand, the boundary conditions present for EMA tests performed in the laboratory are not reflective of the real world conditions in which vehicles operate, considering the fact that a vehicle has non-linear sub components such as the suspension system. Use of Operational Modal Analysis (OMA) methods which require only responses to be measured dramatically improve the ability to study the structure in real operational conditions.

The application of OMA techniques to automotive structures is however, quite different from other applications. The basic assumptions of broadband and spatially distributed excitations do not hold true in real operating conditions for a vehicle, as it does for a civil structure. Reasons include the presence of engine and other strong rotational harmonics, and the fact that road-induced operational forces are partially filtered out by the suspension system. It is also to be noted that these road-induced inputs can excite the system primarily through the four wheels.
only. This again is not a spatially well distributed excitation. Operational inputs are not broadband and the forced operating vectors cannot be easily separated from the modal vectors.

A response-only OMA test on the vehicle in a laboratory using excitation methods such as random impacts using hammers or shakers would serve as a logical first step in attempting to customize OMA methods for automotive applications. Due to its closer agreement with OMA assumptions, it would yield better results than testing the vehicle in more realistic operational conditions as on a test rig (Road Simulator) or on a test track. Keeping these views in mind, it must be noted that the work done with the OMA approach in this thesis is based upon response-only data, but not in truly operational conditions.

1.4 Research Objectives

The major goals of this thesis revolve around the experimental methods adopted for implementing and validating OMA techniques on an automotive structure. This work attempts to obtain modal parameters of a truck chassis based on specialized response data and to validate the modal parameters with results obtained from well-established EMA methods. This structure poses a few challenges in that it is moderately damped by the suspension system and is known to have closely spaced modes. Further, the presence of the suspension system is expected to involve non-linearities [Hermans et al., 1998]. Focus is kept on the rigid body modes of the suspension, such as pitching, yawing and rolling and the structural modes in the 0 - 30 Hz spectral range. Power spectra obtained by processing response time histories will be used as the basis for parameter estimation under the OMA framework [Chauhan et al., 2008] and will be validated with the FRF-based EMA methods.
The above mentioned aspects are summarized below as specific goals of the thesis:

1. Obtaining a standard set of modal parameters using well established EMA methods.
2. Application of OMA approach to automotive modal testing.
3. Validation of OMA results by comparing with baseline EMA estimates using standard validation tools.
4. Study of OMA results when one or more of the basic assumptions of OMA are violated.

1.5 Thesis Outline

Chapter 1 gives an introduction to both Experimental and Operational Modal Analysis and the role played by Modal Analysis in the automotive development stage. It states the motivation for the study undertaken and reiterates the research goals of the thesis.

Chapter 2 delves into the details of Modal Analysis, starting with the inception of the field of Operational Modal Analysis (OMA). It further discusses OMA algorithms, the mathematical framework and various processing techniques required for parameter estimation.

Chapter 3 introduces the structure under study. It describes the components of the structure and the instrumentation involved in the testing. Modal concepts involved in test setup and instrumentation are highlighted.

Chapter 4 explains each test performed in detail. Starting from data acquisition parameters up to MAC plots are listed for each test in separate sections.

Chapter 5 compares the OMA test results with the conventional EMA results. Comparison is also made between the shaker and impact hammer based tests in order to achieve various research goals of the thesis.
Chapter 6 summarizes the results obtained after comparisons, with detailed description of mode shapes. It further looks into the future areas of interest for this thesis work and recommends suitable research goals for furthering this line of work.
2 Literature Review

Operational modal analysis (OMA) started gaining significance from the 1990’s with its usage in civil applications such as off-shore platforms, buildings, bridges, etc. Also known as ambient, natural-excitation or output-only modal analysis, OMA utilizes only response measurements of the structures in operational condition subjected to natural excitation to obtain modal parameters of the system. The last 20 years have seen research focused on development of its workability on civil structures and also extending its scope to more applications such as industrial machinery, aerospace, automobiles, etc. Most of the algorithms and processing techniques for OMA have been developed from existing EMA based models. The common mathematical formulation of the Unified Matrix Polynomial Approach (UMPA) [Allemang et al., 1994] for EMA has also been modified to accommodate for usage on OMA based techniques [Chauhan et al., 2007].

2.1 OMA Algorithms

One of the first algorithms for OMA was the NExT (Natural Excitation Technique) [James et al., 1995]. This technique is based on the auto and cross-correlation functions calculated between the responses. The method then uses traditional EMA time based algorithms for parameter estimation. Some of the other popular algorithms are the Auto-Regressive Moving Average (ARMA) based Prediction Error Method (PEM) [Andersen, 1997] and Instrument Variable (IV) method [Peeters, De Roeck, 2001]; the Covariance-driven Stochastic Realization-based algorithms (SSI-COV) [Peeters, De Roeck, 1999]; the Data-driven Stochastic Realization-based algorithms (SSI-DATA) [Brincker, Andersen, 2006; Zhang et al., 2005; Peeters, De
Roeck, 2001]; Spatial Domain algorithms [Allemang, Brown, 2006]; Frequency Domain algorithms such as the Polyreference Least Square Complex Frequency algorithm (Polymax) [Peeters et al., 2005], etc. A detailed study of OMA algorithms can be found in the Ph.D. dissertation work by Chauhan, (2008).

2.2 Mathematical Framework for OMA

The mathematical framework for OMA can be developed from the basic Experimental Modal Analysis model. EMA can be expressed in terms of its input-output model. If \( \{X(\omega)\} \) is the measured output and \( \{F(\omega)\} \) is the input force, the relationship between them can be used to define the transfer function \([H(\omega)]\) as [Bendat, Piersol, 1986]:

\[
\{X(\omega)\} = [H(\omega)]\{F(\omega)\}
\]  

(2.1)

\([H(\omega)]\) is known as the frequency response function (FRF) and this equation is the basis of EMA in its most basic form. The FRF contains all necessary information from which modal parameters of a system can be extracted. This can be observed by expressing the frequency response functions in terms of modal parameters as

\[
H_{pq}(\omega) = \sum_{r=1}^{N} \frac{Q_{r}}{j\omega - \lambda_{r}} + \frac{Q_{r}^{*}}{j\omega - \lambda_{r}^{*}}
\]  

(2.2)

Eq. (1.2) shows the frequency response function \(H(\omega)\) for a particular input location \(q\) and output location \(p\) being expressed in terms of the modal parameters; mode shape \(\psi\), modal scaling factor \(Q\) and modal frequency \(\lambda\). This model is referred to as the partial fraction modal model. Modal
parameter estimation using EMA involves the extraction of these parameters from the measured FRF data.

Now Eq. (2.1) can be written as

$$\{X(\omega)\}^H = \{F(\omega)\}^H [H(\omega)]^H$$

(2.3)

In the OMA approach, there is no input force measurement made. Without measuring the input, FRF formulation as in the case of EMA cannot be done. Instead, power spectra of response measurements are used as the basis for parameter estimation. The EMA framework explained in Eqn. (2.1) can be used to derive the mathematical model for OMA as shown below:

Multiplying Eq. (2.1) and Eq. (2.3)

$$\{X(\omega)\} \{X(\omega)\}^H = [H(\omega)] \{F(\omega)\} \{F(\omega)\}^H [H(\omega)]^H$$

with averaging,

$$[G_{xx}(\omega)] = [H(\omega)][G_{FF}(\omega)][H(\omega)]^H$$

(2.4)

where $[G_{xx}(\omega)]$ is the output response power spectra matrix and $[G_{FF}(\omega)]$ is the input force power spectra matrix. Eq. (2.4) forms the basis of Operational Modal Analysis.

Under the basic OMA assumptions, $[G_{FF}(\omega)]$ is constant and hence $[G_{xx}(\omega)]$ can be expressed in terms of frequency response functions as

$$[G_{xx}(\omega)] \propto [H(\omega)][I][H(\omega)]^H$$

(2.5)

The partial fraction model of $GXX$ for a particular response location $p$ and reference location $q$ is given by
\[ G_{pq}(\omega) = \sum_{k=1}^{N} \frac{R_{pqk}}{j\omega - \lambda_k} + \frac{R_{pqk}^*}{j\omega^* - \lambda_k^*} + \frac{S_{pqk}}{j\omega - \lambda_k} + \frac{S_{pqk}^*}{j\omega^* - \lambda_k^*} \]  

(2.6)

Here, \( \lambda_k \) is the pole and \( R_{pqk} \) and \( S_{pqk} \) are the \( k^{th} \) mathematical residues. These residues are different from the residue obtained using a frequency response function based, partial fraction model since they do not contain the modal scaling factor.

### 2.3 OMA Processing Techniques

Both EMA and OMA work on essentially the same algorithms in the parameter estimation step. The fundamental difference lies in the type of raw data that is being used for estimation. While the EMA algorithms work on impulse response or frequency response functions, the OMA methods work on correlation functions or power spectra. Processing techniques are required to obtain power spectra from raw time history data [Chauhan et al., 2006]. The several techniques through which power spectra can be obtained are presented in the following sections.

#### 2.3.1 Welch’s Periodogram Method

The Welch’s Periodogram method [Stoica, Moses, 1997] begins with dividing output time histories into overlapping segments. A window function is then applied to each segment before computing its periodogram. The power spectra estimates are then averaged to obtain the estimated power spectra. Averaging reduces the variance of the estimates while the overlap allows for more averages. The bias errors are taken care of by the introduction of the windowing function. These concepts can be found in detail in textbooks on modal theory [Bendat, Piersol, 1986].

2.3.2 Correlogram Based Method

Another way of obtaining power spectra from time histories is the correlogram [Stoica, Moses, 1997] based approach. In this method, correlation functions are estimated from the output data segments and then Fourier transformed to get the power spectral density. Sometimes an exponential window is applied to the correlations before applying Fourier transform. This is done to reduce the bias errors, similar to application of exponential windows to impulse response functions. Another alternative to this approach is estimation of covariance [Stoica, Moses, 1997] which is essentially correlation with the mean removed.

2.3.3 Power spectra with Windowing, Overlap Processing & Cyclic Averaging

Obtaining power spectra by utilizing cyclic averaging [Allemang, Phillips, 1996] along with the overlap processing and windowing operations is a more traditional approach used in EMA methods. The primary advantage of cyclic averaging is the reduction of leakage errors.

The above mentioned data processing techniques have been observed to result in very similar spectral matrices and result in modal parameters that compare very well with each other [Chauhan et al., 2006].

2.4 Positive Power Spectra

The order of a power spectrum based model is twice that of a FRF based model, (from Equation 2.5). This makes the usage of frequency domain based algorithms more difficult as they inherently suffer from numerical conditioning problems [Phillips, Allemang, 2004]. With the time domain based algorithms, this does not pose a serious issue due to the numerical properties of the correlation function upon which they work. The correlation function is a symmetric
function with essentially the same information in both the decaying and growing exponential portions. This said, the decreasing exponential portion alone is sufficient for parameter estimation and the negative poles or the increasing exponential portion can be sieved off in the estimation process as illustrated in Figure 2.1

![Figure 2.1 Correlation Function for a measurement using OMA method](image)

This higher order model consisting of positive and negative poles forms the basis of the positive power spectrum [Chauhan et al., 2007] which is defined in the frequency domain by the following equation.

\[
G_{pq}^+(\omega) = \sum_{k=1}^{N} \left( \frac{R_{pqk}}{j\omega - \lambda_k} + \frac{R_{pqk}^*}{j\omega - \lambda_k^*} \right)
\]

(2.6)

In the positive power spectra method, the power spectrum is first inverse Fourier transformed to obtain the associated correlation function. Then the negative lag portion of the correlation function is removed. The resultant function is Fourier transformed back to obtain the positive power spectrum. The advantage of positive power spectrum is that it has the same order as the frequency response functions and also contains all the information necessary for parameter estimation.
estimation. This results in better numerical conditioning for frequency domain, partial estimation methods. It is to be noted that positive power spectra is not used in data processing in this thesis and the information above is only provided for a complete description of OMA processing techniques.
3 Test Structure, Instrumentation and Test Setup

3.1 Test Structure

The structure used for testing is a small truck chassis Figure 3.1 available at the Structural Dynamics Research Laboratory (SDRL), University of Cincinnati. The truck has a frame with the engine and gearbox mounted and is supported by independent double wishbone suspensions in the front and solid axle leaf springs at the rear. There is no cab in the truck. For the purpose of this thesis, the effect of tire dynamics is not explored, considering the tires to be linear within the scope of the excitation. The presence of sub components and the moderate level of damping of the structure make the implementation of OMA on this truck challenging.

Figure 3.1 Test Structure with Sensors Mounted
For choosing the response positions, the sub-components of the structure are studied, namely, the frame, the suspensions, the gearbox and the engine. In the double wishbone suspension system in the front, three points are selected each on the upper control arm (UCA) (Figure 3.2) and lower control arm (LCA), and one point near the kingpin (Figure 3.3), for each side of the suspension.

**Figure 3.2 Sensors on Upper Control Arm (UCA)**

**Figure 3.3 Sensors on Kingpin and Lower Control Arm (LCA)**

Figure 3.4 shows few other sub-components with some of the sensors visibly mounted on them. Four sensors are distributed along the leaf of the rear suspension system on either side. Eight points are chosen on the engine as response locations to better understand the nature of its
interaction with the frame and other components. The frame is extensively covered with eighteen sensors distributed evenly, including three points on the transaxle and two on the transmission. A total of fifty tri-axial accelerometers are distributed across the structure. Details of sensor distribution on all sub components are listed in Table 3.1, further in the chapter.

![Sensors on structure](image)

**Figure 3.4 Clockwise from left: Sensors on (a) Rear Leaf Springs, (b) Engine, (c) Transaxle, and (d) Transmission**

Points on the test structure are numbered using a nomenclature rule. The points on the chassis are given direct numbers from 1-18, which includes points 13, 14 and 15 on the transaxle. For the rest of the sub-components, the first letter of the part name is taken and depending on the order of appearance of the letter in the English alphabet, a specific series is chosen. For example, E being the 5\textsuperscript{th} letter of the alphabet is given the 500 series. Hence the
eight points on the engine are numbered from 501-508. Similarly, 1200 series is used for the leaf springs, 700 series is used for the gearbox and 400 series for the double wishbones.

<table>
<thead>
<tr>
<th>Part name</th>
<th>No. of points</th>
<th>No. of channels</th>
<th>Nomenclature</th>
<th>Point number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chassis</td>
<td>15</td>
<td>45</td>
<td>S</td>
<td>1-12, 16-18</td>
</tr>
<tr>
<td>Transaxle</td>
<td>3</td>
<td>9</td>
<td>T</td>
<td>13-15</td>
</tr>
<tr>
<td>Front Double Wishbones</td>
<td>14</td>
<td>42</td>
<td>F</td>
<td>401-414</td>
</tr>
<tr>
<td>Rear Leaf Springs</td>
<td>8</td>
<td>24</td>
<td>L</td>
<td>1201-1208</td>
</tr>
<tr>
<td>Gearbox</td>
<td>2</td>
<td>6</td>
<td>G</td>
<td>701-702</td>
</tr>
<tr>
<td>Engine</td>
<td>8</td>
<td>24</td>
<td>E</td>
<td>501-508</td>
</tr>
</tbody>
</table>

Table 3.1 Point and Channel information

The right hand rule is followed to set the global co-ordinates for the vehicle. When seen from the vehicle, the X axis runs in the lateral direction, with the positive x axis pointing from left to right. The Y axis runs longitudinally to the structure, with positive y being from rear to front. The positive Z axis is pointing up in the vertical direction. Local co-ordinates vary in accordance to the way each accelerometer is mounted on to the structure. The channel information and global direction is corrected at the time of calibration and data acquisition.

A complete geometry of the test structure is shown in Figure 3.5 along with point numbers and the global co-ordinate axes. The blue arrows indicate the positions of the two shakers mounted to the structure for the shaker excitation tests. The red circles show the points on the structure which are excited in the EMA based impact hammer test.
3.2 Sensors and Other Hardware

A data acquisition system with a capacity of 160 channels is set up for the tests. The main board consists of 16 channel digitizers. Due to hardware availability, some of the channels are routed through a dedicated signal conditioner while the rest are routed through ICP boxes (which do not need further signal conditioning). Complete hardware details including make, model number and specifications are listed in Table 3.2 below.
<table>
<thead>
<tr>
<th>Hardware</th>
<th>Make and Model</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Digitizer</td>
<td>Model E 1432 A</td>
<td>16 channels</td>
</tr>
<tr>
<td>ICP Boxes</td>
<td>PCB</td>
<td>6 Nos. 48 channels (Channel 113-160)</td>
</tr>
<tr>
<td>Signal Conditioners</td>
<td>PCB</td>
<td>112 channels (Channel 1-112)</td>
</tr>
<tr>
<td>Mainframe</td>
<td>VXI (HP) 75000 Series C</td>
<td>160 channels</td>
</tr>
</tbody>
</table>

**Table 3.2 Hardware Information**

### 3.3 Testing Conditions

The test frame stands on a concrete inertia mass at the SDRL, measuring 15’ wide, 25’ long and 12’ deep, throughout the tests performed. This ensures uniform boundary conditions across all tests. To verify the time invariance of the structure, an impact test is conducted at the very beginning and at the very end of testing. Data from these two tests are processed using EMA methods and are found to be consistent.

### 3.4 Modal Considerations

The choice of number of response locations determines the ability to study the modal behavior of the structure. There is always a trade-off between the spatial resolution of the response locations and the logistics involved with the test. While a large number of sensors increase the observability of the modes, the resulting large number of channels poses a challenge considering hardware availability and instrumentation. In terms of data processing, a large number of sensors lead to an over-determined model. Methods like Singular Value
Decomposition (SVD) and Eigen-Value Decomposition (EVD) are used to compress the over-determined model to a reasonable size to optimize valuable computing time and effort in real situations. Data storage also becomes an issue with large file sizes. For these reasons, it is prudent to choose the right sensor locations required to completely define the modal model of the structure, and a reasonable frequency range and resolution.

A background study on the structure usually helps in the choice of sensor locations. Previous studies on the same structure have been useful in determining response locations, reference points along the frame, and selection of frequency range of interest. This particular structure is difficult to study due to presence of various sub-components and also close modes. Sensor locations are chosen such that most of the sub-components are observable. This is essential for observing the phase difference between various components of the structure, especially in the case of close modes. For example, the frame might have a torsion mode at two different frequencies, but with the engine rocking in longitudinal direction in one of the modes and in the lateral direction in the other. Without enough response locations on the engine, both the modes would appear the same, even though their modal frequencies might be different. This would be observed in the MAC [Allemang 1980; Heylen et al., 1995] plots too, where the lack of spatial resolution would result in both modes having a high MAC value.
4 Data Acquisition and Modal Parameter Estimation

Data acquisition is probably the most important step from a modal perspective, in the experimental study of dynamics of any structure. It involves a thorough understanding and implementation of several concepts, discussed in Chapter 2, that affect the test data collected and consequently, the parameter estimates. Data acquisition is unique for each structure type and is a function of these concepts, thereby influencing the quality of data.

Modal parameters are often used to validate FE models in product design and development. A similar approach could be taken for this case too, where results from OMA tests can be validated against a FE model of the truck. But with the goal of the thesis being applicability of OMA on automotive structures rather than updating FE models, it is more appropriate to validate it against a well established experimental technique. Keeping this in mind, four tests are performed on the structure so as to achieve the goals of the thesis. The tests are listed in order below.

1. Conventional FRF-based EMA tests
   a. EMA based test using shaker excitations.
   b. EMA based test using impact hammer excitations.

2. Power-spectra-based (output-only) OMA tests
   a. OMA based on response time histories from shaker excitations.
   b. OMA based on response time histories from random impact excitations.

Data acquisition parameters and further estimation procedures for the above mentioned tests are discussed in detail in this chapter.
4.1 Conventional FRF-Based EMA Tests

EMA has been successfully used in the past for studying the modal behavior of automotive structures. In this thesis work, two conventional EMA tests are conducted initially to obtain modal parameters of the structure. The results from these tests are used as a baseline for comparison and validation of results from the OMA tests. Data acquisition for the EMA tests is done using the X-Acquisition and MRIT softwares available at the SDRL, University of Cincinnati. While the responses remain the same for all tests, the references change according to nature of excitations. The two tests done using the EMA methodology are described below.

4.1.1 Shaker Test

Electrodynamic shakers are ideal for exciting automotive structures. Shakers are preferred for their ability to impart consistent excitations. They are versatile in terms of the several types of input excitation signals that can be used. The signal can be chosen according to the nature of the structure and its physical properties such as damping, etc. On the down side, shakers are expensive and sometimes difficult to handle. They require careful setup in order to impart the desired levels of excitation to the system under study, and also to protect the shaker coils from permanent damage.

Some of the factors to consider in shaker testing are the mounting locations of the shaker to the structure and the type of input signal and other signal processing techniques related to it. One testing philosophy suggests that shakers should be mounted at such locations that would excite the maximum number of modes in a single test configuration [Allemang, 1999]. If a shaker is mounted to the structure at the node of a mode, it will not excite that mode. Further, due to the size and weight of shakers and the way it is mounted, it is not easy to reposition or
move around the shakers once they are fixed to the structure. After being set, excitations are also limited to primarily that direction alone. Hence the choice of position of shakers becomes all the more important.

For exciting modes in different directions, two approaches can be used. In one case, multiple shakers can be used in more than one direction to excite all the modes. A typical configuration would consist of two vertical shakers and one in the horizontal direction. A similar result can be obtained by using a combination of horizontal or vertical shakers along with a shaker set up at a skewed angle to the test structure [Allemang, 1999].

The second important factor to consider in shaker excitation based tests is the type of input signal. The choice of input signal is a function of the nature of the structure, damping characteristics of the structure, the frequency range of interest, observability of the transient, etc. Each type of signal has its inherent advantages and shortcomings. It is important to choose the right input signal in order to obtain good data. The various types of random input signals [Allemang, 1999] are:

1. Pure random
2. Pseudo random
3. Periodic random
4. Burst random
5. Slow random
6. Hybrid random signals
   a. Burst pseudo random
   b. Burst periodic random
For the purpose of this thesis, two shakers are used in the vertical direction at point numbers 2 and 12 as shown in Figure 4.1. One shaker is placed at the front end of the truck and the other at the rear end. Both shakers are mounted on the left side overhang of the frame, enabling better excitation of the modes owing to the asymmetry. The overhang also reduces the chances of exciting at the node of a mode. Random forces are used as excitation functions and responses are measured at 150 locations distributed over the structure (refer to Figure 3.5).

The data acquisition parameters for this test have been summarized below.

- Sampling Frequency : 125 Hz
- Frequency Resolution : 0.0625 Hz
- 20 RMS averages with 4 cyclic averages for each RMS average
- Window : Hanning
- Excitation degrees of freedom: 2
- Response degrees of freedom: 150
The FRF data so obtained is used as the basis for parameter estimation. The Polyreference Time-domain (PTD) algorithm [Vold et al., 1982; Allemang et al., 1994] is used to estimate the modal parameters. Being a higher order algorithm, it uses more temporal information than spatial information. It is also better suited to handle systems that have a large number of response channels compared to the references [Chauhan et al., 2007]. Due to the above reasons and for maintaining consistency, all parameter estimation is done using the PTD algorithm throughout the thesis.

A consistency or stabilization diagram [Allemang, 1999; Maia, Silva, 1997] for one of the estimates using this algorithm is shown in Figure 4.2. The blue diamonds in the diagram represent both poles and vector consistency, indicating physical modes that stabilize over increasing model order. Often there are just poles or frequencies estimated as shown by other shapes described in the figure, which do not stabilize to estimate the vector. These are mostly the computational modes generated due to numerical characteristics of the algorithm and the noise on the data. Only the stabilized modes are chosen for additional processing. This way, the computational modes are removed from the estimation process. Further, the size of the blue diamond represents the Modal Phase Colinearity (MPC), an indicator of the consistency of linear relationship between real and imaginary parts of each modal coefficient, or in other words, the measure of normal mode characteristics. When the MPC is low, the size of the blue diamond is smaller, indicating a complex mode. For a normal mode, MPC should be 1.0 (100 percent).

For the EMA shaker test data, it can be observed that the system modes consistently show up over increasing model order.
The Modal Assurance Criterion (MAC) plot is a validation tool for establishing linear independence of mode shapes. It can be used to identify multiple estimates of the same mode which may be due to an observability problem. A MAC plot for the estimates from the EMA shaker test is shown in Figure 4.3. The independent modes can be observed by the presence of unity coefficients along the diagonal (shown in red), and their absence off the diagonal (shown in blue). A total of 19 modes are estimated based on this test, which are summarized later in Error! Reference source not found. in Chapter 6.
The second procedure in the series of EMA tests involves impact hammer excitations. A roving hammer type approach is used for this test, with all response locations fixed. In this method, the hammer is moved from one reference point to another, exciting the system at a particular location for each measurement. The other approach is the roving sensor method, which is not suitable for a large number of sensors.

A medium size hammer with a semi–hard rubber tip is used for testing the truck frame. Choice of hammer size and tip depend on physical properties of the structure such as stiffness and damping and also the frequency range of interest. It should be able to impart sufficient energy to the structure to excite the maximum number of modes in that range of frequencies.
the higher frequency modes very well. On the other hand, very hard tips impart energy for high frequency modes, but fail to excite the low frequency modes. The usage of a semi-hard rubber tip in this case sufficiently excites most of the modes in the frequency range of interest, with the exception of the very low frequency modes below 5 Hz.

Seven points in the truck frame are chosen as reference locations (refer Figure 3.5). The reference points are chosen such that all sub components of the truck are well excited. These include the engine, suspension system, gearbox, etc. Due to the complex nature of the structure, it is not feasible to excite in all the three directions at every reference point. Hence, at each of these locations, the structure is excited in at least two directions. It is a combination of an X and Z direction, or Y and Z direction. The directions at each reference point are chosen so as to excite the vertical and lateral modes of that part of the structure. For example, a lateral beam of the chassis would have majority of deflections in the Y and Z directions, and not in the X direction. Similarly, a longitudinal member would deflect more in the X and Z directions, and relatively less in the Y direction. A total of fourteen measurements are made, impacting in two directions at each of the seven reference points.

The data acquisition parameters for this test are listed below.

- Sampling Frequency : 125 Hz
- Frequency Resolution : 0.125 Hz
- RMS averages : 3
- Excitation degrees of freedom: 14
- Response degrees of freedom: 150
With the system being moderately damped, response vibrations damp out well within the chosen time period of 8 seconds, which explains a relatively coarser frequency resolution of 0.125 Hz. For the same reason, the use of an exponential window is not needed.

A sample consistency diagram for an estimate using the PTD algorithm is shown in Figure 4.4. The consistency in the estimation of the modes over increasing model orders is shown by the blue diamonds. As in the previous case, the other poles and frequencies are left out and only the stabilized vectors are picked for further parameter estimation.

![Consistency Diagram](image)

**Figure 4.4 Consistency Diagram for an Estimate for EMA Impact Test**

The MAC plot shown in Figure 4.5 again highlights the linearly unrelated mode shape vectors of the modes. A total of 17 modes are estimated from this test. The modal estimates are summarized later in [Error! Reference source not found.](#) in Chapter 6.
4.2 Power-Spectra-Based (Output-Only) OMA Tests

The next two tests conducted in this thesis involve OMA methods which do not measure the input forces going into the system. Instead, simulated operational conditions are attempted in laboratory. Both shakers and impact hammers are used as excitation sources, in order to vary the level of adherence of the tests to OMA assumptions. The results from these tests are compared with respective baseline EMA estimates for validation.

Since these tests do not measure FRFs but power spectra instead, the data acquisition procedure starts with recording raw time histories using the VTI Instruments DAC Express software. The time histories are processed to obtain power spectra using the Welch Periodogram method [Stoica, Moses, 1997]. Power spectra have different numerical characteristics compared to the conventional Frequency Response Functions. As explained in Chapter 2, the order of the
power spectrum model is twice that of the FRF based model and the data contains both positive and negative poles [Chauhan, 2007]. The presence of negative poles can be explained by the correlation function, which is the time domain equivalent of power spectrum. The positive poles give rise to the decaying exponential portion of the correlation function and the negative poles are represented by the growing exponential portion. Only the positive decaying half of the correlation function is selected for further processing, which is sufficient to estimate the required modal parameters.

4.2.1 OMA Based on Response Time Histories from Shaker Excitations

In this test, two shakers are employed at the same locations as used for the EMA test for exciting the structure (refer Figure 3.5). The purpose of this test is to study the nature of estimates knowing the excitations to be uncorrelated and random but with limitations on the spatial distribution and direction of inputs. Response time histories are collected over 150 channels, and processed to obtain power spectra data for OMA. The following data acquisition and processing parameters are used.

- Sampling Frequency : 160 Hz
- Duration of data acquisition : 20 minutes (191488 time points)
- Number of excitation locations : 2
- Cyclic Averaging over 3 ensembles with 66.6% overlap processing employed for noise reduction
- Hanning window employed for reduction of leakage errors

A sampling frequency deviant from the earlier tests is used since a different software package is used to record time-histories, with 160 Hz being the nearest sampling frequency that could
have been chosen under the requirements of this study. Given the constraints on computing capabilities, only the first 102400 time points are used in obtaining the power spectra.

The PTD algorithm is again employed to estimate the modal parameters for the structure, with the algorithm using power spectra information instead of frequency response functions as the basis for parameter estimation [Chauhan, 2007]. The references are chosen by observing their spectral content from the auto power spectra plots of each channel and the nature of the associated correlation of each channel time history with the other channels. Parameters are estimated from different combinations of reference channels over narrow frequency bands covering the entire frequency range of interest.

The recurring presence of modes for varying model orders can be observed from a sample consistency diagram shown in Figure 4.6.

![Figure 4.6 Consistency Diagram for an OMA Estimate Based on Shaker Excitations](image)
The modal frequencies obtained for this test are shown alongside the EMA shaker test estimates in Error! Reference source not found.. From the MAC plot for this set of estimates shown in Figure 4.7, modes at 14 and 18.7 Hz might seem to indicate partial linear dependence. But visual inspection of the corresponding mode shapes and the fact that they are well-separated on the frequency scale confirm them to be distinct modes.

![MAC Plot for OMA Estimates Based on Shaker Excitations](image)

**Figure 4.7 MAC Plot for OMA Estimates Based on Shaker Excitations**

### 4.2.2 OMA Based on Response Time Histories from Random Impact Excitations

This test is conducted to study the nature of estimates knowing the excitation to be spatially well-distributed and assumed to be random and broadband in the absence of force measurements. Multiple hammers are employed to excite the structure with random impact excitations covering most parts of the structure in all directions. The data acquisition and processing parameters are similar to those described in the previous section.
• Sampling Frequency: 160 Hz

• Number of excitation locations: Multiple locations uniformly spread across the structure.

• Duration of data acquisition: 20 minutes (191488 time points)

• Cyclic Averaging over 3 ensembles with 66.6% overlap processing employed for noise reduction

• Hanning window employed for reduction of leakage errors

A sample consistency diagram for estimates from this test using the PTD algorithm has been shown in Figure 4.8. The presence of modes over varying model orders highlights the consistency of the modes estimates.

![Consistency Diagram](image)

**Figure 4.8 Consistency Diagram from PTD Estimates for OMA Random Impact Excitations**
The modal estimates for this test are again listed in Error! Reference source not found. in Chapter 6. Figure 4.9 shows the MAC plot for this set of estimates. Modes 13.5 Hz and 13.8 Hz seem to show a certain amount of similarity. A study of the mode shapes also indicates a high level of similarity. These modes around 13-14 Hz are predominantly engine modes, and might not have been excited well with the random impacts. Modes at 22.7 Hz and 24.58 Hz however are seen to be distinct physical modes in spite of a possible indication of linear dependence by the MAC plot. The rest of the modes appear to be linearly unrelated.

Figure 4.9 MAC Plot for OMA Estimates Based on Random Impact Excitations
5 Comparison Between Estimates – Modal Validation

When data is measured and processed using several numerical techniques and estimation algorithms, almost every time, computational modes are generated that affect the quality and consistency of estimates. Differentiating between system modes and computational modes becomes difficult, especially when there are a large number of channels and also in the case of close modes. Modal validation forms an essential part of any modal parameter estimation procedure.

The Modal Assurance Criterion [Allemang, 1980] is a useful validation tool commonly used in identifying system modes from those generated due to the numerical characteristic of the algorithm. The MAC coefficient is calculated based on the linearity between mode shapes obtained from estimates. AutoMAC establishes the linear independence of modes, and is useful in identifying repeated estimates of the same mode at a particular frequency. CrossMAC coefficient is calculated by comparing mode shapes between two different set of estimates. By employing this, a new estimate can be validated against a more established set of results.

Though the MAC coefficient is simple to calculate and is effective in identifying system modes, it is not a complete validation tool. It depends heavily on the observability of modes. In cases where there are not enough sensors on a test structure, this makes it a misleading tool in the hands of an inexperienced user. Two modes might appear to have the same mode shape, even though they are at different frequencies. The MAC number calculated between them would be high, implying that the modes are the same. But it is only a limitation in usage of MAC, due to lack of observability of those modes.
Another method for validation used in this thesis is through visual inspection of the mode shapes. Where MAC coefficients cannot observe differences between mode shapes of different modes, the visual inspection technique helps. The displacement of sub-components in-phase and out-of-phase with each other can be better exhibited using this. It is however a subjective technique and requires keen judgment on the part of the user.

The above mentioned methods are used in validating results of the OMA tests with estimates from the EMA tests. CrossMAC coefficients between respective impact tests and shaker tests from both OMA and EMA are plotted and discussed in detail in this chapter. The crossMAC plots between impact test and shaker test within each methodology are also shown below, evaluating how the excitation technique influences the results, and also to ascertain the effect of violation of OMA assumptions. Visual inspection is also done extensively, and the summarized mode shape animations are discussed in the next chapter.

5.1 Cross-MAC Plot Between Two EMA Tests

Figure 5.1 shows the cross-MAC plot between the estimates from the two EMA-based tests. The low-frequency 3.8 Hz mode seen in the shaker test is not estimated in the impact test. Impact tests have been known to have limitations with exciting very low frequency modes, which would explain the absence of the 3.8 Hz mode. The 20.66 Hz, 23.7 Hz and 24.58 Hz modes are predominantly in the lateral direction. Since the shakers are mounted to the truck frame in the vertical direction alone, these modes might not have been excited properly. While these modes show up well in the EMA impact test and, as will be shown later, in the OMA test based on random-impact excitations, they are estimated poorly in the tests involving shaker excitations. The 28 Hz mode is a torsion mode that has been fairly difficult to excite using...
hammer impacts. Hence this mode does not figure in the EMA impact test estimates. The rest of the rigid-body and structural modes have been observed to be consistently estimated in both the EMA tests.

![Figure 5.1 EMA Shaker Test Estimates vs. EMA Impact Test Estimates](image)

**5.2 Cross-MAC Plot Between EMA & OMA Tests with Shaker Excitations**

Estimates from the EMA and OMA tests with shaker excitations have been compared in the cross-MAC plot in Figure 5.2. The low-frequency modes around 3-4 Hz and the higher order modes at 17 Hz and between 20-25 Hz, being lateral modes, are poorly estimated due to the violation of the OMA requirement of spatially well-distributed excitations in all directions. The closely lying modes around 14 Hz have not been estimated very distinctly in the OMA methods. Other prominent rigid-body and structural modes are estimated well across both tests.
Figure 5.2 EMA Shaker Test Estimates vs. OMA Shaker-Excitation Estimates

5.3 Cross-MAC Plot Between EMA & OMA Tests with Impact Excitations

Figure 5.3 compares the modal estimates from the OMA test based on random impact excitations with the EMA impact test. It can be readily seen that more modes match well with each other in the impact-excitation based tests since the random-impact excitations follow the OMA requirements more closely than the tests with shaker excitations discussed earlier. As in the previous case, the two modes at around 14 Hz are not well estimated. The mode at 17.4 Hz is a lateral sway mode lying close to a very dominant 18.9 Hz mode and does not lend itself very well to estimation in the power spectra-based estimation methods. The high-order complex torsion mode at 26.4 Hz might not have been well-excited with the random impacts and hence does not show up in the OMA estimate. Most of the other modes from the OMA-based estimates match up with corresponding modes from the EMA test with high modal consistency.
Comparison of estimates from the two response-only tests reflects the fact that one of these methods does not fully conform to the OMA assumption of uniform spatial distribution across the structure, as shown in Figure 5.4. A series of low cross-MAC coefficients for modes at 3.8 Hz, 20.03 Hz, 23.4 Hz and 24.7 Hz can be ascribed to this violation of OMA assumption. Again, barring the close modes around 14 Hz, majority of modes that appear in both estimates compare well with each other.
Figure 5.4 Cross-MAC Between OMA (Shaker Excitations) & OMA (Random Impact Excitations)

From the above discussions, it can be concluded that most of the modes show a high degree of similarity and consistency across the EMA and OMA estimates both in terms of the MAC coefficients and in terms of the nature of the physical mode shapes. Estimates from the OMA based shaker test compare fairly well with the EMA shaker test, though the results obtained are relatively better when OMA assumptions are met more closely, as in the case of the impact based tests.
6 Summary of Results, Conclusions & Scope for Future Work

From the previous chapter, it is clear that the estimates are fairly consistent over the entire range of tests performed. To summarize, the EMA shaker test results are listed down below, as it is reflective of the entire set of estimates.

6.1 Summary of Results

The rigid-body modes start in the 4 Hz range and go up to 10 Hz. The modes at frequencies 4.9 Hz and 6.7 Hz are observed to be the rigid-body pitching modes. In the 4.9 Hz mode shown in Figure 6.1, pitching is observed predominantly at the front. The 6.7 Hz mode has a similar mode, but with pitching observed at the rear (Figure 6.2).

![Figure 6.1 Pitching Mode (front) at 4.9 Hz](image)
Figure 6.2 Pitching Mode (rear) at 6.7 Hz

Rigid-body yawing and rolling modes are observed to lie at 5.7 Hz and 10.0 Hz respectively.

Figure 6.3 Yaw Mode at 5.7 Hz
Figure 6.4 Rolling Mode at 10.0 Hz

Figure 6.5 Transaxle Bending Mode at 10.5 Hz
The first torsion mode appears at 11.7 Hz and the first frame bending mode appears at 18.9 Hz.

Figure 6.6 First Torsion Mode at 11.7 Hz

Figure 6.7 First Frame Bending Mode at 18.9 Hz
The EMA and OMA test results are listed separately. Table 6.1 lists the results from the two EMA tests. The mean frequencies and standard deviations from the EMA tests have been computed and tabulated in the table. It can be observed that except for two modes, the standard deviation between the two EMA methods is very low, indicating good consistency of estimates across the tests.

Figure 6.8 Lateral Bending Mode at 30.9 Hz
<table>
<thead>
<tr>
<th>EMA Shaker - PTD</th>
<th>EMA Impact - PTD</th>
<th>Average Frequency – EMA (Hz)</th>
<th>Std. Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Freq (Hz)</td>
<td>Damp (%)</td>
<td>Freq (Hz)</td>
<td>Damp (%)</td>
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<td>-</td>
</tr>
<tr>
<td>3.93</td>
<td>2.18</td>
<td>3.89</td>
<td>1.28</td>
</tr>
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<td>30.94</td>
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</table>

Table 6.1 Modal Estimates from the two EMA tests

Similarly, Table 6.2 lists the results from the two OMA tests. The average frequencies and standard deviation values are again computed. It is to be noted that out of the two OMA tests, the shaker based test is known to give poor results due to the violation of OMA assumptions. Hence the standard deviation is bound to be high for this comparison. In addition to that, as has been the general case with OMA, damping ratios are overestimated for a few modes.
<table>
<thead>
<tr>
<th>OMA Shaker -PTD</th>
<th>OMA Impact – PTD</th>
<th>Average Frequency - OMA (Hz)</th>
<th>Std. Deviation (%)</th>
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Table 6.2 Modal Estimates from the two OMA tests

Table 6.3 is the comparison of average frequencies between the OMA and EMA methodologies. As can be observed, the standard deviation is relatively high for most of the modes. This represents the fact that there are some deviations in frequency values between the EMA and OMA estimates, though the mode shapes are comparable.
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<th>Average Frequency - OMA (Hz)</th>
<th>Average Frequency - EMA vs. OMA</th>
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Table 6.3 Comparison of Average values between OMA and EMA results

6.2 Conclusions

The potential for usage of OMA methods for modal analysis on a truck chassis have been demonstrated and discussed in detail. Several excitation methods were employed with different levels of adherence to the OMA assumptions of uncorrelated randomness with temporal consistency and spatial coverage across the structure. The results so obtained were validated against conventional EMA-based methods. While the results are reasonable, presence of
localized inputs, as in the OMA shaker test, hinders the estimation of certain modes. Likewise, limitations in spatial resolution affect the quality of the estimated modal vectors as measured by the cross-MAC values between some vector estimates. Complexities in system geometry and issues with suspension non-linearities are other reasons why some modes were not excited or estimated well. Excitation methods for OMA of this structure have been compared and limitations for each method have been discussed. The shaker approach, with a limited spatial distribution of inputs, does not match the input assumptions of OMA as precisely as the randomized impact excitation. Average frequency comparisons also indicate the similarity between EMA baseline estimates, while highlighting the differences between the OMA results and also their comparison with the baseline estimates.

It is significant to understand that a lot of effort has been involved in extracting modal parameters from the OMA tests and to match them with the EMA estimates in order to get the above comparisons. This eventually leads to the conclusion that as long as the inputs can be reasonably measured, OMA is not a good alternative to EMA in the testing of the automotive frame. Only in situations where the measurement of input excitations is extremely difficult or impractical would OMA serve as a viable method to estimate the modal frequencies and mode shapes of the system.

6.3 Scope for Future Work

As a continuation of this research work, the applicability of using a 4-post road simulator for exciting the structure similar to operating conditions has been studied and published in the form of a conference paper [Sharma et al., 2009]. This forms the basis for the thesis work of a colleague at SDRL, University of Cincinnati. This is operational modal analysis in a much closer
sense in that the vehicle's on-road behavior is simulated in the laboratory. However, the spatial distribution of inputs is still limited and frequency content is broadband unlike the harmonic nature of engine excitations and other inputs when operating at a limited engine speed and vehicle velocity. The purpose of the second study is the incremental change from this thesis work, to observe the effectiveness of the excitation method with inputs coming only from the four wheels, and with the suspension acting as a mechanical filter.

Also, further work needs to be completed in developing a selection process to identify the most suitable reference channels of response that are utilized in OMA-based estimates, especially for studies involving a large number of responses. If some sort of model is available, the optimal response sensor locations can be selected using singular value decomposition of the modal matrix, as is done in current EMA test methods. If no \textit{a priori} model exists, a purely experimental method based upon information theory and an initial test using all response sensors will be considered to optimize the parameter estimation process based on power-spectra information directly.
References


