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A New Multiple Input Random Excitation Technique Utilizing Pneumatic Cylinders

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A New Multiple Input Random Excitation Technique Utilizing Pneumatic Cylinders

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Abstract

As an alternative to traditional shaker and impact hammer excitation, a new excitation technique using single acting pneumatic cylinders is investigated. In this technique, the structure is excited at multiple locations by successive impacts from the piston of pneumatic cylinders on which different types of tip can be mounted. The force transducer is attached directly to the structure or can be mounted on the piston of the pneumatic cylinder. The configuration in which the force transducer is attached to the structure gives better results than the other configuration. Solenoid valves regulate air flow through the cylinders, thus controlling the timing and duration of impacts. An electrical circuit consisting of a transistor switch is used to actuate the solenoid valves. A system of independent, random duration and interval pulses (digital time series) is sent to each exciter system to be able to estimate multiple input (MI) frequency response functions (FRFs). In this way, the excitation signal is more like a random input than an impact input (where only one impact per measurement ensemble is acquired). Since, the random pulse sequences sent to each exciter system are uncorrelated, and the exciters and structure are not connected, it will be always possible to uncouple the forces involved. This setup can also be used for single input impact testing as there will be better repeatability in the location, direction and magnitude of impacts. In order to validate this method, tests are conducted on a rectangular steel plate and trimmed truck body and the resulting FRFs and modal parameters are compared with those obtained from traditional excitation methods.
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1 Introduction

1.1 Motivation

Frequency response function (FRF) estimation is the most important step towards finding the modal parameters of a structure since the accuracy of the computed modal parameters relies heavily on the quality of FRF data used as an input to different modal parameter estimation (MPE) methods. The traditional techniques employed for exciting the structure for FRF estimation are shaker and impact testing in which electromechanical shakers and impact hammers are used for excitation, respectively. The choice of excitation technique for a particular test depends on the type of structure, time available for testing, equipment availability etc. Shaker testing is generally used in a multiple input multiple output (MIMO) excitation case. Multiple inputs ensure uniform excitation of the structure and reduce the effect of nonlinearities present in the system; but the mechanical attachment of shakers to the structure gives rise to other problems [1]. Due to the input power spectra matrix inversion involved in the MIMO case for estimating FRFs, it is necessary that the inputs used to excite the structure should not be perfectly correlated at any frequency. But, as a result of the electrodynamic interaction between the shakers and structure, input forces may become partially coherent, especially at resonance frequencies. As long as the forces do not become fully coherent, this generally does not cause a numerical problem. Additionally, the electrodynamic interaction also reduces the force transmitted by the shaker to the structure causing the force drop-off phenomenon [2, 4]. Force transmissibility between the shaker and structure plays an important role in controlling signal to noise ratio [5]. Although, the impact excitation method by virtue of a single roving input and absence of mechanical attachment does not have these problems, the repeatability of the location, magnitude and direction of input force is susceptible to human error [6]. These issues give motivation to develop a new excitation technique, which has only the positive characteristics of both shaker and impact testing.
1.2 Thesis Outline

The thesis is divided into six chapters:

Chapter One discusses the motivation for developing a new excitation technique for FRF estimation and gives an outline of the work presented in the rest of the sections of the thesis.

Chapter Two provides a brief overview of the existing excitation techniques: shaker and impact testing. Some of the positive and negative issues associated with these methods are also mentioned in this chapter.

Chapter Three introduces the pneumatic excitation setup and the principle behind this technique. The schematic, working and the different configurations of the setup are described in detail. A stepwise process of generating a pneumatic excitation signal is also presented in this chapter.

Chapter Four is focused on the study of the effects of various parameters on the measured excitation and FRFs. The parameters, which are discussed in this chapter, include pulse width and time delay of the excitation signal, air pressure and exciter tip. The study is done on both configurations and comparisons are made.

Chapter Five discusses the test results obtained from the pneumatic testing of a rectangular steel plate and trimmed truck body. These structures are also tested with traditional excitation methods; the FRFs and modal parameters are compared with the new technique.

Chapter Six concludes the thesis by deliberating the issues and areas of improvement associated with pneumatic testing.

After Chapter Six, there is a list of reference material used while completing this thesis. Appendices with the MATLAB code, list of pneumatic setup components and demonstrative videos follow the references.
2 Theoretical Background

2.1 Frequency Response Function Estimation

Frequency response function is estimated for studying the dynamic characteristics of a system by describing its input-output relationships in the frequency domain. The system is assumed to be linear and time-invariant, though it may not always be correct. For a mechanical system, FRF is formulated as the ratio of displacement over force. The acceleration or velocity of the system (structure) can also be used in place of displacement as they can be synthetically integrated to give the equivalent displacement over force ratio. Theoretically, the displacement and force at any frequency are related as shown in equation (2-1).

\[ X_p = H_{pq} F_q \]  

(2-1)

where \(X_p\) is the displacement, \(H_{pq}\) is the frequency response function and \(F_q\) is the force applied to the structure at that particular frequency. However, the above equation does not account for the noise (errors) present in the measurement due to various sources. Some of the sources of noise are the digital signal processing errors like quantization, leakage and aliasing error; the error due to cable motion and other instrumentation related errors; incorrect calibration and errors due to non-linearities present in the structure. This requires a method to calculate the best estimate of FRF in the presence of noise. The most common and reasonable approach for it is the use of least squares (LS) or total least squares (TLS) techniques. The main aim of these techniques is to find a function such that the sum of the squared errors (difference between assumed and actual value) is minimized. The algorithms (based on these techniques) used for FRF estimation are summarized in Table 2-1. The main difference between the algorithms is in the assumption of where the noise enters the measurements. In the absence of noise, all the algorithms will give the same answer. In the following sections, only \(H_1\) technique is used for representation and FRF estimation.
Table 2-1: Summary of FRF Estimation Models

<table>
<thead>
<tr>
<th>FRF Estimation Models</th>
<th>Technique</th>
<th>Solution Method</th>
<th>Assumed Location of Noise</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Force Inputs</td>
<td>Response</td>
<td></td>
</tr>
<tr>
<td>( H_1 )</td>
<td>LS</td>
<td>noise</td>
<td>noise</td>
</tr>
<tr>
<td>( H_2 )</td>
<td>LS</td>
<td>noise</td>
<td>no noise</td>
</tr>
<tr>
<td>( H_v )</td>
<td>TLS</td>
<td>noise</td>
<td>noise</td>
</tr>
</tbody>
</table>

2.1.1 Single Input FRF Estimation

In this case, the force is applied to the structure at only one degree of freedom (DOF), while the response can be taken at a single or multiple DOFs. Assuming that the noise (\( \eta_p \)) enters the measurements through the output, the response at any DOF on the structure is related to the input force as shown in equation (2-2).

\[
X_p - \eta_p = H_{pq} F_q
\]  

(2-2)

The sum of squared errors in the response values as the same measurement is taken multiple times is depicted by \( S \) in equation (2-3) where the digits in the superscripts represent the number of averages.

\[
S = (H_{pq}F_q^n - X_p^n)^2 + \ldots \ldots (H_{pq}F_q^{n_{avg}} - X_p^{n_{avg}})^2
\]  

(2-3)

For \( S \) to be minimum, its derivative with respect to \( H_{pq} \) should be zero. After setting the derivative of \( S \) to zero and rearranging the terms, the frequency response function can be calculated as shown in equation (2-4) where \( G_{XFpq} \) is the cross power spectrum between the response and force and \( G_{FFqq} \) is the force auto power spectrum.

\[
H_{pq} = \frac{\sum_{1}^{n_{avg}} X_p F_q^{*}}{\sum_{1}^{n_{avg}} F_q F_q^{*}} = \frac{G_{XFpq}}{G_{FFqq}}
\]  

(2-4)

2.1.2 Multiple Input FRF Estimation

In this case, forces are applied to the structure at multiple DOFs and response may be taken at a single or multiple DOFs. The response taken at each DOF is generated by multiple input forces though their
individual contribution to the overall response generally differ. The procedure for multiple input FRF estimation consists of similar steps that are used for single input FRF estimation. However, there is difference in the mathematical formulation as matrices are used for ease of calculation. The issues related to the mathematical inverse of the input auto power matrix depicted in equation (2-5) is described in detail in section 2.2.1.

\[
\begin{bmatrix}
H_{11} & \cdots & H_{1N_i} \\
\vdots & \ddots & \vdots \\
H_{N_o1} & \cdots & H_{N_o N_1 N_o \times N_i}
\end{bmatrix}
= \begin{bmatrix}
G_{XF_{11}} & \cdots & G_{XF_{1N_i}} \\
\vdots & \ddots & \vdots \\
G_{XF_{N_o1}} & \cdots & G_{XF_{N_o N_1 N_o \times N_i}}
\end{bmatrix}
\begin{bmatrix}
G_{FF_{11}} & \cdots & G_{FF_{1N_i}} \\
\vdots & \ddots & \vdots \\
G_{FF_{N_o1}} & \cdots & G_{FF_{N_o N_1 N_o \times N_i}}
\end{bmatrix}^{-1}
\] (2-5)

### 2.1.3 Coherence

The coherence is a dimensionless frequency domain function that estimates the extent of linear dependence between the input and output at each frequency, and hence indicates the degree of correlation in a frequency response function. It is the fraction of the measured output power directly related to the measured input power, and therefore bounded between zero and one. A coherence value of unity indicates perfect correlation. If any amount of measured output power is generated by noise, then the coherence value is less than one at that frequency. The non-linearities also reduce the overall value of coherence as they violate the assumption of linearity. Nevertheless, it should be noted that if the value of coherence is low at any frequency, it does not necessarily imply that the FRFs are of poor quality, but simply means that more averaging is needed for reliable results [1,7]. In the case of a single input as in an impact test, ordinary coherence function depicted by equation (2-6) is used.

\[
O_{COH_{pq}} = \frac{G_{XF_{pq}}}{G_{FF_{pq}} G_{XX_{pp}}} = \frac{|G_{XF_{pq}}|^2}{G_{FF_{pq}} G_{XX_{pp}}}
\] (2-6)

If there is only random noise present in the system and enough averages are taken, then a coherence value of unity can be achieved through the whole frequency range of interest except at very low frequencies. The accuracy of piezo-electric transducers, which are normally used in a vibration test, decreases at very low frequencies. Generally, bias error is also present in the system due to leakage or some other sources of
correlated noise. The bias errors cannot be reduced by averaging and special methods are used to counter them such as cyclic averaging is used to reduce leakage error. It is especially important to eliminate or minimalize the leakage error as the poles of FRF are very sensitive to it.

In addition, there may be instances when along with the measured input forces, unintentionally, some unmeasured forces are also applied to the system or the force transducers measure forces which are not actually applied to the system. In both the cases, the estimated FRFs will be erroneous and the coherence will be low. These problems cannot be solved by averaging or other DSP techniques, especially, if they are correlated with the measured inputs and their contribution to the measured output and input power is relatively substantial. The best method to rectify these errors is to eliminate their source.

However, it should be emphasized that even if all the inputs to the structure are measured, still the ordinary coherence function cannot be used for calculation as it is only formulated for the single input configuration. The multiple coherence shown in equation (2-7) is used when the structure is subjected to multiple inputs. There is a multiple coherence function for every output that determines the correlation between the output and all the measured inputs. In this thesis, generally, the word “coherence” is used for both the ordinary and multiple coherence. It should be understood that for the multiple input configuration (shaker and pneumatic test) in the text, multiple coherence and for the impact test, ordinary coherence is used

\[
M_{COHP} = \sum_{q=1}^{N_q} \sum_{i=0}^{N_i} \frac{H_{pq} G_{FFq} H_{pt}^*}{G_{XXpp}}
\]  

(2-7)

2.2 Existing Excitation Techniques

2.2.1 Shaker Testing

The shaker excitation method, particularly multiple input shaker testing, is very appealing due to the shortest testing time and consistency of the computed modal parameters [8]. A common shaker is an electrodynamic system in which an armature moves back and forth due to a strong magnetic field. An alternating current (AC) voice coil is attached to the armature to facilitate this movement. Depending on
the shaker’s size, either a high strength permanent magnet or an electromagnet is used for generating the magnetic field. The power amplifier used to drive the shaker may function either in voltage or current mode; however, voltage mode is more common. A variety of signals such as swept sine, random, periodic random, burst random etc. can be generated by altering the input. Current mode is usually used in sine or swept sine test signals or to study non-linearities in the system. Generally, two to four shakers are sufficient for most tests and five shakers are rare. Moreover, the number of shakers that could be used in a test is limited by the total number of output sources in the data acquisition system. There are other types of shakers that operate in the same way but utilize different physics, such as hydraulic shakers.

Shakers used in modal tests (modal shakers) often have a through-hole armature design (based on ideas from UC-SDRL implemented first by MB Dynamics, Inc., shown as exploded view in Figure 2-1) which gives great flexibility in attaching the shaker to the structure [8]. This flexibility is not found in the traditional shakers used for general vibration testing. In order to dynamically decouple the shaker and structure, and to protect the shaker from side loads, they are mechanically attached through stingers. The stingers have very low stiffness in the off-axis directions due to their material and geometry. This ensures that the structure is not excited through forces which are not measured by the force transducers. In addition, due to its limited stiffness in the on-axis direction, a stinger acts as a mechanical fuse and prevents any large force from being transmitted between the shaker and structure.

Figure 2-1: Exploded view of a shaker (modal exciter) based on through-hole armature design [8]
The procedure of computing FRFs in a MIMO configuration involves the mathematical inversion of the input power spectra matrix \([G_{FF}]\) shown in equation (2-8).

\[
[G_{FF}] = \begin{bmatrix}
G_{FF_{11}} & \cdots & G_{FF_{1N_i}} \\
\vdots & \ddots & \vdots \\
G_{FF_{N_i1}} & \cdots & G_{FF_{N_iN_i}}
\end{bmatrix}
\]  

(2-8)

There can be situations in which \([G_{FF}]\) matrix is nearly singular and therefore, its inverse may be numerically marginal [1]. First, if one of the input auto power spectra is zero then the corresponding rows and columns of the \([G_{FF}]\) matrix would also be zero thus making it singular. This can be an indication of a shaker or force transducer not working as demonstrated in Figure 2-2 [9]. Second, if any two of the input forces are perfectly correlated at a particular frequency then the inverse of \([G_{FF}]\) matrix will not exist at that frequency. In order to make all the input forces sufficiently uncorrelated, independent random signals are sent to each exciter. Nevertheless, due to electrodynamic interaction between the shaker and structure, input forces may still remain partially correlated at the natural frequencies [1]. This can be observed in the form of dips in the middle curve of Figure 2-2. It does not generally cause a problem but is undesirable as the coherent information could not be used for calculating the inverse.

\[
[G_{FF}] = [V][\Lambda][V]^H
\]  

(2-9)

Principle (virtual) force analysis is done to investigate this phenomenon which involves eigenvalue decomposition of the \([G_{FF}]\) matrix at each frequency as showed in equation where \(V, \Lambda\) and \(V^H\) represent left eigenvectors, eigenvalue matrix and right eigenvectors respectively [9] Eigenvalues associated with the corresponding inputs determine their contribution and they should be approximately same since the eigenvectors are unitary. If one eigenvalue is significantly smaller than the others, then either one of the inputs is not present or any two inputs are correlated.
2.2.2 Impact Testing

In the 1960s, impact testing evolved as one of the first applications of the fast Fourier transform (FFT) algorithm in the field of vibration testing [6, 9]. Before that period, only the sinusoidal excitation method was used which was very time consuming and cumbersome. Impact excitation comes in the category of transient excitation techniques and uses a roving impact hammer to excite the structure. During that early period, it was a common practice to mount the force transducers on the test structure. This configuration added mass to the structure at each driving point and it was also not possible to impact a rotating structure. Due to these issues, an impact hammer embedded with a force transducer began to be used. The frequency band of impact excitation is controlled by hardness of the tip, hammer’s mass and the velocity of hammer strike. A soft tip excites the structure at low frequencies relatively better while a hard tip is used for exciting the structure at high frequencies. The windows used in this technique are also different from other excitations due to transient characteristics of the force and response. Exponential windows are applied so
that the response ends within the time block, hence minimizing the digital signal processing error referred to as leakage. Although, the exponential window virtually increases the damping of the structure, the damping values can be corrected after computing the modal parameters. Apart from the exponential window, a force window is also used on the input to remove the portion of the time block after the impact, thus increasing signal to noise ratio[6]. It should be noted that the exponential window is applied on both the force and response while the force window is applied only on the force channel.

Figure 2-3: Impact hammers with different sizes (source: www.avenirtechnologies.co)

The impact excitation method does not have problem of correlated inputs found in shaker testing due to the application of a single impact at a time. However, it is difficult to maintain the repeatability of location, direction and magnitude of impacts for various measurements at a degree of freedom. Furthermore, a single impact is not sufficient to uniformly excite the structure which allows the nonlinearities present in the structure to distort the FRF measurement. Over the years, considering the evolution of analog to digital conversion (ADC) hardware and software, the problem of overloaded time domain signals has become comparatively more critical [6] due to the transient nature of the signals. In the delta-sigma analog digital converters (ADCs), even after auto-ranging, there can be an undetected overload because of significant energy in the frequency bands greater than the desired Nyquist frequency. This demands greater control over the frequency band of excitation which is relatively difficult in impact excitation as compared to shaker excitation.
3 Pneumatic Excitation Technique

3.1 Principal Idea

In this technique, a new type of random excitation signal using multiple impacts is investigated. The principal idea behind the new method is the fact that any random signal can be theoretically represented by successive impacts with random magnitude. Various DSP parameters (windows, averaging methods etc.) can be applied in the same way as in a typical random excitation. An example of this type of excitation and response is demonstrated in Figure 3-1.

![Figure 3-1: Typical multiple impact excitation force (left) and response (right)](image)

In order to verify the principle, an exercise was conducted in which three students repeatedly excited a structure (an aluminum circular plate) at three different points with impact hammers. It can be seen in Figure 3-2 that apart from being relatively noisy, the FRF measurement obtained from the multiple impact test matches with the traditional shaker test. At that point of time, there was not much concern for the relatively higher noise in the multiple impact test as the method was in its nascent stage.
3.2 Experimental Setup

Figure 3-2: FRF and Coherence comparison of traditional shaker and multiple impact test

Figure 3-3: Pneumatic excitation setup schematic
The schematic of the pneumatic excitation setup is described in Figure 3-3 and the physical setup is shown in Figure 3-4. Pneumatic cylinders powered by compressed air are used to excite the structure through successive impacts. However, this new method is different from impact testing because the structure is excited simultaneously at multiple locations (three in the existing setup). This method is more like a multiple input random excitation using continuous, randomly spaced impacts at each location to create independent random inputs. This method is a sort of hybrid between shaker and impact excitation techniques, possessing the positive characteristics of both. The absence of mechanical attachment between the structure and exciter makes it possible to always uncouple the input forces. This lack of mechanical attachment also means that lateral motion of the shaker connection is not present and cannot distort the measured force signal. Presence of multiple inputs also ensures uniform excitation of the structure [1]. Also note, there is high repeatability of the magnitude, direction and location of input forces as there is no direct human intervention in generating the impacts.

Figure 3-4: Physical pneumatic excitation setup

Solenoid valves are used to control the timing and duration of impacts at each location by regulating the air flow to the pneumatic cylinders. A system of independent pulses having random duration and interval (digital time series, shown in Figure 3-5) is sent to a simple switching circuit through the output channels (sources) of the data acquisition system. This digital time series dictates the opening and closing of solenoid valves through the switching circuit. A pressure regulator is used to control the air-pressure which in turn
controls the magnitude of impacts. As the air pressure increases, the velocity at which the piston of the cylinder hits the structure also increases thus, increasing the magnitude of impacts. Digital time series also affects the magnitude and characteristics of impacts as the stroke of the cylinder is dependent on the pulse width and time delay between the pulses. The pulse width and time delay values are represented in multiples of sampling interval which is generally denoted as $\Delta t$.

![Digital time-series](image)

**Figure 3-5: Digital time-series**

The piston of the pneumatic cylinder acts as an impact hammer with the force transducer mounted either on the piston of the pneumatic cylinder or the structure. The location of the force transducer significantly affects the quality of FRF data; hence both the configurations are studied in detail in the following sections.

### 3.2.1 Force Transducer on Exciter (FTE) Configuration

In this configuration, the force transducer is mounted on the piston of the pneumatic cylinder (Figure 3-6). Different types of tips can be attached to the force transducer to vary the frequency band of excitation. One clear advantage of this configuration is that there is no mass loading of the structure due to the force transducer. However, the movement of the force transducer with the piston gives rise to other problems. During the return (exhaust) stroke, the piston is quickly pulled back to its initial position due to a spring
force in the pneumatic cylinder and it hits the end cap with a high velocity (hard seating). This impact force is not transmitted to the structure but the force transducer still registers the force. There is no response in the structure caused by this force and hence, it is essentially noise and should be minimized. Furthermore, the tip mounted on the force transducer is essentially a single degree of freedom (SDOF) system that vibrates while in motion which augments the false force signal produced due to hard seating. As the mass of the tip increases, the false force signal produced due its vibration also increases (Section 4.4.1).

Figure 3-6: Force Transducer on Exciter (FTE) configuration

It can be seen in Figure 3-7 that there is considerable amount of force measured by the force transducer (up to 0.15 Volt) even when the structure is not excited. This error can be minimized by avoiding the impact caused by the spring force by balancing it with some other force. The back pressure at the outlet of the solenoid valve, regulated by a screw adjustable flow control valve (Figure 3-8), balances the spring force, thus acting as a cushion (soft seating) during the return stroke of the cylinder. The reduction of noise by soft seating and its effect on the estimated FRF can be observed in Figure 3-9 and Figure 3-10 respectively.
Also note the changes in multiple coherence, in Figure 3-10, representing undesirable MIMO FRF estimation.

![Flow control valve](image)

**Figure 3-8: Flow control valve**

However, the exhaust back pressure changes the characteristics of the impact as the exciter remains in contact with the structure for a relatively longer period of time. In order to correctly measure an impact force, the piston should be pulled back quickly after it makes contact with the structure. This is because a piezoelectric force transducer cannot measure static force due to charge drift phenomena and the signal conditioner is operated in the AC coupling mode to eliminate DC bias. Both these factors result in errors in the measurement of the overall force applied to the structure. Although, on a lightly damped structure, good results can be obtained in this configuration if the excitation force is kept relatively high and appropriate signal processing parameters are employed. High excitation force is necessary to maintain a good signal to noise ratio as the noise floor is comparatively higher than the other configuration. FRF measurements taken on a lightly damped rectangular steel plate are discussed in detail in the next chapter. On a moderately damped structure, good measurements could not be obtained in this configuration.

![Soft seating](image)

**Figure 3-9: Soft seating (structure is not excited)**
3.2.2 Force Transducer on Structure (FTS) Configuration

In this configuration, the force transducer is mounted on the structure. There is no need to employ back pressure to avoid hard seating as the force transducer does not move with the piston. Also, as a result of this, the piston could be pulled back quickly; hence, the error present in the measurements due to force transducer’s movement and piezoelectric drift is minimized or eliminated. Still, there is some mass loading of the structure caused by mounting the force transducer on it. Since the force transducers are mounted at fixed locations, this mass loading can be accounted for in later data processing. Also, the setup in this configuration is acoustically noisier than the other configuration. It is interesting to note that traditional impact testing was first conducted in the UC-SDRL in the late 1960s using the configuration where the force transducer was placed on the test object and struck with a common hammer.
It can be seen in Figure 3-11 that the force transducer is glued to the structure with a plastic disc and impact tip is mounted on the piston. The side of the force transducer which is facing the tip is also provided with a similar plastic disc so that impacts can be made on that surface. The typical excitation signal obtained in this configuration is shown in Figure 3-12. This configuration gives better results than the FTE configuration on both a lightly and moderately damped structure as demonstrated by Figure 3-13 and Figure 3-14 respectively. It is difficult to improve the FRF quality with FTE configuration on a moderately damped structure as higher signal to noise ratios are hard to achieve. Furthermore, relatively large and heavier tips in FTE configuration worsens the quality of FRF measurements in the testing performed to this point.

![Figure 3-12: Typical excitation force in FTS configuration](image-1)

![Figure 3-13: FRF comparison between FTS and FTE configuration on a lightly damped structure](image-2)
Generating an Appropriate Excitation Signal

The factors that affect the magnitude and characteristics of the pneumatic excitation signal are air pressure, pulse width and time delay of the digital time series. Also, the factors which are crucial in a traditional impact test like hammer tip, mass and velocity have an effect on the pneumatic excitation signal. The aim is to create an appropriate impact which can be duplicated, with some variations, to produce a multiple impact excitation. This process of generating a multiple impact signal is applicable only to the FTS configuration whereas it could not be applied successfully to the FTE configuration due to the requirements of soft seating. Generally, the value of parameters (pressure, pulse width and time delay) finalized through this process on the FTS configuration are taken as a starting point for the FTE configuration. Finally, it should be kept in mind that this process is very sensitive to the variations in parameters and setup with time; and small adjustments in parameter values may be frequently required for consistency in the generated excitation signal. The process is described in detail as follows:

- The test is started with an arbitrary value of pulse width, time delay and pressure for all the three exciters. In order to minimize the back pressure, the screw adjustable flow control valve should be kept completely open. It would be good to keep a similar distance between the exciter tip and force...
transducer at all the three driving points so that there is uniformity in the setup and comparisons can be made. However, if needed, it can be altered at each pneumatic cylinder to affect the magnitude of the force.

- An impact should be made at all the three driving points. This initial attempt may not generate a proper impact excitation and pulse width may need to be adjusted. Increasing the pulse width increases the time for which the solenoid valve is open; thus increasing the stroke of the piston and vice-versa. It is better to adjust the pulse width of one exciter at a time. However, the other two exciters should not be turned off while adjusting the first exciter as all of them are connected to a common pressure line which results in interdependence between them. This issue can be eliminated by providing separate compressed air sources or by additional accumulators on each source line.

- Generally, in the first attempt, either the exciter tip does not hit the force transducer or it remains in contact with the force transducer too long. Both of these conditions are not desirable and require change in the current pulse width. In order to find the optimum value some iterations need to be performed. After finalizing the pulse width for all the exciters for a proper impact, a recheck should be done as the exciters are slightly interdependent. During the pulse width iterations, if the air pressure is changed to increase or decrease the force then the whole activity needs to be repeated all over again.

- The excitation signal (pulse train) is synthesized taking the finalized pulse width as input. The MATLAB code that is used to synthesize the pulse train is written such that the maximum pulse width is equal to the input value. The pulse width values range randomly from 0 to the maximum pulse width (max_PW). This will ensure that either the intended impact is made or the tip doesn’t make contact with the force transducer. There is a small amount of inherent variability in the system due to the motion of piston which takes care of fully randomizing the magnitude of impacts. The other parameter affecting the excitation signal is the randomized time delay between the pulses. Its value can be varied to change the number of pulses in a time block of data. However, the minimum value of time delay (min_Td) should be such that the return stroke of the piston is fully completed.
after every pulse. This will minimize the dependence between successive impacts for a particular exciter. The time delay values range from $\text{min}_Td$ to twice of its value.

- It has been observed that even if there are double or triple impacts in the excitation, the final results does not get affected. This is due to the fact that as long as the excitation force is dynamic and completely measured by the force transducer, the quality of FRF and coherence should not deteriorate. It gives the freedom of keeping the input pulse width a little higher than its maximum value and the input time delay a little lower than its minimum value.

### 3.3.1 Effect of Pulse Width on Impact Force Characteristics

The pulse width is represented as multiples of the sample time interval ($\Delta t$). The number designation for the pulse width in the following examples represents the number of samples ($\Delta t$) that the pulse width includes. The experiment is conducted in the FTS configuration on two configurations: 1) a lightly damped (rectangular steel plate) and 2) a moderately damped structure (trimmed truck body). It should be clear that this section focusses on the effect of pulse width on a single impact; Section 4.1 is dedicated to the multiple impact case.
3.3.1.1 Lightly damped structure

The encircled portions in Figure 3-15 are shown in detail in the plots shown on the right.

Figure 3-15: Effect of pulse width on the impact characteristics of a lightly damped structure
As the pulse width increases, the auto power of the input force distorts due to double or multiple impacts. The disturbances indicated by the circle in the auto power spectra consistently occur at natural frequencies of the structure (Table 5-2) even after changing the pulse width. Also, in the time domain, it can be observed that the impacts are succeeded by oscillations of very low magnitude which are generally not seen in a traditional impact test. The reason for these observations is that the force transducer being attached to the structure (FTS configuration) continuously vibrates and is affected by the structure’s dynamics. The interaction between the structure and force transducer generates a force which appears as noise in both the frequency and time domain. However, this force does not distort the FRF measurements as it is correlated with the response of the structure.

3.3.1.2 Moderately damped structure

![Figure 3-16: Effect of pulse width on the impact characteristics of a moderately damped structure](image)
As the pulse width increases, the width of the impact also increases, as expected (Figure 3-16) The magnitude of the impact force is also growing with pulse width; however, the energy is increasing only at lower frequencies. It is because the exciter remains in contact with the structure for comparatively longer period of time with increasing pulse width. Furthermore, due to relatively high damping of the system, the disturbances that are present in the lightly damped system could not be observed here. An extreme case with pulse width 512 is also presented in the fourth plot to demonstrate the effect of piezoelectric drift and AC coupling mode of the signal conditioner. Ideally, the force signal in the time domain should look similar to a square wave pulse but due to the above mentioned factors, static forces cannot be measured. It means that after some point, the increase in pulse width will induce errors in the measurement. It should also be noted that apart from the difference in damping, there are other differences in the setup of the lightly and moderately damped structure like the characteristics of supports, size of pneumatic cylinder etc. These factors will also slightly effect the measurements.

### 3.3.2 Effect of Pressure on Impact Force Characteristics

In order to study the effect of pressure, the experiment is conducted at 20, 25 and 30 PSI. With the increase in pressure, there is an increase in the velocity with which the exciter tip hits the structure; therefore, the time required for the piston to cover the distance between the exciter and structure also decreases. Again, it should be clear that this section focusses on the effect of pressure on a single impact; Section 4.3 is dedicated to the multiple impact case.

#### 3.3.2.1 Lightly damped structure

There is not much difference observed between the measurements at different air pressures apart from a slight increase in the distortion in auto power with increasing pressure.
Figure 3-17: Effect of pressure on the impact characteristics of a lightly damped structure

### 3.3.2.2 Moderately Damped Structure

The width and magnitude of impact increases with pressure. Additionally, the time taken by the piston to reach the structure decreases with increasing pressure as shown in the zoomed sections of Figure 3-18. The piston comes in contact with the structure relatively early, hence the impact width increases. The magnitude of the impact force is larger due to the increase in momentum of the piston.
Figure 3-18: Effect of pressure on the impact characteristics of a moderately damped structure
4 Effect of Various Parameters on Measurements

The effect of pneumatic excitation parameters on the measurements (excitation force and FRF) is studied in this chapter. The parameters studied are pressure, pulse width and time delay. Additionally, various types of tips are also tested during the course of this experiment. The structure on which these tests are conducted is a lightly damped rectangular steel plate of dimensions 34 x 22.5 x 0.243 in. The tests are conducted with both the FTE and the FTS configuration. While analyzing the effect of variation in one parameter, all other pneumatic as well as DSP parameters are kept constant. Results associated with only one exciter are used for representation.

Table 4-1: Initial Values of Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of frequency lines</td>
<td>1600</td>
</tr>
<tr>
<td>Sampling Frequency (Hz)</td>
<td>1024</td>
</tr>
<tr>
<td>Maximum frequency (Hz)</td>
<td>400</td>
</tr>
<tr>
<td>No. of synchronous averages</td>
<td>20</td>
</tr>
<tr>
<td>No. of cyclic averages</td>
<td>4</td>
</tr>
<tr>
<td>Windows function</td>
<td>Hanning</td>
</tr>
<tr>
<td>Trigger</td>
<td>Free</td>
</tr>
<tr>
<td>Pulse width (no. of data points)</td>
<td>35 for all exciters</td>
</tr>
<tr>
<td>Time delay (no. of data points)</td>
<td>30 for all exciters</td>
</tr>
<tr>
<td>Pressure (PSI)</td>
<td>20</td>
</tr>
</tbody>
</table>

4.1 Effect of Pulse Width

FRF is estimated with three different values of maximum pulse width (max_PW); and for all the exciters, max_PW is kept same. The values of max_PW used in the experiment are 35, 200 and 400. However, in the FTE configuration one of the exciter’s max_PW is kept 45 instead of 35 because the exciter barely moves with a value of 35. This is due to the additional mass of force transducer on the exciter in the FTE configuration as compared to the FTS configuration.
4.1.1 FTE Configuration

As the value of max_PW increases, intuitively, the number of impacts in the same time block should decrease. But rather a change in the max_PW from 35 to 200 increases the number of impacts in the time block. It is due to the randomness of pulse width which can take any value between 0 and max_PW. The probability of the exciter to reach the structure is more in case of max_PW 200 than 35. However, the number of impacts again decreases with max_PW of 400 as the two phenomenon mentioned above work against each other. It is also found that a min_Td of 30 is not sufficient for max_PW values of 200 and 400. With this combination, the exciter does not have sufficient time to move away from the structure and it continuously remains in contact with the structure producing a distorted force signal of very low magnitude shown in Figure 4-2. Higher value of time delay is used for max_PW 200 and 400.

Figure 4-1 : Excitation(left) and principal forces(right) with max_PW 35 and min_Td 30

Figure 4-2: Excitation with max_PW 200 and min_Td 30

Figure 4-3: Excitation (left) and principal forces (right) with max_PW 200 and min_Td 300
It can be seen in the graphs of principal forces (Figure 4-1, Figure 4-3 and Figure 4-4) that there are dips at natural frequencies. These dips indicate partial coupling of the input forces and the magnitude of dips is almost equal in the three cases. In the FTE configuration, the exciters remain in contact with the structure for relatively longer period of time, thus the probability of the forces becoming coupled to the structure increases. There is no considerable difference in the FRF curves obtained with the three values of max_PW, though the coherence curve with max_PW of 35 is less smooth than those obtained with others. The drop in coherence at peaks indicates input correlation or leakage.

4.1.2 FTS configuration

It is observed that FTS configuration follows a similar trend as FTE configuration in terms of the number of impacts in a time block (Figure 4-6, Figure 4-7 and Figure 4-8). However, the dips in the principal forces at natural frequencies are far less deep than the FTE configuration, especially with max_PW 35. It is
because of the fast retreat of the piston after hitting the structure. Also, it should be closely noticed that the dips are deeper in the case of max_PW 200 and 400 as compared to max_PW 35 because as the pulse width increases the extent to which forces are coupled also increases due to extended contact between the exciter and structure. As opposed to FTE configuration, min_Td of 30 is sufficient for all the three cases.

The FRFs acquired with different max_PW values lie closely on top of each other (Figure 4-9). The coherence and FRF curves are much smoother than the FTE configuration. Also, the dips in coherence at the peaks are much less than those in the FTE configuration. All this can be attributed to the high signal to noise ratio and minimal coupling of input forces in the FTS configuration.
4.2 Effect of Time Delay

In this section, the effect of time delay on the excitation and estimated FRFs is studied. The test is conducted with three different cases: \( \text{min}_T 30 \) for all exciters, \( \text{min}_T 100 \) for all exciters, \( \text{min}_T 150, 200 \) and 100 for exciters at location no. 3, 5 and 12 respectively. The values of \( \text{max}_P \) are kept as 35, 35 and 45 at location no. 3, 5 and 12 respectively for all the three cases.

4.2.1 FTE Configuration

The increase in time delay prolongs the time for which the exciter does not move (rests at its initial position), thus decreasing the number of impacts in a time block as clear by observing the Figure 4-10, Figure 4-11 and Figure 4-12. This can be thought of as a minimum hold off time. The noise generated by the movement of force transducer is also reduced with higher values of \( \text{min}_T \). In comparison to \( \text{min}_T 100 \) and 150, the dips in the coherence functions at natural frequencies are deeper in the case where minimum time delay is 30. With larger values of time delay, the impacts generated by an exciter have higher probability of being relatively more separated in time with respect to the other exciters, which decreases the extent to which the input forces are coupled.
Figure 4-10: Excitation(left) and principal forces(right) with min_T_d 30

Figure 4-11: Excitation(left) and principal forces(right) with min_T_d 100

Figure 4-12: Excitation(left) and principal forces(right) with min_T_d 150

There is a very small deviation between the three FRF curves as can be noticed in Figure 4-13. Nevertheless, the dips in the coherence curve are deeper for min_T_d 30, since its excitation signal is noisier as compared to others. In order to improve the coherence, the time delay should not be made too large such that it is not possible to sufficiently excite the structure.
4.2.2 FTS Configuration

The number of impacts in a time block decreases as the value of min_T increases (Figure 4-14, Figure 4-15, Figure 4-16) which is same as in the FTE configuration. There is no significant difference in the characteristics of the principal forces with increase in time delay because even with min_T of 30 there is minimal coupling of the input forces.
There is very little variation in the estimated FRFs with different min_Td values. Overall, the FRFs and coherences are of good quality. It should be noted that even with highest min_Td it is possible to sufficiently excite the structure and achieve a good signal to noise ratio. However, in certain setups where signal to noise ratio is already low, high min_Td values may not work.

4.3 Effect of Pressure

The structure is tested at three different values of pressure in both the configurations. The first value is kept as 20 PSI as the range of operation of the pressure regulator is 20 PSI – 130 PSI. The three values at which the test is conducted are 20 PSI, 25 PSI and 30 PSI. Pressures higher than this do not seem necessary.

4.3.1 FTE Configuration

The velocity of the piston or exciter should increase with pressure, thereby enhancing the momentum and resulting in higher values of the excitation force. Nevertheless, the excitation force is not greater at 25 PSI
in comparison to 20 PSI (Figure 4-18 and Figure 4-19); also, the number of impacts in a time block seems to follow a decreasing trend with pressure. The reason for this anomalous behavior is that a min_Td of 30 is not sufficient at 25 PSI and 30 PSI for the piston to consistently move away from the structure a sufficient distance to produce an increase in the momentum or excitation force. A value of min_Td 100 solves this problem as demonstrated in Figure 4-21 and Figure 4-22. This observation suggests that in order to produce an intended effect (e.g., high excitation force), more than one parameter may have to be altered.

The FRF and coherence curves become smoother with increasing pressure as a result of high signal to noise ratio (Figure 4-23). This is due to the fact that with an increase in pressure, the excitation force also increases, but the noise floor remains approximately at the same level. The coherence drop at natural
frequencies is also less in 25 PSI in comparison to 20 PSI for the same reason. However, there is not much difference in coherence drop between 25 PSI and 30 PSI.

4.3.2 FTS configuration

In the FTS configuration, there is no need to increase the min_Td value of 30 to 100 to get a consistent excitation force signal at higher pressures. However, there is marginal increase in the magnitude of excitation with pressure.
There is good agreement between the graphs of principal forces obtained at different pressure values (Figure 4-24, Figure 4-25 and Figure 4-26). The FRFs estimated at various pressures match very closely, thus indicating that the signal to noise ratio is adequate even at 20 PSI. These FRFs are quite good with notably good quality coherences (Figure 4-27).

### 4.4 Effect of Exciter Tip

In order to see the effect of various tips on excitation forces and FRFs, tips harder and softer than the regular tip (the tip used in all the other tests) are tested. The different types of tips used in the test are same for FTE and FTS configurations (Figure 4-28).
4.4.1 FTE Configuration

It is known from the traditional impact testing that the hammer tip affects the frequency band of excitation. In order to analyze this effect, it is better to study the input auto power rather than the principal forces. The reduction in auto power at higher frequencies is noticeable in the case of a regular (softer) tip as compared to the harder tip as shown by Figure 4-29 and Figure 4-30. As the frequency increases, the augmentation in the gap between the auto power curves is more while using a regular (softer) tip. The effect of tip hardness on the frequency band of excitation can also be observed in the zoomed section of Figure 4-33 where coherence curve of the harder tip is better than the regular tip. A harder tip excites the structure relatively well through a larger frequency range of interest.

Figure 4-28: Different types of exciter tips (from left to right: softer, harder and regular tip)

Figure 4-29: Excitation (left) and auto power curves (right) with regular tip

Figure 4-30: Excitation (left) and auto power curves (right) with harder tip
The excitation and auto power graph of the softer tip seem remarkably different from the other tips. It may appear by observing Figure 4-31 that the number of impacts in a time block are comparatively more in the case of softer tip, though there is no apparent reason for this phenomenon. The reason for this is understood when force transducer signal is measured without exciting the structure. In that condition, the exciter reciprocates without hitting the structure and the force transducer signal mainly represents the noise induced in the system due to the FTE configuration (Figure 4-32). The magnitude of this noise is comparable to the actual excitation force in case of softer tip and it is difficult to distinguish the two signals as can be seen in Figure 4-31. However, it should be carefully noted that the softness of the tip has no correlation with the induced noise, rather the vibration of the tip is the actual source of noise. Apart from exciting the structure, the tip also vibrates during the rest of the stroke. This vibratory motion generates forces which are sensed by the force transducer mounted beneath the tip; though they are not being transferred to the structure. These forces are not causing any response within the structure and thus distort the measurements. As the mass of the tip increases, the magnitude of the forces generated by its vibration also increases, and hence the softer tip being heaviest of all produces more noise than the other tips. Also, the softer tip exerts lower excitation force and thus the response is also lower. The extent to which this noise affects the FRF measurements can be assessed by comparing the FRFs and coherences of the softer tip with other tips as shown in the Figure 4-33.
4.4.2 FTS Configuration

In the FTS configuration, there is no problem of noise due to vibration of the tip as the force transducer is not mounted on the exciter. This makes it possible to take good measurements even with the heavier tip. The behavior of various tips at higher frequencies is same as expected. There is a downward trend in the magnitude of auto power with increasing frequency when a softer tip is used; the regular tip follows the same trend though to a less extent than the softer tip; the decrease in magnitude is barely visible in case of the harder tip. Also, it can be noticed from Figure 4-34, Figure 4-35 and Figure 4-36 that the magnitude of excitation force is highest with the softer tip unlike the FTE configuration. This may be due to relatively higher velocity of the softer tip (heavier tip) in the FTS configuration, since there is no impediment to its motion due to back pressure. With every pulse, the softer tip covers the whole stroke and generates higher momentum that is not possible in the case of FTE configuration. The high momentum of the tip offsets the
reduction in the excitation force due to softness of the tip. The FRFs estimated with the three tips have good coherences and closely match each other (Figure 4-37).

Figure 4-34: Excitation (left) and auto power curves (right) with the regular tip

Figure 4-35: Excitation (left) and auto power curves (right) with the harder tip

Figure 4-36: Excitation (left) and auto power curves (right) with the softer tip

Figure 4-37: FRF estimation with different tips (FTS configuration)
5 Experimental Validation of Pneumatic Excitation Technique

5.1 Overview

In order to experimentally validate the pneumatic excitation technique, tests are conducted on a lightly damped rectangular steel plate and a moderately damped trimmed truck body. The FTS configuration is used because the results obtained with it are better than the FTE configuration as shown in the previous sections. The estimated results (FRF and modal parameters) are compared with the traditional shaker and impact test. Note that some variation in the methods is to be expected as the mass loading due to fixed sensors and the impedance loading due to attached shakers will contribute to small differences. It should also be noted that only shaker and pneumatic test are conducted on the trimmed truck body as an impact test is more susceptible to noise, rattles and small non-linearities in the trimmed vehicle. This is somewhat a function of the single input aspect of impact testing.

5.2 Rectangular plate

A steel plate having dimensions 34 x 22.5 x 0.243 inch is used for the test (Figure 5-1) as in previous examples. The plate is excited at three degrees of freedom (DOFs) locations and in the vertical direction, and response is taken at 16 locations. The same DOFs are used in all the tests whether it is an impact, shaker or pneumatic test. The data acquisition hardware, software and DSP parameters used in the tests are listed in the following sections. The hardware exclusive to the pneumatic setup is given in the Appendices.
5.2.1 Data Acquisition

5.2.1.1 Hardware and software

- Interface: VXI Technology, CT-310A VXI Mainframe, 16 channels per card, 3 sources
- Accelerometer: PCB Model no. TLD352A56, Sensitivity 100mv/g, Quantity -16
- Force transducer: PCB Model no. 208C03, Sensitivity 50 mv/lbf, Quantity - 3
- Impact hammer: PCB Model no. 086C03, Sensitivity 10mv/lbf, Quantity - 1
- Shaker: The Modal Shop Model no. K2004E01, Force output 7lbs, Quantity - 3
- Software: X-Modal III, UC-SDRL
- Algorithm generation and results: MATLAB™

5.2.1.2 DSP parameters

<table>
<thead>
<tr>
<th>Table 5-1: DSP Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling rate</td>
</tr>
<tr>
<td>Frequency resolution</td>
</tr>
<tr>
<td>Block size</td>
</tr>
<tr>
<td>Power spectrum averages</td>
</tr>
<tr>
<td>Window</td>
</tr>
<tr>
<td>Trigger</td>
</tr>
<tr>
<td>Tip</td>
</tr>
</tbody>
</table>
5.2.1.3 Pneumatic excitation parameters

- Number of exciters: three at DOF no. 3, 5 and 12
- Pulse width: 35 for all exciters
- Time delay: 30 for all exciters
- Air pressure: 20 PSI

5.2.2 FRF Estimation

![Figure 5-3: Pneumatic excitation](image)

Pneumatic excitation forces applied to the rectangular plate by the three exciters are shown in Figure 5-3.

The disturbance which appears as a thick line at 0 Volt (more noticeable with the 2nd exciter) is not noise; it is the force that the transducers are sensing due to being attached to the structure (see Section 3.3.1.1 for detail). The force exerted by each exciter consists of multiple impacts with random magnitude and time delay. It ensures that the input forces from the three exciters are not correlated with respect to each other which is clear by observing the virtual forces (principal components of the auto-power matrix of excitation forces (Figure 5-4)). In addition, the dips in principal components at the natural frequencies are less for pneumatic excitation as compared to shaker excitation (Figure 5-5).
Figure 5-4: Principal forces obtained from pneumatic excitation

Figure 5-5: Principal forces obtained from shaker excitation

Figure 5-6: Response obtained at a driving point in the pneumatic test

The response obtained in the pneumatic excitation appears like a typical response generated by applying random input forces through shakers on a structure. In Figure 5-7, the FRF acquired at a driving point on the rectangular plate by the shaker and pneumatic test is compared. Both the FRF curves closely match each other throughout the whole frequency range. There are significant coherence dips at natural frequencies in both the methods which indicate leakage due to the time truncation FFT error.
In order to reduce leakage, cyclic averaging is used with four and eight averages. It is well-known that cyclic averaging diminishes the non-periodic part and enhances the periodic part of the signal in the time block, thus reducing leakage [11].
Due to cyclic averaging, the leakage has almost become zero as evident from the multiple coherence plots of Figure 5-8 and Figure 5-9. The multiple coherence is approximately one in both the figures; however, eight cyclic averages are marginally better than four cyclic averages. Also, it should be noted that eight cyclic averages require twice as much acquisition time as needed for four cyclic averages. In Figure 5-9, the FRF measurement from the impact test (no cyclic averaging) is also shown. The coherence obtained in the impact test is also one at the peaks, but it drops significantly at some of the anti-resonances. It is because the impact test is prone to small errors caused by the variability in location and direction of the applied force, and coherence at the anti-resonances is most affected by these errors due to minimal response at these frequencies. However, for the same reason, this problem is trivial as the frequencies of interest are only those where the response peaks to its local maximum value.

Figure 5-10: Pneumatic burst random excitation and response with 75% burst length
In addition to the random excitation, results from burst random excitation with 75% burst length are also analyzed for both the shaker and pneumatic (Figure 5-10) testing method. In both the methods, 75% burst length, four cyclic averages and uniform window are used while estimating the FRFs [12]. The FRF curves match relatively well (Figure 5-11) and the multiple coherence is at par with the test in which eight cyclic averages are used. Additionally, the acquisition time for burst random excitation with four cyclic averages is half than that of the time used for eight cyclic averages.

![Multiple Coherence](image)

**Figure 5-11:** FRF comparison of shaker and pneumatic test with burst random excitation

### 5.2.3 Modal Parameter Estimation

The next step in the validation process is to compare the modal parameters obtained from the impact, shaker and pneumatic testing technique. In the pneumatic test, one of the non-driving point response accelerometer did not give correct data and therefore, that DOF was sieved from all the three tests while computing the modal parameters. For modal parameter estimation, the Poly-reference Time Domain (PTD) method is used for all the cases and autonomous modal parameter estimation technique is used to select the poles. The number of modes present in the frequency band of interest (0 - 400 Hz) is 17. In these modes, a repeated root is also present at around 45 Hz. While computing the FRFs for modal parameter estimation in the case of shaker and pneumatic test, the same window (Hanning), number and type of averages (20 power
spectrum, eight cyclic) are used for consistency. Considering the nature of impact test, window (Force-rectangular), number and type of averaging (5 power spectrum, no cyclic) cannot be kept same as the shaker and pneumatic test.

Figure 5-12: Frequency and damping comparison between impact, shaker and pneumatic test
It can be noticed in Table 5-2 that the natural frequencies obtained from the three methods are very close to each other. Apart from the first three modes (rigid body modes of the plate on its supports), the percentage difference between pneumatic and other tests is below 0.5%. Ignoring the rigid body modes, the maximum difference between the pneumatic and other tests occurs at fourth and fifth mode for the impact and shaker test respectively.
### Table 5-3: Damping comparison between the three methods

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Impact test</th>
<th>Shaker test</th>
<th>Pneumatic test</th>
<th>% difference between pneumatic and impact test</th>
<th>% difference between pneumatic and shaker test</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0976</td>
<td>0.0808</td>
<td>0.1046</td>
<td>7.172</td>
<td>29.455</td>
</tr>
<tr>
<td>2</td>
<td>0.1847</td>
<td>0.1417</td>
<td>0.2199</td>
<td>19.058</td>
<td>55.187</td>
</tr>
<tr>
<td>3</td>
<td>0.1832</td>
<td>0.1745</td>
<td>0.2508</td>
<td>36.900</td>
<td>43.725</td>
</tr>
<tr>
<td>4</td>
<td>0.1238</td>
<td>0.1074</td>
<td>0.1529</td>
<td>23.506</td>
<td>42.365</td>
</tr>
<tr>
<td>5</td>
<td>0.142</td>
<td>0.1773</td>
<td>0.1555</td>
<td>9.507</td>
<td>-12.296</td>
</tr>
<tr>
<td>6</td>
<td>0.1713</td>
<td>0.1542</td>
<td>0.1823</td>
<td>6.421</td>
<td>18.223</td>
</tr>
<tr>
<td>7</td>
<td>0.0824</td>
<td>0.1013</td>
<td>0.1155</td>
<td>40.170</td>
<td>14.018</td>
</tr>
<tr>
<td>8</td>
<td>0.0915</td>
<td>0.1179</td>
<td>0.1051</td>
<td>14.863</td>
<td>-10.857</td>
</tr>
<tr>
<td>9</td>
<td>0.1176</td>
<td>0.1302</td>
<td>0.1488</td>
<td>26.531</td>
<td>14.286</td>
</tr>
<tr>
<td>10</td>
<td>0.14</td>
<td>0.1562</td>
<td>0.1723</td>
<td>23.071</td>
<td>10.307</td>
</tr>
<tr>
<td>11</td>
<td>0.1178</td>
<td>0.1194</td>
<td>0.1119</td>
<td>-5.008</td>
<td>-6.281</td>
</tr>
<tr>
<td>12</td>
<td>0.0777</td>
<td>0.0969</td>
<td>0.1027</td>
<td>32.175</td>
<td>5.986</td>
</tr>
<tr>
<td>13</td>
<td>0.0938</td>
<td>0.1381</td>
<td>0.1175</td>
<td>25.267</td>
<td>-14.917</td>
</tr>
<tr>
<td>14</td>
<td>0.131</td>
<td>0.1497</td>
<td>0.1303</td>
<td>-0.534</td>
<td>-12.959</td>
</tr>
<tr>
<td>15</td>
<td>0.1887</td>
<td>0.178</td>
<td>0.1709</td>
<td>-9.433</td>
<td>-3.989</td>
</tr>
<tr>
<td>16</td>
<td>0.1038</td>
<td>0.1195</td>
<td>0.1183</td>
<td>13.969</td>
<td>-1.004</td>
</tr>
<tr>
<td>17</td>
<td>0.292</td>
<td>0.4332</td>
<td>0.3494</td>
<td>19.658</td>
<td>-19.344</td>
</tr>
</tbody>
</table>

The damping values are very small but are not consistent across the three methods (Table 5-3). The variation in damping values obtained from the impact test (see error bars in Figure 5-12) is relatively high as compared to the shaker and pneumatic test. Ignoring the rigid body modes, the pneumatic and shaker test give higher values of damping than the impact test for most of the modes. However, this is anecdotal evidence, as multiple tests for each method are needed for statistical inference.
The Modal assurance criteria (MAC) is the measure of linear dependence between the modal vectors; its value ranges from zero to one where zero signifies that modal vectors are independent of each other and one signifies total linear dependence between them[1, 12]. The modal vectors obtained from all the three tests are shown together in the MAC plot represented in Figure 5-13. The red color denotes MAC value of one and the blue denotes MAC of zero. The red squares along the diagonal indicate that the corresponding modal vectors obtained from the three tests are linearly dependent with respect to each other. Ideally, the plot should consist of only red and blue color but due to lack of observability (insufficient DOFs) intermediate colors are also present in the plots. However, it is not a matter of concern in the context of this validation process. Furthermore, modal vector complexity plots obtained from the three methods are shown.

Figure 5-13: Modal Assurance Criterion (MAC) comparison of the three methods
in Figure 5-14. The modal vectors hardly show any complexity as evident by the fact that all the points in a modal vector lie along the X-axis.

![Modal Vector Complexity Plots](image)

Figure 5-14: Modal vector complexity plots of the three methods

5.3 **Trimmed Truck Body**

In addition to experimentally validating the pneumatic technique on a simple structure like the rectangular plate, it is necessary to test a more complicated, *real world* structure such as a trimmed truck body with this technique. The truck body used for the tests is shown through Figure 5-15 and Figure 5-16.
The truck is excited at three locations, two in the vertical direction and one in the horizontal direction. One vertical and horizontal exciter is on the front side (Figure 5-15) and the other vertical exciter is at the back side (Figure 5-16) of the truck. The response is taken at 8 locations which are chosen such that the accelerometers’ cluster forms a cuboid like shape on the structure. Due to relatively large size of the structure, different pneumatic cylinders that have more capacity (larger bore) than those used to test the rectangular plate were utilized. The ability to adjust the size (force capacity) of the pneumatic cylinders at marginal cost is another attractive aspect of the technique. The force capacity of the shakers is also more
as compared to the shakers used in rectangular plate testing. The shaker setup can be seen in Figure 5-17 (shaker position from left to right: horizontal, front, back). The accelerometers used are also more sensitive than those used with the rectangular plate as it is a moderately damped structure. The hardware exclusively used for the pneumatic setup is not listed in the following section, rather it can be found in Appendices.

![Figure 5-17: Shaker testing set up on the truck body](image)

5.3.1 Data Acquisition

5.3.1.1 Hardware and software

- Interface: VXI Technology, CT-310A VXI Mainframe, 16 channels per card, 3 sources
- Accelerometer: PCB Model no. 356B18, Sensitivity 1000 mv/g, Quantity - 8
- Force transducer: PCB Model no. 208C03, Sensitivity 50 mv/lbf, Quantity - 3
- Shaker: MB Dynamics Modal 50, Force output 50 lbs, Quantity - 3
- Software: X-Modal III, UC-SDRL
- Algorithm generation and results: MATLAB™
5.3.1.2 DSP parameters

Table 5-4: DSP parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling Frequency</td>
<td>128 Hz</td>
</tr>
<tr>
<td>Frequency resolution</td>
<td>0.25 Hz</td>
</tr>
<tr>
<td>Block size</td>
<td>512</td>
</tr>
<tr>
<td>Power spectrum averages</td>
<td>20</td>
</tr>
<tr>
<td>Cyclic Averages</td>
<td>4</td>
</tr>
<tr>
<td>Window</td>
<td>Hanning</td>
</tr>
<tr>
<td>Trigger</td>
<td>Free</td>
</tr>
<tr>
<td>Tip</td>
<td>Soft</td>
</tr>
</tbody>
</table>

5.3.1.3 Pneumatic excitation parameters

- Number of exciters: three at location no. 1, 2 and 3.
- Pulse width: 13, 12 and 11 for exciters at location no. 1, 2 and 3 respectively
- Time delay: 15 for all exciters
- Pressure: 25 PSI

5.3.2 FRF Estimation

The process of estimating FRFs on the truck body is more challenging than the rectangular plate. One of the reasons is the low frequency span (0-50 Hz) in which measurements are to be made. The excitation of higher modes is not required as the goal here is not to assess the complete dynamics of truck body; rather, the test is conducted only to validate the pneumatic excitation technique. Additionally, the higher modes may be more affected by the non-linearities present in the structure. The restriction of energy in such a low frequency band requires a very soft tip. If a hard tip is used, it will sufficiently excite the frequencies higher than the maximum frequency leading to high frequency overload [6]; which results in compromising the amplitude resolution of signal in the frequency band of interest. Many tip iterations are conducted before
finalizing the soft tip shown in Figure 5-18. In addition to the rubber tip, foam is also attached on both the exciter and force transducer to widen the force pulse and limit the higher frequency energy.

![Figure 5-18: Tip used in truck testing](image)

It can be assessed from the response signal in Figure 5-19 that this structure is relatively more damped than the rectangular plate as evident from the shorter duration of responses due to individual impacts. In the case of rectangular plate, these individual responses merge into each other and the overall response appears more like a typical random vibration.

![Figure 5-19: Excitation (left) and response (right) at one of the driving points](image)

![Figure 5-20: FRF comparison between the shaker and pneumatic test](image)
There is very good agreement between the FRF curves of the shaker and pneumatic test as shown in Figure 5-20. The coherence is almost one through the whole frequency range except there are little dips at the anti-resonances. The peaks are less sharp in comparison to the rectangular plate FRFs due to the structure being moderately damped.

![Figure 5-21: Principal forces obtained in the pneumatic test](image1)

![Figure 5-22: Principal forces obtained in the shaker test](image2)

The principal forces obtained in both the tests are uncorrelated and remain approximately same through the whole frequency range as demonstrated by Figure 5-21 and Figure 5-22. Unlike the rectangular plate testing, there are no dips at natural frequencies even in the shaker test except at one of the modes. The reason behind it may be that the truck body is relatively more complex and bigger than the rectangular plate.

### 5.3.3 Modal Parameter Estimation

The Rational Fraction Polynomial with Z mapping (RFPZ), a time domain modal parameter estimation method, and autonomous pole picking is used for estimating modal parameters for all the cases. The number of modes found in the frequency range (0 – 50 Hz) is 13.
Figure 5-23: Frequency and damping comparison between shaker and pneumatic test

Table 5-5: Frequency comparison

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Shaker test</th>
<th>Pneumatic test</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.801</td>
<td>2.77</td>
<td>-1.11</td>
</tr>
<tr>
<td>2</td>
<td>3.785</td>
<td>3.78</td>
<td>-0.08</td>
</tr>
<tr>
<td>3</td>
<td>4.573</td>
<td>4.61</td>
<td>0.85</td>
</tr>
<tr>
<td>4</td>
<td>5.392</td>
<td>5.42</td>
<td>0.45</td>
</tr>
<tr>
<td>5</td>
<td>6.085</td>
<td>6.13</td>
<td>0.74</td>
</tr>
<tr>
<td>6</td>
<td>12.37</td>
<td>12.40</td>
<td>0.21</td>
</tr>
<tr>
<td>7</td>
<td>14.71</td>
<td>14.72</td>
<td>0.09</td>
</tr>
<tr>
<td>8</td>
<td>15.684</td>
<td>15.80</td>
<td>0.74</td>
</tr>
<tr>
<td>9</td>
<td>17.833</td>
<td>17.93</td>
<td>0.52</td>
</tr>
<tr>
<td>10</td>
<td>20.383</td>
<td>20.44</td>
<td>0.29</td>
</tr>
<tr>
<td>11</td>
<td>32.585</td>
<td>32.57</td>
<td>-0.04</td>
</tr>
<tr>
<td>12</td>
<td>35.423</td>
<td>35.34</td>
<td>-0.23</td>
</tr>
<tr>
<td>13</td>
<td>37.51</td>
<td>37.46</td>
<td>-0.15</td>
</tr>
</tbody>
</table>

Table 5-6: Damping comparison

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Shaker test</th>
<th>Pneumatic test</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.076</td>
<td>0.0687</td>
<td>-9.605</td>
</tr>
<tr>
<td>2</td>
<td>0.119</td>
<td>0.1268</td>
<td>6.376</td>
</tr>
<tr>
<td>3</td>
<td>0.142</td>
<td>0.1351</td>
<td>-4.792</td>
</tr>
<tr>
<td>4</td>
<td>0.183</td>
<td>0.2149</td>
<td>17.239</td>
</tr>
<tr>
<td>5</td>
<td>0.159</td>
<td>0.1586</td>
<td>0.000</td>
</tr>
<tr>
<td>6</td>
<td>0.281</td>
<td>0.2865</td>
<td>2.066</td>
</tr>
<tr>
<td>7</td>
<td>0.216</td>
<td>0.2126</td>
<td>-1.574</td>
</tr>
<tr>
<td>8</td>
<td>0.419</td>
<td>0.3587</td>
<td>-14.371</td>
</tr>
<tr>
<td>9</td>
<td>0.408</td>
<td>0.3787</td>
<td>-7.227</td>
</tr>
<tr>
<td>10</td>
<td>0.359</td>
<td>0.3561</td>
<td>-0.697</td>
</tr>
<tr>
<td>11</td>
<td>0.097</td>
<td>0.1171</td>
<td>20.971</td>
</tr>
<tr>
<td>12</td>
<td>0.180</td>
<td>0.2378</td>
<td>24.474</td>
</tr>
<tr>
<td>13</td>
<td>0.160</td>
<td>0.1765</td>
<td>10.381</td>
</tr>
</tbody>
</table>
All the modes match relatively well as shown in Figure 5-23. Ideally, there should be six rigid body modes in a three-dimensional structure, but in this case, there are only five rigid body modes as the structure was excited in only two directions. Ignoring the first five modes, the maximum percentage difference in frequency is 0.74 %, which occurs at the eighth mode. As expected, the difference in damping is comparatively high between the two methods. However, this is anecdotal evidence, as multiple tests for each method are needed for statistical inference. The maximum percentage difference in damping occurs at the 12th mode and it is 24.474% (Table 5-6). The damping values of the modes, considering only flexible modes from the two methods, range from 0.097 to 0.419 Hz. This clearly indicates that the truck body is moderately damped.

Figure 5-24: MAC comparison between shaker and pneumatic test
The red squares along the diagonal indicate that the corresponding modal vectors obtained from the two tests are linearly dependent with respect to each other. (Figure 5-24). Some of the frequency modes have significantly high cross MAC values due to lack of observability (very limited number of response sensors).

It can be seen in Figure 5-25 and Figure 5-26 that the modal vector complexity for both the tests is very low. However, it is not a matter of concern in the context of this validation process. The results obtained for the plate and truck prove that the pneumatic excitation gives good results in all type of settings whether, academic or industrial.

![Modal Vector Complexity Plots](image)

Figure 5-25: Modal vector complexity plot of shaker test
Figure 5-26: Modal vector complexity plot of pneumatic test
6 Summary

6.1 Conclusions

Based upon work to date, the pneumatic excitation method is a good alternative to traditional excitation methods, especially shaker testing. The experimental setup is made up of simple, low cost components and can be easily customized (sized) for different testing requirements. In order to generate different levels of excitation forces, tests can be conducted with pneumatic cylinders of various capacities and at various pressure values. Due to the compact size of pneumatic cylinders, it may be possible to excite certain portions of a complicated structure like an automobile which otherwise may be inaccessible to shakers. Another attractive feature of this technique is the low cost associated with its setup which may be a factor of 5-10 times cheaper than small shakers. As the force capacity and number of shakers increases, the cost factor will also increase further. A small problem with this method is that it is relatively noisy (acoustically) as compared to shaker testing. The source of noise is the hard metal to metal impact of the piston of pneumatic cylinder and the plunger of solenoid valve with their respective end caps (seats). The expulsion of air from the exhaust port of solenoid valve also generates some noise. In order to reduce the noise due to the impacts, some vibration isolator or cushioning can be used. The flow control valve in the current setup cannot be used as it affects the entire return stroke, which distorts the excitation force. Another way to minimize the noise is the sound insulation of the whole setup by packaging it in some kind of a noise reduction box.

6.2 Recommendations for future work

There is scope for improving the FRF measurements in the FTE configuration if a force transducer having both the dynamic and static measurement capability and a signal conditioner with DC coupling is used. The motivation for it is that the pneumatic setup operates relatively quietly in the FTE configuration as compared to the FTS configuration. However, the mass of the tip will remain a limitation for the FTE configuration. It will also be interesting to explore the possibility of generating other types of excitations like periodic random, pseudo random with the pneumatic testing technique, although these signal types are expected to
give similar results a for conventional shaker testing. A considerable amount of time is spent in preparing the setup if the excitation direction is different from vertical, similar to conventional shaker testing. In order to reduce the setup time, a sturdy fixture that can rotate the exciter in three dimensions as well as adjust its height could be developed. The method that is currently used for adjusting the distance between exciter and structure is relatively crude as it is done by placing rubber pads or some other vibration isolator beneath the exciter mounting. The relatively lower weight of the pneumatic exciter should make this easier than for conventional shakers.
7 References


8 Appendices

Appendix A - MATLAB scripts

A.1 - Code for generating a single impact for finalizing pulse width

clc;
clear all;
close all;
Flines=input('Enter number of positive frequency lines:');
if isempty(Flines), Flines=1600, end;
Fmax=input('Enter max frequency: ');
if isempty(Fmax), Fmax=400, end;

df=Fmax/Flines % Frequency resolution
T=1/df % Total time
Fnq=Fmax/0.78125 % Nyquist frequency
Fsamp=2*Fnq % Sampling frequency
dt=1/Fsamp % Sampling time interval (dt)
bs=T/dt % Block size
pwidth1=input('Enter pulse width for 1st exciter: '); % Pulse width is in multiples of dt
if isempty(pwidth1), pwidth1=10, end;
t=linspace(0,T,bs);
xdata=t;
ydata=zeros(3, bs);
ydata(1,1:pwidth1)=5;
figure;
plot(t,ydata(1,:));
pwidth2=input('Enter pulse width for 2nd exciter: '); % Pulse width is in multiples of dt
if isempty(pwidth2), pwidth2=10, end;
t=linspace(0,T,bs);
xdata=t;
ydata(2,1:pwidth2)=5;
figure;
plot(t,ydata(2,:));
pwidth3=input('Enter pulse width for 3rd exciter: '); % Pulse width is in multiples of dt
if isempty(pwidth3), pwidth3=10, end;
t=linspace(0,T,bs);
xdata=t;
ydata(3,1:pwidth3)=5;
figure;
plot(t,ydata(3,:));
cd 'D:/'
save Impact_excitation xdata ydata
A.2 - Code for generating the pulse train

clc;
clear all;
close all;
% DSP parameters
Flines=input('Enter number of positive frequency lines:');
if isempty(Flines), Flines=1600, end;
Fmax=input('Enter max frequency');
if isempty(Fmax), Fmax=400, end;
df=Fmax/Flines  % Frequency resolution
T=1/df  % Total time
Fmax=Fmax/0.78125 % Nyquist frequency
Fsamp=2*Fmax  % Sampling frequency
dt=1/Fsamp  % Sampling time interval
bs=T/dt  % Block size
SynAvg=input('Enter number of synchronous averages:');
if isempty(SynAvg), SynAvg=30, end;
CycAvg=input('Enter number of cyclic averages(greater than 1):');
if isempty(CycAvg); CycAvg=4,
end;
Burst=input('Enter burst length in percentages : ');
if isempty(Burst), Burst=100, end;
BS=bs*SynAvg*CycAvg;
T=T*SynAvg*CycAvg;
t=linspace(0,T,BS);

% Generation of random pulse train for 1st exciter
S1=zeros(1,BS);
maxpul1=input('Enter maximum pulse width for 1st exciter in multiples of dt');
if isempty(maxpul1), maxpul1=35, end;
mintd1=input('Enter minimum time delay for 1st exciter in multiples of dt');
if isempty(mintd1), mintd1=30, end;
seed1=RandStream('mt19937ar','Seed',1); % Seed has been set to 1;
ii=0;jj=0;t1=0;t2=0;iigap=0;jjgap=0;temp1=0;kk=0;
for kk= 1:SynAvg
    ii=0;
    temp1=zeros(1,CycAvg*bs);
    while ii<(Burst/100)*CycAvg*bs
        ii=ii+1;
        iigap=maxpul1-ceil(rand(seed1)*maxpul1); % Values of pulse width will be always
        temp1(ii:ii+iigap-1)=5;  % less than maxpul1
        if size(temp1,2)>(CycAvg*bs),
            temp1=temp1(1,1:CycAvg*bs);
            break
        end
        t1=S1(ii:ii+iigap-1);
        jj=size(t1,2)+ii;
        jjgap=mintd1+ ceil(rand(seed1)*mintd1); % Values of time delay will be always
        temp1(jj:jj+jjgap-1)=0;  % more than or equal to mintd1
    end
end
if size(temp1,2)>(CycAvg*bs),
    temp1=temp1(1,1:CycAvg*bs);
    break
end

t2=S1(jj:jj+jjgap-1);
ii=size(t2,2)+jj-1;
end
S1(((kk-1)*(CycAvg)*bs+1):kk*CycAvg*bs)=temp1;
end

% Plot the pulse train for 1st exciter
figure;
plot(t,S1)
title('Voltage Vs time for 1st exciter')
ylabel('Magnitude(Volt)');
xlabel('Time (sec)');

% Generation of random pulse train for 2nd exciter
ii=0;jj=0;t1=0;t2=0;iigap=0;jjgap=0;temp1=0;kk=0;
S2=zeros(1,BS);
maxpul2=input('Enter maximum pulse width for 2nd exciter in multiples of dt');
if isempty(maxpul2), maxpul2=35, end;
mintd2=input('Enter minimum time delay for 2nd exciter in multiples of dt');
if isempty(mintd2), mintd2=30, end;
seed2=RandStream('mt19937ar','Seed',2); % Seed has been set to 2;

for kk= 1:SynAvg
    temp1=zeros(1,CycAvg*bs);
    ii=0;
    while ii<(Burst/100)*bs*CycAvg
        ii=ii+1;
        iigap=maxpul2-ceil(rand(seed2)*maxpul2); % Values of pulse width will be always less than or equal to maxpul2
        if size(temp1,2)>(CycAvg*bs),
            temp1=temp1(1,1:CycAvg*bs);
            break
        end
    t1=S2(ii:ii+iigap-1);
    jj=size(t1,2)+ii;
    jjgap=mintd2+ceil(rand(seed2)*mintd2); % Values of time delay will be always more than or equal to mintd2
    if size(temp1,2)>(CycAvg*bs),
        temp1=temp1(1,1:CycAvg*bs);
        break
    end
    t2=S2(jj:jj+jjgap-1);
    ii=size(t2,2)+jj-1;
end
S2((kk-1)*CycAvg*bs+1:kk*CycAvg*bs)=temp1;
% Plot the pulse train for 2nd exciter
figure;
plot(t,S2)
title('Voltage Vs time for 2nd exciter')
ylabel('Magnitude(Volt)');
xlabel('Time (sec)');

% Generation of random pulse train for 3rd exciter
ii=0;jj=0;t1=0;t2=0;iigap=0;jjgap=0;temp1=0;kk=0;
S3=zeros(1,BS);
maxpul3=input('Enter maximum pulse width for 3rd exciter in multiples of dt ');
if isempty(maxpul3), maxpul3=35, end;
mintd3=input('Enter minimum time delay for 3rd exciter in multiples of dt');
if isempty(mintd3), mintd3=30, end;
seed3=RandStream('mt19937ar','Seed',3); % Seed has been set to 3;

for kk= 1:SynAvg
    ii=0;
    temp1=zeros(1,CycAvg*bs);
    while ii<(Burst/100)*bs*CycAvg
        ii=ii+1;
        iigap=maxpul3-ceil(rand(seed3)*maxpul3); % Values of pulse width will be always
        temp1(ii:ii+iigap-1)=5; % less than or equal to maxpul3
        if size(temp1,2)>(CycAvg*bs),
            temp1=temp1(1,1:CycAvg*bs);
            break
        end
        t1=S3(ii:ii+iigap-1);
        jj=size(t1,2)+ii;
        jjgap=mintd3+ceil(rand(seed3)*mintd3); % Values of time delay will be always
        temp1(jj:jj+jjgap-1)=0; % more than or equal to mintd3
        if size(temp1,2)>(CycAvg*bs),
            temp1=temp1(1,1:CycAvg*bs);
            break
        end
        t2=S3(jj:jj+jjgap-1);
        ii=size(t2,2)+jj-1;
    end
    S3((kk-1)*CycAvg*bs+1:(kk*CycAvg*bs))=temp1;
end

% Plot the pulse train for 3rd exciter
figure;
plot(t,S3)
title('Voltage Vs time for 2nd exciter')
ylabel('Magnitude(Volt)');
xlabel('Time (sec)');
xdata=t;
ydata=zeros(3,BS);

ydata(1,:)=S1(1:BS);
ydata(2,:)=S2(1:BS);
ydata(3,:)=S3(1:BS);
% Choose location for saving the data
cd 'D:\'
save pneumatic_hammer xdata ydata
Appendix B – Components used in the pneumatic setup

1. Pneumatic Cylinder (single acting, spring return)

Cylinders of two sizes are used in the setup for exciting different structures.

Table 8-1: Pneumatic cylinder specifications

<table>
<thead>
<tr>
<th>Bore (in.)</th>
<th>Stroke (in.)</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’ site</th>
</tr>
</thead>
<tbody>
<tr>
<td>7/16</td>
<td>2</td>
<td>automation direct</td>
<td>A07020SP</td>
<td>3</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
</tr>
<tr>
<td>3/4</td>
<td>3</td>
<td>automation direct</td>
<td>A12030SP</td>
<td>3</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
</tr>
</tbody>
</table>

Figure 8-1: Pneumatic cylinder

2. Solenoid valve

In this set up, the type of solenoid valve needed is 3 way, 2 position. However, 5 way, 2 position valves already available in the lab are used for this purpose. The extra exhaust port in the valves is closed by plugs.

Table 8-2: Solenoid valve specifications

<table>
<thead>
<tr>
<th>Valve function</th>
<th>Port size (in.)</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’ s site</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 way 2 position</td>
<td>1/8</td>
<td>The Pneumatic Store</td>
<td>081SA43AM00061</td>
<td>3</td>
<td><a href="http://www.thepneumaticstore.com">www.thepneumaticstore.com</a></td>
<td>Close the extra exhaust port with a plug</td>
</tr>
</tbody>
</table>
3. Push to connect pneumatic fitting (male elbow)

Each pneumatic cylinder requires a different pneumatic fitting due to different sizes of threads.

Table 8-3: Pneumatic fitting specifications

<table>
<thead>
<tr>
<th>Tube Outer Dia. (O.D.) (in.)</th>
<th>Thread size</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’s site</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td># 10-32 MNPT</td>
<td>automation direct</td>
<td>ME14-10N</td>
<td>3</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
<td>To be fitted on 7/16 inch bore cylinder</td>
</tr>
<tr>
<td>1/4</td>
<td>1/8 in MNPT</td>
<td>automation direct</td>
<td>ME14-18N</td>
<td>9</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
<td>To be fitted on 3/4 inch bore cylinder and solenoid valves</td>
</tr>
</tbody>
</table>

4. Nylon tubing

Table 8-4: Tubing specifications

<table>
<thead>
<tr>
<th>Tube O.D. (in.)</th>
<th>Tube I.D. (in.)</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’s site</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>0.180</td>
<td>automation direct</td>
<td>N14NAT100</td>
<td>1 (100 ft. roll)</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
</tr>
</tbody>
</table>
5. Screw adjustable flow control valve

Table 8-5: Flow control valve specifications

<table>
<thead>
<tr>
<th>Tube Outer Dia. (O.D.) (in.)</th>
<th>Thread size (in.)</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’s site</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>1/8</td>
<td>automation direct</td>
<td>FVS14-18N</td>
<td>3</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
<td>To be fitted on the unplugged exhaust port of the solenoid valve</td>
</tr>
</tbody>
</table>

Figure 8-4: Flow control valve

6. Air pressure regulator and filter

One end of the pressure regulator is connected to the compressor with hoses and the other end is connected to the solenoid valves with tubes. Appropriate couplings as shown in Figure 8-5 should be used for connections.

Table 8-6: Pressure regulator specifications

<table>
<thead>
<tr>
<th>Operating pressure (PSI)</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Company’ site</th>
</tr>
</thead>
<tbody>
<tr>
<td>20 - 130</td>
<td>automation direct</td>
<td>AFR-2233-D</td>
<td>1</td>
<td><a href="http://www.automationdirect.com">www.automationdirect.com</a></td>
</tr>
</tbody>
</table>
7. Electronic circuit board

The electronic circuit can actuate five solenoid valves at a time and its schematic is shown in Figure 8-6. (Reference: https://playground.arduino.cc/uploads/Learning/solenoid_driver.pdf). However, in the current setup only three valves are used. While designing the circuit, the space occupied by the heat sink is not taken into account which makes it impossible to package five transistors on the circuit board. If more than three valves have to be actuated, then it is recommended to modify the current circuit board design (Figure 8-7) for packaging. ExpressPCB program is used for designing the board.

<table>
<thead>
<tr>
<th>Component</th>
<th>Make</th>
<th>Part no.</th>
<th>Quantity</th>
<th>Remarks</th>
<th>Company’ site</th>
</tr>
</thead>
<tbody>
<tr>
<td>Printed circuit board</td>
<td>expresspcb</td>
<td>-</td>
<td>1</td>
<td>-</td>
<td><a href="http://www.expresspcb.com">www.expresspcb.com</a></td>
</tr>
<tr>
<td>Transistor</td>
<td>Digikey</td>
<td>TIP102-ND</td>
<td>5</td>
<td>Heat sink should be used with every transistor</td>
<td><a href="http://www.digikey.com">www.digikey.com</a></td>
</tr>
<tr>
<td>Diode</td>
<td>Digikey</td>
<td>1N4004FSCT-ND</td>
<td>6</td>
<td>-</td>
<td><a href="http://www.digikey.com">www.digikey.com</a></td>
</tr>
<tr>
<td>Resistor</td>
<td>Standard 1</td>
<td>KΩ resistor</td>
<td>5</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Capacitor</td>
<td>Standard 10μF capacitor</td>
<td>-</td>
<td>1</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
Figure 8-6: Schematic of the electronic circuit

Figure 8-7: Circuit board design {only red and green color curves (traces, pads and letters) should be etched}
Appendix C – Demonstration of the pneumatic testing technique

The demonstrations of the pneumatic technique on a rectangular plate and trimmed truck body are posted on YouTube. The following links can be used to watch the demonstrations:

Rectangular plate: https://www.youtube.com/watch?v=_Q5_vsYsZRe

Truck body: https://www.youtube.com/watch?v=9kDDbO0bCu0