



**MARMARA UNIVERSITY**  
**INSTITUTE FOR GRADUATE STUDIES**  
**IN PURE AND APPLIED SCIENCES**



**DESIGN OF A NOVEL VIBRATION  
ISOLATION FOR LARGE DIESEL ENGINES**

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**AND ELECTRICAL CABINETS**

**TUNCER EVCI**

**MASTER THESIS**

Department of Mechanical Engineering

**Thesis Supervisor**

Prof. Dr. Aykut Kentli

ISTANBUL, 2019



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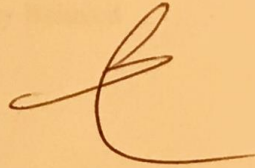
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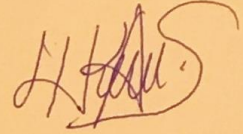
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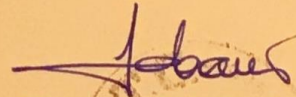
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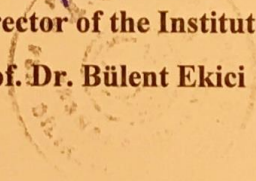
**APPROVAL**

Marmara University Institute for Graduate Studies in Pure and Applied Sciences Executive Committee approves that Tuncer EVCI be granted the degree of Master of Science in Department of Mechanical Engineering on 13.11.2019

Resolution no: 2019/23-02



**Director of the Institute**  
**Prof. Dr. Bülent Ekici**





To my Beloved

## **ACKNOWLEDGMENTS**

This thesis has been a great adventure for me, and I have gained a lot of experiences that are crucial for my career.

I would like to thank my supervisor Prof. Dr. Aykut Kentli who had patience and made great suggestions for me to complete this thesis. He always presented his deep knowledge and useful recommendations.

Then, I would like to thank Vibrol Oy and Wärtsilä for providing me possibility to do this thesis project. I am so grateful to Kari Saine, from Wärtsilä, for his supervision and useful recommends throughout the thesis project. I would also like to express my sincere thanks to Jarkko Keinänen, Kalle Vehviläinen and Aki Kinnunen, from Vibrol, for their great helps. It would be impossible to complete this project without their special helps.

Finally, I would like to present my all grateful feelings and special love to Tülay Yılmaz, to be always with me. Her support and love made me always motivated and strong.

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# ÖZET

## Büyük Dizel Motorlar ve Elektrik Kabinleri İçin Titreşim İzolasyonu Tasarımı

Titreşim ve gürültü problemleri her zaman mühendisler için zorlu bir uğraş olmuştur. Mühendisler hala bu sorunlara çözüm bulmaya devam ediyorlar. Gemicilikte ve enerji üretiminde kullanılan büyük dizel motorları için de çözülmeyi bekleyen pek çok titreşim sorunu var.

Bu çalışmanın ana amacı gemicilikte ve enerji santrallerinde kullanılan sekiz silindirli bir dizel motor için motor takozu ve motor üzerinde yer alan elektrik kabini için titreşim izolasyon elemanları geliştirmek ve tasarlamaktır. Bunun için hem motor hem de elektrik kabininde çelik kablolu titreşim izolatörleri kullanılmıştır. İki farklı sistem için farklı parçalar tasarlanmış olsa da bu iki parça için temel titreşim teorisi geçerli olduğu için aynı tez içerisinde incelenmiştir.

Bu çalışmada, öncelikle, katı cisim doğal frekansları için matematiksel hesaplamalar yapılmıştır. Elde edilen doğal frekans değerleri yalıtım elemanı seçiminde kullanılmış ve deneysel sonuçlar ile karşılaştırılmıştır.

Çelik kablolu yalıtım elemanları için Vibrol, Tuusula'da temel katı cisim doğal frekans ölçümleri yapılmıştır. Daha sonra Wärtsilä, Vaasa'da motor testleri ve hidrolik sarsıcı testleri gerçekleştirilmiştir. Son olarak, elektrik kabini için geliştirilen yalıtım elemanları için Vaasa Üniversitesi titreşim laboratuvarında elektromanyetik sarsıcı ile deneyler yapılmıştır.

Bu tez çalışması boyunca, çelik kablolu titreşim izolatörleri ve bu parçaların dizel motor ve elektrik kabini üzerindeki uygulaması incelendi. Bunun için karmaşık deney sonuçları incelendi ve analiz edildi.

# **ABSTRACT**

## **Design of a Novel Vibration Isolation for Large Diesel Engines and Electrical Cabinets**

Noise and vibration problems have been always challenging for engineers in many different areas. Engineers still have these problems to solve them.

The main purpose of this thesis is to develop new vibration isolators for Wärtsilä's large diesel engines and electric cabinets that are on the engines. Steel wire rope vibration isolation elements have been used to develop and design engine mounts and isolator for electric cabinet. Two different elements have been studied in this thesis project because basic vibration theory is same for both engine mount and electric cabinet isolators.

In this study, firstly, mathematical model of the systems has been prepared for estimating rigid body modes. Then, experimental results have been obtained and compared the mathematical results.

Basic rigid mode experiments for the wire rope elements have been conducted in Vibrol, Tuusula. Engine tests and hydraulic shaker test for the engine mounts have been performed in Wärtsilä, Vaasa. On the other hand, electromagnetic shaker tests for electric cabinet isolators have been conducted in the vibration laboratory of University of Vaasa.

Throughout this thesis project, wire rope vibration isolators and their application on large diesel engines and electric cabinet was investigated. Finally, Complicated results of the experiments has been discussed.

## SYMBOLS

|            |                                |
|------------|--------------------------------|
| $\omega_n$ | Natural frequency              |
| $\omega_d$ | Damped Natural Frequency       |
| $C_c$      | Critical Damping               |
| $\zeta$    | Percentage of Critical Damping |
| $[M]$      | Generalized mass matrices      |
| $[C]$      | Generalized damping matrices.  |
| $[K]$      | Generalized stiffness matrices |
| $K_v$      | Dynamic Stiffness              |
| $K_s$      | Static Stiffness               |

## ABBREVIATIONS

SDOF : Single Degree of Freedom

MDOF : Multi Degree of Freedom

ODS : Operational Deflection Shape

WRI : Wire Rope Isolator

NVH : Noise, Vibration and Harshness

IC : Internal Combustion

FFT : Fast Fourier Transform

FRF : Frequency Response Function

TF: : Transfer Function

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# 1 INTRODUCTION

## 1.1 Vibration of Mechanical Structures

Understanding of single degree of freedom (SDOF) systems is essential to understand the basics of modal analysis and mechanical vibrations. Most physical structures can be described with mass, spring and damper with an excitation however they are continuous. Mass and spring stores the energy of the system that comes from excitation as potential energy and kinetic energy. Damper, on the other hand, symbolizes the dissipation of the energy. Mass, spring and damper are called idealized elements of a physical system. [1]

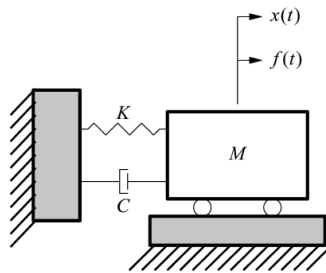


**Figure 1-1** SDOF Mass-Spring System

Equation of motion equation is used for calculation of natural frequency and damping factor of the system. Idealized elements – mass, spring and damper – affect the natural frequency and damping factor. Equation of motion also indicates this effect.

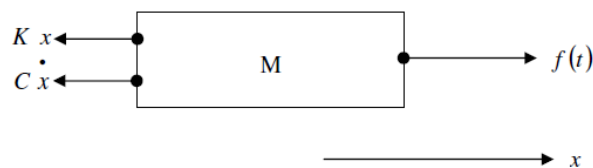


**Figure 1-2** Free Body Diagram for Equation of Motion

$$M \ddot{x} + C \dot{x} + Kx = 0 \quad \text{Equation of motion for Free Damped Vibration}$$

$$(1.1) \quad M \ddot{x} + C \dot{x} + Kx = f(t) \quad \text{Equation of motion for Forced Damped Vibration}$$

(1.2)

$$\omega_n = \sqrt{\frac{k}{m}} \quad ; \text{ Natural frequency} \quad (1.3)$$

$$\zeta = \frac{c}{\sqrt{2km}} \quad ; \text{Percentage of Critical Damping} \quad (1.4)$$

$$\sigma = \zeta \omega_n \quad ; \text{Damping factor} \quad (1.5)$$

$$C_c = 2m\omega_n \quad ; \text{Critical Damping} \quad (1.6)$$

$$\omega_d = \omega_n \sqrt{1 - \zeta^2} \quad ; \text{Damped Natural Frequency} \quad (1.7)$$

Natural frequency,  $\omega$ , is generally written in two different units. These are Hertz (Hz) and rad/s. Critical damping is a phenomenon that the system has no oscillation at this level. Damping factor can be shown as percent of critical damping. If damping in the system is equal to critical damping, the system has no oscillation and arrives the equilibrium position at the shortest time. When a system oscillates at natural frequency, damping ratio is equal to zero. It means that there is no damping at natural frequency.

## 1.2 Frequency Response Functions (FRFs)

The frequency response functions (FRFs) are complex functions so they cannot be shown on a single two-dimensional plot. FRFs can be indicated in different type of formats. The Bode plot and the Nyquist plot are the some widely used forms of the frequency response functions. The Bode plot indicates the magnitude and phase in one plot on the other hand the Nyquist plot shows real vs. imaginary parts of the frequency response functions. The Bode plot has a peak in amplitude at natural frequency and phase angle is equal to  $90^\circ$ . The Nyquist plot has a circle shape on the graph. [2]

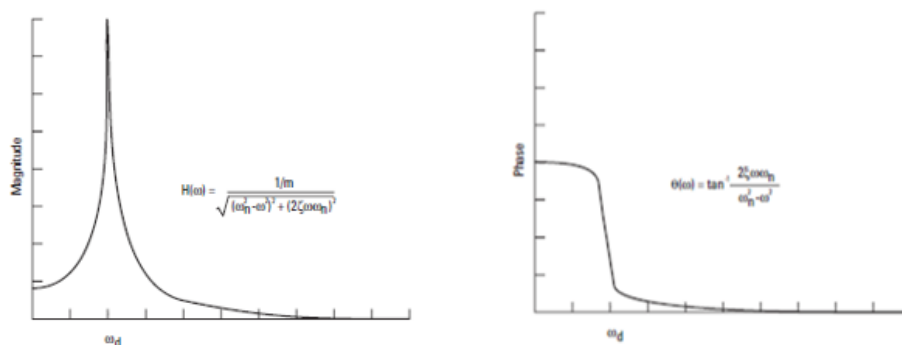


**Figure 1-3** The Bode Plot of SDOF System

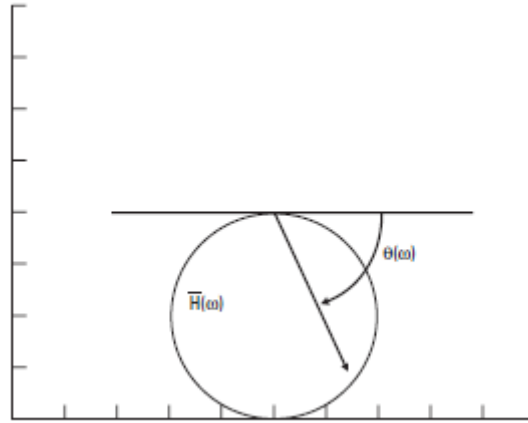


**Figure 1-4** The Nyquist Plot of SDOF System

FRFs are the ratio of the force and response. It can be described basically ratio of output and input. We can assume that displacement, velocity or acceleration are outputs or responses and excitation force is the input of the system. Therefore, we have three types of FRFs. These are ratio of displacement and force, velocity and force and finally ratio of acceleration and force. Inverse of the ratios can be defined as FRF. Hence, we have six type of FRFs.

$$\alpha(\omega) = \frac{\text{displacement}}{\text{force}} \quad \text{Compliance (Receptance)} \quad (1.8)$$

$$Y(\omega) = \frac{\text{velocity}}{\text{force}} \quad \text{Mobility} \quad (1.9)$$

$$A(\omega) = \frac{\text{acceleration}}{\text{force}} \quad \text{Accelerance (Inertance)} \quad (1.10)$$

**Table 1-1** Definition of FRFs

| Response Type | FRF                      | Inverse FRF          |
|---------------|--------------------------|----------------------|
| Displacement  | Compliance or Receptance | Dynamic Stiffness    |
| Velocity      | Mobility                 | Mechanical Impedance |
| Acceleration  | Accelerance or Inertance | Apparent Mass        |

Velocity is mathematically derivation of displacement and acceleration is also mathematically derivation of velocity or second derivation of displacement. Due to the fact that displacement, velocity and acceleration are mathematically interrelated to each other, any one of FRF can be transformed to any other FRF. [2]

### 1.2.1 MDOF Systems

It is not easy to find a SDOF real structure since most structures using in daily life or real systems or structures have an infinite number of degrees of freedom. Multi Degree of Freedom (MDOF) systems can be described as simply linear superposition of SDOF systems. Concept of SDOF extend to a more common form, MDOF, with n degrees of freedom to obtain MDOF systems. Each SDOF system has an equation of motion to describe the system. A MDOF system with n degree of freedom contain n SDOF system so it means that the system has n equations of motions. Therefore, in order to define a MDOF system we need more than one equation. It is described by matrix of equations. If the matrix of equations equal to zero, it means that there is no excitation and it is called as free vibration. If there is force excitation, it is called as forced vibration. [3]

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f\} \quad (1.11)$$

$[M]$  is NxN generalized mass matrices.

$[C]$  is NxN generalized damping matrices.

$[K]$  is NxN generalized stiffness matrices. All generalized matrices are symmetric.

Where  $\{\ddot{x}\}$ ,  $\{\dot{x}\}$  and  $\{x\}$  are Nx1 column matrices and time dependent responses.  $\{\ddot{x}\}$  is vector of acceleration response,  $\{\dot{x}\}$  is vector of velocity responses and  $\{x\}$  is vector of displacement responses.  $\{f\}$  is time dependent excitation force.

If we assume that the system is undamped and free vibration, the equation of motion can be written as:

$$[M]\{\ddot{x}\} + [K]\{x\} = 0 \quad (1.12)$$

Assume a solution  $\{x(t)\} = \{\bar{X}\}f(t)$

Here  $\{\bar{X}\}$  is a vector of constants, i.e.  $\{\bar{X}\} = \begin{Bmatrix} \bar{X}_1 \\ \bar{X}_2 \\ \vdots \\ \bar{X}_n \end{Bmatrix}$

$$\{x(t)\} = \{\bar{X}\}e^{i\omega t}$$

Noting that  $e^{i\omega t} \neq 0$  for any t, we obtain,

$$[[K] - \omega^2[M]]\{\bar{X}\} = 0 \quad (1.13)$$

In order to obtain a trivial solution, we should satisfy

$$\det[[K] - \omega^2[M]] = 0, \text{ it is the determinant of } [[K] - \omega^2[M]].$$

This equation is called as characteristic equation and it is also algebraic. Eigenvalues of this system give us N possible real solution which are  $\omega_1^2, \omega_2^2, \dots, \omega_n^2$ . These are undamped natural frequencies of the system. Eigenvectors of this system give us N possible vectors, and these are the mode shapes of the system.  $(\psi_r)$  (r = 1, 2, 3..., N). Each eigenvector contains N elements.

Solution of free vibration is stated in two NxN matrices. These are

$$[\Psi] = [\{\psi_1\} \ \{\psi_2\} \ \dots \ \{\psi_N\}]$$

And

$$[\omega_r^2] = \begin{bmatrix} \omega_1^2 & 0 & \dots & 0 \\ 0 & \omega_2^2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & \omega_N^2 \end{bmatrix} \quad (1.14)$$

These equations or matrices defines the system by way of natural frequencies and mode shapes or in other words modal properties. Now, we have two different models. First model is commonly known as Spatial Model which contains two matrices  $[M]$  and  $[K]$  (generalized

mass and stiffness matrices). Second model is usually known as Modal Model which is constituted by two eigenmatrices  $[\omega_r^2]$  and  $[\Psi]$ . [2]

It should be noted that the eigenvalue matrix is unique, but the eigenvector is not. It is very important to state at this step. Although natural frequencies are constant values or quantities, mode shapes are dependent on an indeterminate scaling factor. Shape of the mode is not influenced by this factor; it affects only amplitude of the vibration. It means that two matrices, A and B, written below exactly defines the same mode of vibration but in different amplitudes.

$$A = \begin{bmatrix} 1 \\ 2 \\ 3 \end{bmatrix}, B = \begin{bmatrix} 3 \\ 6 \\ 9 \end{bmatrix} \quad (1.15)$$

In this phase, it is very critical to know that Modal Model has very significant properties known as Orthogonal Properties.

Orthogonal properties of an undamped free MDOF system is constituted on the Modal and Spatial Model relationship.

$\{\Psi\}^T [M] \{\Psi\} = [m_r]$  and  $\{\Psi\}^T [K] \{\Psi\} = [k_r]$  where m and k are known as modal mass and stiffness of rth mode of the structure or system. Because of the orthogonality properties, being linearly independent of mode shape vectors generates a basis in the N-dimensional space.

$$\frac{\{\Psi\}^T [K] \{\Psi\}}{\{\Psi\}^T [M] \{\Psi\}} = \frac{[k_r]}{[m_r]} = \omega_r^2 \quad (1.16)$$

and

$$\{\psi_s\}^T [M] \{\psi_r\} = 0 \quad (1.17)$$

$$\{\psi_s\}^T [K] \{\psi_r\} = 0 \quad (1.18)$$

Because of the orthogonality properties, the eigenvector matrices (mode shape vectors) are linearly independent. Thus, in the N-dimensional basis, mode shape vectors generate a basis. [4]

Mode shapes of a MDOF system are linearly independent. It means that each mode shape can be written without linear dependence of other mode shapes. In order to indicate this linearly independence condition of mode shapes, orthogonality of the mode shapes could be used. Therefore, orthogonality of the mode shapes is significant to ensure a unique solution of a system.

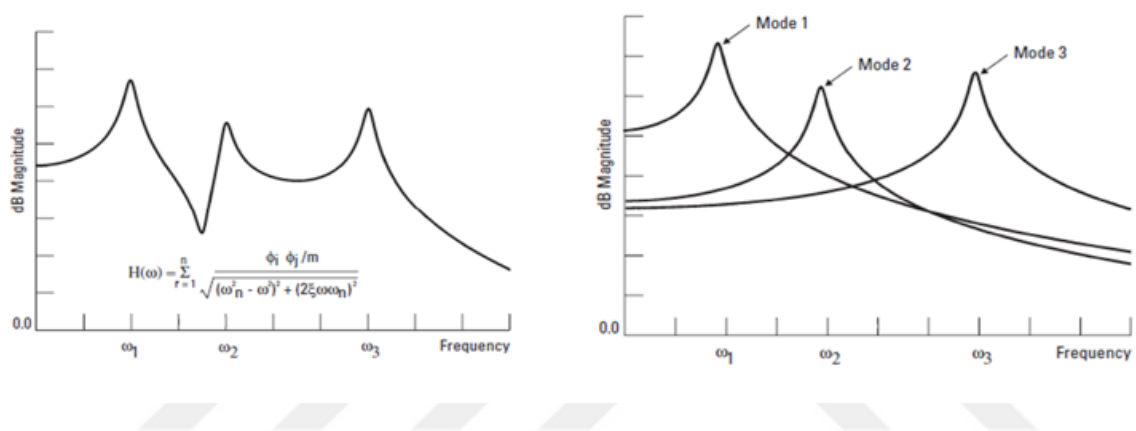


**Figure 1-5** MDOF System FRF and SDOF modal Contributions to MDOF system FRF

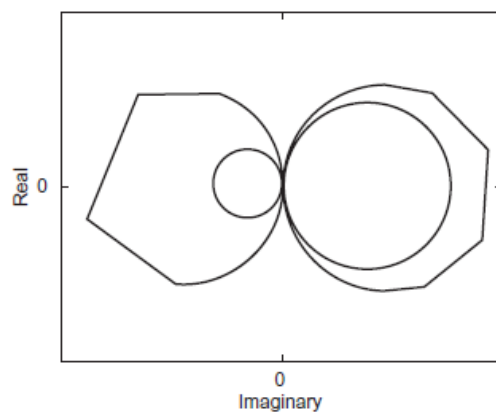


**Figure 1-6** The Nyquist Plot of a FRF of the MDOF system

### **1.3 Natural Frequencies of Structures**

Every physical substances and structures have their unique natural frequencies. When a certain external force, which causes vibration at a natural frequency, to a body, mass or structure, it will be prone to vibrate even after stop applying the force. Natural frequencies depend on stiffness and mass. SDOF systems have only one natural frequency but MDOF systems with  $N$  degree of freedoms have  $N$  natural frequencies.

Name of the phenomenon vibrating a structure at its natural frequency is resonance. Even small forces could cause high level vibration response when the structure is in resonance. Calculating or determining of the natural frequencies are important in aspect of structural engineering since the larger the displacements, the higher the stresses created in the structures. Therefore, high amplitude vibration responses while resonating can cause failure in the structures, bodies or objects. [5]

### **1.4 Elastic and Rigid Body Modes**

Structures, objects or systems have different mode shapes at different frequencies. we can define a mode shape as a unique pattern of motion performed by structures at a specific frequency. These mode shapes can be calculated mathematically, obtained from a finite element software or measured by vibration measurements. It is important to note that modal analysis does not give displacement, velocity or acceleration of a mode shape, it only gives us frequency of a mode and its shape. Acceleration, velocity and displacement could change with amplitude of excitation force. Mode shapes can be grouped under two different headings. These are elastic mode shapes and rigid body mode shapes. Rigid body modes occur at lower frequencies than the elastic modes. Therefore, this situation is a simple way to separate rigid body modes and elastic modes. [6]

Elastic modes are also known as normal modes. General shapes of elastic modes are bending and torsional but in higher frequencies, these mode shapes are very complex. For these mode shapes, every point of systems or structures move relatively to each other so as a result of this relative motion, internal deformation or stress will be observed. [7]

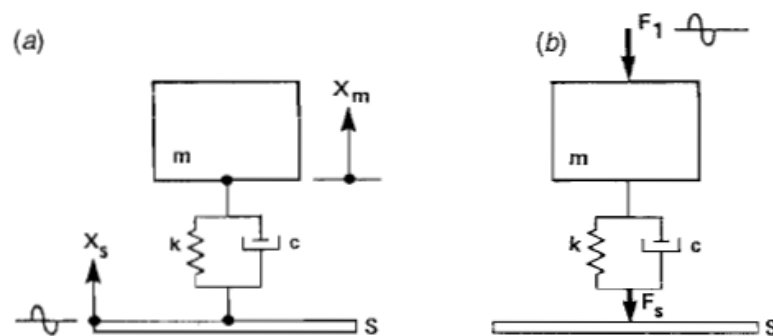
### **1.5 Structure-borne Vibration**

The vibration creating by a vibratory source flows through structures as mechanical energy. This mechanical energy flow is the result of vibratory motion, dynamic stresses and interaction between them. The instantaneous energy flow rate per unit area in a specific

direction is called structure-borne intensity (SI) which is a vectoral quantity. This vector of intensity is generally different in different directions and this intensity vector has different magnitude for every direction in time domain. In order to calculate total energy flow rate in a section, normal component of intensity vector should be integrated. In order to determine the location of vibratory sources and vibration propagation path, experimental measurements should be performed. Not only vibratory sources and vibration propagation paths but also the structure intensity or energy flow rate can be obtained from with experimental measurements. On the other hand, the structure intensity or energy flow rate can be calculated by computation depending on the design or problem. Computational methods could be used for predicting noise and vibration paths and levels during design stage. Besides, experimental measurements are significant supplementary methods or tools for vibration diagnostics or source and path identification to solve engineering problem. [8]

## 1.6 Vibration Isolation

The main purpose of the vibration isolation is to reduce vibratory motions or forces that are transferred from one mechanical component to another component or structure. The elements



that are using for vibration isolation are called vibration isolators and these vibration isolators consist of rubber (elastomeric) elements, of springs or of combination of both elastomeric elements and springs. As distinct from dampers, vibration isolators are using for attenuating the transfer of vibration. To protect sensitive components or equipment from structural vibration and to reduce the vibrations creating by a vibratory source which is in or attached are usually main challenges of engineers. Vibration isolation commonly is applied to the structures or machines to overcome these problems. In terms of integrity of the systems, comfort of the customers and control of radiation of structural vibration and noise, reduction of vibration might be desired. In more sensitive applications, two-stage isolation would be

**Figure 1-7** Vibration Isolator Representation Mass, Spring and Damper [10]

better solution for greater attenuation [9],[25].

One dimensional, ideally linear and translational mass-spring-damper system would be a useful example for better understanding of vibration isolation that is shown in Figure 1-7, vibration isolator can be represented with combination of massless damper, with viscous damping coefficient  $c$ , and massless spring, with stiffness  $k$ . Mass  $m$  is non-rotational rigid body and move only vertically. [26]

In condition (a), the assumption is that bottom support  $S$  vibrates vertically, and purpose of the isolator is to transmit vibratory motion in minimum level from support to mass  $m$ . Therefore, the mass  $m$  will have resulting vibratory motion adequately small. In condition (b), a force  $F_1$  is applied to mass  $m$  vertically and the main purpose of the vibration isolator is to keep the force  $F_s$  which is acting on the support in minimum level. Therefore, support  $S$  will have resulting vibratory motion acceptably small. [10]

To get a better understanding of vibration isolation, we need to clarify one term called *transmissibility*. Transmissibility is the ratio of the employed excitation that is transferred to the component which is protected from vibratory motion or forces. In the case of (a) corresponding to Fig. 7a, the interested issue is the motion that is transmitted to the mass. Therefore, transmissibility of the motion can be defined as  $T_m = X_m/X_s$  where  $X_m$  refers to resulting amplitude of displacement of the mass  $m$  and  $X_s$  refers to amplitude of displacement of the base support  $S$ . Besides, in the case of (b) corresponding to Fig. 7b, transmissibility of the force can be defined as  $T_f = F_s/F_1$  where  $F_1$  refers to acting force on the rigid mass and  $F_s$  is the transmitted from mass  $m$  to support  $S$ .

Formulation of *transmissibility* can be investigated for two different cases. The first case is the undamped isolator which means that there is no damping. The second case is considered that isolator have viscous damping and this damping would be characterized with loss factor,  $\eta$ .

The expression of transmissibility for undamped isolator is

$$T = \frac{1}{\left| \left( \frac{f}{f_n} \right)^2 - 1 \right|} \quad (1.19)$$

where  $f_n$  represents to natural frequency of mass and spring system without any damping and  $f$  refers to excitation frequency. The expression of the natural frequency,  $f_n$ , is that

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (1.20)$$

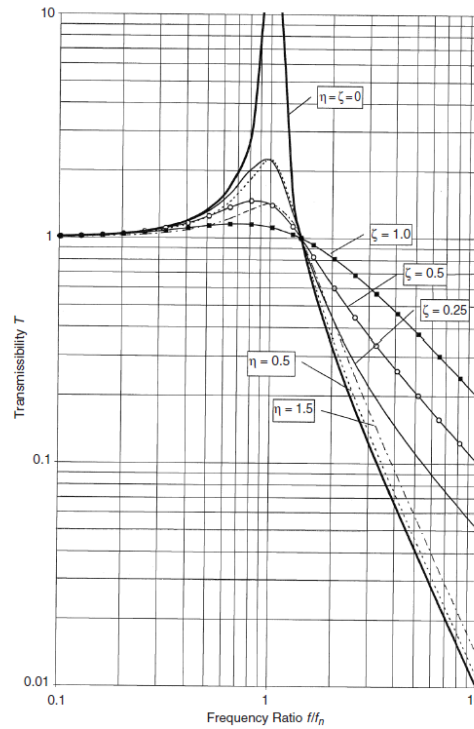
As we described before,  $k$  is the stiffness of the spring and  $m$  refers to rigid mass  $m$ .

In practical applications, it is not easy to experience an undamped condition. So, to gain a more general formulation which is closer to real condition than undamped conditions, damping should be taken into consideration. Therefore, expression for more general condition where the vibration isolator have damping characterized by loss factor  $\eta$ ,

$$T = \sqrt{\frac{1 + \eta^2}{\left[\left(\frac{f}{f_n}\right)^2 - 1\right]^2 + \eta^2}} \quad (1.21)$$

where the loss factor  $\eta = \frac{2\pi f c}{k} = 2\zeta\left(\frac{f}{f_n}\right)$ . In this formulation,  $\zeta$  is called damping ratio and it is ratio of damping factor of the system  $c$ , and critical viscous damping coefficient of the system,  $c_c$ . So,  $\zeta = c/c_c$ . In the Fig.8, you can see how the transmissibility changes with frequency ratio in a logarithmic scale graph. [10]

**Figure 1-8** Transmissibility vs. Frequency ratio  $f/f_n$



### 1.6.1 Isolation Efficiency

To define isolation performance, isolation efficiency  $I$ , could be used sometimes. Isolation efficiency is calculated by  $I = 1 - T$ . For example, if we have 0.5 transmissibility, isolation efficiency is 0.95. Isolation efficiency is commonly stated by percent. So, in this example, isolation efficiency is 95% and transmissibility is 5%. [10]

### 1.7 Active Vibration Isolation

Active vibration control means that producing and applying of forces in an equal and reverse trend against the forces offended by external vibration. Low levels of vibration are required to reduce stresses, effects of fatigue and high-level noise and vibration and to operate smoothly for most structures, machines and engines etc. More precise processes can be obtained and sustained with noise isolation or cancellation. Application of passive noise control is generally used for high frequency vibrations; it is rarely applied for low frequencies. Active vibration isolation systems can meet requirements of lower frequency applications.

Main principle of active vibration control is based upon creating identical force in opposite phase to the imposed vibration. As a result of this cancellation process, imposed vibration will be eliminated.

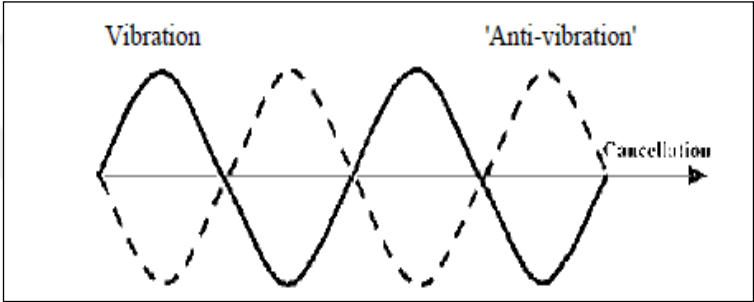
Let's assume that we have sinusoidal vibration:

$$F_1 = X \sin(\omega t) \quad (1.21)$$

The force of actuator would be:

$$F_a = -(X + x) \sin(\omega t + \theta) \quad (1.22)$$

where  $x$  and  $\theta$  are amplitude and the phase difference between sinusoidal and actuator forces. Using with these two forces, resultant force could be calculated by relation:



$$F_1 + F_a = X \sqrt{4 \left(1 + \frac{x}{X}\right) \sin^2\left(\frac{\theta}{2}\right) + \left(\frac{x}{X}\right)^2} \quad (1.23)$$

Different types of sensors and actuators can be used for active vibration isolators as well as that different levels of stiffness can be applied. The selection of elements and most satisfied strategy is decided according to application. Important parameters for this selection process would be interested frequency range, required isolation level, available design space, environmental properties, precision of the feedback loop and the budget. On the other hand,

**Figure 1-9 Active Vibration Control [11]**

active isolators can be separated by three different types according to stiffness levels as low, medium and stiff strategies. Each type has advantages for different applications. Although soft active isolation strategy is more responsive to external disturbances than the others, it provides advantage of broadband isolation. For the medium strategy, high stability and passive isolation in high frequencies are main advantages.

However, it has low dynamic stiffness and compatibility for positioning processes. Main advantage of high stiff active isolation is the high robustness to external disturbances. But then again, robust connection between active isolator and ground should be coupled to get rid of stability problems. [12]

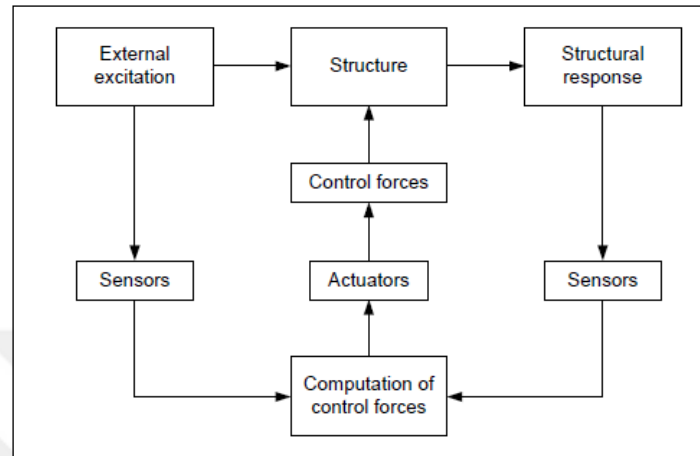


**Figure 1-10** Basic Schema of Active Vibration Control [11]

### 1.8 Passive Vibration Isolation

In not only different mechanical systems, but also engines and suspension systems, passive vibration isolators are commonly used nowadays. General aim of using these isolators is to absorb both external disturbances forces at low frequencies and vibratory motions at high frequencies. Even though passive vibration isolators are commonly less efficient than active vibration isolators, passive isolators aren't required any excitation, sensors or external power. As a result of this, passive vibration isolators provide the opportunity of low-cost robust and modular designs to the engineers. Therefore, passive isolators are common and essential application for solving vibration problems. [13]

As discussed above, active control and passive control are distinguished from each other by the required power or energy. Active control systems use external power or energy for their actuator systems. Besides, passive systems don't require any added energy or power. However, electromechanical controls can be used for passive systems for adjusting but still these systems have passive isolation mechanisms. [27]

Isolation is provided by passive methods as restricting the ability of vibrations to be connected to the system, part or object that is necessitated isolation. This is applied with practicing a mechanical connection that absorb or dissipate the vibrational energy to prevent vibration transfer. Generally, springs, fluids, negative stiffness components and elastomers are used in

passive isolation systems [14]. Dampers and absorbers would be used for passive vibration control but in this study, we will examine the wire rope spring elements.

### 1.8.1 Wire Rope Vibration Isolators

Wire-rope vibration isolators are type of passive isolator and they are isolation and mounting elements that are made of stranded wire cables that are in frictional contact with each other. This wire cable is generally coiled in the helical or polycal shape and fastened by metal parts to maintain helical or polycal shape. Wire rope isolators have typically non-linear hysteretic behavior. They are generally and widely used in a good number of industrial applications to avoid both vibration and shock isolation. Machineries, military applications, aviation and



**Figure 1-11** Helical and Polycal Shape Wire Rope Isolators

space industry, electronic and other sensitive equipment are examples of some of these applications. Although, helical shape isolations are commonly used for heavy applications, polycal shape isolators are preferred for relatively sensitive conditions like electronics and micro mechanical systems. These isolators provide the option of mounting in any direction and orientation for the isolation of protected systems, parts etc. because of their isolation ability in all directions and three planes. [15]

By means of the damping that comes inherently from friction between the strands of wire rope isolator, they are effective in isolation and control of vibration and shock. Restoring hysteretic force of the isolation elements is created by internal sliding and extruding substantially. In the case of that isolator is enforced to change in shape due to the vibratory or shock forces, friction will be come out of interaction between individual cables. Kinetic energy is converted to heat energy with this friction. Thus, wire rope isolator dissipates or absorbs the unintended vibration. On the other hand, both damping and stiffness characteristics of a wire cable isolator can be specified by number of the strands, cable diameter, cable length, number of the loops, diameter of the wire rope, wire material, height and width of the isolator and lay angle of the strands.

Main advantage of these type isolators is resistance of environmental conditions such as extreme temperatures, radiation, salt, grease etc. Primarily wire rope isolators can retain their performance in operations that have wide range of temperature between  $-100^{\circ}\text{C}$  to  $250^{\circ}\text{C}$  [16], [17], [18].

### 1.8.2 Behavior of Wire Rope Elements

Steel wire rope insulators are consistent mounting sets of steel wire ropes twisted with steel cables generally held by metal restraints. Their own natural damping is supplied by flexure hysteresis, for example friction between the thin steel cables of the wire rope. Geometric characteristics of the wire rope elements such as the diameter of the cable, the number of coils, length, height and width of the wire rope isolators affect the wire rope isolators hysteretic behavior. [28]

The phenomenon of hysteresis can be observed in several systems. It can be seen, for example, in dry friction systems, wire rope isolators and ferro-magnetic systems. Hysteresis is the main reason of the energy dissipation. Although hysteresis can be modeled in front, firstly, properties of the wire rope elements should be defined. This is possible by mapping the applied force to wire rope spring and deflection of the wire rope spring. In Figure 1-12,  $w$  represents the applied force to the spring and  $u$  the deflection of the wire rope isolator under force. By doing this force-deflection measurement, a hysteresis loop is obtained. In case of increasing the deflection from  $u_1$  to  $u_2$ , applied force is the curve ADC in Figure 1-12. This is the loading curve of hysteresis loop. On the other hand, when deflection moves from  $u_2$  to  $u_1$  applied force shows the behavior of the curve CBA and this is the unloading curve of hysteresis loop. [28,29]

Rate independency is one of the other properties of the hysteresis. It means that hysteresis

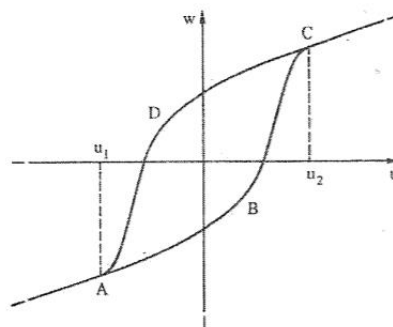


**Figure 1-12** Example of a Stiffness Curve of a Wire Rope Element [28]

curve is unvarying with respect to time. At any time while force is applying to the wire rope

element, force is only dependent to displacement. In addition to that, it is independent from derivative of the deflection. In other words, it is not dependent to velocity of the deflection. Therefore, we can say that hysteresis is rate independent [28]. Moreover, hysteresis model of the wire rope elements can help for getting the loss of rate independent energy because of sliding friction and deformation of the material. In addition, energy loss data can be used to calculate the fatigue life. [30]

Finally, wire rope isolators hysteretic behavior is independent of loading frequency and deflection rate. However, loading amplitude has a strong effect on the hysteresis curve.[31]

### **1.9 Operational Deflection Shape (ODS)**

Vibrational motions of a structure under operating conditions are determined in Operational Deflection Shape (ODS) analysis. Unlike modal analysis, there is no input force in ODS analysis, therefore acting forces or excited forces are not measured for ODS analysis. Acceleration, velocity or displacement can be the output of the system. An animation is being obtained by using the output data [32,33].

ODS is generally compared with modal analysis and ODS analysis has some advantages with respect to modal analysis. These are:

- Input forces are not needed.
- The investigated structure or system examine the actual acting forces.
- Any linear model is not necessary.
- Actual boundary conditions are applied to analysis.

Besides the advantages, ODS has some disadvantages with respect to modal analysis. These are:

- ODS can only present the cyclic motion at a specific frequency. It is not possible to obtain motions of different frequencies.
- Dynamic mode cannot be obtained in principal.

Acceleration can be used for measured output in case of the running machine and vibrational motion of the machine can be animated with this acceleration. Structures are modeled as wire frame structures in ODS software. In this study Dewesoft is used for ODS analysis to get animations. Firstly, measurement points are determined and then structure is modeled by connecting these points with lines. Collecting signals from measurement points are used as

input data and motion of the structure at measured frequency is animated. Different measurements are needed for the motion of different frequencies.

In order to get an ideal ODS analysis result, all points should be measured simultaneously under constant operating conditions. Measuring all point simultaneously depends on the number of the points and directions that needs to measure and channel number of data acquisition system. Since these channels might not be enough to measure all point at the same time. In that condition, measurements can be done separately under the same operating conditions. One of the accelerometers should be kept in a fixed location as a reference point during the separate measurements. By this way, complete animation of the motion of the structure can be obtained. But it should be considered that separate measurement technic is valid for only frequency or order domain measurements. It is not useful for time domain measurements.

Different types of ODS can be performed. These are:

- Time Domain ODS
- Run Up/Down ODS
- Spectral ODS

Spectral and Run Up/Down ODS analyses generally are used for investigating the vibrational motion at a specific frequency. Continuous rotating machine measurements are typical examples for this type of ODS's. For the diesel engines, constant rpm working conditions should be measured to make spectral of run-up/down ODS analysis. On the other hand, time ODS is preferred to understand the vibrational motion of a structure as a function of time. In case of movements with high accelerations and decelerations, time ODS is a useful method since it can be seen how motion of the structure changes by time [32].

Consequently, operating deflection shape of the structures is constituted by the measured responses such as acceleration, displacement etc. of a vibrating structure at a specific frequency or at a specific point in time. Linearity of the structure's response is assumed to express the ODS of a vibrating structure in terms of its mode shapes or its FRF's [34].

### **1.10 Mass and Stiffness Matrix**

In recent years, the study of the dynamics of elastic structures is very popular working area. Despite of the fact that most of the initial work focused on fatigue and lifespan testing, the most recent efforts have focused on analytical modeling and simulation of mechanical

structures. In order to predict dynamic characteristics of mechanical structures, accurate mathematical models are necessary. Mathematical models can be obtained both analytically and experimentally. For the complex structures, mathematical models could be inaccurate due to difficulties in real conditions such as boundary conditions, damping, acting forces etc. [36]

Recently, it is relatively easier to get quick, accurate and complete measurements of mechanical structure's behavior under different vibrational conditions by using computers and computational techniques such as The Fast Fourier Transform (FFT) algorithm. Even complex mode shapes can be identified rather than depending on and being constrained by normal mode concept and modal responses of many modes of mechanical structures can be easily measured simultaneously. Moreover, response matrix of an entire system can be measured too. This response matrix consists of mass, stiffness and damping matrices of the equivalent model. In this equivalent model, main assumptions are that stiffness is proportional to displacement, damping is proportional the velocity and mass is a lump mass [35]. On the other hand, despite the new developments in experimental techniques, experimental methods can be faced with problems because of limited measurement points and limited frequency range [36].

In order to understand techniques of obtaining mass and stiffness matrices, theoretical background that is explained in below is needed.

We can assume that motion of a linear physical system can be defined by  $n$  set of second order differential equations which is for each degree of freedom in time domain,

$$M \ddot{x} + C \dot{x} + Kx = f(t) \quad (1.24)$$

where the dots represent the derivation with respect to time.  $f(t)$  is the vector of applied force, and  $x = x(t)$  is the resulting displacement vector. On the other hand,  $M$ ,  $C$ , and  $K$  are the mass, damping, and stiffness matrices. These matrices are  $n$  by  $n$  matrices.

By taking the Laplace transform of the equations set of the system,

$$B(s)X(s) = F(s), \quad (1.25)$$

$$\text{where } B(s) = Ms^2 + Cs + K \quad (1.26)$$

$s$  is the universal Laplace variable.  $X(s)$  is the displacement vector in the Laplace domain and  $F(s)$  is the applied force vector in Laplace domain.  $B(s)$  is the system matrix.

Let's define a transfer matrix that is  $H(s)$  and it is defined as

$$H(s) = B(s)^{-1} \quad (1.27)$$

After the transfer function assumption, displacement matrix can be written with respect to applied force matrix. That is,

$$H(s)F(s) = X(s) \quad (1.28)$$

Now, it can be inferred that transfer matrix  $H(s)$  is a transfer function. The elements of system matrix  $B$  are quadratic functions of  $s$ . Because of  $H = B^{-1}$ , the elements of transfer function  $H$  are rational fractions in  $s$ .

In case of the assumption that the poles of  $H$  are of unit multiplicity, then  $H$  can be written as

$$H(s) = \sum_{k=1}^{2n} \left( \frac{a_k}{s-s_k} \right) \quad (1.29)$$

The poles happen when  $s = s_k$  that is zeros of determinant of  $B$ . In this system each pole has an  $n$  by  $n$  residue matrix  $a_k$  defining its spatial behavior. There is always  $2n$  poles for an  $n^{\text{th}}$  order oscillatory system, but they will appear in complex conjugate pairs [35].

To write the mass and stiffness matrix, let's recall that  $B(s) = Ms^2 + Cs + K$  and we can write that [35]:

$$\text{Stiffness matrix is } K = B(0) = H(0)^{-1} \quad (1.30)$$

and

$$\text{Mass matrix is } M = \frac{1}{2} B''(0) \quad (1.31)$$

### 1.11 Octave Bands

In order to perform a frequency analysis of time variable signals, there are many different instruments. Frequency analysis bands are standardized to make easier the comparison of measurements and analyses. The most common method is the octave band analysis for frequency analysis. Upper frequency band is approximately two times higher than the lower frequency band for the octave bands. Center frequency that is the geometric mean of the upper and lower limit of the band is used for defining the octave bands [37]. In this study, one-third

octave analysis has been used for structural borne analysis. Therefore, only one-third octave is explained.

The one-third octave band center frequencies are selected with respect to their logarithms that are one-tenth decade numbers. For example, logarithms of 31.5, 40 and 50 are 1.5, 1.6 and 1.7. In Figure 1-13, center frequencies for preferred octave bands are presented.

| <b>Band number</b> | <b>Octave band center frequency</b> | <b>One-third octave band center frequency</b> | <b>Band limits</b> |              |
|--------------------|-------------------------------------|-----------------------------------------------|--------------------|--------------|
|                    |                                     |                                               | <b>Lower</b>       | <b>Upper</b> |
| 14 }               | 31.5                                | 25                                            | 22                 | 28           |
| 15 }               |                                     | 31.5                                          | 28                 | 35           |
| 16 }               |                                     | 40                                            | 35                 | 44           |
| 17 }               | 63                                  | 50                                            | 44                 | 57           |
| 18 }               |                                     | 63                                            | 57                 | 71           |
| 19 }               |                                     | 80                                            | 71                 | 88           |
| 20 }               | 125                                 | 100                                           | 88                 | 113          |
| 21 }               |                                     | 125                                           | 113                | 141          |
| 22 }               |                                     | 160                                           | 141                | 176          |
| 23 }               | 250                                 | 200                                           | 176                | 225          |
| 24 }               |                                     | 250                                           | 225                | 283          |
| 25 }               |                                     | 315                                           | 283                | 353          |
| 26 }               | 500                                 | 400                                           | 353                | 440          |
| 27 }               |                                     | 500                                           | 440                | 565          |
| 28 }               |                                     | 630                                           | 565                | 707          |
| 29 }               | 1000                                | 800                                           | 707                | 880          |
| 30 }               |                                     | 1000                                          | 880                | 1130         |
| 31 }               |                                     | 1250                                          | 1130               | 1414         |
| 32 }               | 2000                                | 1600                                          | 1414               | 1760         |
| 33 }               |                                     | 2000                                          | 1760               | 2250         |
| 34 }               |                                     | 2500                                          | 2250               | 2825         |
| 35 }               | 4000                                | 3150                                          | 2825               | 3530         |
| 36 }               |                                     | 4000                                          | 3530               | 4400         |
| 37 }               |                                     | 5000                                          | 4400               | 5650         |
| 38 }               | 8000                                | 6300                                          | 5650               | 7070         |
| 39 }               |                                     | 8000                                          | 7070               | 8800         |
| 40 }               |                                     | 10000                                         | 8800               | 11300        |
| 41 }               | 16000                               | 12500                                         | 11300              | 14140        |
| 42 }               |                                     | 16000                                         | 14140              | 17600        |
| 43 }               |                                     | 20000                                         | 17600              | 22500        |

**Figure 1-13** Preferred Octave and One-Third Octave Frequency Bands [37]

1/3 Octave Band Filters are very similar in principal to the other preferred Octave Band filters. The only difference is that each of the octave bands is split into three sections to get a more comprehensive description of the frequency content of the noise or vibration. Despite of

the fact that it is seldomly used for occupational noise measurement, one-third octave analysis is a useful tool in many environmental, building acoustics and noise control applications in addition. Although it is used for generally noise measurements, one-third octave analysis can also be used for vibrational measurements.



## 2 LITERATURE SURVEY

There are huge number of studies in the vibration isolation area. Engineers and scientists have been contributing for over the years academically. In this chapter, academic studies that are related to vibration isolation of diesel engines are explained. In addition to marine and power generation, other type of diesel engine has also been investigated since basic vibration theory is same. Then, wire rope vibration isolator applications will be written.

### 2.1 Vibration Isolation for Diesel Engines

Large diesel engine for marine and power generation applications have many rubber mounts examples. Lee at al. [19] investigated a resilient rubber mount application for a diesel engine that is installed on a ship. In their study, both theoretical and experimental studies have been conducted and a design criterion has been proposed in the end of their work.

Yehia at all. [20] conduct a study on six-cylinder marine diesel engine and prepared guidelines that are helpful for the researchers. In addition to engine, propulsion and shafting system have been also analyzed. To avoid resonance problems, they offered a procedure for FEM analysis.

Besides, Ramos [21] made a useful research with a rubber automotive engine mount to have a better understanding and create a basis for new developments. He examined the non-linear behavior of isolation mounts and remarked dynamic properties of rubber engine mounts and made comparison between mathematical and experimental works. In the end, he observed satisfactory performance of the mounts.

Adhau and Kumar [22] were conducted a research to solve mounting problems and eliminate the negative effects of vibrational excitations coming from an internal combustion engine with a new design of engine mount. Furthermore, they were aimed to obtain deep understanding of measuring damping and dynamic properties of the engine while designing an engine mount.

Tienhaara [23] studied vibration behavior of marine diesel engines and his aim is to keep under control the vibration of the marine diesel engine. In his study, he prepared a finite element model to estimate vibration levels close to real life condition. With the simulation results, he aimed to provide a guideline for design process to keep vibration levels under acceptable vibration levels.

## 2.2 Wire Rope Isolator Applications

Wire rope isolators are used in many different applications by mechanical engineers to isolate and reduce unintended vibration. Space, aviation, defense, energy and marine industries can be counted between usage area of these isolators. There are plenty number of application examples with wire rope isolation elements in academic literature. In this section, these applications that have been viewed will be provide to readers to form a better understanding of wire rope isolators.

Balaji et al. [16] have conducted a review study with wire rope element. They provided behavior of wire rope elements under both cyclic and monotonic loading conditions. In the study, hysteretic nonlinear nature of wire rope elements has been represented in addition to cyclic and monotonic loading conditions. Bouc-Wen Model has been used to explain this hysteretic behavior.

Leenen [24] has been performed a research study to identify and analyze the hysteretic behavior of wire ropes under both static and dynamic conditions. In this study, static and dynamic stiffness experiments have been carried out to figure out the wire rope's nature and a favorable model has been proposed. With the dynamic tests, sine sweep excitation has been performed and the Modified Bouc-Wen model has been used for this study. Finally, results of experiments have been discussed and a new model has been presented.

Tinker and Cutchins [17] carried a study to figure out dynamic characteristics of helical wire rope isolation elements. With the difference of other wire rope elements studies written above, phase trajectories and FRF curves have been achieved in addition to stiffness and hysteretic curves. Effect of the dry friction between steel wires on damping has been observed in this study. Proportional the relation between dissipated energy and excitation amplitude has been figured out.

Ni et. al [18] performed an experimental study for not only vertical direction but also shear and roll behavior of the wire rope vibration isolator. The differences between compression and shear, roll stiffnesses of the wire rope elements. For the shear and roll mode of the elements, similar hysteretic curves have been obtained experimentally although compression behavior of the wire rope were different as expected.

### 3 METHODS AND MATERIALS

In this chapter, in addition to mathematical calculation software and results, stiffness tests of wire rope elements, engine measurements, hydraulic shaker experiment for the engine and electromagnetic shaker experiment of electric cabinet will be explained. Calculation results will be presented under this section. However, experimental result will be analyzed in next chapter.

#### 3.1 Mathematical Calculations

In this section, mathematical calculations of engine mount and electric cabinet will be examined. To perform the calculations, WinRemo which is an internal calculation software has been used. In WinRemo, rigid body natural frequencies can be calculated with correct stiffness and mass moment of inertia values. After the comparison between experimental results, it is observed that calculated natural values has a promising match with experimental values. Moreover, calculated natural frequencies are hints to understand which frequency refers to which rigid body mode.

##### 3.1.1 Mathematical Calculations of Engine Mounts

In the beginning of the calculations catalogue values has been used to get natural frequencies and with the help of these calculations' possible suitable elements could be selected. After completing the stiffness tests of selected wire rope elements, calculations have been updated with experimental stiffness values.

Mass moment of inertia values for the engine has been taken from Wärtsilä and hand calculations have been done for the electric cabinet.

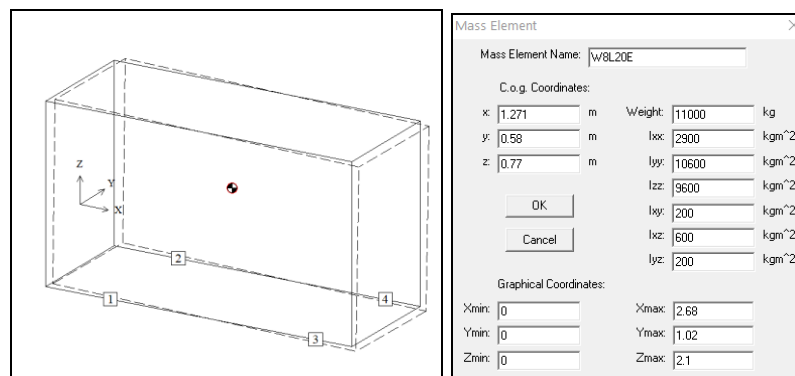


**Figure 3-1** Engine Model and Center of Gravity and Inertial Properties of the Engine

In Figure 3-1, symbolic engine model and inertial properties of the engine in WinRemo is presented. Calculation model is basically mass and spring system. The engine has been considered as a rigid mass in shape of rectangular prism. On the other hand, wire rope elements have been considered as springs with linear and uniform stiffness values.

After using catalogue values, natural frequencies obtained from calculations are available in the Table 3-1. According to iterative calculations with many wire rope elements, wire rope element has been selected. Therefore, WinRemo is a beneficial tool to make engineering calculations and selecting suitable isolators for large diesel engines.

From the catalogue values, compression values are used for vertical stiffness values and shear and roll stiffness values are used for longitudinal and transversal stiffness values. Because of that shear and roll stiffness values are very close to each other, it is assumed that these stiffness values are identical.

| Compression |             |                             |                            |                                      |                                  | Shear/Roll |             |                             |                            |                                      |                                  |
|-------------|-------------|-----------------------------|----------------------------|--------------------------------------|----------------------------------|------------|-------------|-----------------------------|----------------------------|--------------------------------------|----------------------------------|
| Curve       | Model       | Max Static Load<br>lbs. (N) | Max Deflection<br>in. (mm) | Kv<br>(vibration)<br>lbs./in. (kN/m) | Ks<br>(shock)<br>lbs./in. (kN/m) | Curve      | Model       | Max Static Load<br>lbs. (N) | Max Deflection<br>in. (mm) | Kv<br>(vibration)<br>lbs./in. (kN/m) | Ks<br>(shock)<br>lbs./in. (kN/m) |
| 1           | WR20-200-08 | 1,450 (6 450)               | 1.25 (31,8)                | 9,570 (1 676)                        | 4,850 (849)                      | 1          | WR20-200-08 | 790 (3 514)                 | 1.35 (34,3)                | 2,990 (524)                          | 2,990 (524)                      |
| 2           | WR20-300-08 | 1,230 (5 471)               | 1.50 (38,1)                | 7,190 (1 259)                        | 3,480 (609)                      | 2          | WR20-300-08 | 680 (3 025)                 | 1.60 (40,6)                | 2,140 (375)                          | 2,140 (375)                      |
| 3           | WR20-400-08 | 1,140 (5 071)               | 1.65 (41,9)                | 6,310 (1 105)                        | 2,880 (504)                      | 3          | WR20-400-08 | 590 (2 624)                 | 1.70 (43,2)                | 1,760 (308)                          | 1,760 (308)                      |
| 4           | WR20-600-08 | 945 (4 204)                 | 1.95 (49,5)                | 4,690 (821)                          | 2,030 (356)                      | 4          | WR20-600-08 | 480 (2 135)                 | 2.05 (52,1)                | 1,230 (215)                          | 1,230 (215)                      |

**Figure 3-2** Catalogue Table of Wire Rope Elements for Engine Mount

After making iterative calculations with trial error method, suitable wire rope element has been selected and this wire rope element's stiffness values are used for WinRemo calculations. Three wire rope elements have been used for each engine mount because of the load capacity of the wire rope elements. Therefore, stiffness values have been multiplied by three for both static and dynamic condition.  $K_s$  is shock vibration stiffness and it is used as static stiffness value.  $K_v$  is dynamic stiffness value. Dynamic stiffness values have more realistic results since they are considered in working conditions in general.

**Table 3-1** Rigid Body Mode Natural Frequencies Obtained from Mathematical Calculations in WinRemo with Dynamic and Static Stiffness Values

| Rigid Body Mode | Natural Frequency - ( $K_v$ ) | Natural Frequency - ( $K_s$ ) |
|-----------------|-------------------------------|-------------------------------|
| Longitudinal    | 3.4 Hz                        | 2.9 Hz                        |
| Transversal     | 2.8 Hz                        | 2.3 Hz                        |
| Vertical        | 6.7 Hz                        | 5.0 Hz                        |
| Rolling         | 10.2 Hz                       | 8.9 Hz                        |
| Pitching        | 7.9 Hz                        | 6.4 Hz                        |
| Yawing          | 4.4 Hz                        | 4.3 Hz                        |

After completing selection process, stiffness tests have been done and calculations have updated with experimental values. Updated values of natural frequencies are available in Table 3-2. Stiffness values taken from stiffness tests are available in stiffness tests section.

Updated stiffness values and calculated stiffness values will be compared with experimental engine measurement's results. 1mm Dynamic measurement stiffness values have been used as dynamic stiffness and static stiffness test results have been used for calculations with static stiffness. As it can be observed from Table 3-2, dynamic stiffness values make a dramatic change on natural frequencies during calculation process.

**Table 3-2** Updated Rigid Body Mode Natural Frequencies Obtained in WinRemo with Dynamic and Static Stiffness Values Obtained from Stiffness Tests

| Rigid Body Mode | Natural Frequency - ( $K_v$ ) | Natural Frequency - ( $K_s$ ) |
|-----------------|-------------------------------|-------------------------------|
| Longitudinal    | 6.1Hz                         | 3.1 Hz                        |
| Transversal     | 4.8 Hz                        | 2.6 Hz                        |
| Vertical        | 11.9 Hz                       | 6.7 Hz                        |
| Rolling         | 17.6 Hz                       | 9.6 Hz                        |
| Pitching        | 14.3 Hz                       | 7.8 Hz                        |
| Yawing          | 7.5 Hz                        | 3.8 Hz                        |

### 3.1.2 Mathematical Calculations of Electric Cabinet Isolators

In the beginning of the calculations, catalogue values have been used to get natural frequencies and with the help of these calculations' possible suitable elements could be selected. After completing the stiffness tests of selected wire rope elements, calculations have been updated with experimental stiffness values as well as the engine mount calculations. Two different wire rope element has been selected from the catalogue.

Mass moment of inertia values for the electric cabinet has been obtained by hand calculations. Although geometric properties are constant, weight of the cabinet has been changed. Three different weights are calculated for both isolators of the electric cabinet. 57kg, 75kg and 90kg weights are calculated for softer wire rope element and 90kg, 120kg and 150kg weights are calculated for stiffer wire rope elements. By changing the mass, different types of electric cabinet are calculated at the same time. Therefore, these calculations can make for not only the selected engine's electric cabinet but also the cabinets of different size engines.

Mass moment of inertia calculations process has been explained in related section. As it is expected, increasing the mass with constant geometric size increases the mass moment of the structure as it can be seen in Table 3-3.

**Table 3-3** Mass Moment of Inertias for Electric Cabinet

|                          | 57kg | 75kg | 90kg | 120kg | 150kg |
|--------------------------|------|------|------|-------|-------|
| Ixx (kg.m <sup>2</sup> ) | 12.3 | 16.2 | 19.5 | 26    | 32.5  |
| Iyy (kg.m <sup>2</sup> ) | 7.8  | 10.2 | 12.3 | 16.4  | 20.5  |
| Izz (kg.m <sup>2</sup> ) | 14.1 | 18.5 | 22.2 | 29.6  | 37    |
| Ixy (kg.m <sup>2</sup> ) | 5    | 6.6  | 7.9  | 10.5  | 13.1  |
| Ixz (kg.m <sup>2</sup> ) | 2.8  | 3.7  | 4.5  | 6     | 7.5   |
| Iyz (kg.m <sup>2</sup> ) | 4    | 5.2  | 6.3  | 8.4   | 10.5  |

As well as the engine calculation process, calculations started with iterative trial error method for different type of wire rope elements. After the iterative calculations, two different wire rope elements have been selected from the catalogue and initial calculation process has been finished by this way.

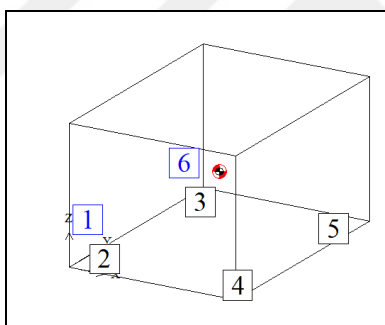


**Figure 3-3** Electric Cabinet WinRemo Calculation Model

In Fig. 3-3, calculation model of electric cabinet has been presented. Cabinet sizes are 500mm in longitudinal direction, 700mm in transversal direction and 400mm in vertical direction. Six connection points have been selected to fix the electric cabinet on the engine. Four points are located below the cabinet and two points are located behind the electric cabinet. The wire rope elements behind the cabinet are called side stoppers. The term of ‘side stopper’ is used to identify these elements after that point in this thesis.

**Table 3-4** Dynamic and Static Stiffness Values of Electric Cabinet Wire Rope Elements

|                                  | Softer      |            | Stiffer     |            |
|----------------------------------|-------------|------------|-------------|------------|
|                                  | Compression | Shear/Roll | Compression | Shear/Roll |
| Static (K <sub>s</sub> ) – kN/m  | 23          | 7.9        | 58          | 35         |
| Dynamic (K <sub>v</sub> ) – kN/m | 58          | 7.9        | 136         | 35         |

Side stoppers are located to control the vibrant motion in longitudinal direction and to restrict the vibration levels upper side of the cabinet. Walls of the cabinet are thin metal plates. Therefore, there is a possibility that walls behave like a spring under working conditions. For that reason, corners of the cabinet are selected to eliminate this problem.

Table 3-4 is the stiffness table for electric cabinet wire rope isolators and data is taken from catalogue. As so in engine mount wire rope elements, compression values are used for vertical direction and shear/roll values are used for other translational directions. Natural frequencies of the electric cabinet are presented in Table 3-5.

**Table 3-5** Rigid Body Natural Frequencies of Electric Cabinet Obtained from WinRemo with Catalogue Stiffness Values

|              | Natural Fre. For Softer Element (Hz) |      |      | Natural Fre. For Stiffer Element (Hz) |       |       |
|--------------|--------------------------------------|------|------|---------------------------------------|-------|-------|
|              | 57kg                                 | 75kg | 90kg | 90kg                                  | 120kg | 150kg |
| Longitudinal | 13.7                                 | 12.7 | 10.9 | 17.0                                  | 14.7  | 13.5  |
| Transversal  | 8.3                                  | 8.0  | 6.7  | 10.6                                  | 9.2   | 9.1   |
| Vertical     | 11.1                                 | 9.7  | 8.9  | 13.7                                  | 11.9  | 11.8  |
| Rolling      | 4.9                                  | 4.1  | 3.7  | 5.7                                   | 4.9   | 4.4   |
| Pitching     | 9.2                                  | 7.1  | 7.4  | 11.6                                  | 10.0  | 9.8   |
| Yawing       | 5.1                                  | 4.3  | 4.1  | 6.4                                   | 5.5   | 5.0   |

Only dynamic stiffness value calculations have been performed for electric cabinet isolators since static stiffness values are not close to working condition properties. By eliminating static stiff values workload has been reduced.

### 3.2 Stiffness Tests of Wire Rope Elements

In this section, stiffness tests of wire rope elements will be presented for both engine mounts and electric cabinet. Stiffness tests have been conducted in laboratory of Armeka Oy at Valkeakoski, Finland for both engine mount isolators and electric cabinet wire rope elements. The engine mount experiment has been done on April 26, 2019 and electric cabinet isolators experiment has been done on July 9, 2019.

Test machine has 100kN maximum force and 300mm stroke. Total weight of the test machine is 1450kg. The machine has both force and displacement sensor. During applying the force to the tested part, force and displacement are recording at the same time. Testing machine has the capability of both static and dynamic tests to the selected parts. By this way, stiffness

values of wire rope elements are obtained experimentally. Firstly, static tests were conducted then dynamic tests were performed. Isolators are tested in three coordinate system directions. After obtaining stiffness values, calculations have been updated with new values.

### 3.2.1 Stiffness Tests of Engine Mount Isolators and Electric Cabinet Isolators

Engine mount isolators are tested in vertical, longitudinal and transversal directions. Testing machine can only apply the force in vertical direction. Therefore, adapter brackets were used for transversal and longitudinal direction tests. For the vertical direction, only one element has been used. Wire rope elements was fixed to bottom plate and stroke arm was squeezed the part in vertical direction. However, two elements were used at the same time to eliminate any moment for transversal and longitudinal direction tests. In Figure 3-4, testing machine and measurement types are presented.



**Figure 3-4** Stiffness Test Pictures

For the vertical direction static test, maximum displacement is 10mm and maximum applied force is 17551N. In addition to this, maximum is displacements are 20mm for both transversal and longitudinal direction tests. In addition to this, maximum applied force is 12503N for longitudinal direction and 10362N for transversal direction.

After the static tests, dynamic tests were performed. Dynamic tests were done in six different amplitude. These are 0.2mm, 0.3mm, 0.4mm, 0.5mm, 1mm and 2mm. Stiffness values that are obtained from the experiments are presented in Table 3-6.

**Table 3-6** Stiffness Values of Engine Mount Wire Rope Elements from Stiffness Tests

|               | Engine Mount Wire Rope Elements (kN/m) |             |          |
|---------------|----------------------------------------|-------------|----------|
|               | Longitudinal                           | Transversal | Vertical |
| Dynamic-0.2mm | 3150                                   | 2750        | 13500    |
| Dynamic-0.3mm | 2730                                   | 2110        | 10150    |
| Dynamic-0.4mm | 2500                                   | 1970        | 8900     |
| Dynamic-0.5mm | 2430                                   | 1940        | 7400     |
| Dynamic-1mm   | 1800                                   | 1440        | 5400     |
| Dynamic-2mm   | 1250                                   | 1020        | 3800     |
| Static        | 420                                    | 380         | 1750     |

### 3.2.2 Stiffness Tests of Electric Cabinet Isolators

Two different wire rope isolators have been selected for the electric cabinet isolation. Electric cabinet wire rope elements are tested with same machine and in same method with engine mount isolators. Stiffer wire rope element's height is lower than softer element. On the other hand, steel wire thickness is thicker for stiffer wire rope element.

**Table 3-7** Stiffness Values of Electric Cabinet Wire Rope Elements Obtained from Stiffness Tests

|               | WRI905060 (Stiffer) (N/mm) |             |          |
|---------------|----------------------------|-------------|----------|
|               | Longitudinal               | Transversal | Vertical |
| Dynamic-0.2mm | 88.6                       | 96.7        | 275.4    |
| Dynamic-0.3mm | 61.4                       | 73.4        | 154.9    |
| Dynamic-0.4mm | 59.2                       | 74.1        | 165.4    |
| Dynamic-0.5mm | 56.9                       | 69.2        | 163.4    |
| Dynamic-1mm   | 42.4                       | 47.1        | 119.1    |
| Dynamic-2mm   | 31.2                       | 32.5        | 88.4     |
| Static        | 17.2                       | 16.2        | 38.9     |
|               | WRI906090 (Softer) (N/mm)  |             |          |
|               | Longitudinal               | Transversal | Vertical |
| Dynamic-0.2mm | 28.7                       | 66.3        | 101.8    |
| Dynamic-0.3mm | 21.5                       | 39.7        | 85.7     |
| Dynamic-0.4mm | 21.4                       | 37.4        | 82.5     |
| Dynamic-0.5mm | 20.5                       | 33.9        | 80.4     |
| Dynamic-1mm   | 15.6                       | 23.7        | 55.4     |
| Dynamic-2mm   | 11.4                       | 15.9        | 38.1     |

|        |     |     |      |
|--------|-----|-----|------|
| Static | 6.3 | 7.3 | 13.7 |
|--------|-----|-----|------|

In Table 3-7, stiffness values of electric cabinet obtained from stiffness tests are presented. Stiffness differences between two elements can be seen and compared.

Stiffer element is approximately two times stiffer than softer element in vertical and transversal directions. However, stiffer element has almost three times higher stiffness in longitudinal direction. Geometrical differences such as height, rope thickness have great effect on stiffness values of wire rope elements. However, effect of these differences is not investigated in this thesis since it is not the subject of this study.

**Table 3-8** Updated Rigid Body Natural Frequencies of Electric Cabinet Obtained from WinRemo Calculations with Stiffness Values Obtain from Stiffness Tests

|              | Softer Element (Hz) |      |      | Stiffer Element (Hz) |       |       |
|--------------|---------------------|------|------|----------------------|-------|-------|
|              | 57kg                | 75kg | 90kg | 90kg                 | 120kg | 150kg |
| Longitudinal | 12.6                | 11.7 | 10.1 | 15.4                 | 13.2  | 12.2  |
| Transversal  | 8.6                 | 7.6  | 6.8  | 9.6                  | 8.3   | 8.2   |
| Vertical     | 10.5                | 9.2  | 8.5  | 12.7                 | 11.0  | 10.9  |
| Rolling      | 4.7                 | 3.9  | 3.5  | 5.2                  | 4.5   | 4.1   |
| Pitching     | 8.2                 | 6.8  | 6.7  | 10.4                 | 9.0   | 8.9   |
| Yawing       | 4.8                 | 4.1  | 3.9  | 5.8                  | 5.0   | 4.5   |

In Table 3-8, updated WinRemo calculation results are demonstrated. For these calculations, 1mm Dynamic value stiffness values are selected. In the light of experiences gained from the engine mount calculations and experiments, it is observed that 1mm Dynamic test results have closest results to experimental results. Moreover, 1mm Dynamic test values are also closest results to catalogue values. Stiffness test graphs are presented in Appendix.

### 3.3 Dynamic Measurements of the Engine

In order to understand the behavior of the wire rope engine mounts under real condition, dynamic engine measurements are needed. In this section, experimental setup information and engine properties have been written. Data analysis results are presented in Chapter 4.

#### 3.3.1 Information about Experimental Setup

All the measurements of the wire rope elements on the engine was conducted in an engine test cell in Wärtsilä Finland Oy, Vaasa at 14<sup>th</sup> – 17<sup>th</sup> of June 2019. The engine has eight cylinders and cylinder bore diameter is 200mm. So, type of the engine is named as W8L20D in Wärtsilä. Nominal output power of the engine is 1480kW while it is running at 1200rpm.

With different load ratios, engine provides different output power. The engine is connected to a dynamometer for loading. Maximum load of the dynamometer for this engine is 1444kw while engine is running at 1200rpm. Total weight of the engine is 10.5 tons. Therefore, approximate weight on an engine mount is 2.6 tons. Considering that one engine mount has three wire rope elements, one wire rope element has carried approximately 875kg during the engine tests.

Dewe43 has been used for data acquisition system during the tests. Fifteen 1-D piezoelectric accelerometers have been used to measure the acceleration at the specific points on the engine. All accelerometers have  $10 \text{ mV}/(\text{m/s}^2)$  sensitivity. Voltage range of the data acquisition system is  $\pm 10\text{V}$  so measurement range is  $\pm 1000 \text{ m/s}^2$  that is enough to measure all accelerations on the points. Dewe43 has eight analog channels input so two Dewe43 data acquisition systems have been used at the same time. Two Dewe43 provides sixteen channels during one measurement run. Dewesoft X3 SP4 has been used as data acquisition software and Dewesoft X3 SP6 that is an upgraded version has been used for data analyzing software after the measurements. Twelve points have been selected for measuring, four of them are located at the top corner of the engine and four of them are located on the engine mounts which are locating at the bottom corner of the engine. The last four points are on the ground to measure how much energy dissipated by the isolation elements. Three directions, the axial, the horizontal and the vertical directions at every point have been measured and one accelerometer has been used for each direction.



**Figure 3-5 Engine in Test Cell**

Because of the data acquisition channel numbers, five points could be measured in one measurement run. One point has been selected as reference point for ODS analysis and other four points have been changed to collect all data from engine. Point A3a has been selected as a reference point and detailed information about reason of the selection will be given in the further parts of the thesis. Therefore, each measurement type has been finished with three measurement runs.

Rigid body modes generally can be located between 4-30Hz for this type huge engines. Although rigid body modes could be seen in lower frequencies, high frequency range is needed to observe structure borne noise. To measure and observe structure borne noise, 2-4kHz measurement range is required. Therefore, sampling rate for all measurement has been selected 10000Hz. To avoid aliasing that is one of the biggest sampling errors for vibration and noise measurements, anti-aliasing filter of the software has been used for all measurements. You can find detailed information about anti-aliasing filter in previous chapter related to.

### **3.4 Hydraulic Shaker Measurement**

Hydraulic shaker measurement was conducted in Wärtsilä, Vaasa outside the test cell. The hydraulic shaker has 2.2kW electric motor. Shaker is fastened to the engine rigidly by bolts. Engine was excited with sine sweep starting from 3Hz and finishing in 22Hz which is a suitable interval to obtain all rigid body modes from the experiment. Sine sweep time is 1Hz/min. Therefore, sweep time is 19 minutes. In addition to sine sweep, random excitation vibration measurement was conducted. Random vibration experiment was started from 0.5Hz and finished 25Hz. Random vibration measurement duration was 5 minutes.

Twelve points are selected for hydraulic shaker measurement. Six of them are located at the top plane of the engine and other six points are located at the bottom plane of the engine. Top and bottom corners and middle point between corners are selected for the measurement.



**Figure 3-6** Hydraulic Shaker Measurement of the Engine

Six triaxial accelerometers were used during shaker experiment. Only one side of the engine has been measured in one time. Two iterations have been performed for each type of measurement. Oros data acquisition system was used to collect and store the data. Mescopex was used for analyzing the collected data for the hydraulic shaker measurement.

### **3.5 Electromagnetic Shaker Measurement**

Electromagnetic shaker measurement was conducted in Vibration Laboratory of Vaasa University. Electric cabinet was tested with electromagnetic shaker to understand the vibrational behavior of the wire rope elements with the cabinet. In addition to this, obtaining the natural frequencies of rigid body modes was also another aim of the shaker experiment. Two different size wire rope elements were selected according to catalogue values and WinRemo calculations have been done with these elements. After finishing the stiffness tests of the wire rope elements, updated WinRemo calculations have been finished.

After the mathematical calculations, next step is to test the elements. electromagnetic shaker is selected to test the wire rope isolators. The shaker that is used in experiments has 25kW amplifier output and vibrator head frequency range of the shaker is 5-2000Hz. Maximum sine force 20.2kN, maximum random force is 22.2kN and maximum acceleration of the vibrator head is 60g for sine peak values and 50g for random rms values. Maximum speed is 2m/s. and maximum amplitude is 50.8mm.



**Figure 3-7** Electromagnetic Shaker Measurement of Electric Cabinet

Three different type excitations were selected to excite the cabinet. These are sine sweep, random and high frequency tests. Sine sweep measurement has been conducted from 5Hz to 50Hz with three different amplitudes that are 5mm/s, 10mm/s and 15mm/s. In sine sweep measurements, sweeping time is 3.2 minutes. Random vibration measurement has been conducted from 5Hz to 50Hz in random amplitude. Measurement time is 5 minutes for the random vibration. Finally, frequency range of the high frequency test is 10-2000Hz. Measurement duration is 8 minutes. During the high frequency experiment, acceleration is constant and 1g.

Five triaxial accelerometers were used during electromagnetic shaker experiment. Dewe43 data acquisition system was used for data collection. Dewesoft X3 SP6 is used for data collecting, storing and analyzing. Data analyzing results are presented in Chapter 4.

## 4 ANALYSIS OF THE EXPERIMENTAL RESULTS

Analysis of the experimental results are heart of this study. Design process has been mainly formed according to experimental results. Therefore, analyzing carefully of these results is very crucial to develop a suitable isolator mount. In this chapter, dynamic measurements and the hydraulic shaker measurements of the engine will be presented. Moreover, electromagnetic shaker tests of main cabinet will be analyzed. In addition to these experiments, static and dynamic measurements of the wire rope elements are also represented.

### 4.1.1 Experiment Results and Data Analysis of the Engine Tests

In this part, result of the experiments will be shown and analyzing these results will be discussed. In this part some of the measurement results will be given for the explanation and analysis. All engine test measurement results can be found in Appendix.

Engine tests have started with Sweep Up measurement. The engine has begun to run in 756rpm, and engine speed has been increased to 1200rpm for the Sweep Up measurement. Total time is for sweep measurement is seven minutes for each run.

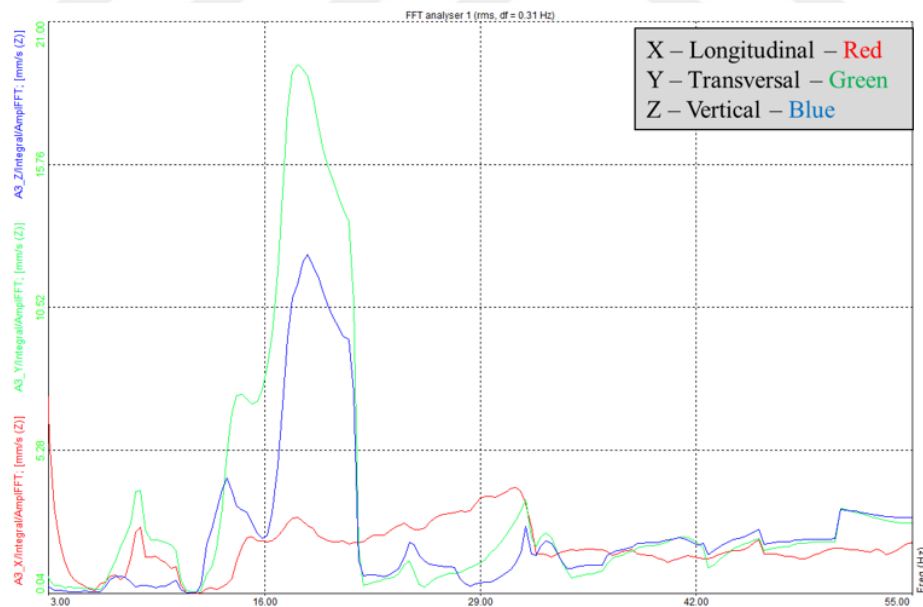


**Figure 4-1** FFT Spectrum Result for Sweep Up Measurement for point A3

In Figure 4-1, red line represents longitudinal direction, green line represents transversal direction and finally blue line shows the vertical direction velocity level. Maximum level of velocity is 19.4mm/s at 18.1Hz in transversal (Y) direction for point A3 which uses wire rope element that is designed for this master thesis project. On the other hand, velocity levels are 12.4mm/s in vertical (Z) direction at 18.7Hz and 2.8mm/s at 18.1Hz in longitudinal (X)

direction. Frequency difference for vertical direction can be negligible and assume that frequency levels are same for these peak values.

In addition, there are other peaks on the spectrum for three directions. For that peak values, maximum level of velocity is 7.3mm/s at 14.6Hz in transversal (Y) direction for point A3 which uses wire rope element that is designed for this master thesis project. On the other hand, velocity levels are 4.3mm/s in vertical (Z) direction at 13.7Hz and 2.1mm/s at 14.9Hz in longitudinal (X) direction.

Velocity levels for transversal and vertical directions are higher than longitudinal direction for both around 18Hz and 14Hz. It could be clue for us to make a deduction for rolling mode since vertical and transversal movement is higher than longitudinal movement for rolling mode. It is not easy to decide which frequency is which mode with only FFT spectrum. Therefore, an analysis method is required to make a precise decision. For that reason, ODS analysis and hydraulic shaker test analysis have been conducted. Results of these analyses will be discussed in further parts of the thesis.

Furthermore, another peak has been observed at 8.5Hz for longitudinal and transversal directions. Velocity levels are 3.8mm/s for transversal direction and 2.4mm/s for longitudinal direction. For 8.5Hz, no peak value has been seen for vertical direction.

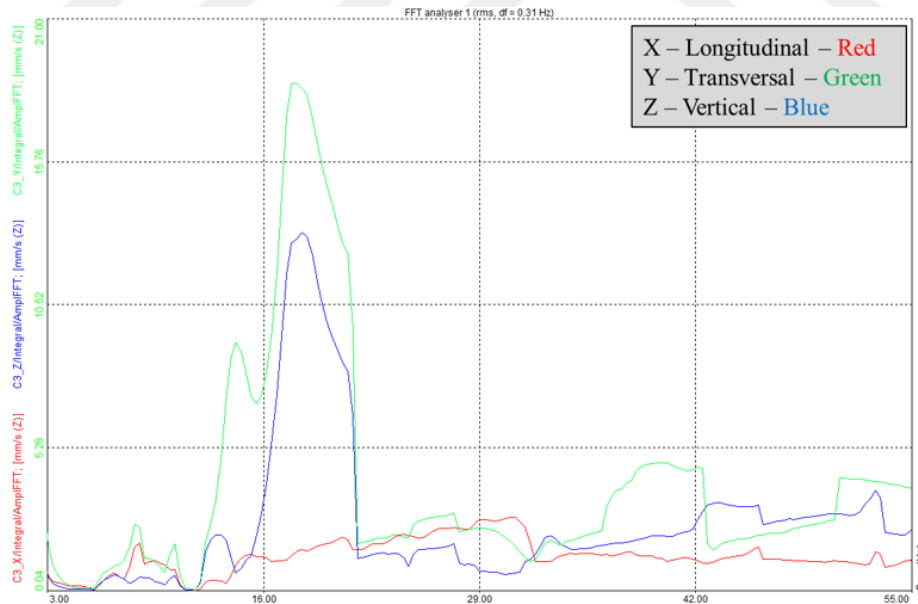


**Figure 4-2** FFT Spectrum Result for Sweep Up Measurement for point C3

In Figure 4-2, you can see the velocity values for point C3 and in the first look, it can be observed that velocity levels and trend of lines are look-alike point A3. Similarity of the graphs says that movement of two points, A3 and C3, is very close to each other. There are only negligible differences between velocity levels. For example, maximum level of velocity is 18.64mm/s at 18.01Hz in transversal (Y) direction for point C3. Moreover, velocity levels are 13.18mm/s in vertical (Z) direction at 18.3Hz and 1.5mm/s at 18.3Hz in longitudinal (X)

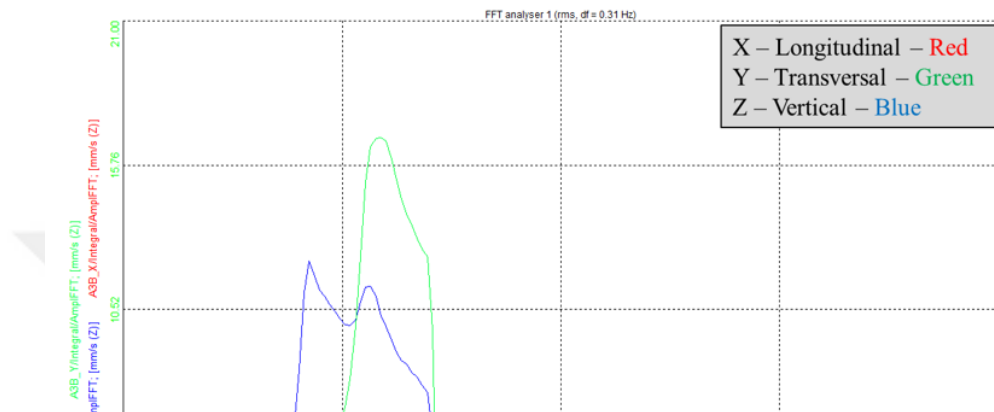
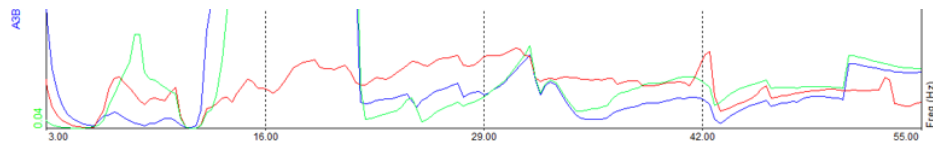


**Figure 4-3** FFT Spectrum Result for Sweep Up Measurement for point C3b



direction.

In Figure 4-3, peaks can be seen in same frequency values with A3 and C3 but velocity level in vertical direction for A3b is higher than A3 and C3 at 14.04Hz. Same behavior was observed for C3b, it can be seen in Appendix. Although velocity levels are quite same for other frequencies, specifically it is higher at 14.04Hz. A3b and C3b is located at the left side of the engine.

Possible rigid body mode frequencies have been observed. These frequencies are 7.3Hz, 8.5Hz, 14.1Hz and 18.1Hz. Although, there are six rigid body modes for every substance, only for peak could be observed. The reason for missing peaks might be mass and the complex structure of engine and excitation levels. Modern diesel engines have a lot of part on engine block such as turbocharger, oil pump or water pump vice versa. Influence of gravity might be the reason of missing peaks also. Therefore, it is not easy to obtain all rigid body modes with running test for the engines. This situation creates another reason for hydraulic shaker test. Since with sweep measurements, the excitation range is from 6.3Hz to 10Hz and from 12.6Hz to 20Hz. Missing frequencies is covering with stopping measurement.

As can be seen from the results, wire-rope vibration isolation elements have good isolation for lower frequencies since this type of elements have more damping for the lower frequencies. Even though longitudinal and transversal stiffness values of wire rope elements are very closed to each other, it can be observed from results that engine mounts have lowest amplitude vibration in X – longitudinal direction on the engine foot. It is because of the engine crankshaft rotation. Crankshaft rotation creates more vibrational forces in transversal and vertical direction. For the transversal and vertical direction isolation performance of the elements are varying according to frequency.

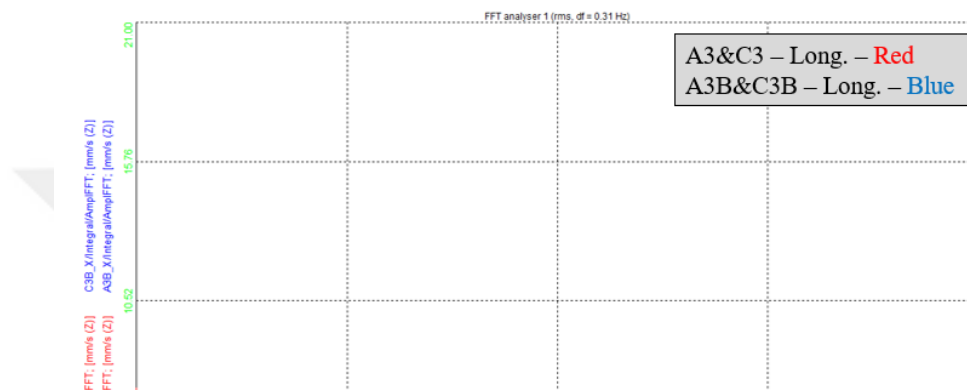
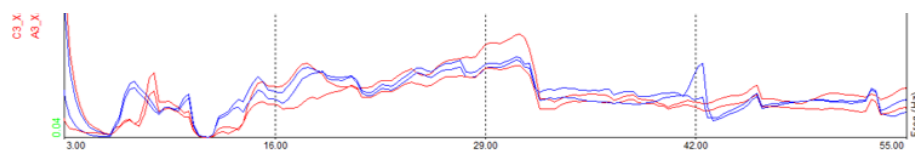


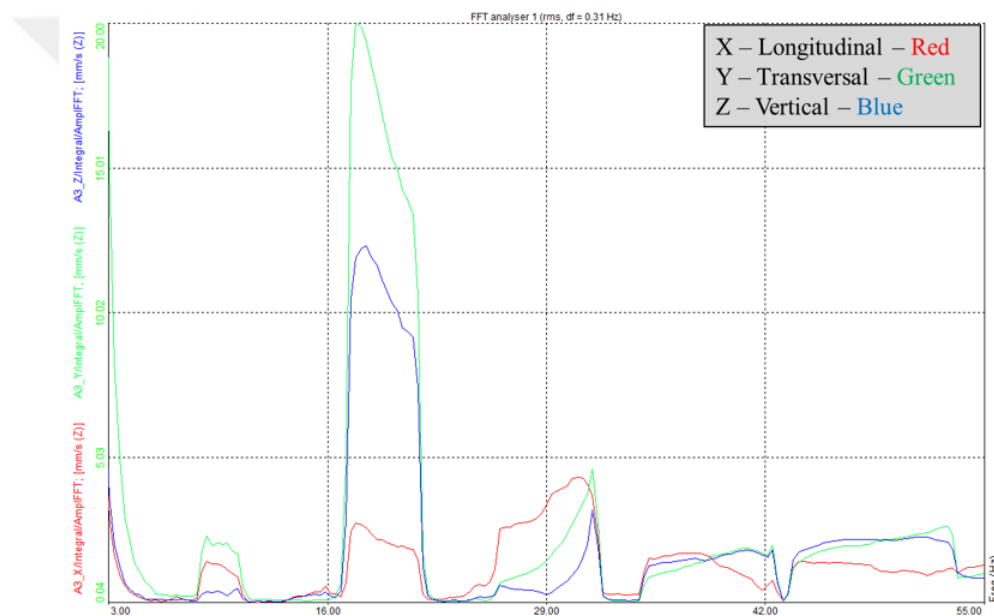
**Figure 4-4** FFT Comparison for Sweep Up Measurement for Longitudinal Directions



In Figure 4-4, FFT spectrum results has been compared for longitudinal directions. In the lower frequency area, between 5-10Hz, a shift on the graphs has been observed. In the points of A3 and C3, there are two different peaks at 7.3Hz and 8.5Hz but there is only one peak in the points of A3B and C3B at 7.3Hz. The has different parts between right and left side. The exhaust pipe, oil and water-cooling pipes and turbocharger pipe are located on the left side of the engine. These engine installation components might cause the change on the FFT results. However, to find the sources of this shift on the FFT is not the aim of this thesis. Therefore, any experiment has not been conducted to figure out this change. On the other hand, this kind of irregularity will create a change also on the rigid body motion of the engine. Thanks to the fact that wire rope vibration isolation elements have great damping property for the lower frequency vibrations, motion of the engine could be kept in reasonable levels. Moreover, as a result of the fact that movement of the engine in longitudinal direction is lower than the transversal and vertical direction, velocity levels of the engine motion in that direction is in lower levels too. Although the SDOF assumption is common for the engine mounts design processes, engine mounts cannot be assumed as a SDOF system. Even one directional force is

applied to an engine mount, it will create reaction forces in other coordinate directions too [21]. Because of the nonlinear behavior of the engine mounts could cause the same shifting result.

As it can be seen in Fig. 4-5, velocity levels on point A3 are close to other points and according to FFT spectrum, peaks of velocity levels are same with other points as expected. That velocity level of transversal direction is higher than vertical and longitudinal directions at 18.01Hz means that frequency of 18.01Hz might be the rolling mode since motion in transversal and vertical directions are higher than the longitudinal direction and because of that inertial forces coming from crankshaft causes higher vibration in vertical direction. As we saw on FFT spectra for other points, that's a clue for rolling mode. We can see the motion



with ODS analysis to confirm that 18.01 is the rolling mode.

**Figure 4-5** FFT Spectrum Result for Sweep Down Measurement for point A3

Sweep down measurement results are close to sweep up measurement results as expected. In Fig. 4-5, sweep down measurement FFT spectrums are available from point A3 and frequency of peaks are very close to Fig. 3-1. This situation is also same for other points. Comparison of the frequency levels of peaks on FFT spectrum can be seen in Table 4-1 and Table 4-2 for all type of measurements.

In Sweep down measurement, engine has been running from 1200 rpm to 760 rpm. It is aimed to obtain vibration levels for different frequency levels. Changing in rotational speed of crankshaft provides excitation in different frequencies. In this measurement, it can be seen that the velocity levels of transversal direction in 18.01Hz is higher than other direction's

velocity levels. The velocity levels between 8 and 10Hz for all directions have continuous line instead of peaks and in contrast of sweep up measurement. This continuous line firstly shows us that there is not enough excitation to obtain separate peaks on FFT spectrum. Secondly, it would be deduced that the engine may have more than one rigid body modes in the interval of 8-10Hz. Lastly, half order of the engine at 1200 rpm is 10 Hz and sweep down measurement has been started from 1200 rpm therefore peak at 10Hz may come from half order of the engine at 1200rpm. In addition, first order of the engine at 1200 rpm is 20Hz and that keeps the velocity level high at 20Hz. This situation has been observed in sweep up measurement too.

In Fig. 4-6, FFT spectrum of sweep down measurement for point A3b is presented. Peaks can be seen in same frequency values with A3 and A3b but velocity levels in all direction for A3 is higher than A3b. Same behavior was observed between C3 and C3b, it can be seen in Appendix.

In the lower frequency area, between 5-10Hz, a shift on the graphs has been observed between A3 and A3b. As it is noted previous parts of the thesis, these points are in different sides. In the point of A3, there is a continuous line between 8 and 10Hz and highest point of this interval is at 8.8Hz in longitudinal direction for A3. On the other hand, highest point in this interval for point A3b again in longitudinal direction is at 10.7Hz.

Same behavior has been also observed in sweep up measurements. As it is stated that the engine has different parts between right and left side. The engine installation components like exhaust pipe, oil and water-cooling pipes and turbocharger pipe might cause the shift on the FFT results. However, to find the possible reason of this shift is not the goal of this thesis. On the other hand, this irregular dynamic behavior will create complex motions and rigid body modes between this irregular interval could affect each other.

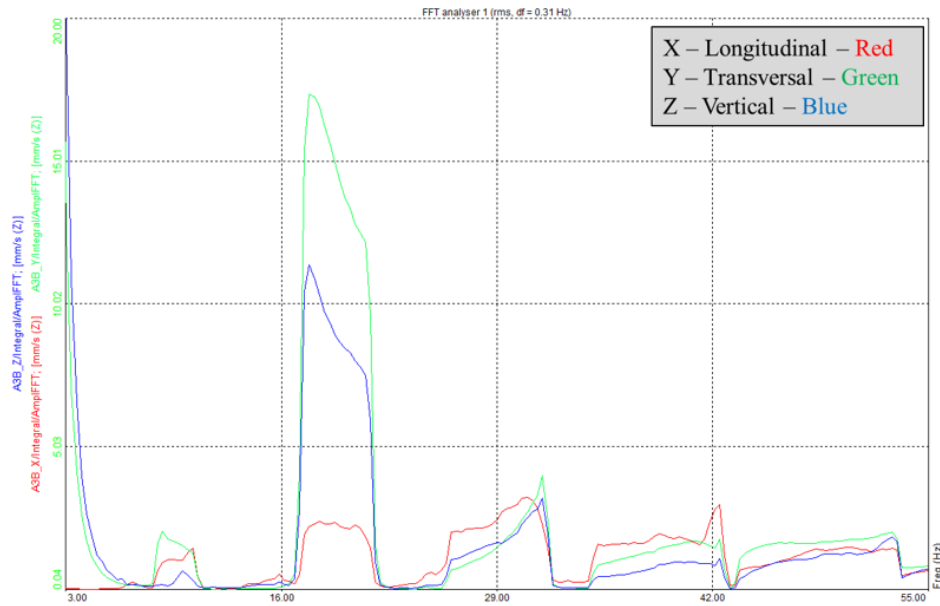


**Figure 4-6** FFT Spectrum Result for Sweep Down Measurement for point A3b

In stopping measurement, the engine has been shut down while running at 1200 rpm. Same frequency values have been observed for stopping measurement also. Different from sweep up measurement, any peak could not be obtained in sweep down measurement in 14.4 Hz however it is possible to see this peak in stopping measurement.

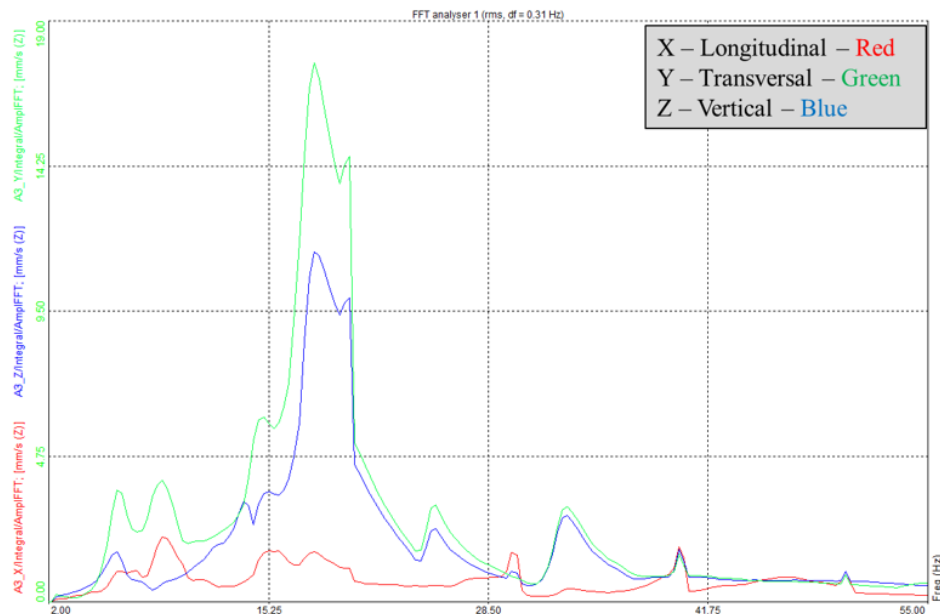


**Figure 4-7** FFT Spectrum Result for Stopping Measurement for point A3

Peaks at 6.1 and 8.8Hz can also be seen in stopping measurement FFT results. At 6.1Hz, velocity levels in transversal and vertical directions, at 8.8Hz velocity levels in transversal and

longitudinal directions have peak values separately different from sweep down measurement. At 18.01Hz and 6.1Hz, both longitudinal and vertical direction motion is dominant to longitudinal direction. To understand which rigid body modes are of these, ODS analysis has been used.

In Table 4-1 all peak points obtained from Sweep up, sweep down and stopping measurements are written for point A3 and A3b. In Table 3-1 all peak points obtained from Sweep up, sweep down and stopping measurements are written for point C3 and C3b. As it can be seen in the tables, there irregularities between possible rigid body mode frequencies. It is because of complex structure of diesel engine and its dynamics. Misalignment of engine and driven equipment, unbalance of rotating parts, resonance from an engine part or system, unbalancing of reciprocating parts or connected parts might be possible reasons for these irregularities too. Dynamic characteristic of the diesel engine is generally a complex mission for vibration engineers. The ability of animating mode shapes using ODS is a useful tool for identifying rigid body modes and decompose them each other.

**Table 4-1** Possible Natural Frequencies from Engine Measurement for Point A3 and A3b

|            | Sweep Up |        |       | Sweep Down |        |       | Stopping |        |       |
|------------|----------|--------|-------|------------|--------|-------|----------|--------|-------|
|            | Long.    | Trans. | Vert. | Long.      | Trans. | Vert. | Long.    | Trans. | Vert. |
| <b>A3</b>  |          |        |       |            |        |       |          | 6.1    | 6.1   |
|            | 7.0      |        | 7.3   |            |        |       |          |        |       |
|            | 8.5      | 8.5    |       | 8.8        | 8.8    | 8.8   | 8.8      | 8.8    |       |
|            |          |        | 13.7  |            |        |       |          |        | 13.7  |
|            | 14.9     | 14.6   |       |            |        |       | 15.3     | 14.9   | 14.9  |
|            | 18.1     | 18.1   | 18.6  | 17.7       | 17.7   | 18.3  | 18.1     | 18.1   | 18.1  |
| <b>A3b</b> |          |        |       |            |        |       |          | 6.1    | 6.1   |
|            | 7.3      |        | 7.1   |            |        |       | 7.9      |        |       |
|            | 9.1      | 8.5    |       | 9.1        | 8.8    | 8.8   |          | 8.8    |       |
|            |          |        | 14.1  |            |        |       |          |        | 14.3  |
|            | 14.8     | 14.5   |       | 15.8       |        |       | 15.8     | 14.9   |       |
|            |          | 18.1   | 17.7  | 18.3       | 17.7   | 17.7  |          | 18.1   | 18.1  |

**Table 4-2** Possible Natural Frequencies from Engine Measurement for Point C3 and C3b

|  | Sweep Up | Sweep Down | Stopping |
|--|----------|------------|----------|
|--|----------|------------|----------|

|            | Long. | Trans. | Vert. | Long. | Trans. | Vert. | Long. | Trans. | Vert. |
|------------|-------|--------|-------|-------|--------|-------|-------|--------|-------|
| <b>C3</b>  |       |        |       |       |        |       |       | 6.1    | 6.1   |
|            | 7.1   |        | 7.1   |       |        |       |       |        |       |
|            | 8.5   | 8.2    |       | 8.8   | 8.8    | 8.8   | 8.8   | 8.5    |       |
|            |       | 14.3   | 13.1  |       |        |       |       |        | 13.7  |
|            | 14.9  |        |       | 15.8  |        |       | 15.5  | 14.6   |       |
|            |       | 17.7   | 18.3  | 18.3  | 17.7   | 17.7  |       | 18.1   | 18.1  |
| <b>C3b</b> |       |        |       |       |        |       |       | 6.1    | 6.1   |
|            | 7.3   |        |       |       |        |       | 7.9   |        |       |
|            |       | 8.2    | 8.5   | 9.1   | 8.8    | 8.8   |       | 8.5    | 8.5   |
|            |       | 14.4   | 14.1  |       |        |       |       |        | 14.3  |
|            | 14.9  |        |       | 15.8  |        |       | 15.8  | 14.9   |       |
|            | 18.1  | 18.3   | 17.4  | 17.7  | 17.7   | 17.7  |       | 18.01  | 17.7  |

Main design aim of engine mount is to create a separation between harmonic orders of the rotational speed of the crankshaft and rigid body modes and to keep vibration levels under the standard or acceptable levels.

Using steel wire rope vibration isolators is an effective way for lower frequencies. As previously noted, the measurement results are combined from a series of identifications in data collected from the engine in a series of operational conditions such as nominal speed, sweep up and down, and shut-down conditions. Although the diesel engine is not in operational conditions inclined to generate low natural frequency excitations, modes relatively are determined below 20 Hz.

This is thought to be due to relatively short data periods during stopping measurements, the analysis results inclined to give mixed results in the sense that, a single measurement set did not include all clearly identified modes. Although the modes are not missed with a high enough order selection, in defining all modes, it would not be possible to obtain greater confidence from the results obtained from a single data period.

The best FFT results is generally acquired from the engine in nominal speed, sweep up, sweep down and shutdown measurements. Notably, engine sweep up measurements have proved to be most important owing to the longer runtime from idle speed to nominal speed. Like sweep

up, sweep down measurements provided also significant contribution to get better understanding for the possible natural frequencies. Despite short measurement time, shut-down measurements provide us valuable FFT results since it is not possible to excite all frequencies with sweep measurements. Therefore, in order to fill this unexcited gap stopping measurement is needed while considering the rigid body modes of the diesel engine. In nominal speed measurement, although excited frequencies are constant significant FFT results can be obtained for structure borne analysis and engine mount isolation performance.

#### 4.1.2 Hydraulic Shaker Measurement Analysis

In this section, hydraulic shaker measurement results have been analyzed. With the engine measurements, every rigid body mode could not be captured for every time therefore hydraulic shaker measurement has been conducted. In addition, the engine has a lot of have installation parts such as pipes, dynamometer etc. and these parts can cause changes on natural frequencies and rigid body mode shapes. To eliminate the effects of these parts, shaker measurement is needed. Furthermore, WinRemo calculation results would be close to shaker measurement results because the engine model in WinRemo has no installation parts too.

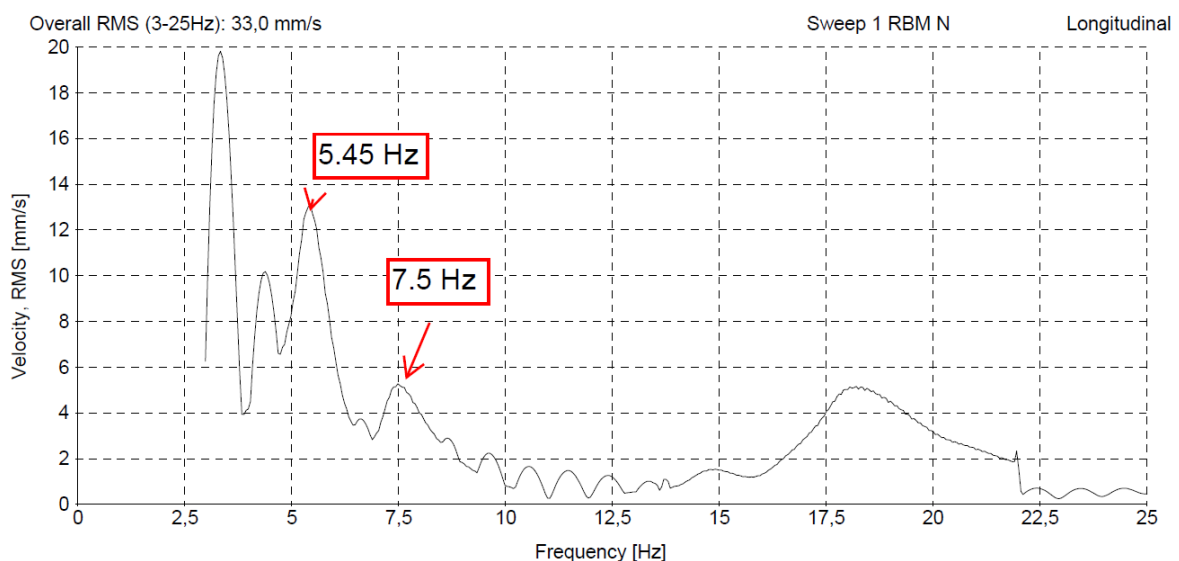


**Figure 4-8** Hydraulic Shaker Test FFT Result for Point A1b in Longitudinal direction

In Figure 4-8, FFT spectrum of point A1 in longitudinal direction has been presented. The spectrum has peaks on 5.45Hz and 7.5Hz. These frequencies might be clue for longitudinal, pitching or yawing since these three rigid body modes have higher level motion in longitudinal direction. Velocity levels of the motion are 13mm/s at 5.45Hz and 5mm/s at 7.5Hz.

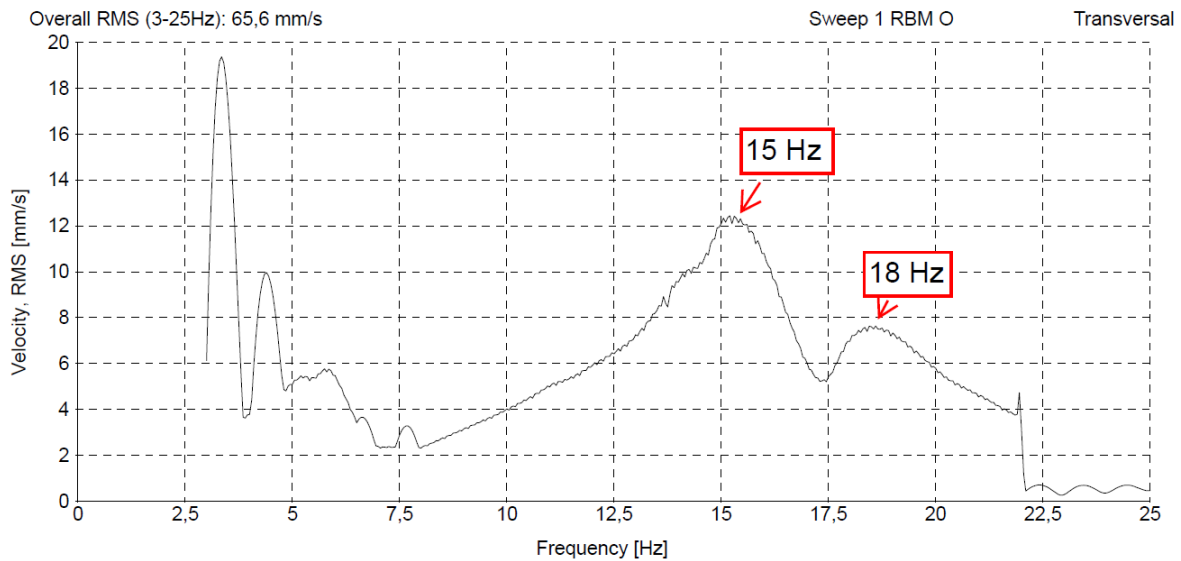


**Figure 4-9** Hydraulic Shaker Test FFT Result for Point A1b in Transversal direction

In Figure 4-9, FFT spectrum of point A1 in transversal direction has been shown and the engine has highest velocity levels at 15Hz and 18Hz. These peaks on FFT spectrum could be sign of transversal, Rolling or yawing rigid body modes frequencies. Since motion in transversal direction is higher for these three modes.

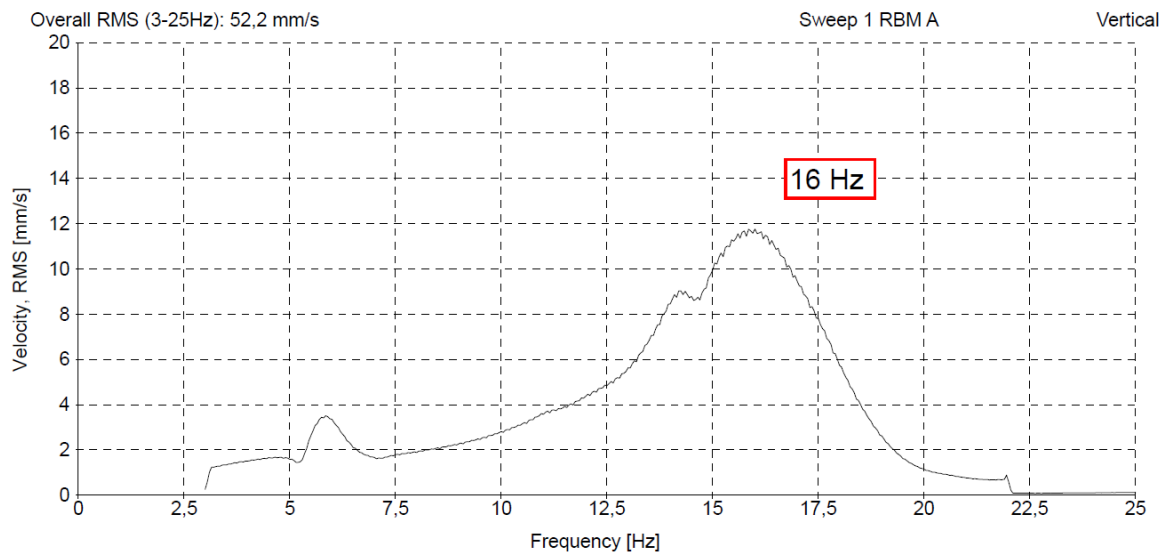


**Figure 4-10** Hydraulic Shaker Test FFT Result for Point A3 in Vertical direction

In Figure 3-10, FFT spectrum of point A1 in transversal direction has been represented. The engine has higher vibration at 14Hz and 16Hz relative to other frequencies. After analyzing hydraulic measurement experiment data, rigid body modes results have been presented in Table 4-3.

**Table 4-3** Rigid Body Modes of Hydraulic Shaker Measurement

| Rigid Body Mode | Frequency (Hz) |
|-----------------|----------------|
| Longitudinal    | 5.85           |
| Transversal     | 5.34           |
| Vertical        | 13.7           |
| Rolling         | 17.2           |
| Pitching        | 14             |
| Yawing          |                |

Hydraulic shaker experiment results have been used for WinRemo calculation iteration to obtain corrected calculation and compare with engine experiment results. By this way, obtained peaks from FFT spectra of engine experiment can be nominated to correct rigid body mode. To have a corrected WinRemo model, vertical mode is selected as a reference natural frequency. Stiffness members in calculation model has been iterated to get coherent results with hydraulic shaker measurement.

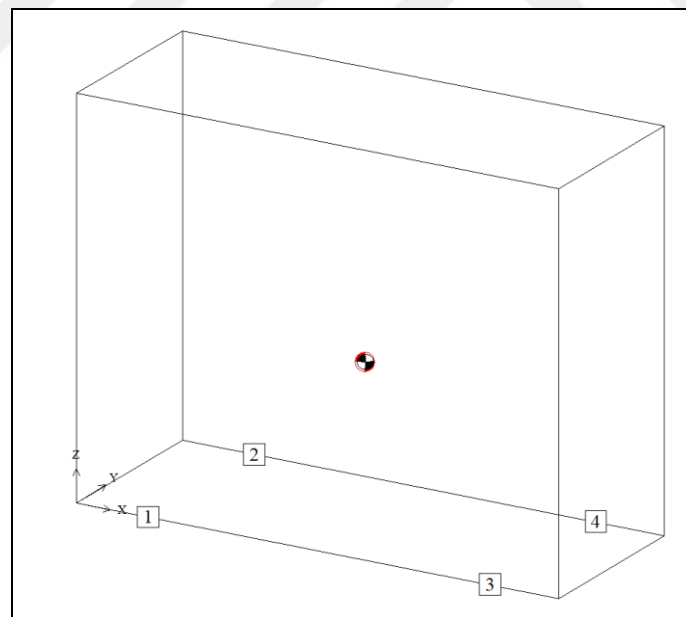


**Figure 4-11** WinRemo Calculation Model of the Wärtsilä Engine

After iterating stiffness values, calculated natural frequencies in Table 4-4 are obtained.

**Table 4-4** Rigid Body Modes of Iterated WinRemo Calculation

| Rigid Body Mode | Frequency (Hz) |
|-----------------|----------------|
|-----------------|----------------|

|              |       |
|--------------|-------|
| Longitudinal | 5.76  |
| Transversal  | 5.40  |
| Vertical     | 13.67 |
| Rolling      | 18.47 |
| Pitching     | 14.88 |
| Yawing       | 7.85  |

In this iterated calculation, linear stiffness element assumption has been used. The stiffness values have been decided 4000, 5000 and 20500 kN/m in order of longitudinal, transversal and vertical. Dynamic coefficient has been assumed that it is 1.05. In Table 4-5, comparison of calculations and hydraulic shaker measurement and engines tests are available. Helping with the comparison, rigid body mode shapes will be decided. In hydraulic shaker measurement, yawing is missing but with the engine test all rigid body modes could be caught.

Engine measurement results have been assigned according to WinRemo calculation results. The difference between calculated and experimental results could be many reasons.

**Table 4-5** Comparison of Calculated and Experimental Results

| Rigid Body Mode | WinRemo | Hydraulic Shaker | Engine Measurement |
|-----------------|---------|------------------|--------------------|
| Longitudinal    | 5.7     | 5.85             | 7.1                |
| Transversal     | 5.4     | 5.34             | 6.1                |
| Vertical        | 13.7    | 13.7             | 14.1               |
| Rolling         | 18.5    | 17.2             | 18.1               |
| Pitching        | 14.9    | 14               | 14.9               |
| Yawing          | 7.8     |                  | 8.8                |

The engine model in Winremo is a regular rectangular solid but the real engine has a different shape. Moreover, engine model is symmetric part, but the real engine is not symmetric, and it has a complex shape. One reason might be that every stiffness element could have different stiffness value since they compressed by the engine asymmetrically and deflection influences stiffness value for the wire rope elements. On the other hand, it is assumed that all isolation elements have same stiffness value. In the engine measurements, the engine has many

connections and installation parts as stated before. Thus, these connections and parts could cause a change on the natural frequencies.

## 4.2 Structure Borne Noise

Engineering acoustics are interested in vibration of mechanical systems and waves in solid structures in the audible frequency range. The name that is structure borne sound/noise can be given to the study of the phenomena of vibrations and sound waves. Therefore, structure borne noise is the study of the mechanical vibrations or waves in structures and they affect and propagate into adjacent media. Sound waves in structures have a significant role in noise control although they are not in audible range and can only be recognized at low frequencies. Before emitting into surrounding environment, many sound waves are generated and transferred through the structures.

In the structure borne noise analysis spectrum, frequencies up to 4kHz has been investigated for all directions and measurement points. In Figure 4-12, structure borne noise analysis from point A3 in X direction is being showed. Other structure borne noise spectra are available in Appendix. Sweep up, sweep down, stopping and nominal speed measurement have been used for structure borne noise analysis. Measurement points have been selected on engine mounts and ground to see the vibration levels between before isolation elements and after isolation elements.



**Figure 4-12** Structure Borne Noise Analysis (1/3 Octave Band Analysis) of Nominal Speed Measurement at Point A3 in Longitudinal Direction

In Figure 4-12, the output values are respectively the velocity values of the engine mount and ground. It can be observed that mounting system can basically effectively isolate the vibration of the engine in vertical direction. thus, it can be said that the mounting system of the engine is in a good condition. The engine produces significant low frequency excitations while running, and the dynamic effect of the wire rope engine mounts cannot be ignored in this condition. To make a better evaluation of the effect of mount participation on vibration isolation, an analytical 3-dimensional engine system is first considered. Vibration velocity curves from the engine mount points of the system while engine is running at nominal speed which is 1200 rpm, it can be seen that, with increasing frequency, the vibration velocity of the point A3 decreased significantly. This situation was also observed for other points.

The engine is resiliently mounted. Vibrations measurements have been performed on the engine mounts, on top of the wire rope engine mounts, and on the basement of the foundation, directly under the wire rope engine mounts.

This provides us to observe the isolation efficiency of the wire rope elements. Vibration levels for all frequency were found higher on the engine mounts than on the basement as it is expected, which classically vibration isolating performance improving with increasing frequency. On the other hand, it is observed that vibration levels between 150Hz and 600Hz are increasing on the basement while they are not increasing on the engine mounts. Engine harmonic are recognized as being multiples of the 0.5 order crank shaft rotation frequency for a four-stroke engine. In this case, the eight cylinders four stroke engine of the Wärtsilä running at 1200 rpm will produce multiples of 10 Hz.

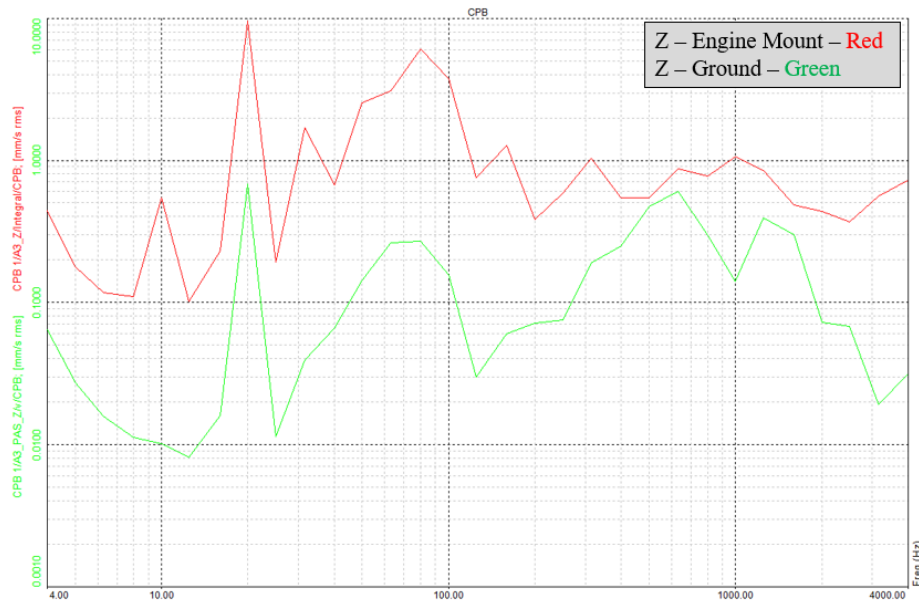


**Figure 4-13** Structure Borne Noise Analysis (1/3Octave Band Analysis) of Nominal Speed Measurement at Point A3 in Vertical Direction

### 4.3 Analysis of Electric Cabinet Measurements

In this part, result of the electric cabinet measurements will be presented and analyses of the measurement results will be discussed. In this section, some of the measurement results will be given as example of the study for the explanation and analysis. All electric cabinet test measurement results can be found in Appendix.

Analysis of the experimental results are the main part of electrical cabinet measurements. Design process is mainly leading according to experimental results of the main cabinet. Thus, analyzing the experimental results carefully is very significant to develop a suitable wire rope isolator for electric cabinet. In this section, electromagnetic shaker tests of main cabinet is analyzed and results of analysis is discussed.

Two different type of wire rope elements with different stiffness have been tested. Both wire rope elements have been tested with three different weights. Softer wire rope element has been tested with 57kg, 75 kg and 90 kg. Stiffer wire rope element has been measured with 90kg, 120 kg and 150kg weights. Both elements tested with 90 kg weight and these measurements will be used for comparison.

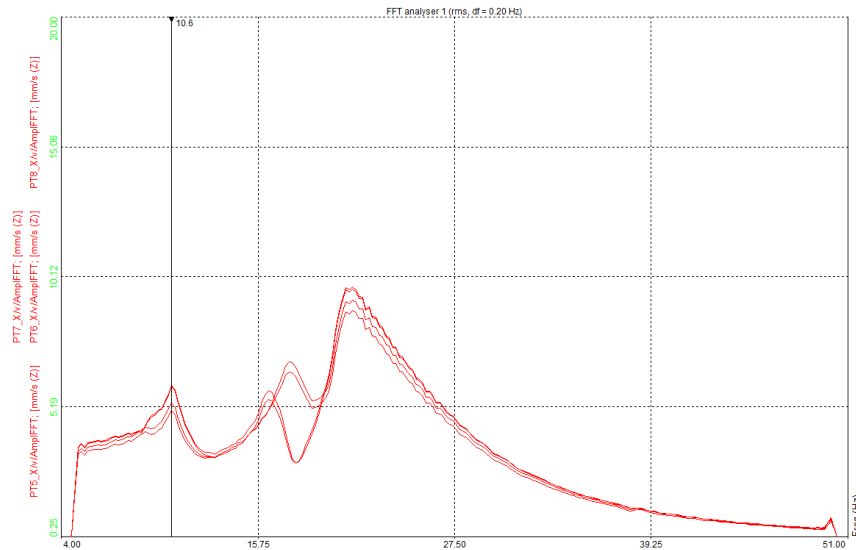


**Figure 4-14** Electric Cabinet Electromagnetic Shaker Experiment

Both elements have been tested with sine sweep measurement, random vibration measurement and high frequency measurement. Sine sweep measurement has been conducted from 5 Hz to 50 Hz with three different amplitude that are 5mm/s, 10mm/s and 15mm/s. In sine weep measurement, sweeping time is 4 minutes. Random vibration measurement has been conducted from 5 Hz to 50Hz in random amplitude.

#### **4.3.1 Softer Element Test Results**

One of the wire rope elements for electric cabinet is softer than other and its capacity of carrying weight is lower than the other elements. Therefore, it is tested with lower weights. In



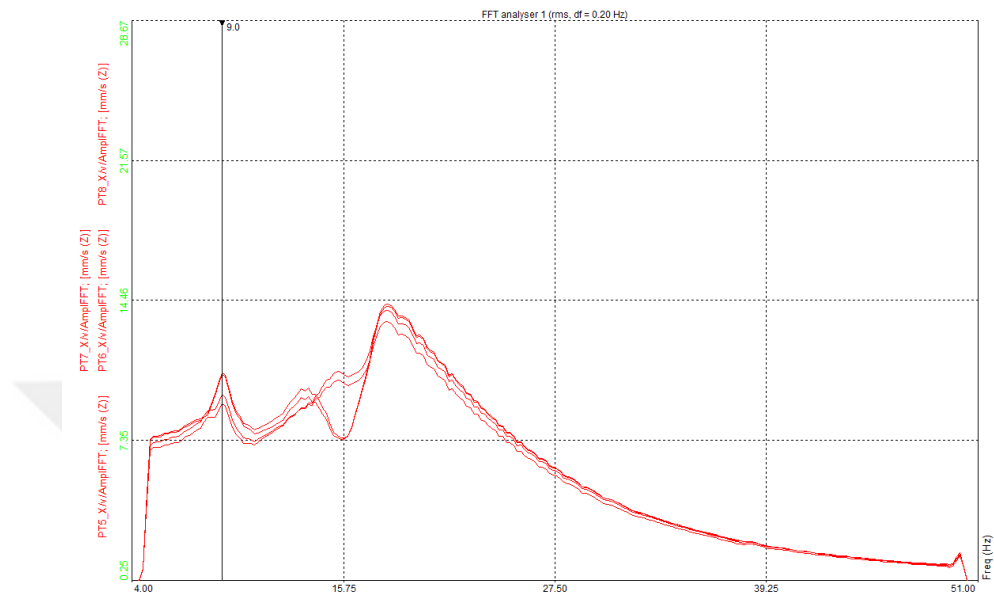
this part, results of softer element will be presented and discussed. Inferences have been shared.

**Figure 4-15** FFT Spectrum of 57 kg Electric Cabinet with Softer Elements in Sine Sweep 5mm/s Longitudinal Measurement in Longitudinal Direction

In Fig 4-15, peak of for all points is at 10.6 Hz with 5mm/s amplitude. Therefore, it is expected that longitudinal mode would be at 10.6Hz. On the other hand, because of the nature of wire rope elements, changing in stiffness with amplitude is expected. Therefore, expectation is that peak on FFT spectrum would be at lower frequency since wire rope elements behaves as a softer isolation element with increasing amplitude.

In Fig. 4-16, FFT analysis spectrum of electric cabinet in sine sweep 10mm/s longitudinal measurement in longitudinal direction has been presented. Peak level is at 9.0Hz with 10.8 mm/s amplitude. Thus, peak level is shifted from 10.6Hz to 9.0Hz because of the characteristic behavior of the wire rope elements. Same situation has been observed in all measurements in all directions. Moreover, all peaks on FFT spectrum has been shifted from higher frequencies to lower frequencies. Other measurement results can be seen in Appendix.

In order to validate FFT spectrum, transfer function of the measurements has been analyzed. By this way, FFT spectrum and transfer function of the measurement has been checked. To obtain transfer function, ODS analysis has been used and shaker table surface is used as a



reference for ODS analysis.

**Figure 4-16** FFT Analysis Spectrum of Electric Cabinet in Sine Sweep 10mm/s Longitudinal Measurement in Longitudinal Direction

In Figure 4-17, transfer function of electric cabinet in sine sweep 5mm/s longitudinal measurement in longitudinal direction has been presented and peak level is at 10.5Hz. It is coherent with FFT spectrum. FFT spectra and FRFs are compared to be sure in natural frequencies. When both graphics are aligned with each other, it is easier to decide the natural frequencies. Therefore, comparison and double check method has been used during the data analyzing of electric cabinet measurements. In addition to this, ODS analysis has been also done to animate the mode shapes.

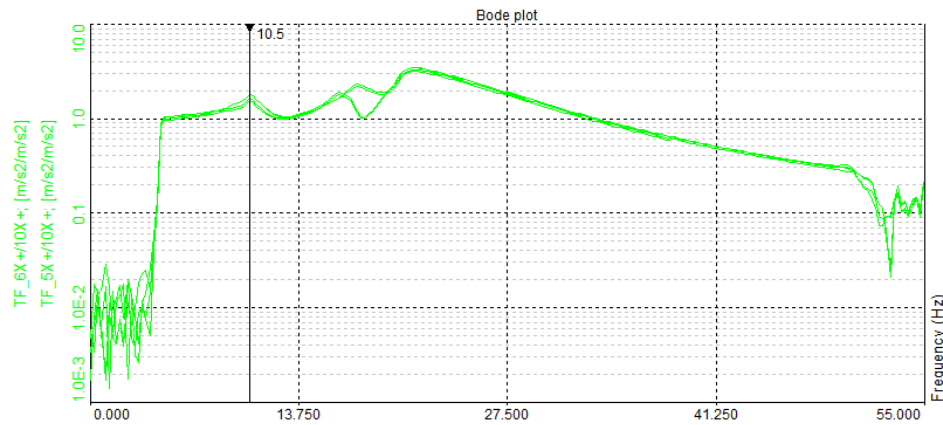


**Figure 4-17** Frequency Transfer Function of Electric Cabinet in Sine Sweep 5mm/s Longitudinal Measurement in Longitudinal Direction

**Table 4-6** Experimental Results of Softer Element with 57kg Cabinet

| 57 Kg           | Sine Sweep |         |         |
|-----------------|------------|---------|---------|
| Rigid Body Mode | 5 mm/s     | 10 mm/s | 15 mm/s |
| Longitudinal    | 10.6 Hz    | 9.0 Hz  | 8.2 Hz  |
| Transversal     | 7.3 Hz     | 6.3 Hz  | 5.4 Hz  |
| Vertical        | 11.0 Hz    | 10.2 Hz | 9.4 Hz  |

In Table 4-6, experimental results for softer elements has been presented. For all rigid body modes, it is observed that increasing in amplitude causes decrease in stiffness of the elements. Although stiffness in vertical direction is higher than other directions, longitudinal mode is higher than vertical mode and transversal mode is very close to vertical mode. It is because of the side wire rope elements. Side wire rope elements are located in longitudinal direction and position of these side wire rope elements increases the stiffness in longitudinal direction. Other measurement results are available in appendix and they can be checked in there.

WinRemo calculations have been also made for electric cabinet measurements and initial calculation results are shown in Chapter 3. However, updated calculations cannot be done successfully because of the nonlinearity of the wire rope elements. Therefore, suitable WinRemo model could not be developed for softer wire rope elements that using for electric cabinets.

Natural frequency diversifies relying on the amplitude of the excitation, as a consequence of the nature of the wire rope isolators, different natural frequencies can be seen under the condition of different amplitudes. The side stoppers have a significant role on the natural frequencies and

vibration levels. If this main cabinet set up would be used, rigid brackets are needed to fasten the side stoppers to work efficiently and under other circumstances the bracket will simply bend and cause a higher level of vibration. Side stoppers also minimize the vibration levels in longitudinal direction. This is mostly because they help in the direction in which they are installed, the impact of the side stoppers is not high in other directions.

**Table 4-7** Experimental Results of Softer Element with 75kg and 90kg Cabinet Models

|                 | WinRemo |      |         |      |         |      |
|-----------------|---------|------|---------|------|---------|------|
| Rigid Body Mode | 5 mm/s  |      | 10 mm/s |      | 15 mm/s |      |
|                 | 75kg    | 90kg | 75kg    | 90kg | 75kg    | 90kg |
| Longitudinal    | 9       | 8.2  | 7.6     | 6.8  | 6.6     | 6.0  |
| Transversal     | 6.1     | 5.5  | 5.2     | 4.6  | 4.6     | 4.6  |
| Vertical        | 10.3    | 9.6  | 9.0     | 9.2  | 8.5     | 7.6  |

Experimental results of natural frequencies are presented in Table 4-6 for 57kg electric cabinet model. In addition to this, 75kg and 90kg electric cabinet models' natural frequencies are presented in Table 4-7. Because of that electromagnetic shaker has excited the electric cabinet in only translational coordinate directions, longitudinal, transversal and vertical rigid body modes could be caught from the shaker experiment. Recalling the WinRemo calculations that were done with stiffness test experimental results, it is understood that rolling and yawing modes are generally very close to or below 5Hz. Shaker has been started to work from 5Hz. Therefore, it is impossible to obtain natural frequencies that are below 5Hz and very hard for natural frequencies that are close to 5Hz.

#### 4.3.2 Stiffer Element Test Results

One of the wire rope elements for electric cabinet is stiffer than other and its capacity of carrying weight is higher than the other element as it is stated in the previous section. Therefore, it is tested with higher weights. In this part, results of stiffer element will be presented and discussed. Inferences from analysis will be shared. Stiffer element has been tested with 90kg, 120kg and 150kg weights.

In Fig 4-18, peak of for all points is at 15Hz with 5mm/s amplitude. Therefore, it is expected that longitudinal mode would be at 15Hz.

As it is stated many times in this thesis, because of the nature of wire rope elements, stiffness changes with amplitude of excitation. Therefore, natural frequency of longitudinal mode of

the electric cabinet under 10mm/s sine sweep excitation is lower than 15Hz since wire rope elements show softer stiffness behavior with increasing amplitude.

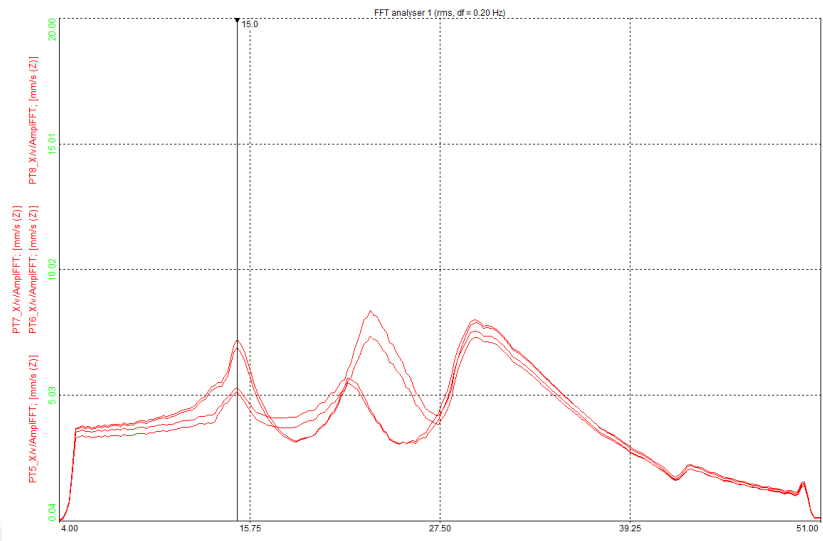


**Figure 4-18** FFT Spectrum of 90 kg Electric Cabinet with Stiffer Elements in Sine Sweep 5mm/s Longitudinal Measurement in Longitudinal Direction

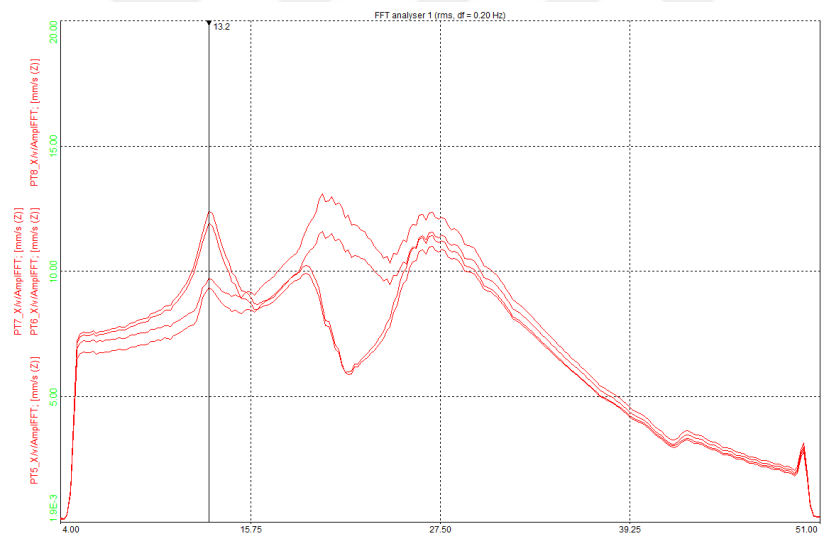


**Figure 4-19** FFT Spectrum of 90 kg Electric Cabinet with Stiffer Elements in Sine Sweep 10mm/s Longitudinal Measurement in Longitudinal Direction

In Fig. 4-19, FFT analysis spectrum of longitudinal direction of 90 kg electric cabinet with sine sweep 10mm/s longitudinal measurement has been presented. Peak level is at 13.2Hz with 12.3 mm/s vibration velocity level. Thus, peak level is shifted from 15.0Hz to 13.2Hz because of the characteristic behavior of the wire rope elements. Same situation has been observed in all measurements and in all directions. Other measurement results for stiffer element can be seen in Appendix.

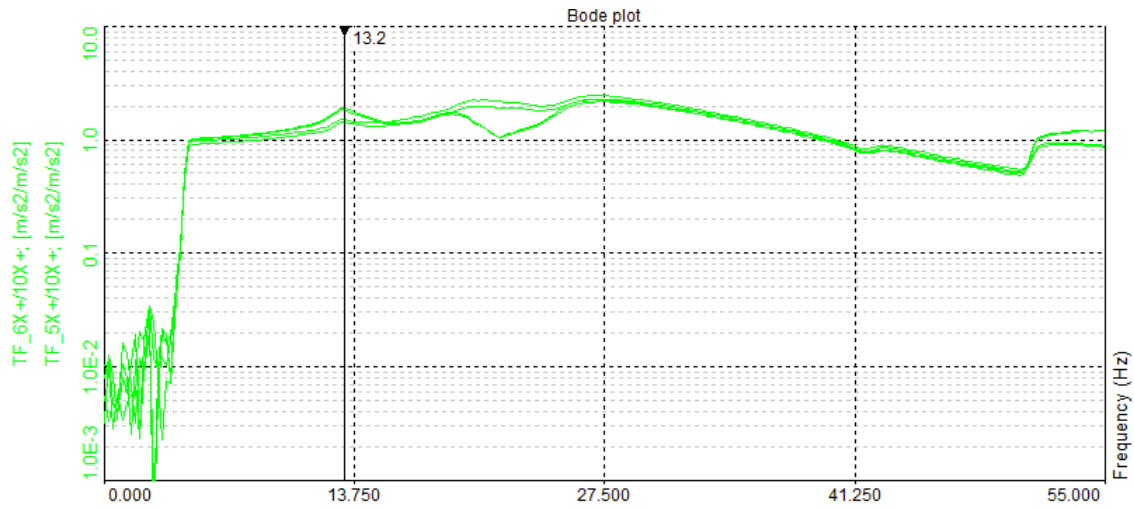


**Figure 4-20** Frequency Transfer Function of 90kg Electric Cabinet in Sine Sweep 5mm/s Longitudinal Measurement in Longitudinal Direction

In Figure 4-20, FRF of 90kg electric cabinet in Sine Sweep 5mm/s is shown. There is a peak at 13.2Hz. Result is consistent with FFT spectra. Therefore, now, it can be said that 13.2Hz is longitudinal mode of 90kg electric cabinet.

#### 4.3.3 Comparison of Electric Cabinet Isolators

Both stiffer and softer wire rope elements were tested with 90kg electric cabinet. Hence, there is a possibility to compare the results between two isolators.

For the softer element, longitudinal mode is at 8.0Hz and vibration velocity level is 5.8mm/s at this frequency. However, longitudinal mode is at 15.0Hz and vibration velocity level is 7.2mm/s. The cabinet has more vibration level with stiffer element in longitudinal direction. On the other hand, highest peak for softer element is at 16.4Hz and vibration velocity level is 8.9mm/s. highest peak for stiffer element is at 23.2Hz and vibration velocity level is 8.4mm/s. Between 5 and 16.4Hz, vibration velocity level is higher for softer element but after that point, vibration levels are decreasing dramatically for softer elements.

For the softer element, transversal mode is at 5.4Hz and vibration velocity level is 11.4mm/s at this frequency. However, transversal mode is at 11.2Hz and vibration velocity level is 14.4mm/s. The cabinet has more vibration level with stiffer element in transversal direction like longitudinal mode. However, vibration levels of transversal direction are higher than longitudinal direction vibration levels. Both elements have better isolation in longitudinal direction than transversal direction.

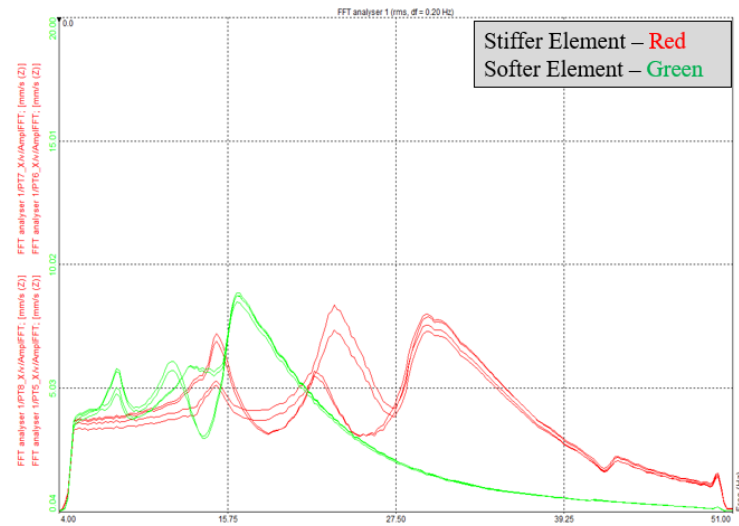


**Figure 4-21** Comparison of Stiffer and Softer Elements in Longitudinal Direction with 90kg Electric Cabinet in 5mm/s Sine Sweep Measurement

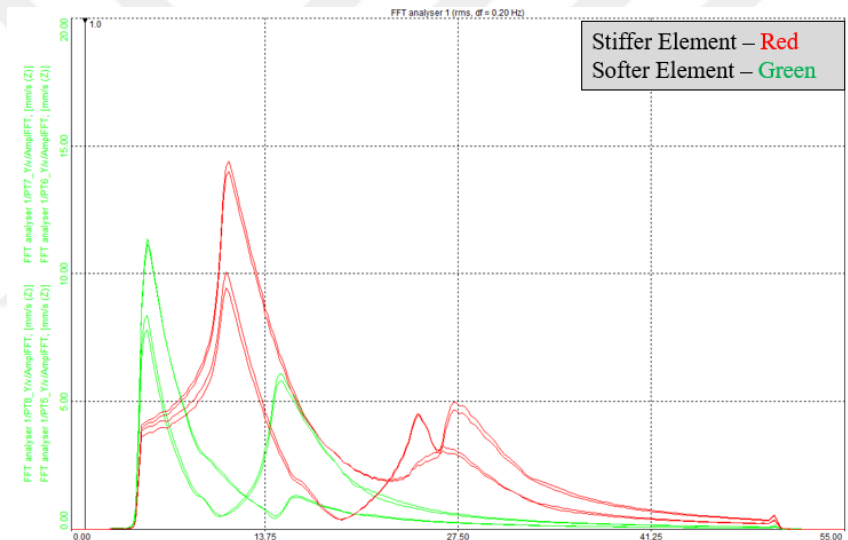


**Figure 4-22** Comparison of Stiffer and Softer Elements in Transversal Direction with 90kg Electric Cabinet in 5mm/s Sine Sweep Measurement

In Figure 4-23, vertical direction shaker measurement is demonstrated. Vibration velocity level is 12mm/s for softer element and 13mm/s for stiffer element. Like as longitudinal and transversal direction measurements, stiffer element has higher vibration levels than softer element. Moreover, both elements have more better isolation property for longitudinal direction than vertical direction.

In conclusion, softer element has better isolation than stiffer element with 90kg electric cabinet. Longitudinal direction is the best isolated direction for both elements. Side stoppers are the main for that situation. Side stoppers have great effect to reduce vibration levels in

longitudinal direction. On the other hand, effect of side stoppers in transversal and vertical direction is less than longitudinal direction.

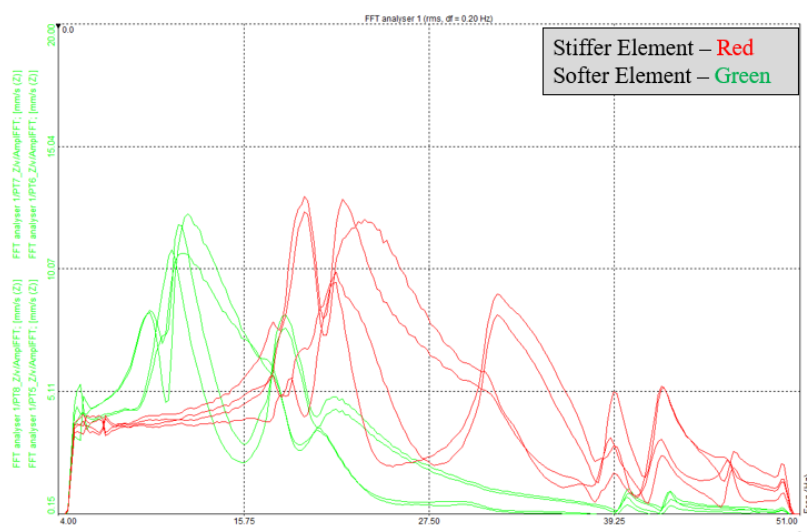


**Figure 4-23** Comparison of Stiffer and Softer Elements in Longitudinal Direction with 90kg Electric Cabinet in 5mm/s Sine Sweep Measurement

#### 4.3.4 Results of Electric Cabinet Measurements

Table 4-8 and Table 4-9 are presented in an order to compare the experimental and calculated results easily. According to results, it can be observed that there is perfect matching between two tables for the stiffer elements results. Experimental results have been corrected with mode shape animations with ODS analysis. With 90kg weight, pitching and transversal modes are very close to 10Hz which is half order of the crankshaft excitation at 1200rpm. With 120kg weight, vertical mode for 15mm/s sine sweep measurement is 9.9Hz. With 150kg weight, vertical modes are also 10.5Hz, 10Hz and 9.5Hz for sine sweep experiments. These values very close to 10Hz too. Therefore, half order of the crankshaft might cause vibration problems with these weights and rigid body modes. In contrast, updated calculation model cannot be developed for softer elements experimental results, as a result of the nonlinear behavior of the wire rope elements.

The side stoppers have a significant role on natural frequencies and vibration levels. However, it should be considered that side stoppers must be fastened with stiff bracket otherwise the bracket would bend, and it might create higher vibration levels. In addition to stiffness of the bracket, natural frequencies of the bracket should be considered during design process. If one of the natural frequencies of the bracket is excited by crankshaft orders, the bracket would resonate and has higher vibration level. Therefore, height of the side stopper bracket is important to have better stiffness and to have lower vibration levels. Moreover, using side stoppers in transversal direction

could be consider for future works to reduce the vibration levels in transversal direction. Wire rope elements should be re-selected to add new side stoppers and design process should start from the beginning since side stoppers would affect the natural frequencies in all directions.

**Table 4-8** Experimental Results for Electric Cabinet

|        |                              | Experimental |             |          |
|--------|------------------------------|--------------|-------------|----------|
|        |                              | Longitudinal | Transversal | Vertical |
|        | Stiffer Element Test Results | Hz           | Hz          | Hz       |
| 90 Kg  | 5 mm/s Sine Sweep            | 15.4         | 11.5        | 12.9     |
|        | 10 mm/s Sine Sweep           | 13.6         | 10          | 11.5     |
|        | 15 mm/s Sine Sweep           | 12.5         | 9           | 11       |
| 120 Kg | 5 mm/s Sine Sweep            | 13.2         | 9.3         | 11.5     |
|        | 10 mm/s Sine Sweep           | 12           | 8.3         | 11       |
|        | 15 mm/s Sine Sweep           | 11           | 7.3         | 10.5     |
| 150 Kg | 5 mm/s Sine Sweep            | 11           | 7.8         | 10.5     |
|        | 10 mm/s Sine Sweep           | 9.5          | 6.8         | 10       |
|        | 15 mm/s Sine Sweep           | 9            | 6.2         | 9.5      |
|        | Softer Element Test Results  |              |             |          |
| 57 Kg  | 5 mm/s Sine Sweep            | 10.6         | 7.3         | 11       |
|        | 10 mm/s Sine Sweep           | 9.4          | 6.3         | 10.2     |
|        | 15 mm/s Sine Sweep           | 8.2          | 5.4         | 9.4      |
| 75 Kg  | 5 mm/s Sine Sweep            | 9            | 6.1         | 10.3     |
|        | 10 mm/s Sine Sweep           | 7.6          | 5.2         | 9        |
|        | 15 mm/s Sine Sweep           | 6.6          | 4.6         | 8.5      |
| 90 Kg  | 5 mm/s Sine Sweep            | 8.2          | 5.5         | 9.6      |
|        | 10 mm/s Sine Sweep           | 6.8          | 4.6         | 9.2      |
|        | 15 mm/s Sine Sweep           | 6            | 4.6         | 7.6      |

**Table 4-9** WinRemo Calculation Results with Updated Stiffness Values

|        |                      | WinRemo      |             |          |         |          |        |
|--------|----------------------|--------------|-------------|----------|---------|----------|--------|
|        |                      | Longitudinal | Transversal | Vertical | Rolling | Pitching | Yawing |
|        | Stiffer Element Test | Hz           | Hz          | Hz       | Hz      | Hz       | Hz     |
| 90 Kg  | 5 mm/s Sine Sweep    | 15.6         | 11.3        | 12.9     | 5.34    | 10.5     | 6.1    |
|        | 10 mm/s Sine Sweep   | 13.9         | 9.9         | 11.4     | 4.7     | 9.5      | 5.4    |
|        | 15 mm/s Sine Sweep   | 13.2         | 8.8         | 10.6     | 4.4     | 9        | 5      |
| 120 Kg | 5 mm/s Sine Sweep    | 13.5         | 9.4         | 11.2     | 4.6     | 9.2      | 5.2    |
|        | 10 mm/s Sine Sweep   | 12.5         | 8.2         | 10.7     | 4.3     | 8.5      | 4.8    |
|        | 15 mm/s Sine Sweep   | 11.5         | 7.2         | 9.9      | 4       | 7.8      | 4.4    |
| 150 Kg | 5 mm/s Sine Sweep    | 11.2         | 7.7         | 10.3     | 3.8     | 8.2      | 4.2    |
|        | 10 mm/s Sine Sweep   | 9.6          | 6.9         | 10.2     | 3.5     | 7.5      | 3.8    |
|        | 15 mm/s Sine Sweep   | 9.1          | 6.2         | 9.5      | 3.3     | 7        | 3.6    |

In Figure 4-24, correlation between stiffness of the elements and amplitude of the excitation. Wire rope elements are natural passive but adaptive vibration isolators. Therefore, relation

between excitation amplitude and stiffness is important to understand the nature and working principle of the wire rope elements.

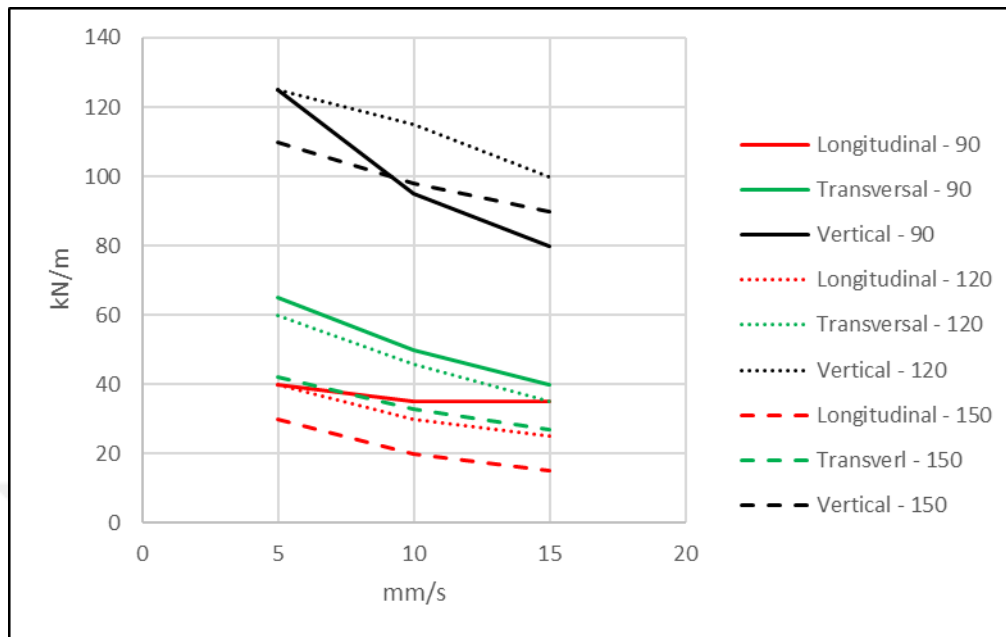


**Figure 4-24** Stiffness Change with Excitation Amplitude

In Figure 4-24, vertical stiffness lines are shown with black color, transversal stiffness lines are shown with green and longitudinal stiffness lines are shown with red. In the addition to colors, 90kg electric cabinet stiffness lines are shown with solid lines, 120kg electric cabinet stiffness lines are shown with round dot lines and 150kg electric cabinet stiffness lines are shown with dash lines. X axis of graph represents the excitation amplitude and unit is mm/s. Y axis of the graph represents the stiffness values of wire rope elements and unit is kN/m. This graph is prepared for only stiffer element since WinRemo model could not be developed for softer elements according to experimental results. For all directions, increasing amplitude decreases the stiffness values of the isolators. Vertical stiffness value of the wire rope element under 90kg electric cabinet has highest change since it decreases dramatically, and amplitude has more effect on wire rope element in this condition. On the other hand, longitudinal stiffness value of the wire rope element under 90kg has the least change. Stiffness values obtained from 120kg and 150kg electric cabinet measurements changes approximately with same rate.

## 5 CONCLUSIONS

In this chapter, obtained results both discussed and concluded. One of the main properties of wire rope elements is non-linear hysteretic behavior of them. It is observed under all experimental results. With the non-linearity of the WRIs, different stiffness values are observed under different excitation amplitudes. It is observed that increasing amplitude of excitation decreases the stiffness value of wire rope elements. this change has a considerable effect on design process. During design process of WRIs, minimum and maximum excitation under real life condition of the application should be considered and in order to avoid clashing between working excitations and rigid body mode frequencies, working condition stiffness values interval should be designed carefully.

Three wire rope elements are used for one engine mount and totally twelve elements have been used by this way. Fixture plates have been designed to make three elements be used together and fix them to engine mount bracket. Engine has been measured four different types of working condition. These are sweep up, sweep down, stopping and nominal speed measurement. FFT spectrum analysis and ODS analysis have been used to catch possible natural frequencies. Hydraulic shaker measurement and WinRemo calculations have been performed to get better understanding to assign correct rigid body natural frequencies.

All rigid body modes were caught from experimental analysis of engine measurements. These results have been compared and analyzed with hydraulic shaker measurement and WinRemo calculations. With the help of this comparison and analysis, peak level frequencies have been nominated. All results are coherent with each other. Rigid body natural frequencies are higher than the initial calculation results since wire rope elements behaves stiffer than the wire rope elements stiffness test results. On the other hand, suitable WinRemo calculation model has been obtained after the engine experiments to get better initial prediction for future works. the vibration levels are higher in Z direction with respect to X and Y direction.

Comparison of hydraulic shaker tests and engine test shows us the there are external effects on rigid body natural frequencies. Since during the engine tests, the engine has a lot of connected parts such as pipes, dynamometer and so on. Moreover, animating mode shapes from engine tests was harder than animating mode shapes from hydraulic shaker experiment. Because time of sweep and stopping measurements is not enough to get meaningful mode shape animation for ODS analysis. Furthermore, there are only two different frequency excitations, which are 10Hz coming from 0.5 crankshaft order and 20Hz coming from 1 order

of crankshaft, were available for nominal speed measurement since engine was working in 1200rpm during in nominal speed. There is no rigid body mode frequency at 10Hz or 20Hz for the wire rope elements. This situation is one of the aims of the design process. If there were any natural frequency at these frequencies, excitation of crankshaft orders may cause an excessive vibrant motion.

After comparing engine tests, hydraulic shaker test and WinRemo calculations, it is observed that WinRemo calculation model is more fitted with hydraulic shaker measurement than engine test for translational rigid body modes. Error percentages of longitudinal, transversal and vertical modes for hydraulic shaker measurement are %2, %1 and %0 although they are %20, %11 and %2 for engine measurements. On the other hand, mathematical calculation results are closer to engine test results than hydraulic shaker measurement results for rotational rigid body mode frequencies. Error percentages of rolling, pitching and yawing are %2, %2 and %11 for the engine test result. However, error percentages of rolling and pitching modes are %7 and %6 for hydraulic shaker measurement and yawing mode could not be obtained from shaker test. Consequently, despite the errors, all rigid body mode natural frequencies are determined for the engine.

Effect of 1st orders of crankshaft rotation frequency was observed in one-third octave analysis. Although wire rope isolators insulate the engine at low frequencies, a peak can be seen at 20Hz on the both engine mounts and ground because of the high-level excitation.

Two different wire rope elements have been selected for electric cabinet and stiffer element has approximately three times higher stiffness. Electromagnetic shaker test has been performed for electric cabinet wire rope elements. Electric cabinet has been fixed in six different points, four of them are under the cabinet and two of them are located on rear side. Moreover, wire rope elements located on rear side were removed to observe the effect of side isolators. Only softer element setup has been tested without side stoppers due to the limited time. Three different weights have been tested for each element. Stiffer element has been tested with heavier weights. Sine sweep and random vibration experiments have been performed. The cabinet were excited in three different directions that are longitudinal, transversal and vertical. Shaker measurements have been started from 5Hz since shaker has own natural frequency at 4Hz.

FFT spectrum analysis and WinRemo calculations were performed for electric cabinet vibration analysis too. In addition to this, ODS analysis has been done to obtain mode shapes.

Translational rigid body mode shapes have been captured with ODS analysis. Rotational rigid body modes could not be obtained because they are under 5Hz.

Natural frequencies were decreased with higher weights as it is expected for both elements. Moreover, it is observed that stiffness of the elements has been changed with amplitude of the excitation. Both elements behave having softer stiffness under higher excitations. Other observation is that rear side has different natural frequency than front side of the cabinet because of side stoppers. Side stoppers increase the natural frequencies on rear side and drop the vibration levels on the elements that are located under the rear side of the cabinet.

WinRemo calculations and experimental results are compared. Well matching WinRemo model could be developed for the stiffer element. In contrast, a suitable mathematical model could not be developed for the softer element due to the non-linear behavior of the wire rope elements.

Both elements for electric cabinet are tested with 90kg and softer element has better isolation than stiffer element with 90kg electric cabinet. Vibration levels are more decreasing with increasing frequency for the softer elements. Longitudinal direction is the best isolated direction for both elements. Side stoppers are the main reason for that situation. Side stoppers have great effect to reduce vibration levels in longitudinal direction. On the other hand, effect of side stoppers in transversal and vertical direction is less than longitudinal direction.

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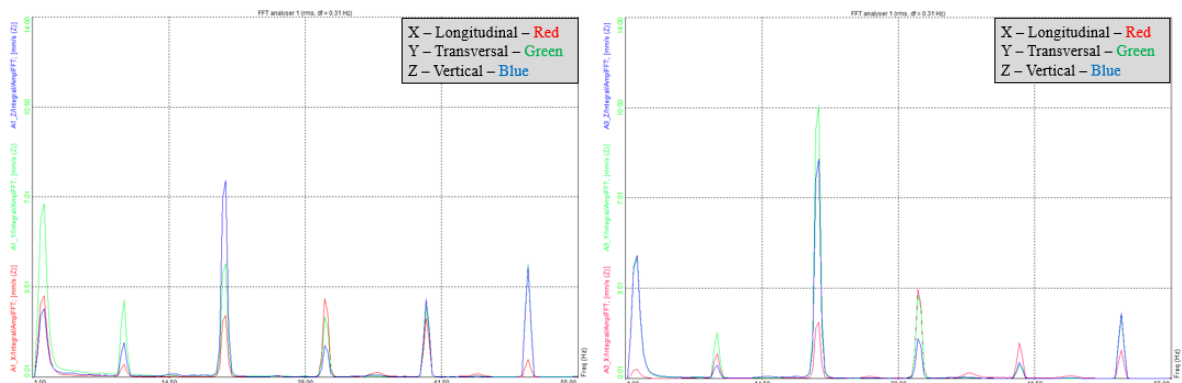
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## 7 APPENDICES

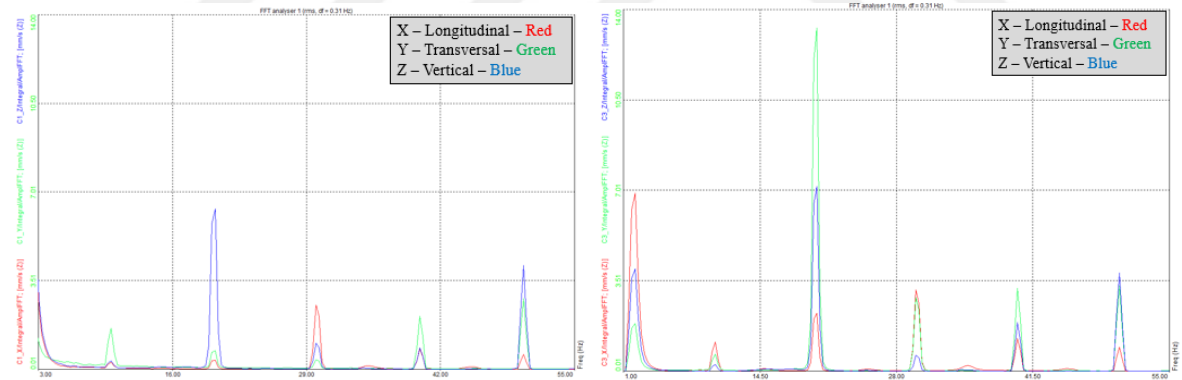
### Appendix A

#### FFT Spectrum Results for 1200RPM Measurements

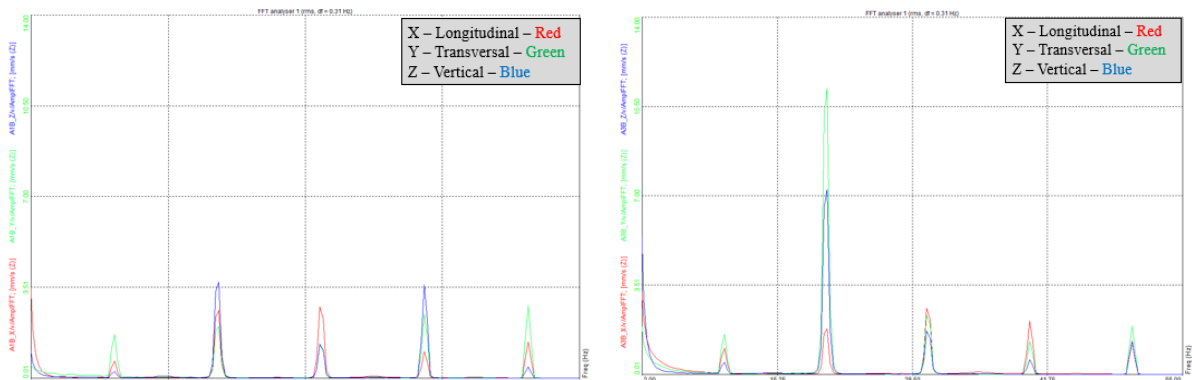
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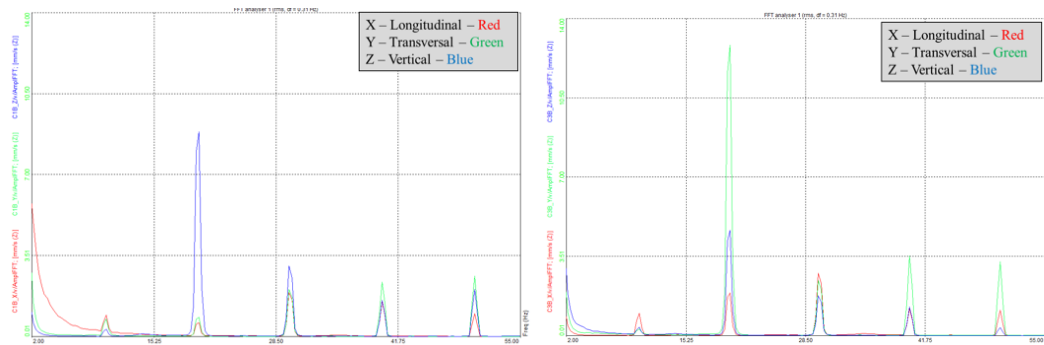
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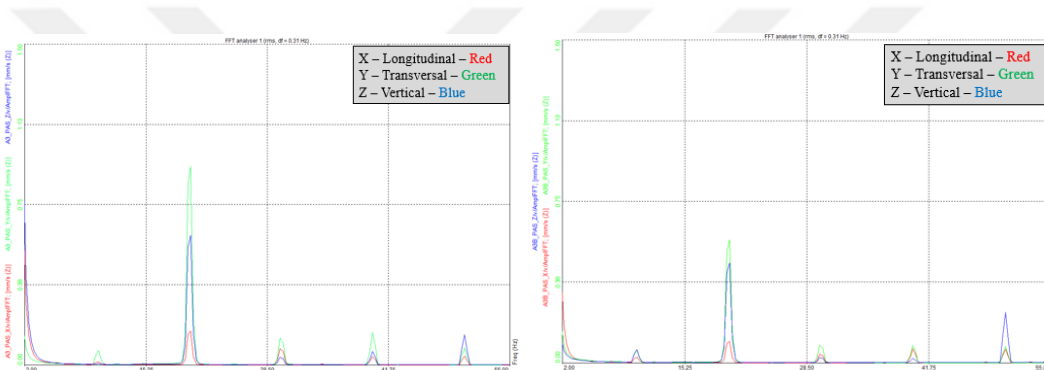
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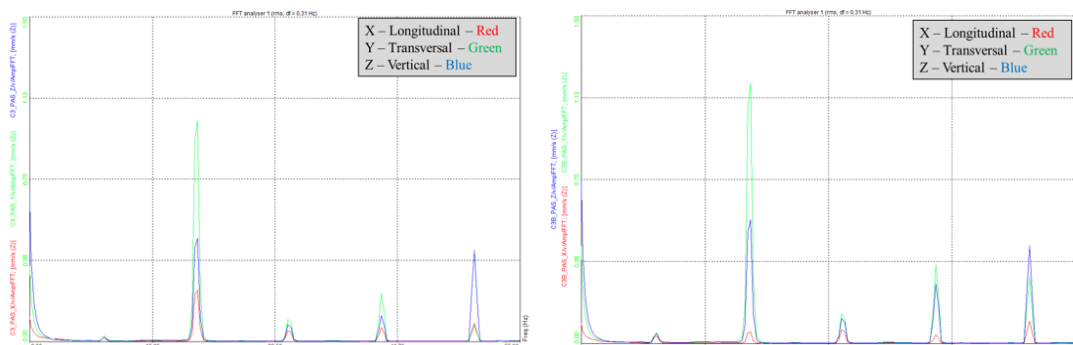
## Points C1b & C3b



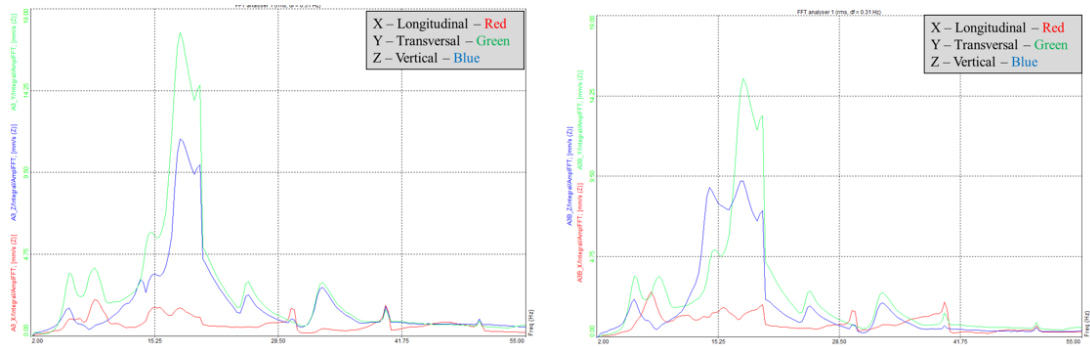
## Points A3-Ground & A3b-Ground



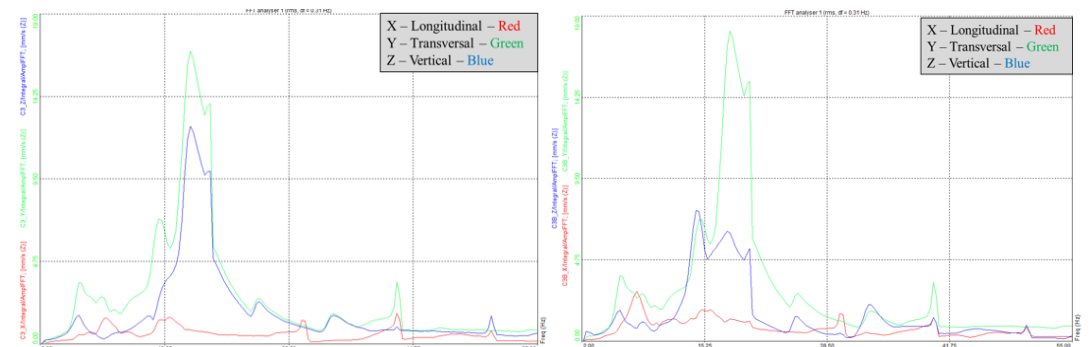
## Points C3-Ground & C3b-Ground



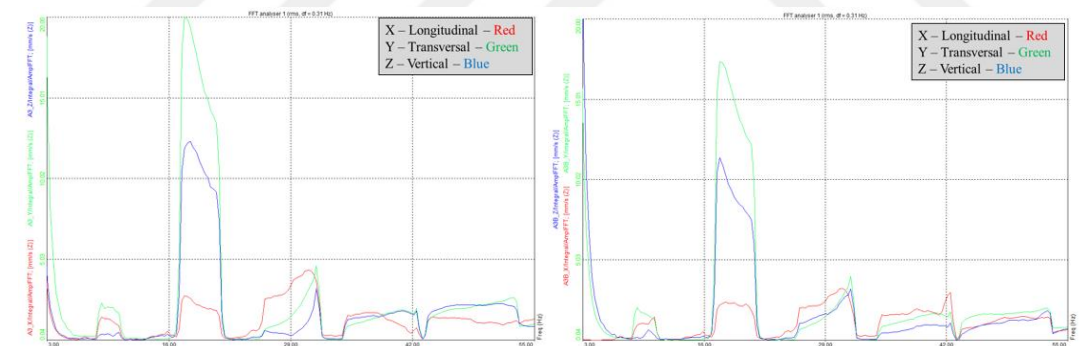
## FFT Spectrum for Shut-Down Measurement – Points A3 & A3b



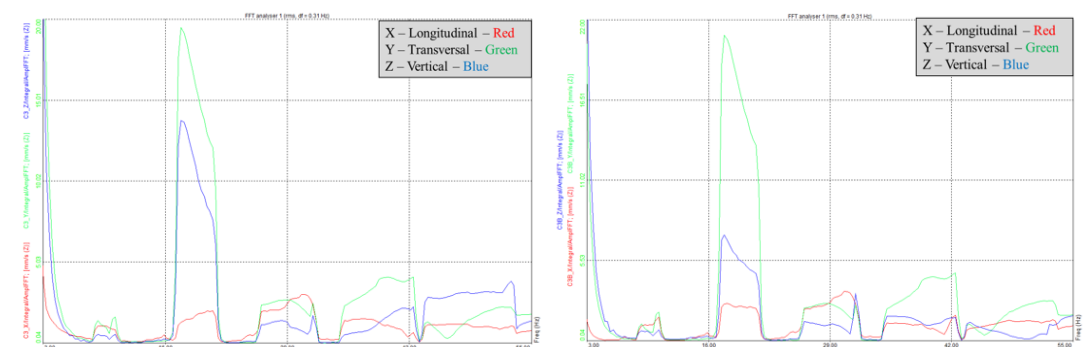
FFT Spectrum for Shut-Down Measurement – Points C3 & C3b



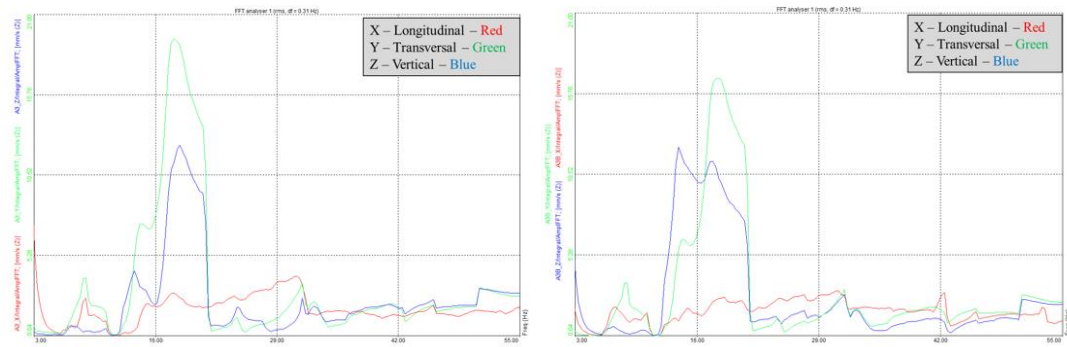
FFT Spectrum for Sweep Down Measurement – Points A3 & A3b



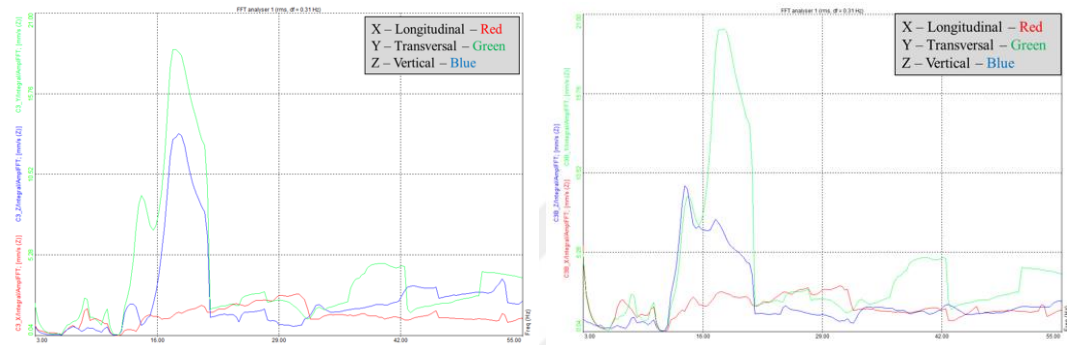
FFT Spectrum for Sweep Down Measurement – Points C3 & C3b



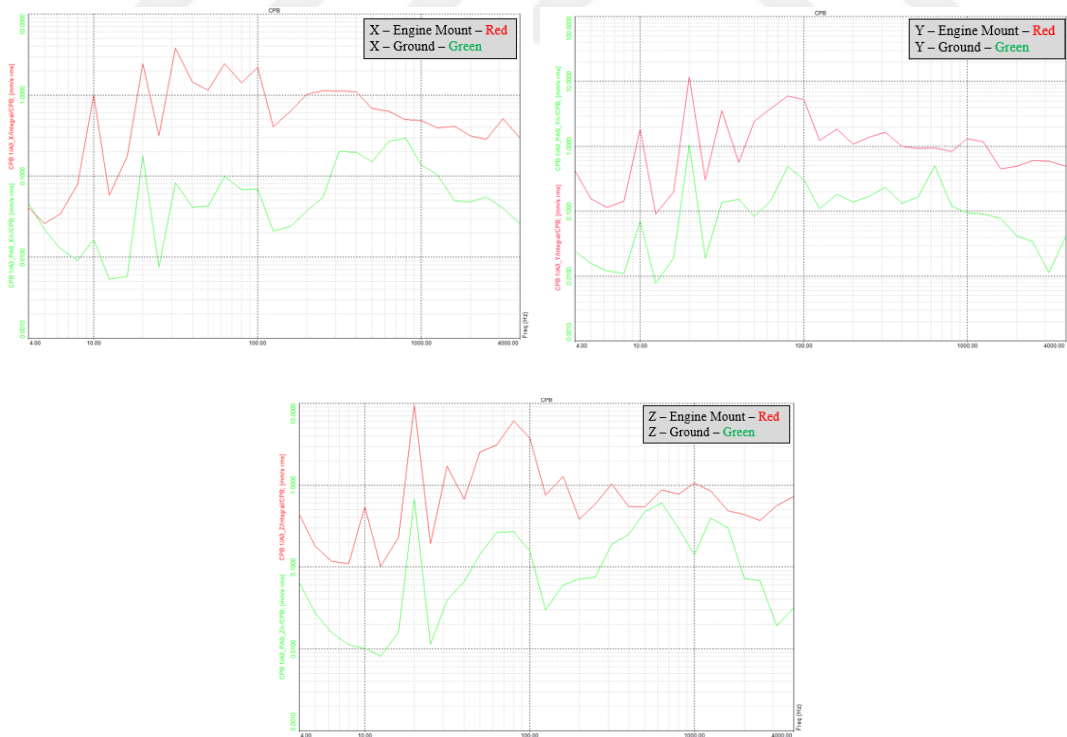
FFT Spectrum for Sweep Up Measurement – Points A3 & A3b



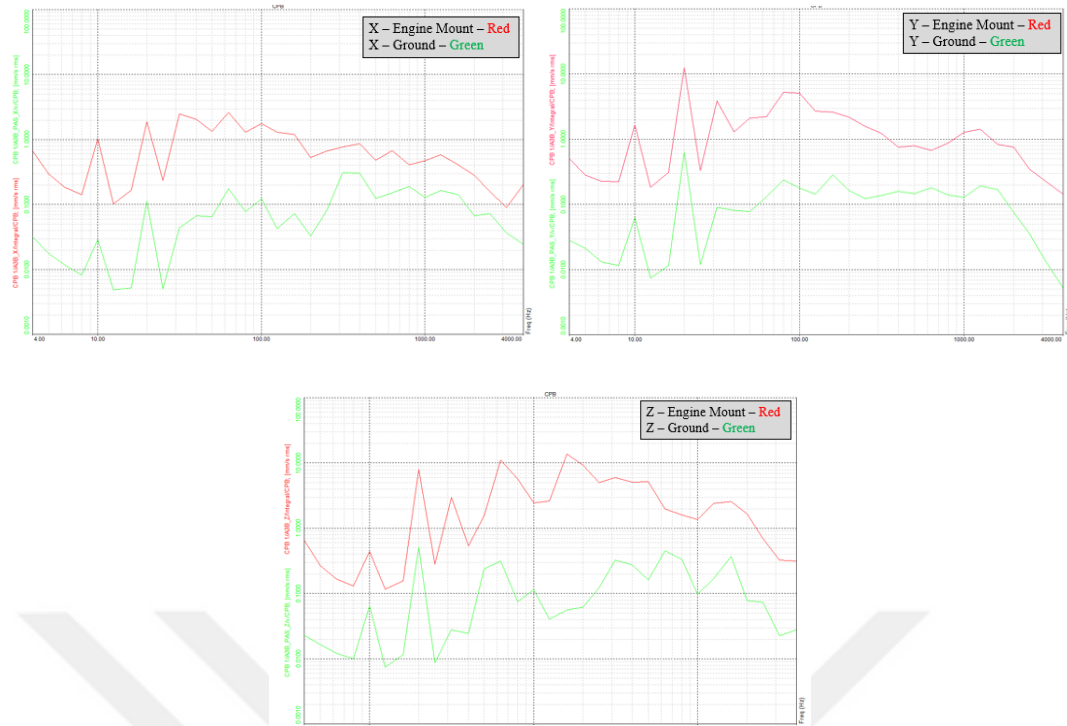
FFT Spectrum for Sweep Up Measurement – Points C3 & C3b



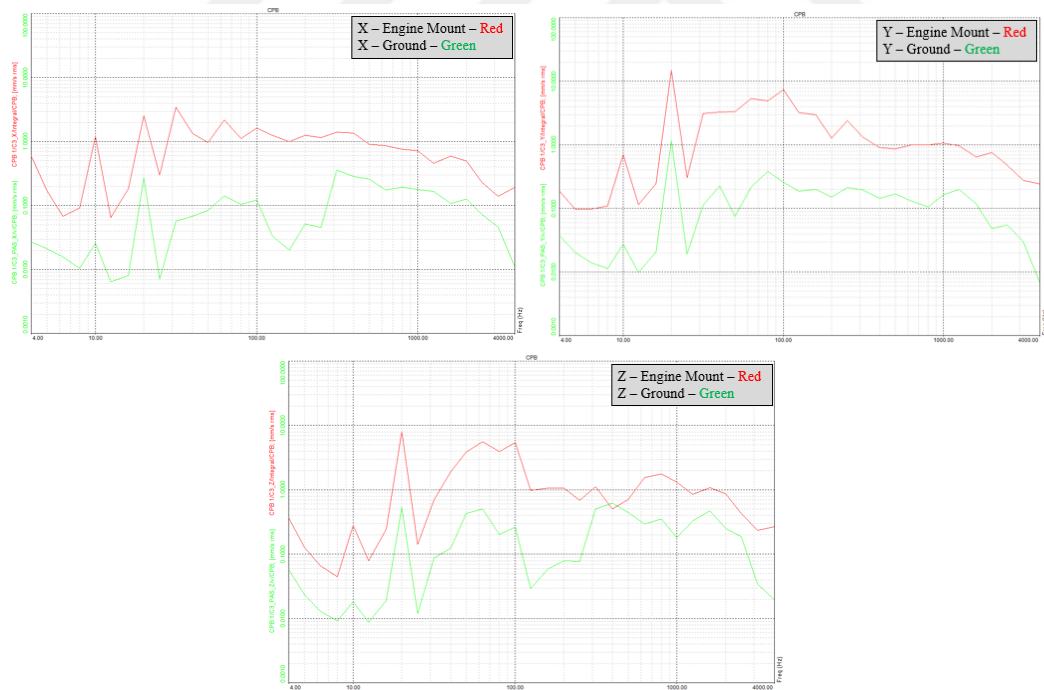
1/3 Octave Analysis for 1200RPM Measurements – Point A3



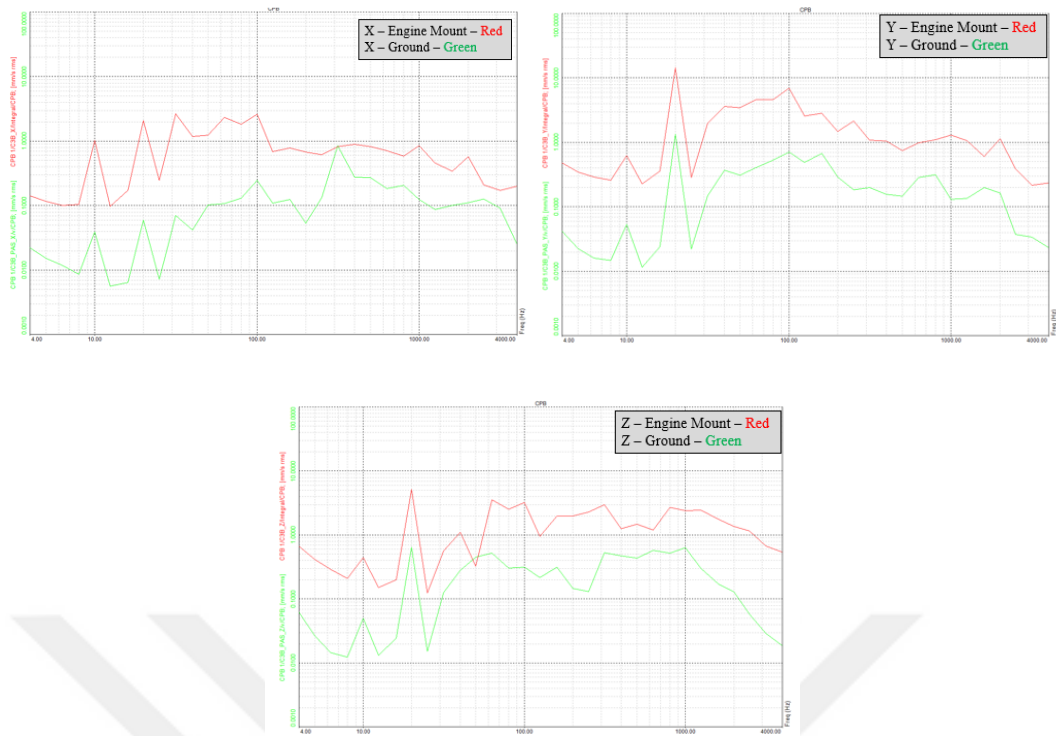
1/3 Octave Analysis for 1200RPM Measurements – Point A3b



1/3 Octave Analysis for 1200RPM Measurements – Point C3



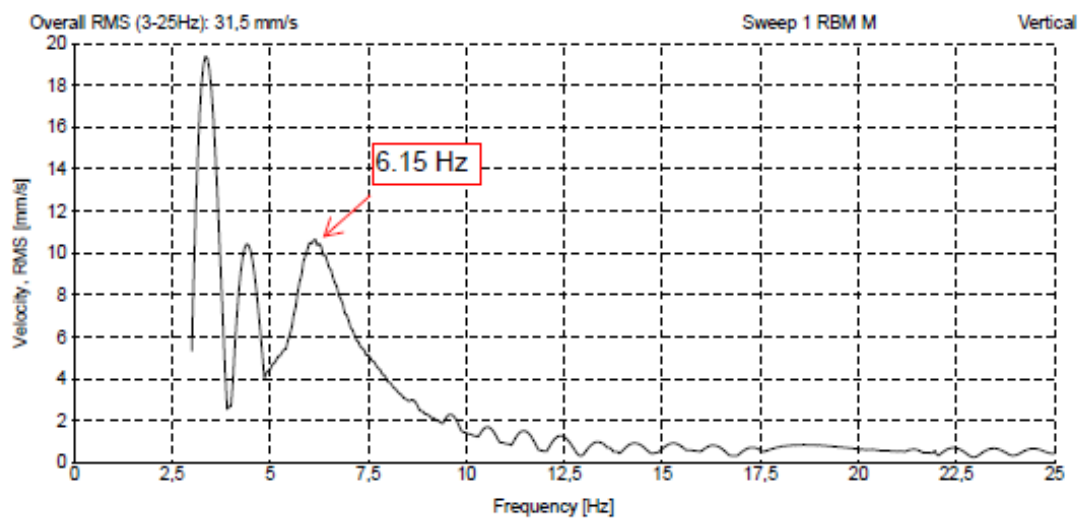
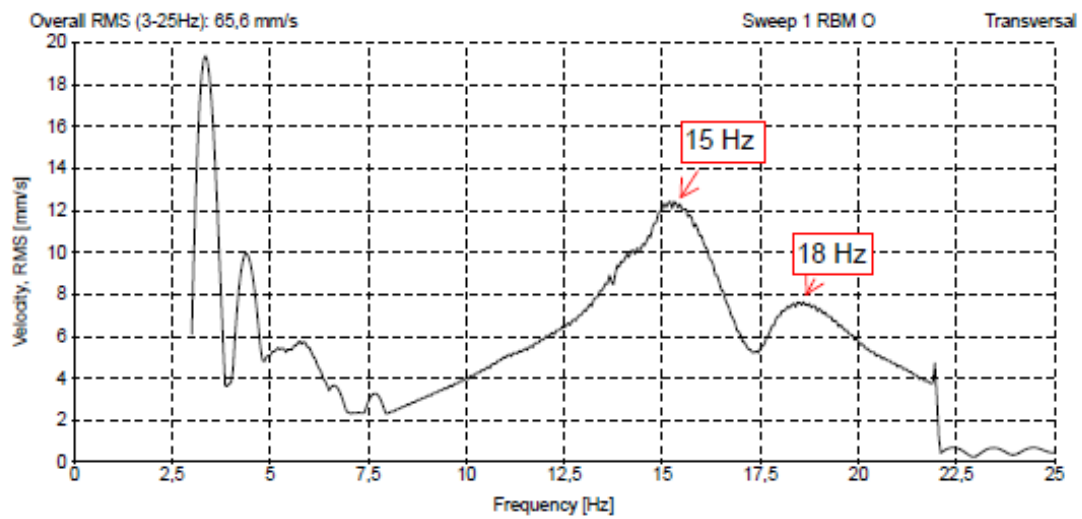
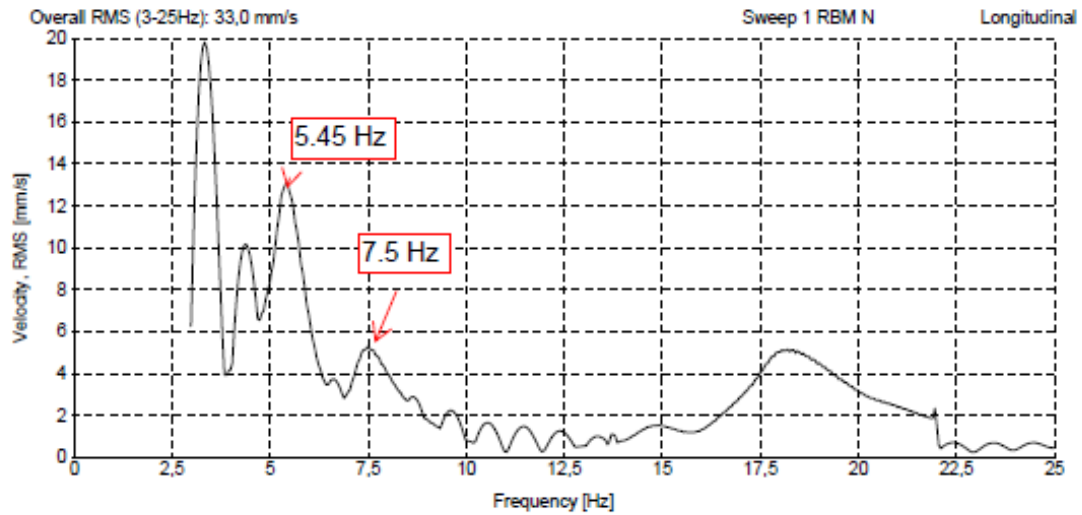
1/3 Octave Analysis for 1200RPM Measurements – Point C3b



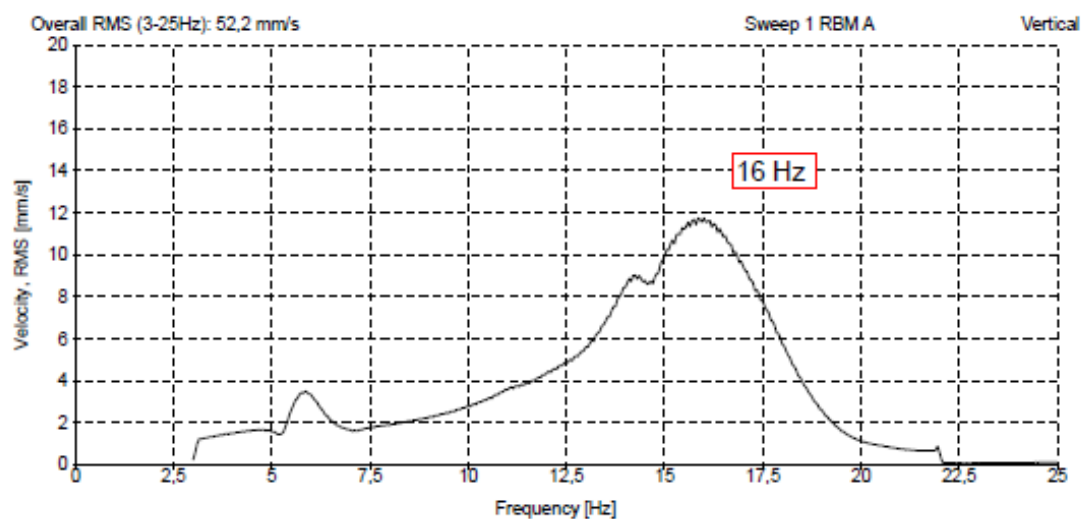
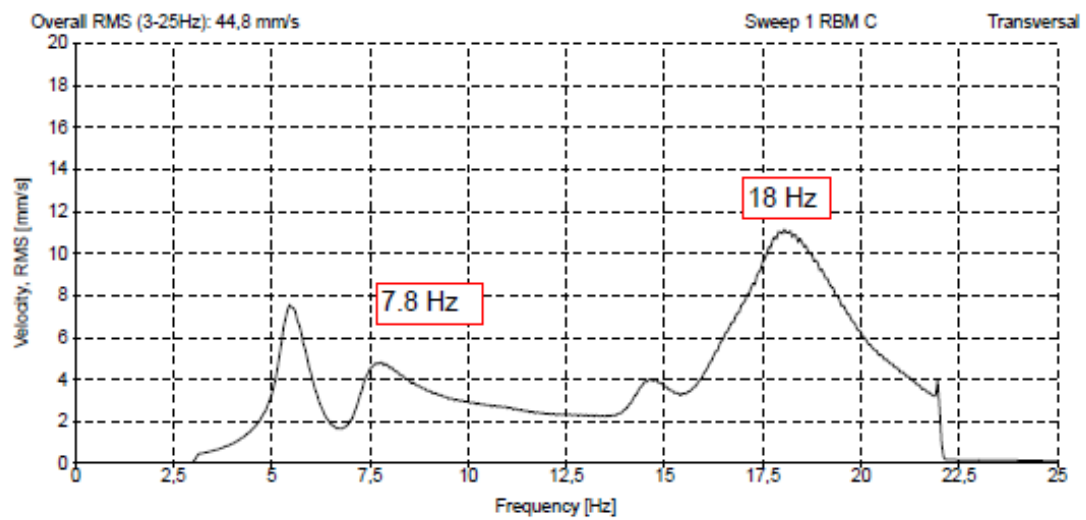
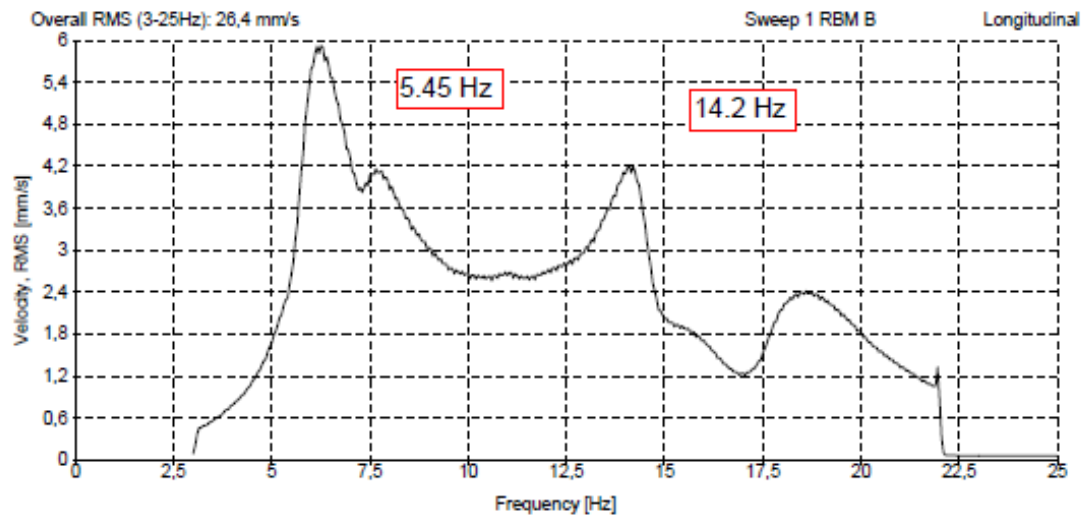
## **Appendix B**

Hydraulic Shaker Measurement FFT Spectrum – Point A1B

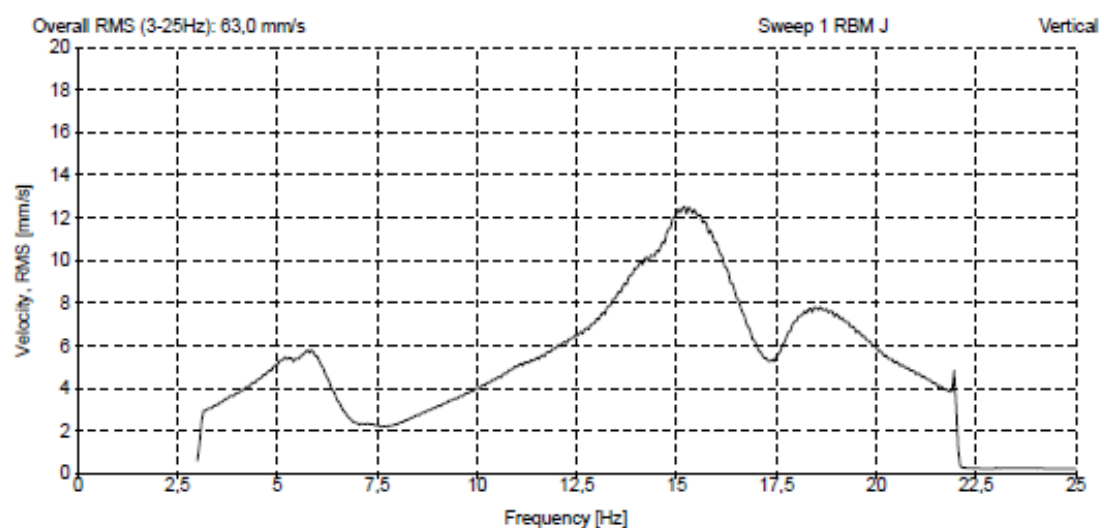
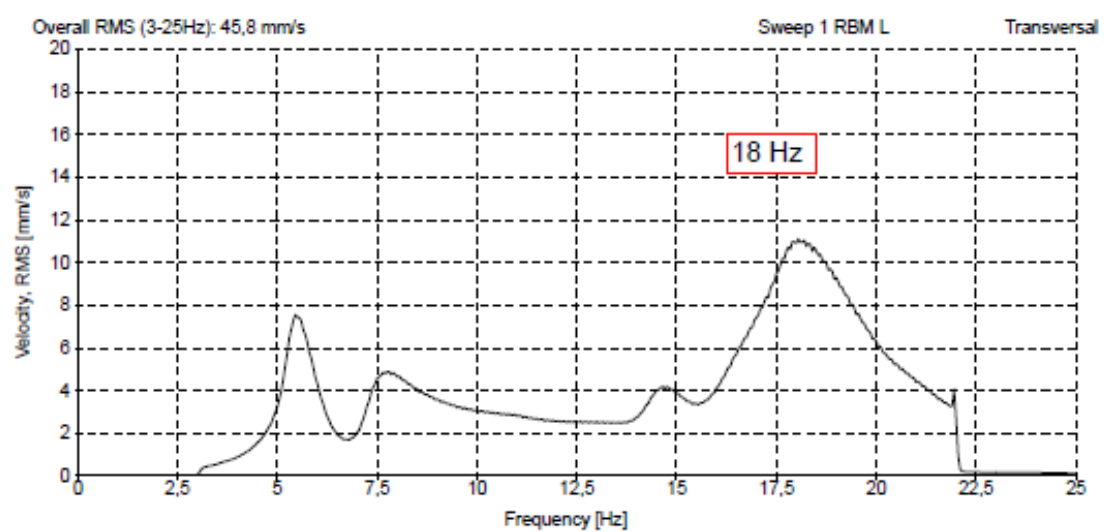
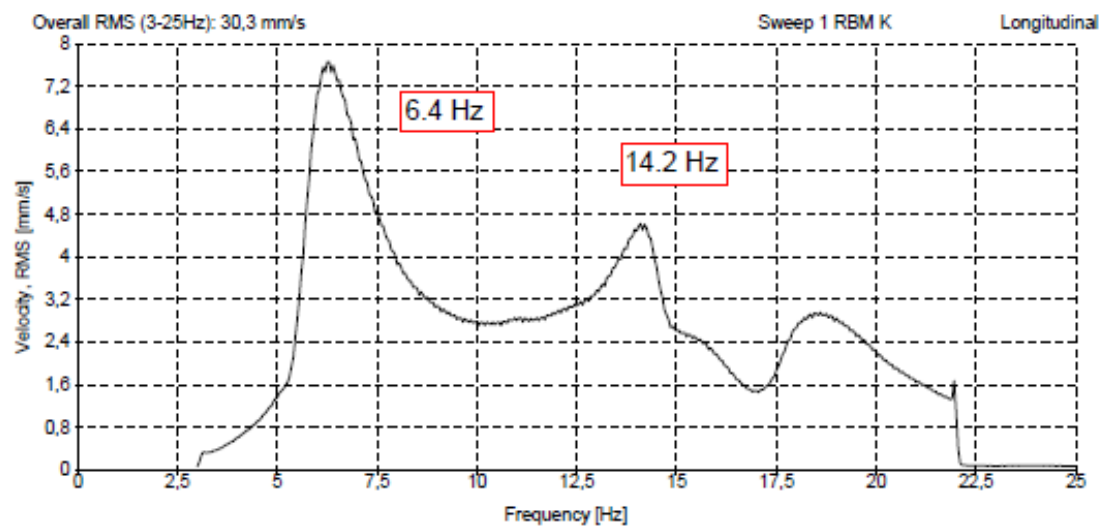




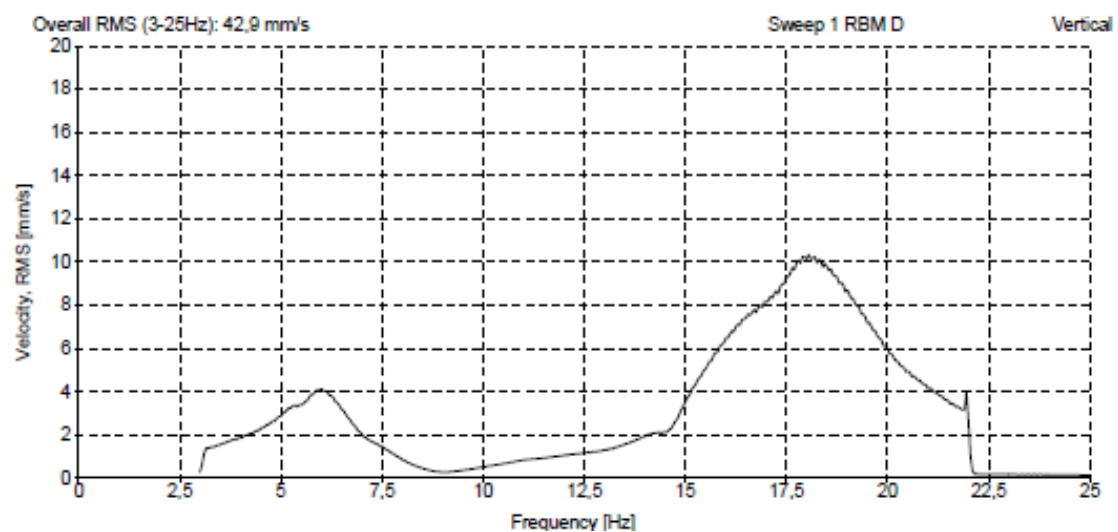
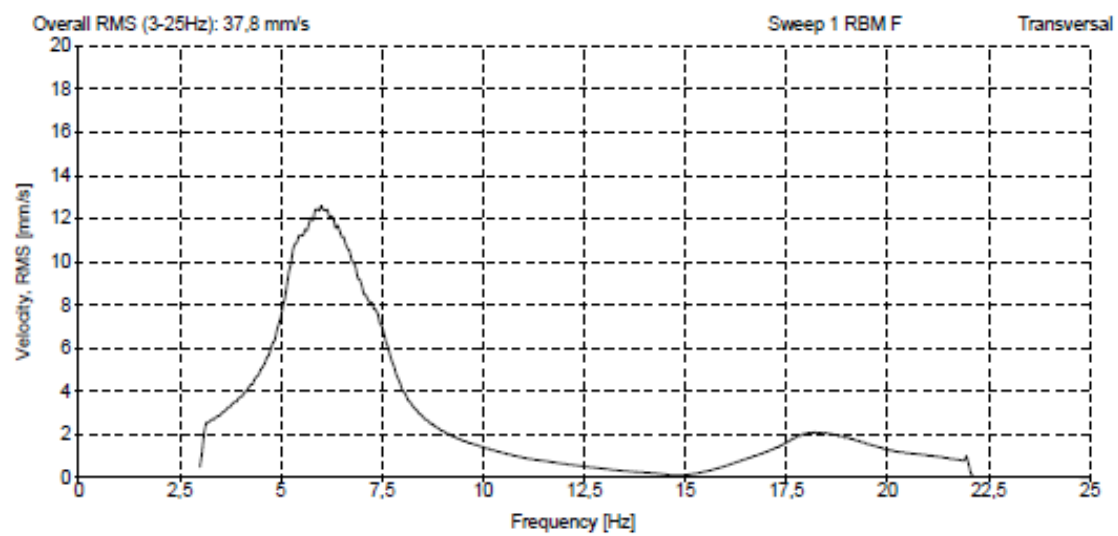
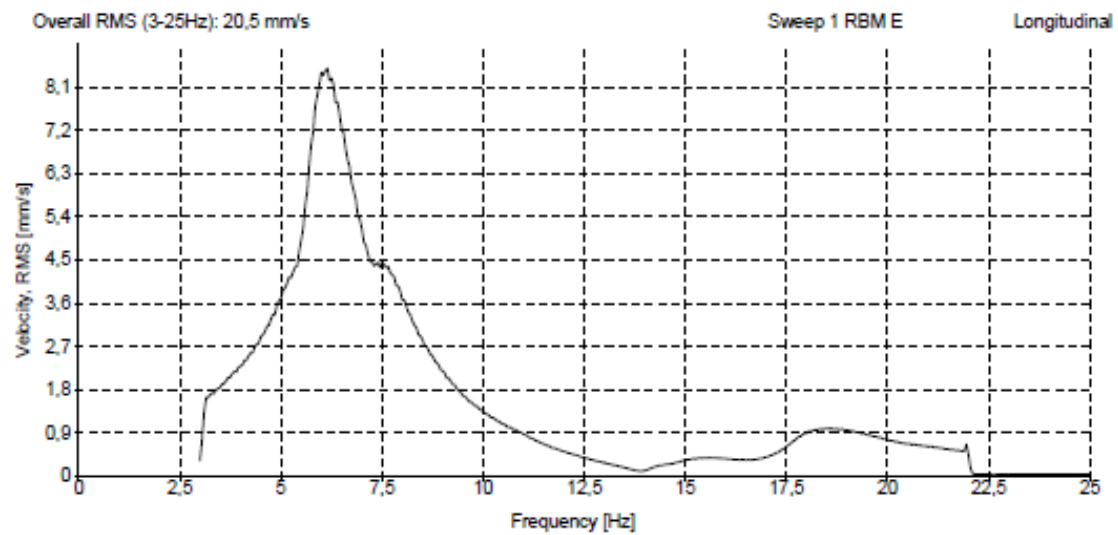
Hydraulic Shaker Measurement FFT Spectrum – Point A3



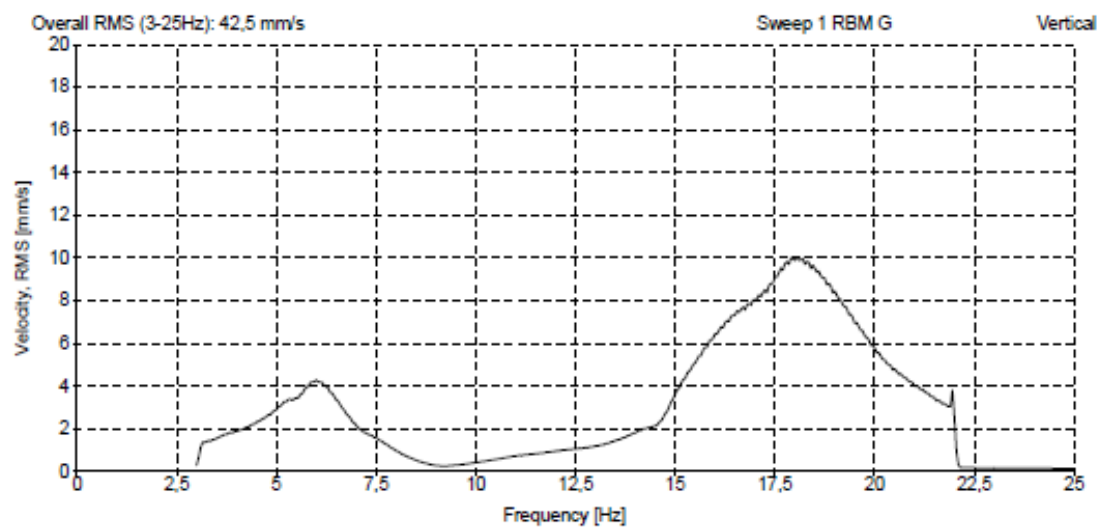
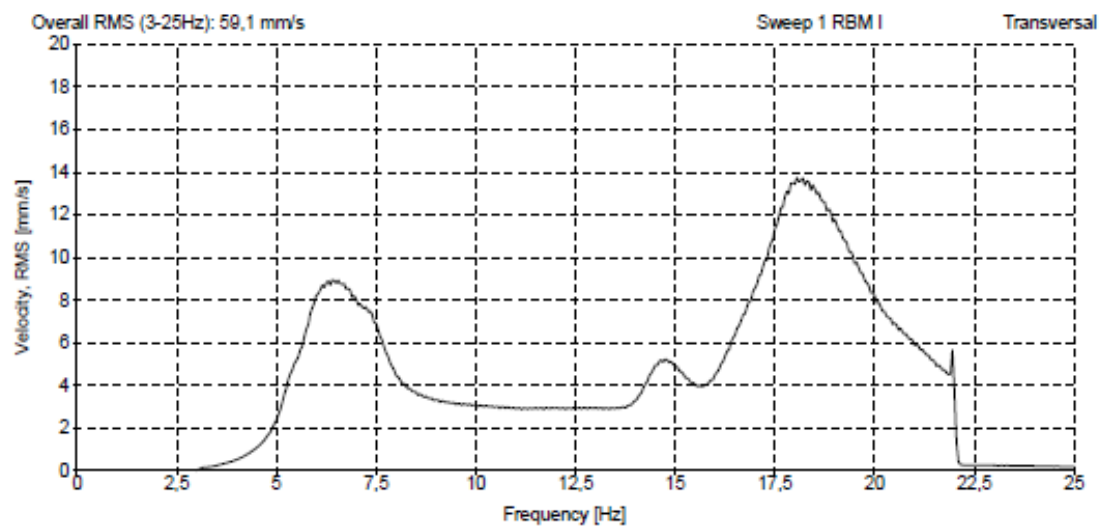
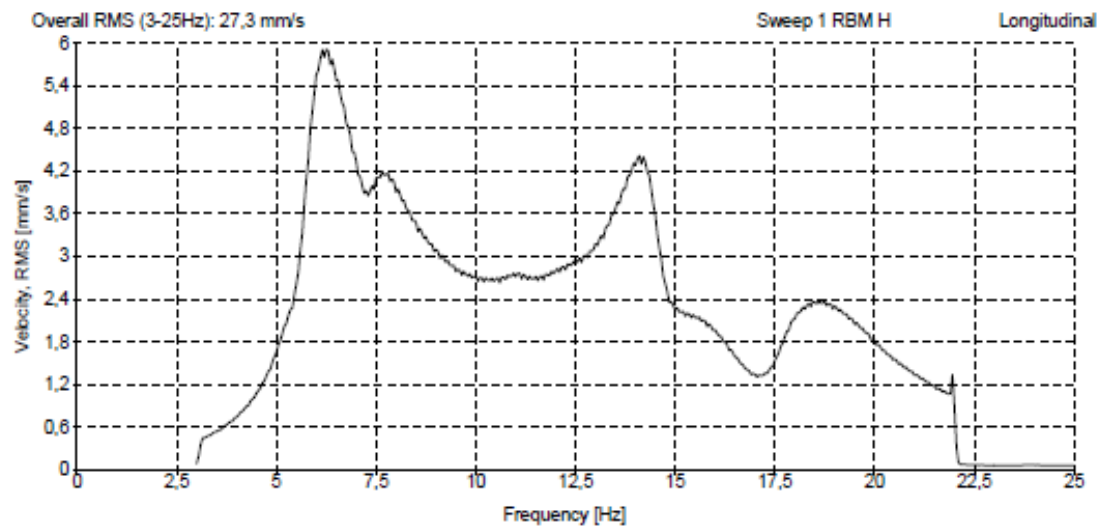
Hydraulic Shaker Measurement FFT Spectrum – Point A3b



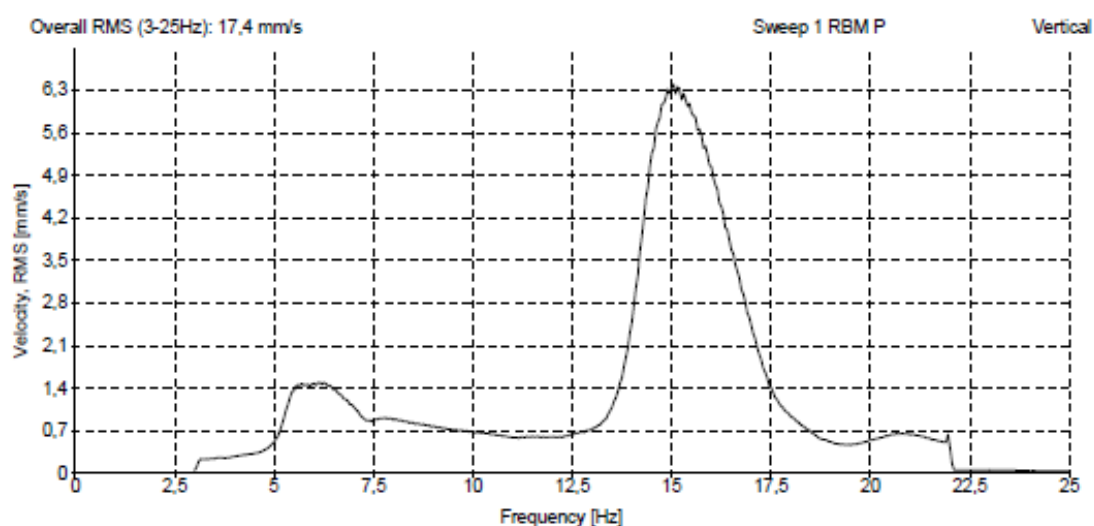
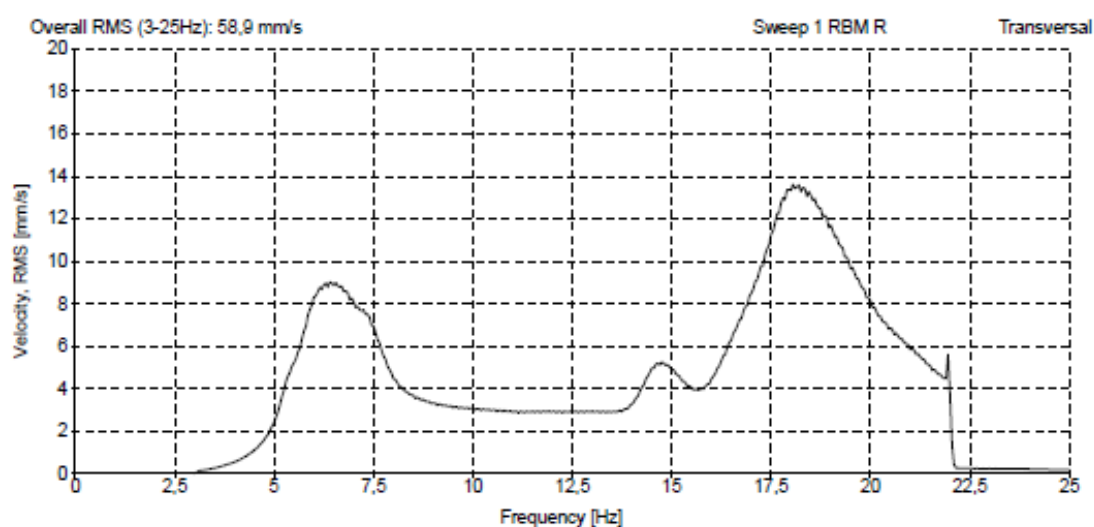
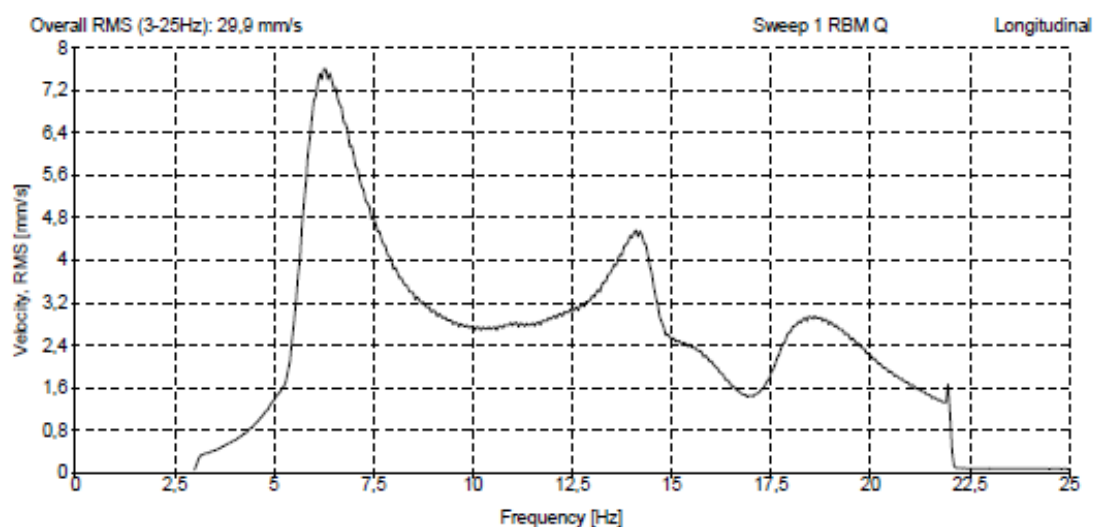
Hydraulic Shaker Measurement FFT Spectrum – Point C1



Hydraulic Shaker Measurement FFT Spectrum – Point C3



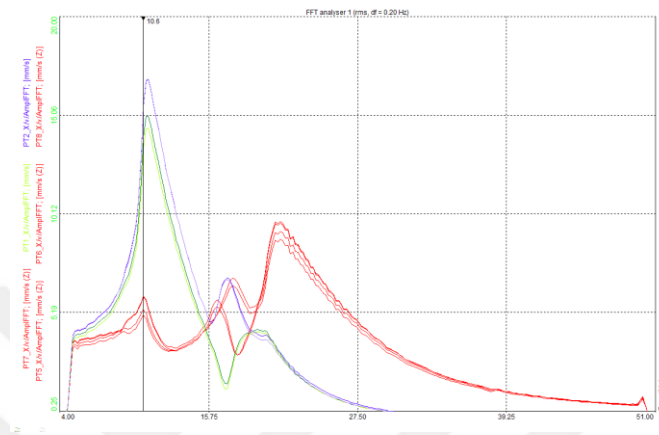
Hydraulic Shaker Measurement FFT Spectrum – Point C3b



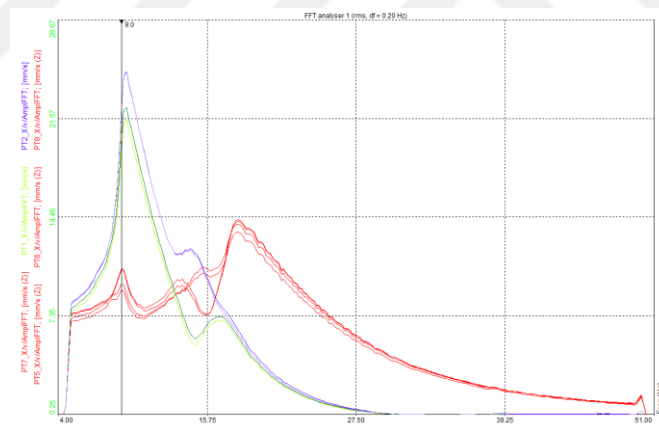
## Appendix C

### Electrical Cabinet Sine Sweep Measurements for Softer Elements

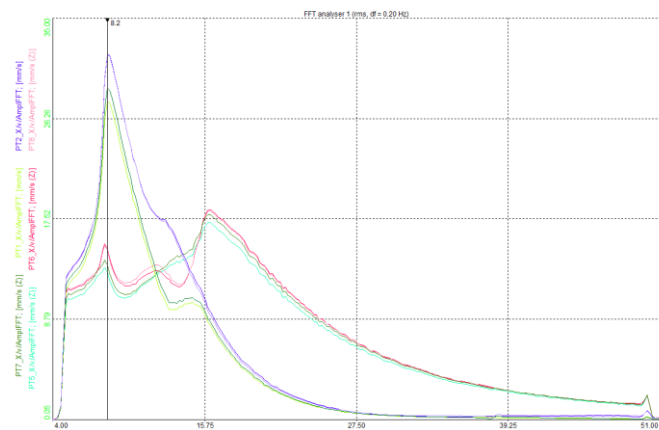
#### 57Kg Longitudinal – 5mm/s Sine Sweep



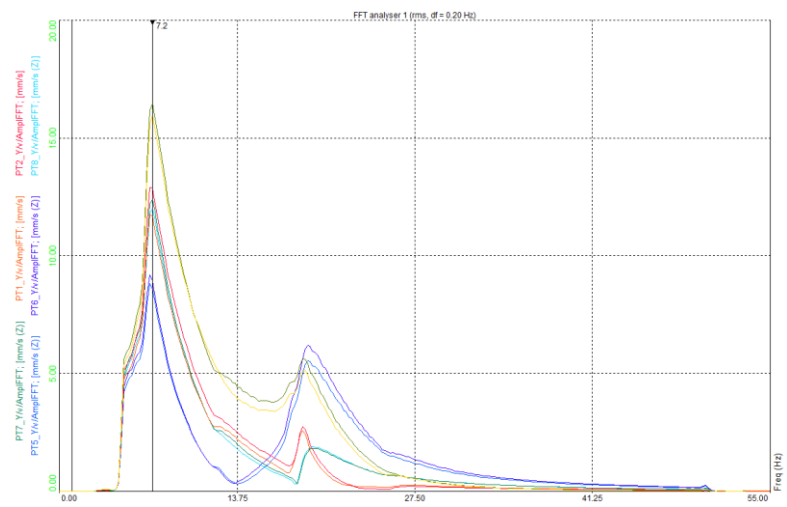
#### 57Kg Longitudinal – 10mm/s Sine Sweep



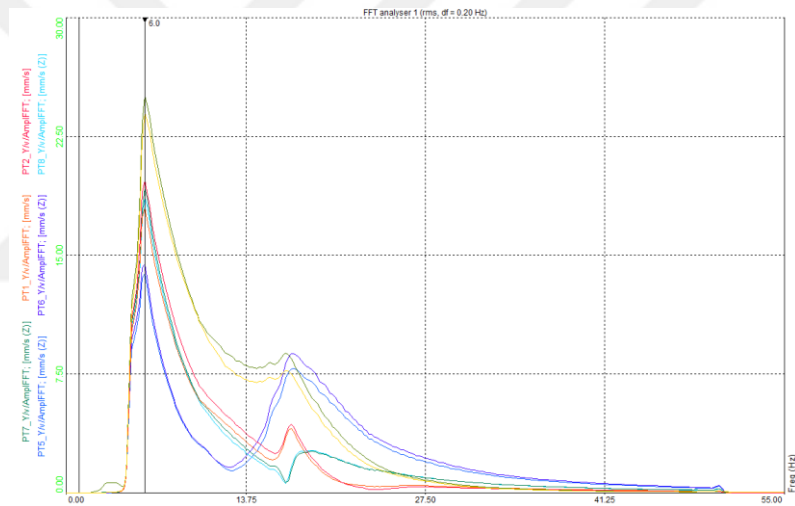
#### 57Kg Longitudinal – 15mm/s Sine Sweep



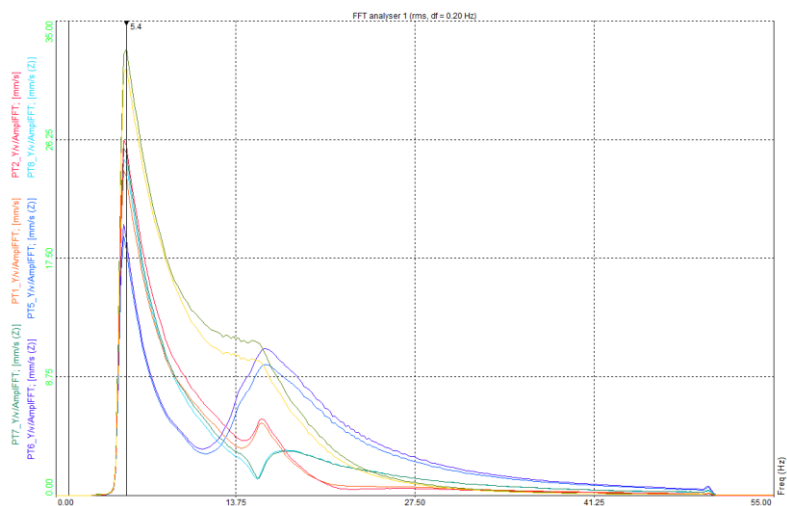
57Kg Transversal – 5mm/s Sine Sweep



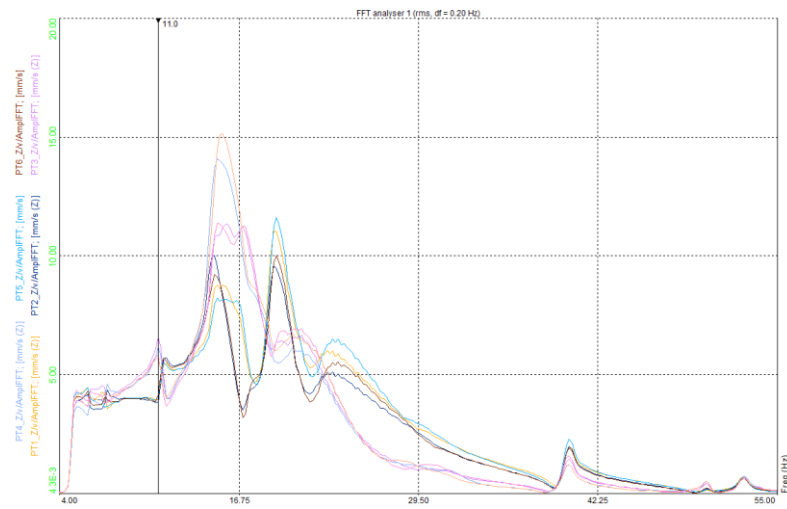
57Kg Transversal – 10mm/s Sine Sweep



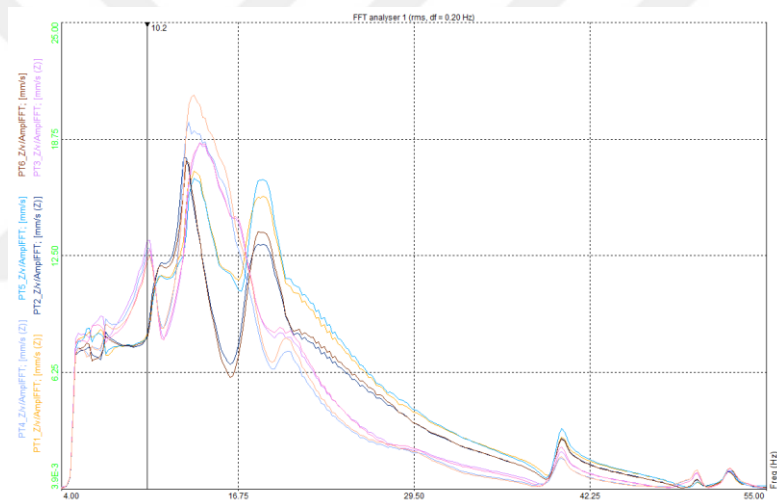
57Kg Transversal – 15mm/s Sine Sweep



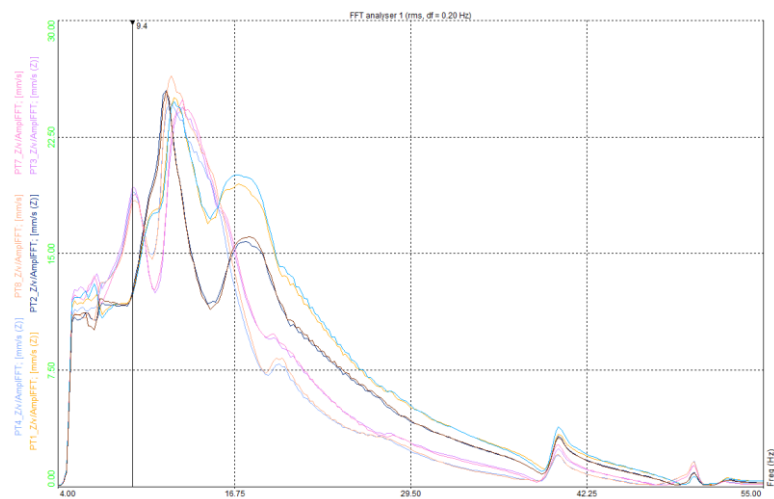
## 57Kg Vertical – 5mm/s Sine Sweep



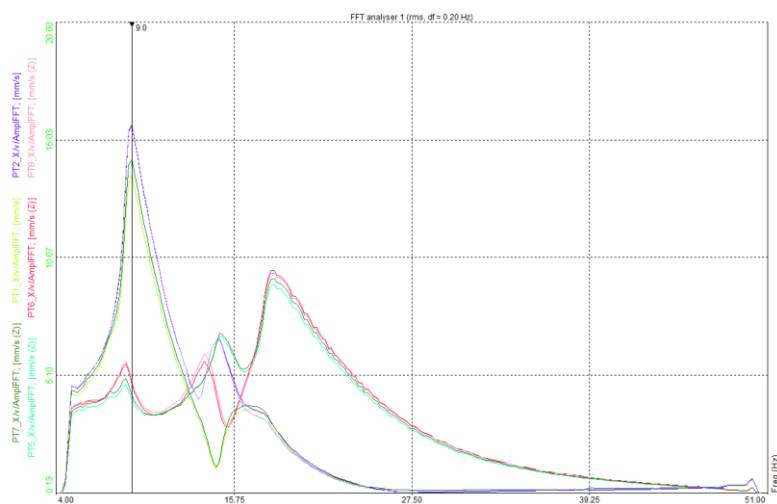
## 57Kg Vertical – 10mm/s Sine Sweep



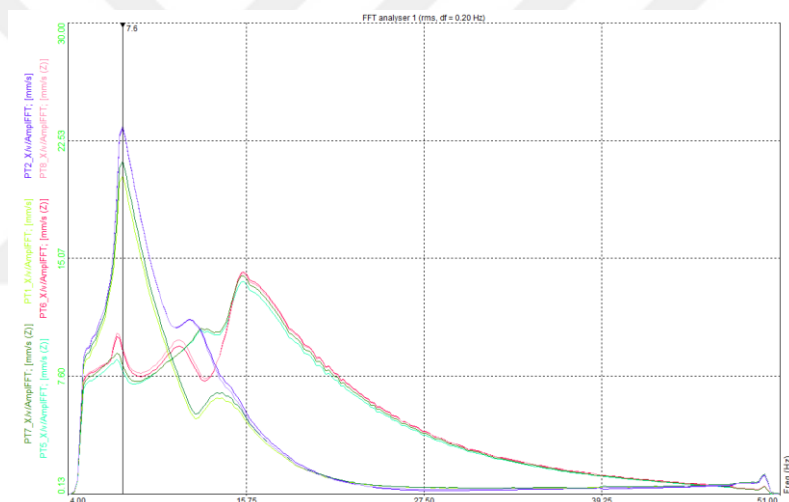
## 57Kg Vertical – 15mm/s Sine Sweep



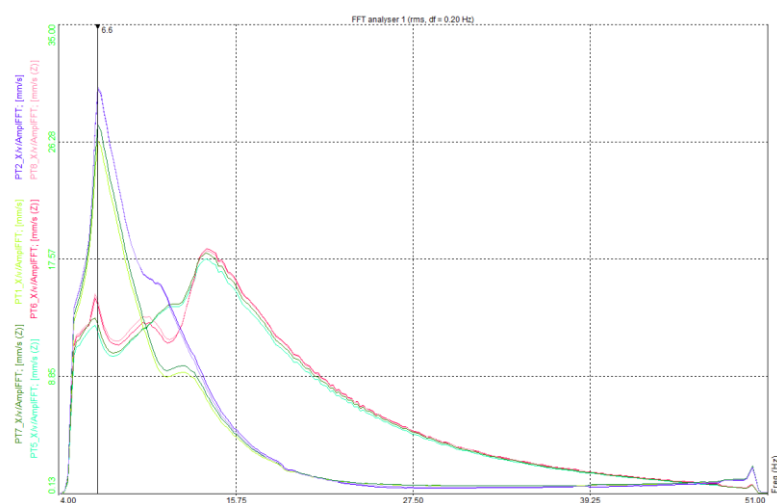
## 75Kg Longitudinal – 5mm/s Sine Sweep



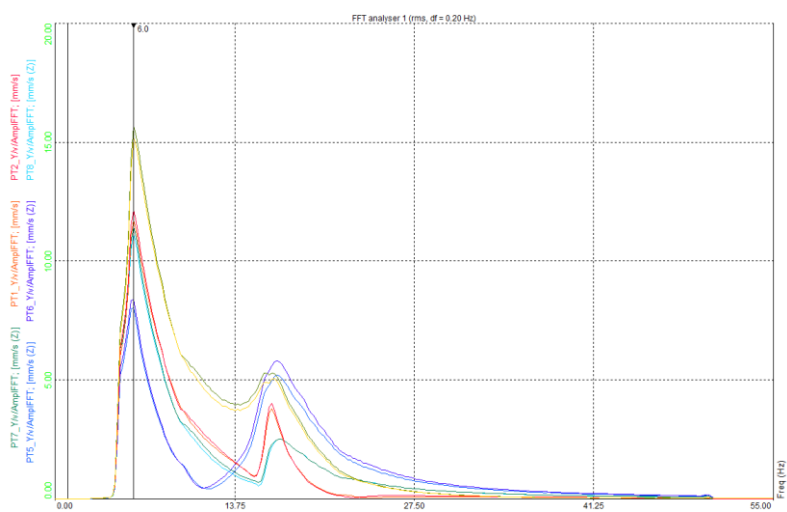
## 75Kg Longitudinal – 10mm/s Sine Sweep



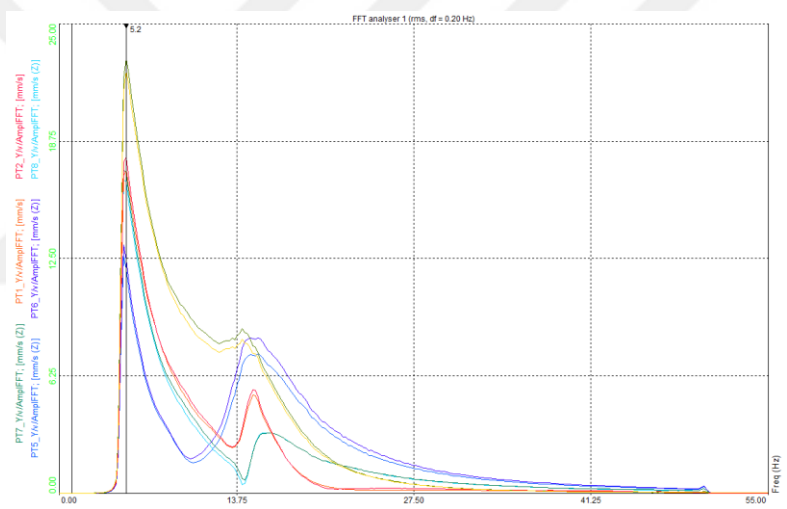
## 75Kg Longitudinal – 15mm/s Sine Sweep



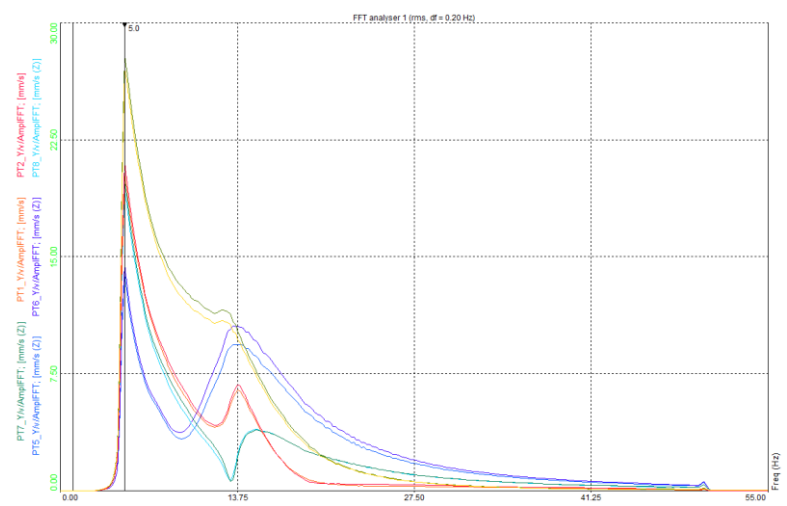
## 75Kg Transversal – 5mm/s Sine Sweep



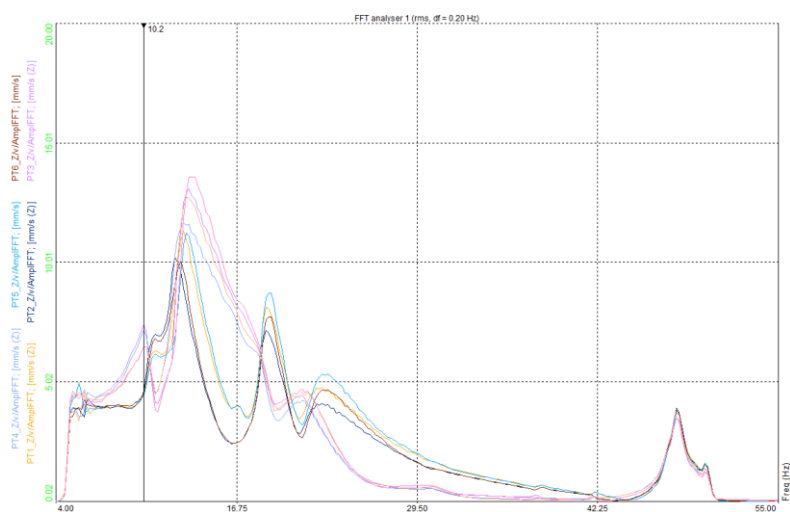
## 75Kg Transversal – 10mm/s Sine Sweep



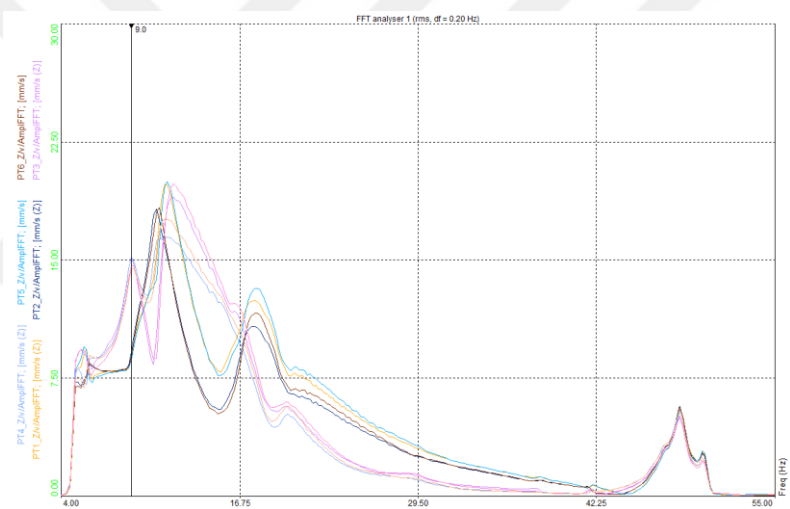
## 75Kg Transversal – 15mm/s Sine Sweep



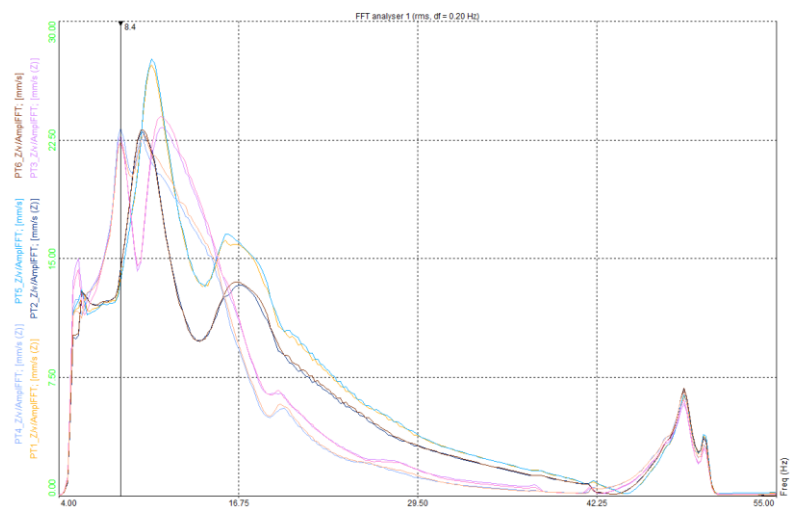
## 75Kg Vertical – 5mm/s Sine Sweep



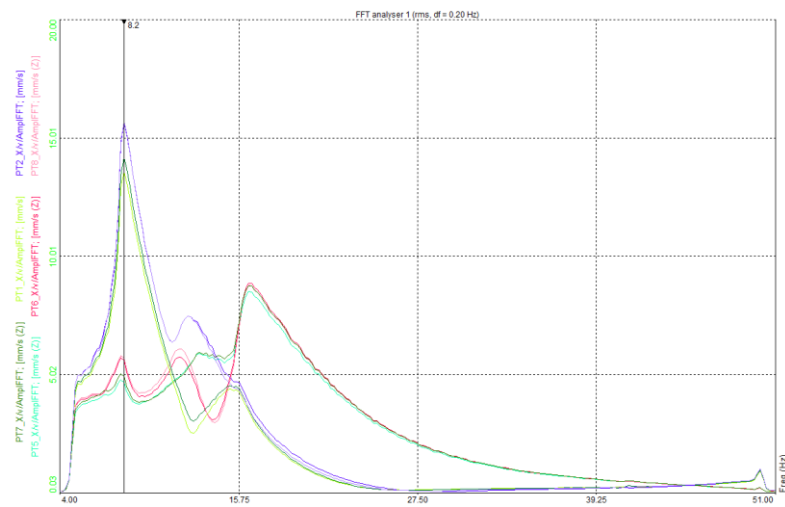
## 75Kg Vertical – 10mm/s Sine Sweep



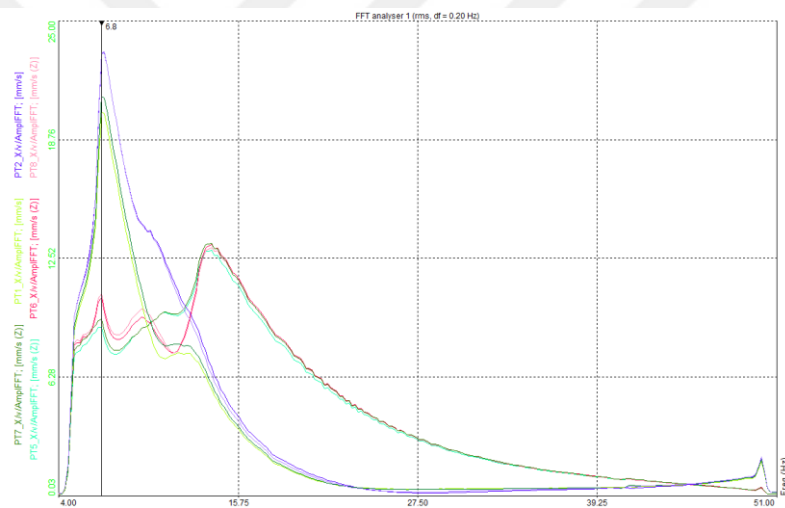
## 75Kg Vertical – 15mm/s Sine Sweep



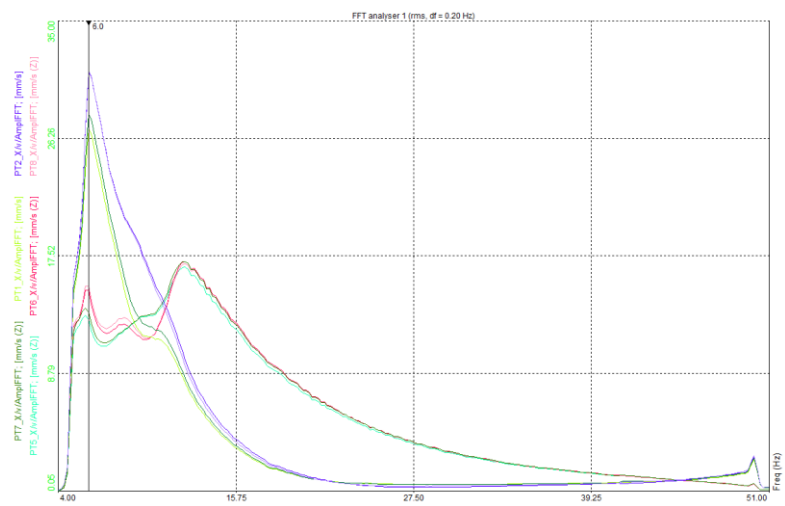
## 90Kg Longitudinal – 5mm/s Sine Sweep



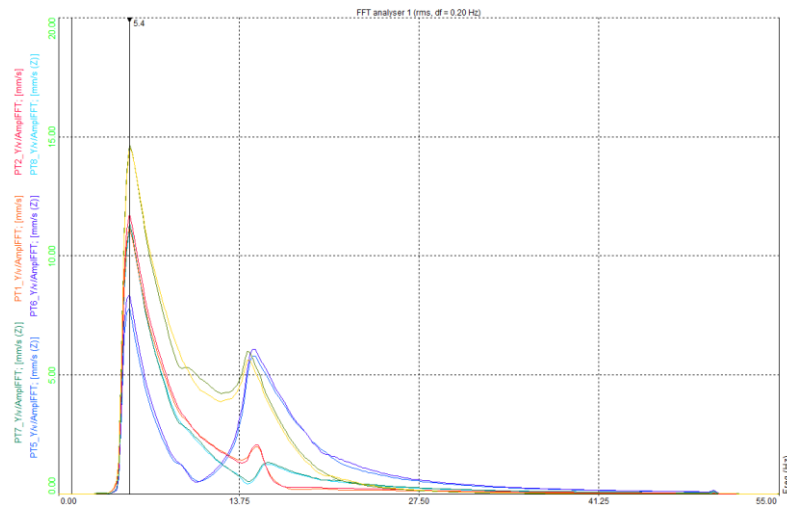
## 90Kg Longitudinal – 10mm/s Sine Sweep



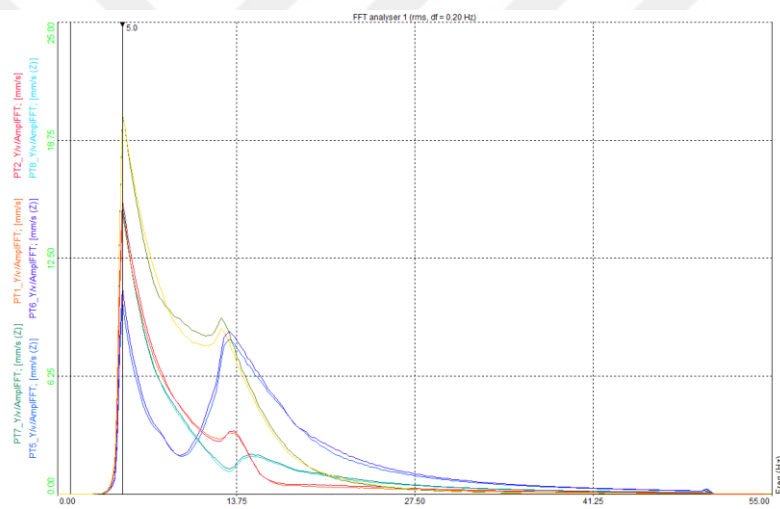
## 90Kg Longitudinal – 15mm/s Sine Sweep



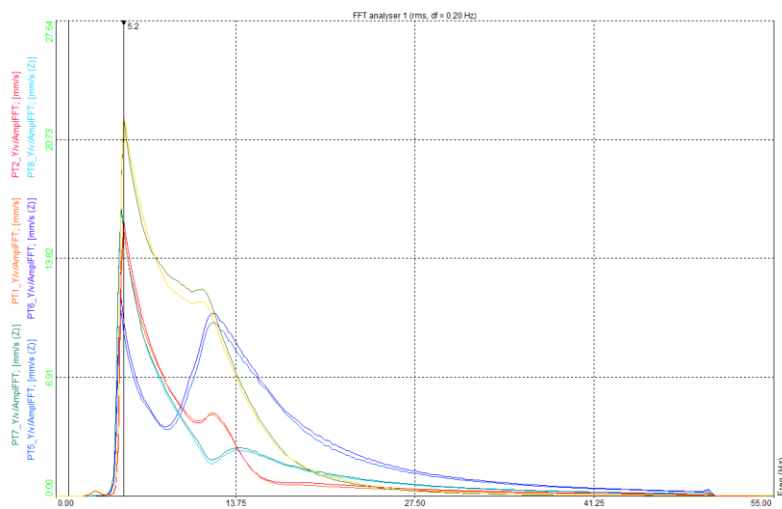
## 90Kg Transversal – 5mm/s Sine Sweep



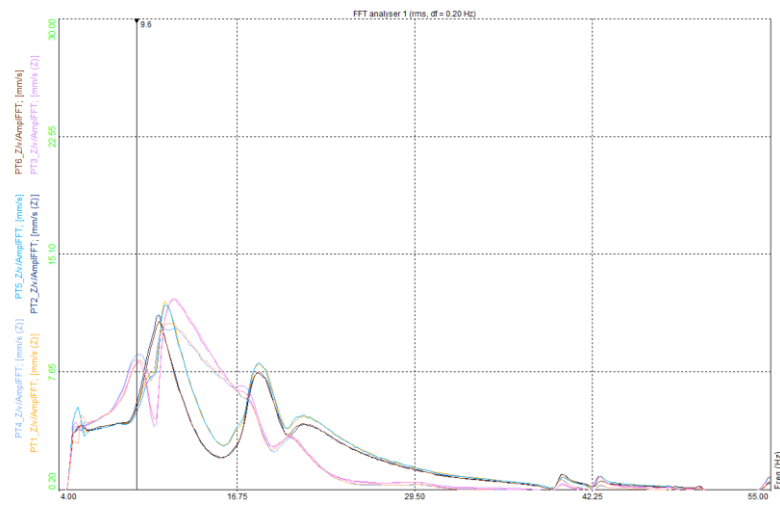
## 90Kg Transversal – 10mm/s Sine Sweep



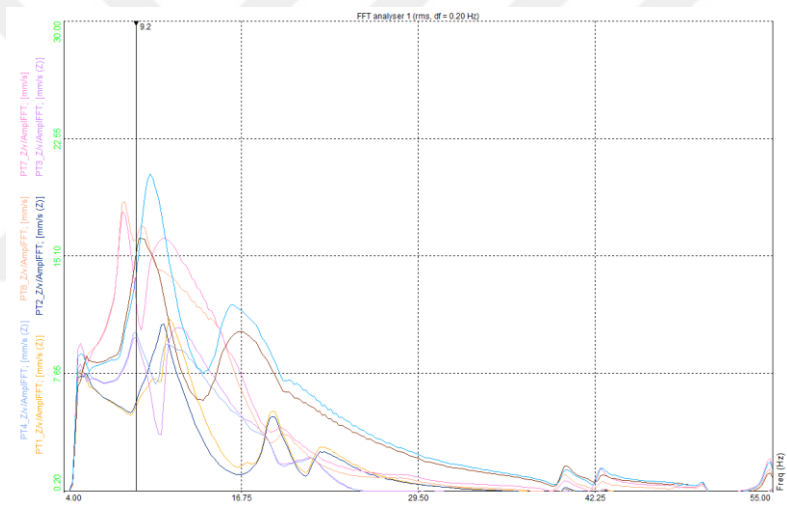
## 90Kg Transversal – 15mm/s Sine Sweep



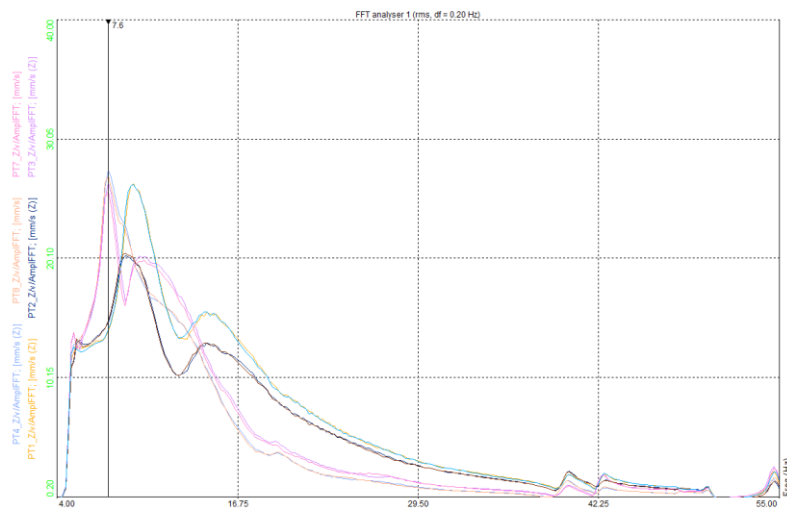
## 90Kg Vertical – 5mm/s Sine Sweep



## 90Kg Vertical – 10mm/s Sine Sweep

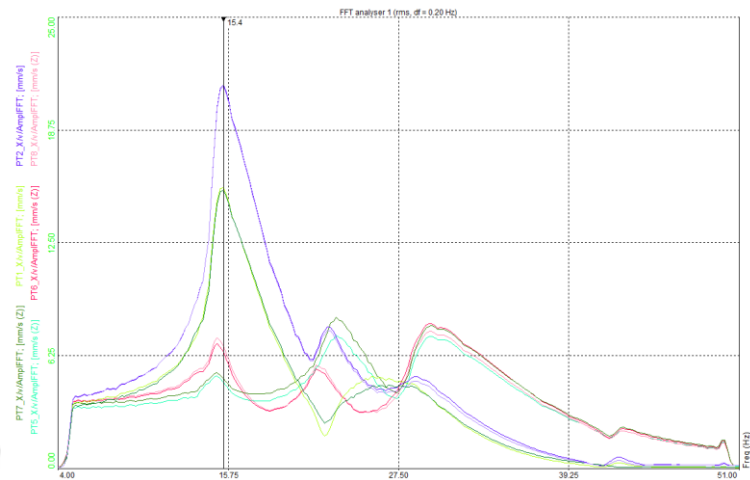


## 90Kg Vertical – 15mm/s Sine Sweep

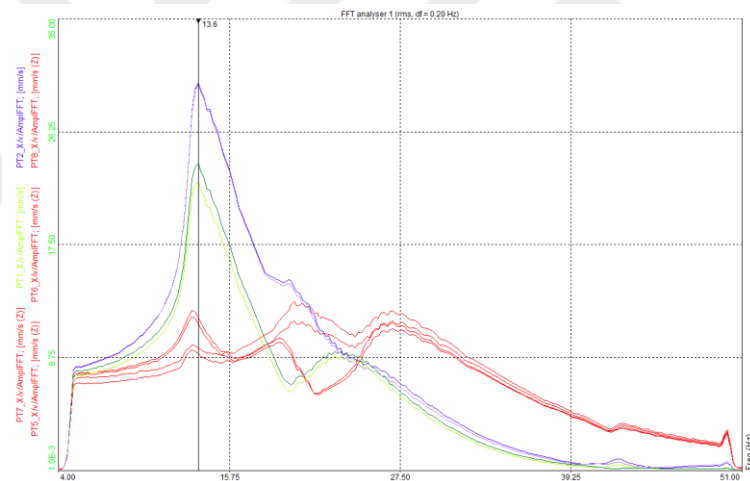


## Electrical Cabinet Sine Sweep Measurements for Stiffer Elements

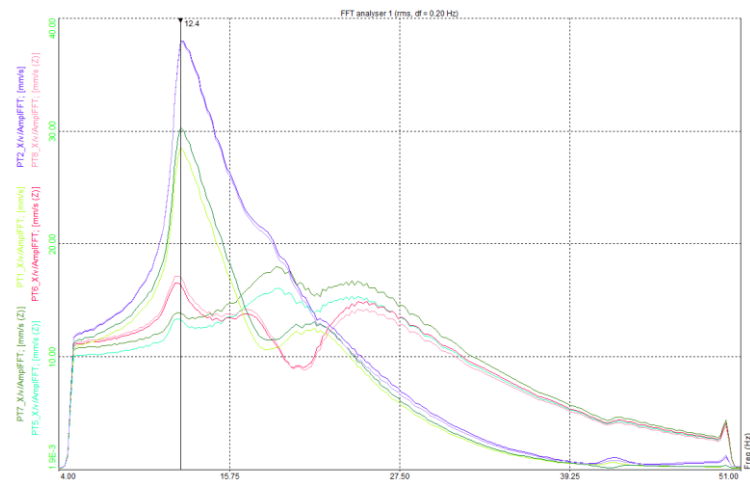
### 90Kg Longitudinal – 5mm/s Sine Sweep



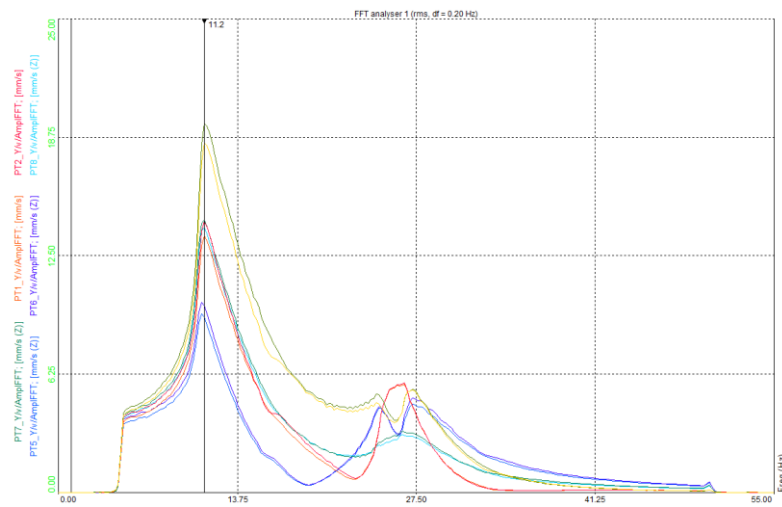
### 90Kg Longitudinal – 10mm/s Sine Sweep



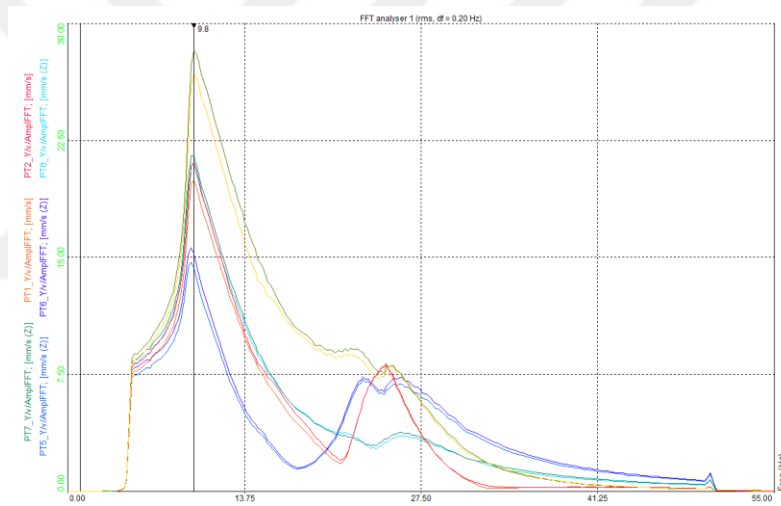
### 90Kg Longitudinal – 15mm/s Sine Sweep



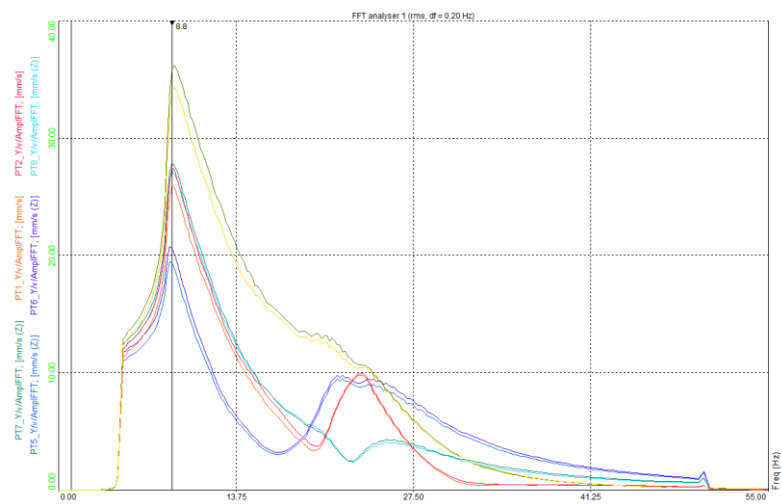
## 90Kg Transversal – 5mm/s Sine Sweep



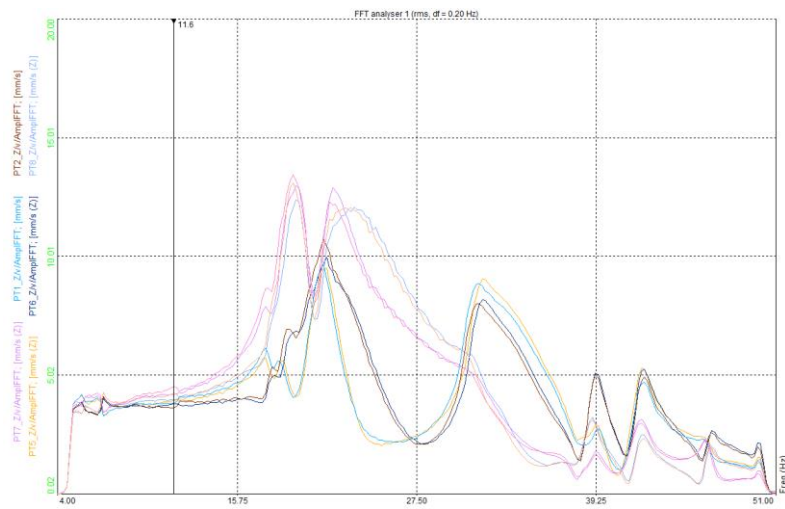
## 90Kg Transversal – 10mm/s Sine Sweep



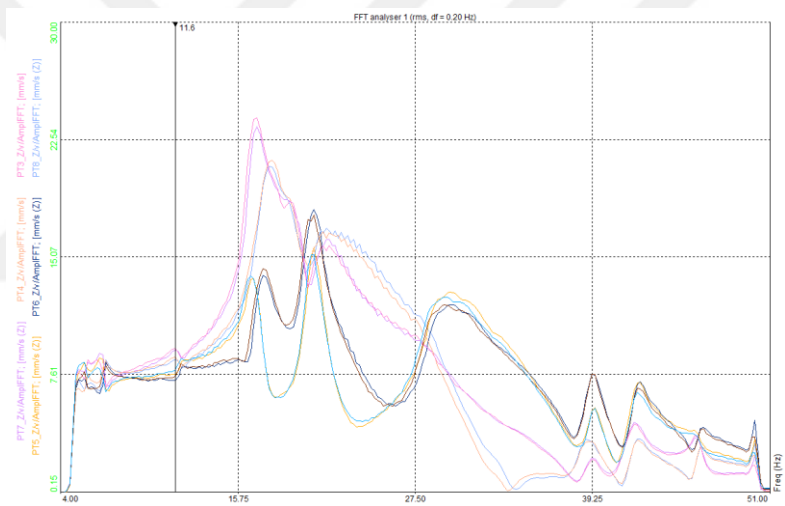
## 90Kg Transversal – 15mm/s Sine Sweep



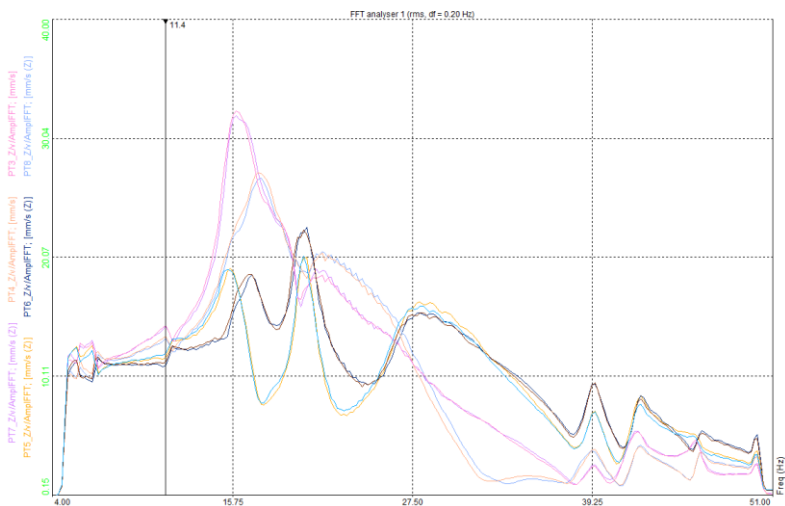
## 90Kg Vertical – 5mm/s Sine Sweep



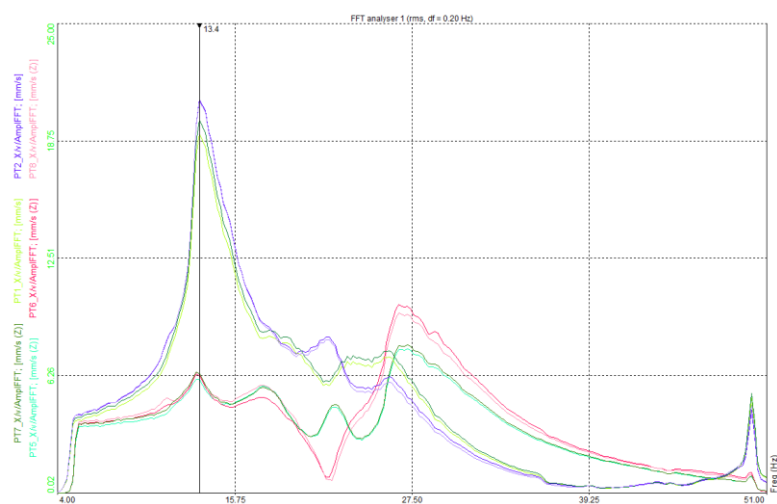
## 90Kg Vertical – 10mm/s Sine Sweep



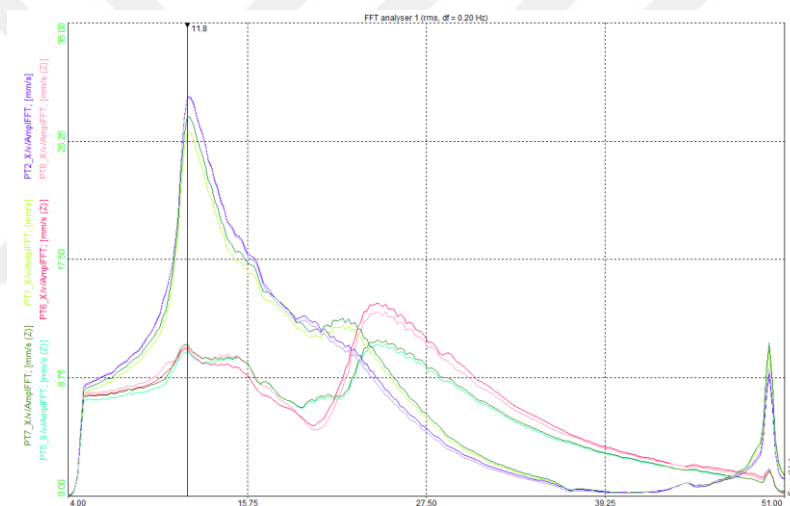
## 90Kg Vertical – 15mm/s Sine Sweep



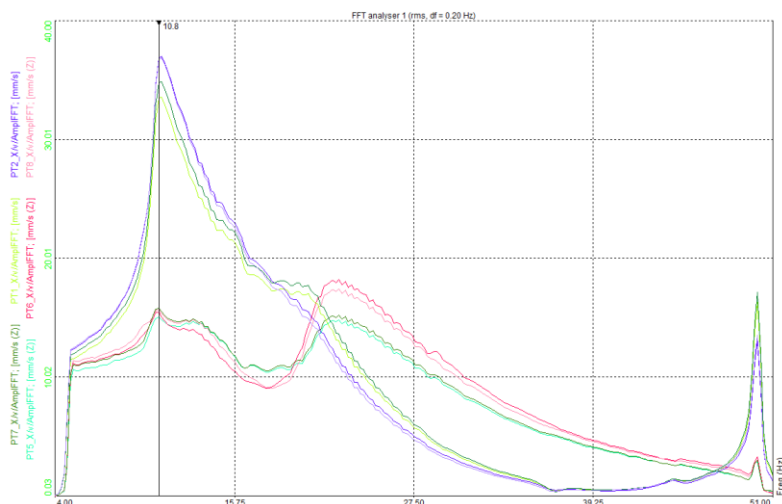
## 120Kg Longitudinal – 5mm/s Sine Sweep



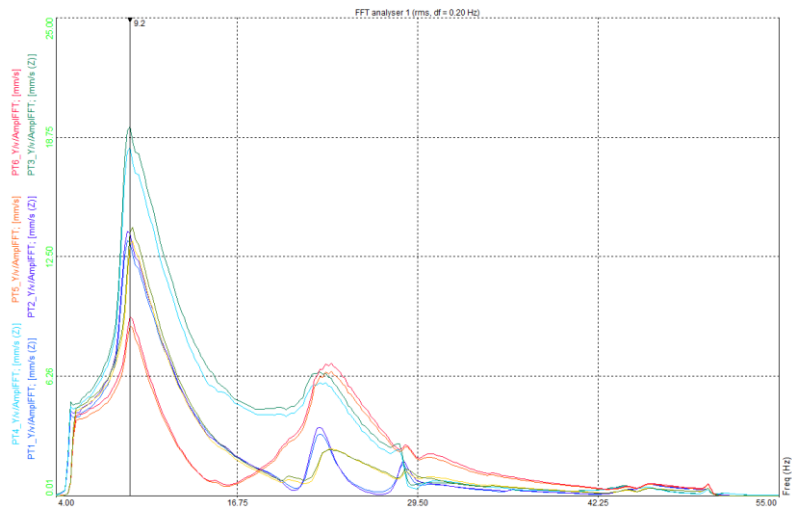
## 120Kg Longitudinal – 10mm/s Sine Sweep



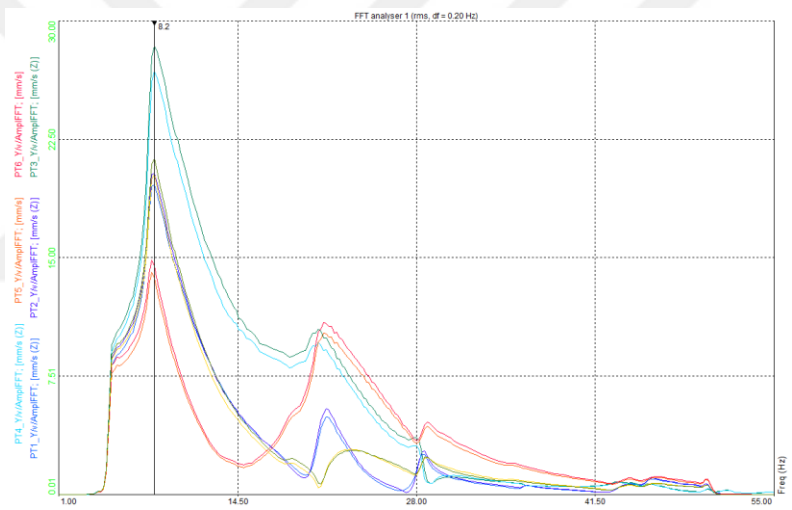
## 120Kg Longitudinal – 15mm/s Sine Sweep



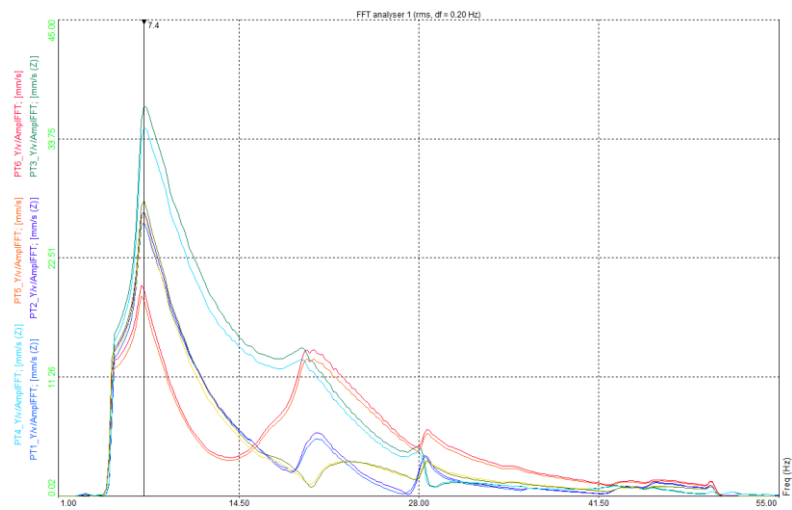
## 120Kg Transversal – 5mm/s Sine Sweep



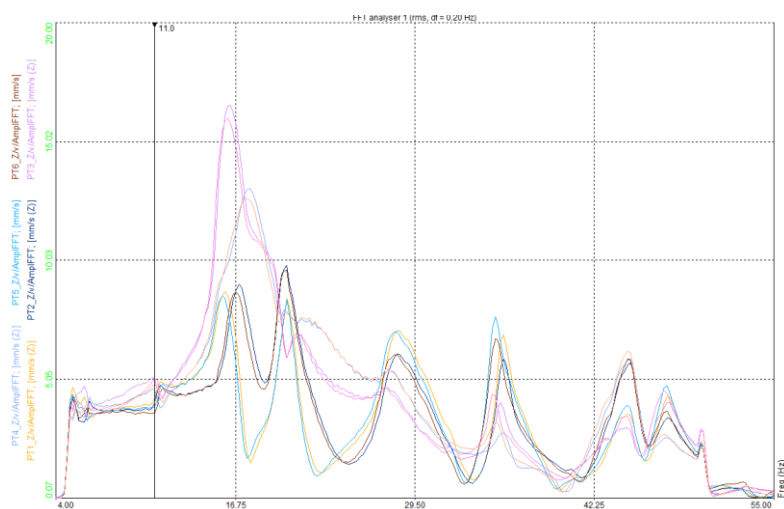
## 120Kg Transversal – 10mm/s Sine Sweep



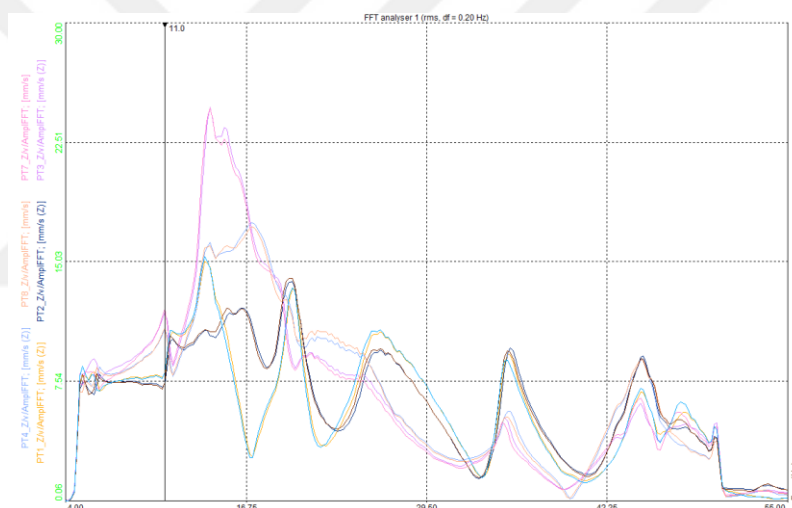
## 120Kg Transversal – 15mm/s Sine Sweep



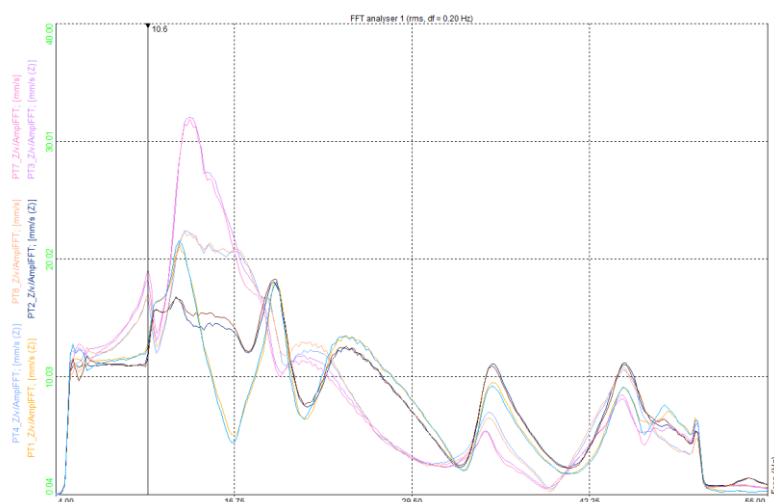
## 120Kg Vertical – 5mm/s Sine Sweep



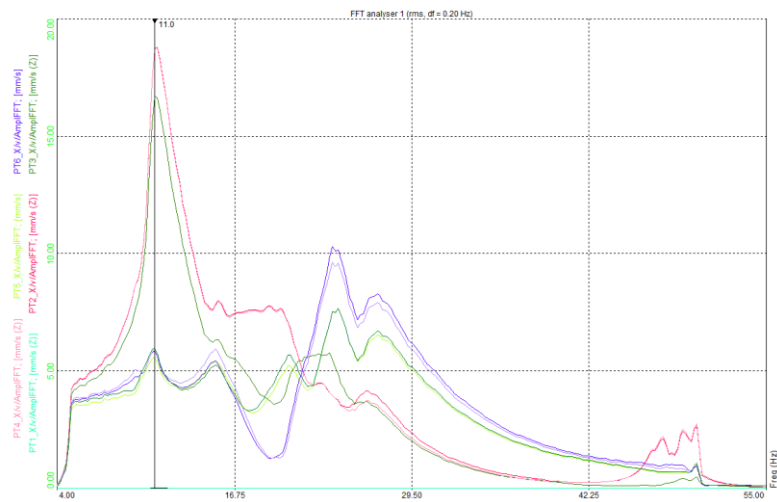
## 120Kg Vertical – 10mm/s Sine Sweep



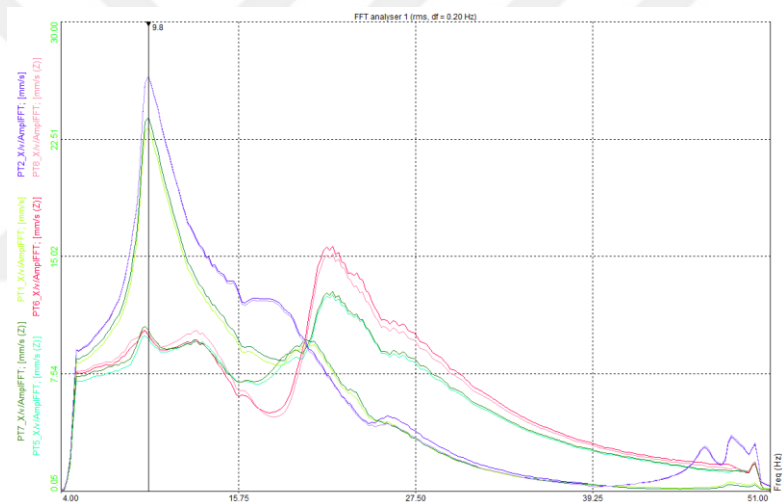
## 120Kg Vertical – 15mm/s Sine Sweep



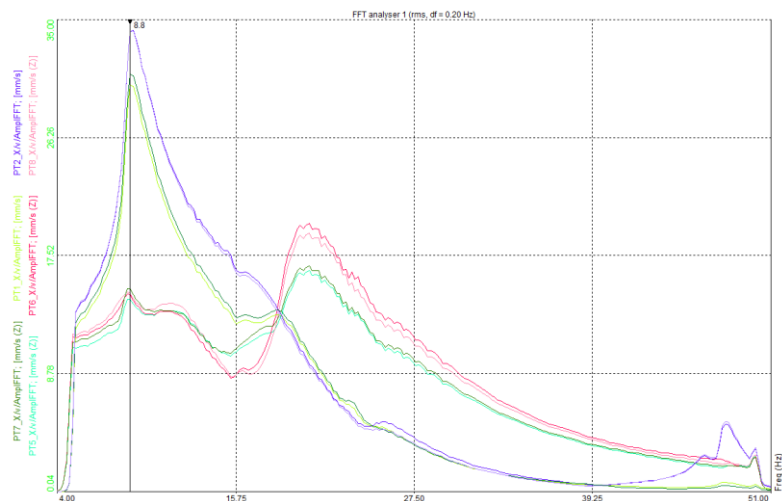
## 150Kg Longitudinal – 5mm/s Sine Sweep



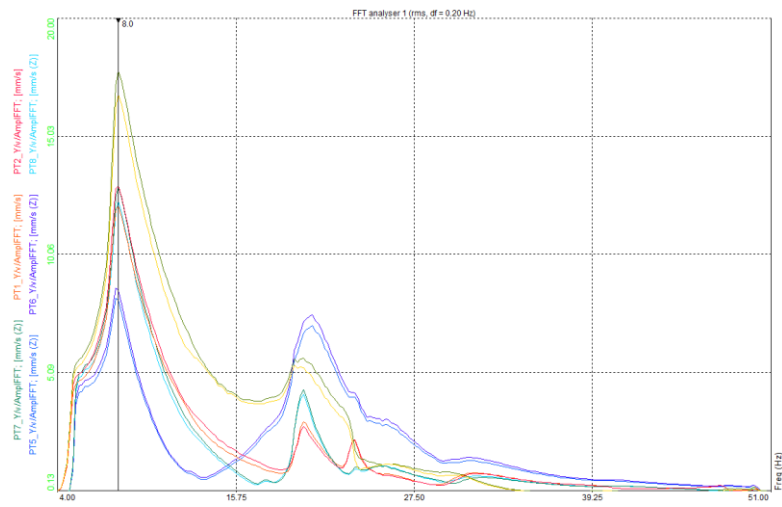
## 150Kg Longitudinal – 10mm/s Sine Sweep



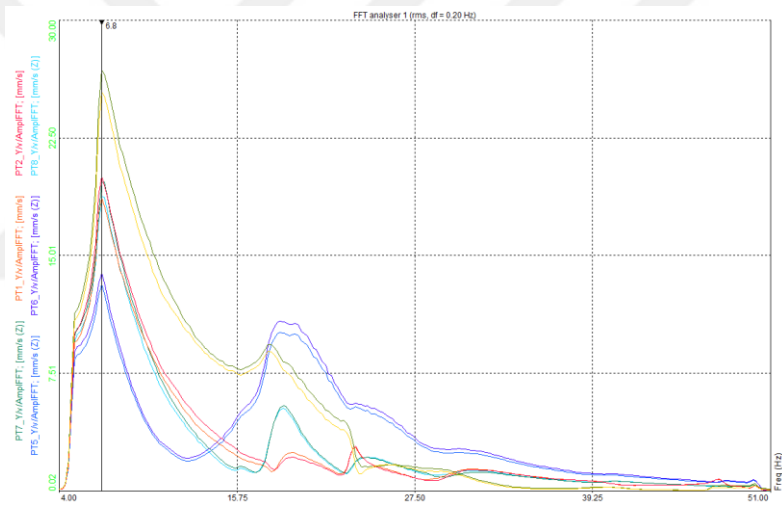
## 150Kg Longitudinal – 15mm/s Sine Sweep



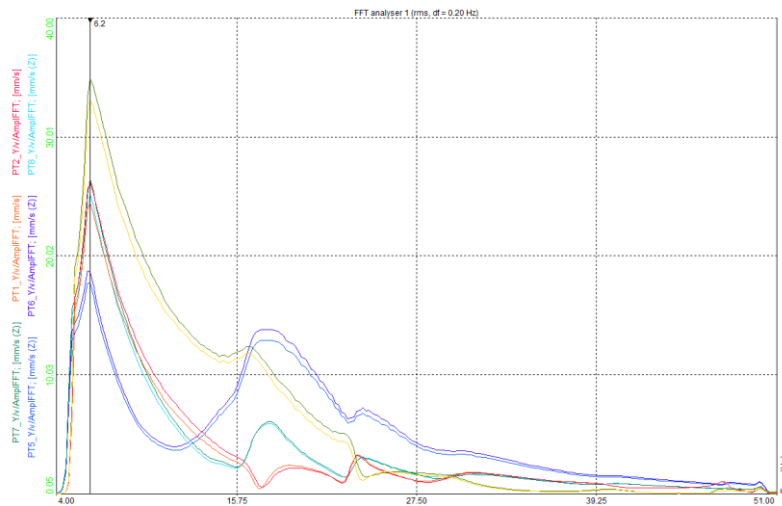
## 150Kg Transversal – 5mm/s Sine Sweep



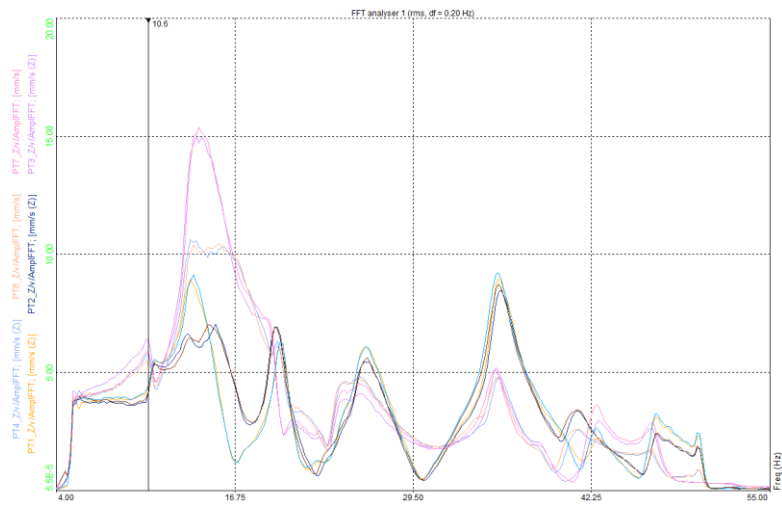
## 150Kg Transversal – 10mm/s Sine Sweep



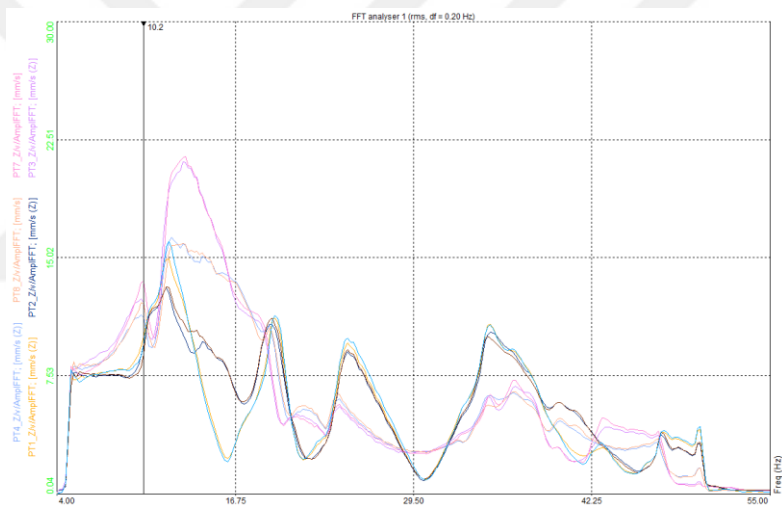
## 150Kg Transversal – 15mm/s Sine Sweep



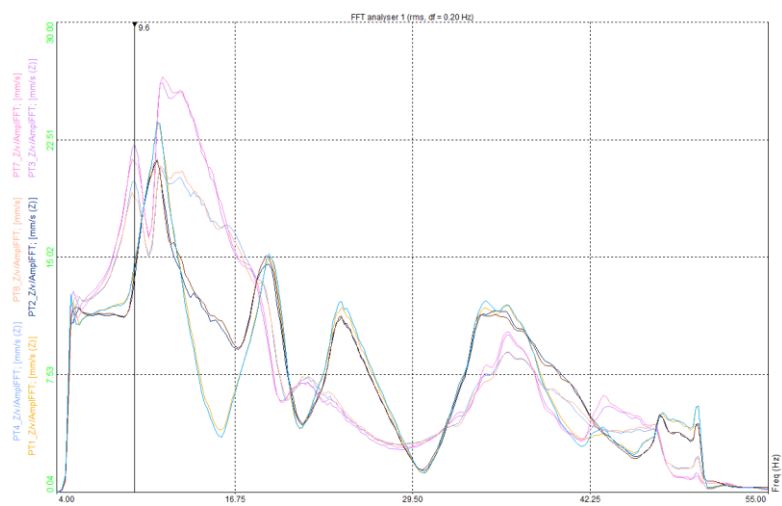
## 150Kg Vertical – 5mm/s Sine Sweep



150Kg Vertical – 10mm/s Sine Sweep

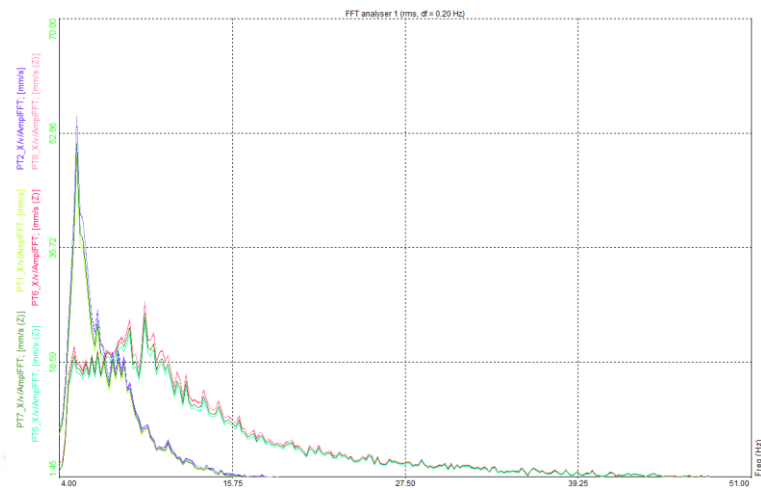


150Kg Vertical – 15mm/s Sine Sweep

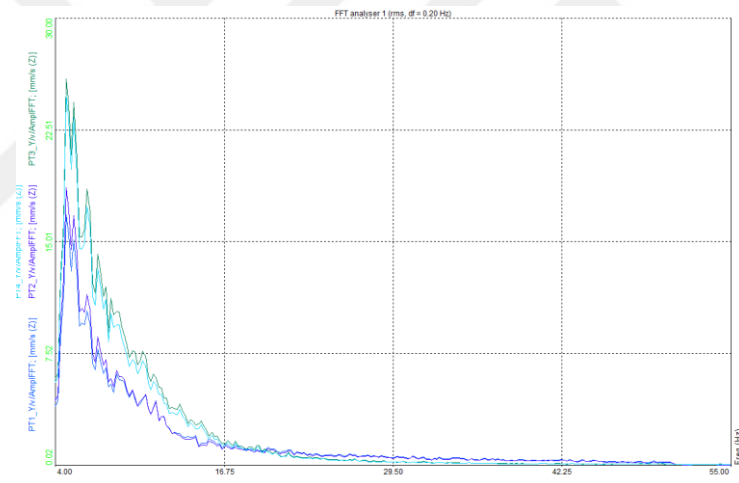


## Electrical Cabinet Random Vibration Measurements for Softer Elements

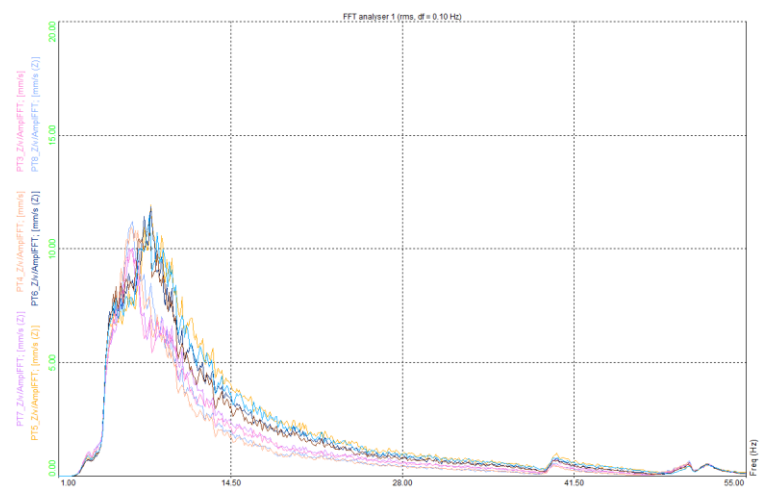
### 57kg Longitudinal – Softer Elements



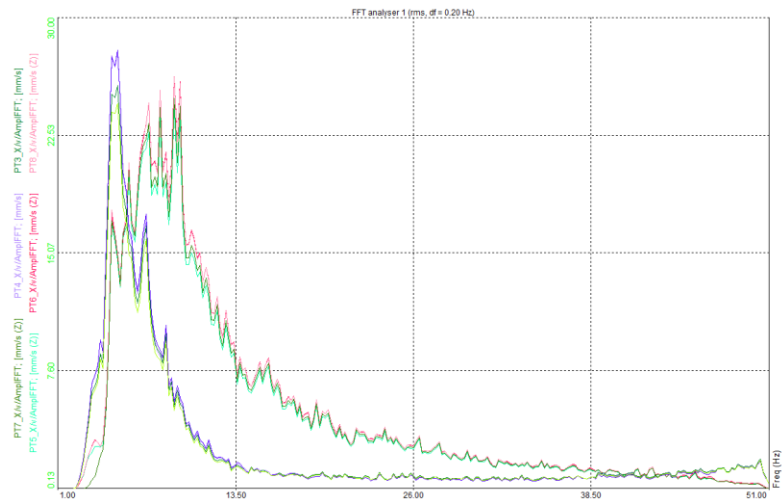
### 57kg Transversal – Softer Elements



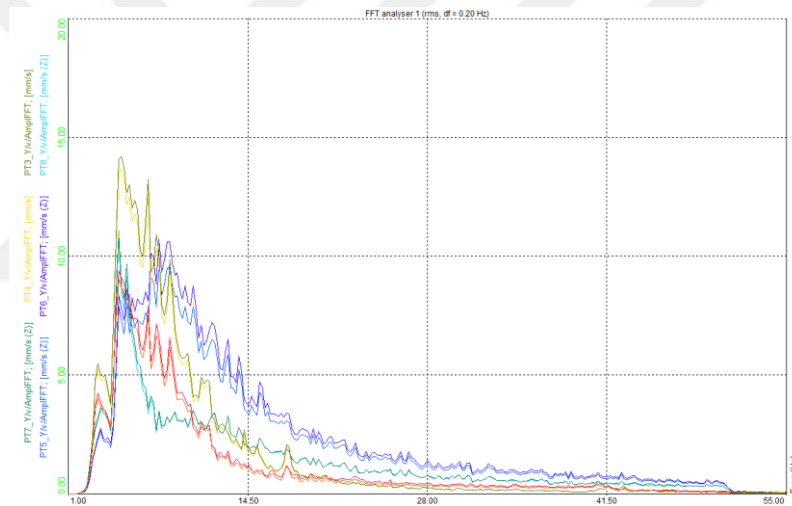
### 57kg Vertical – Softer Elements



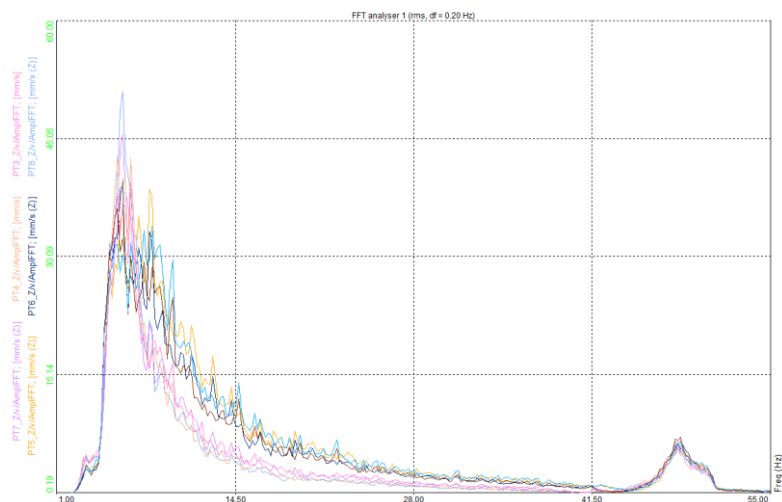
## 75kg Longitudinal – Softer Elements



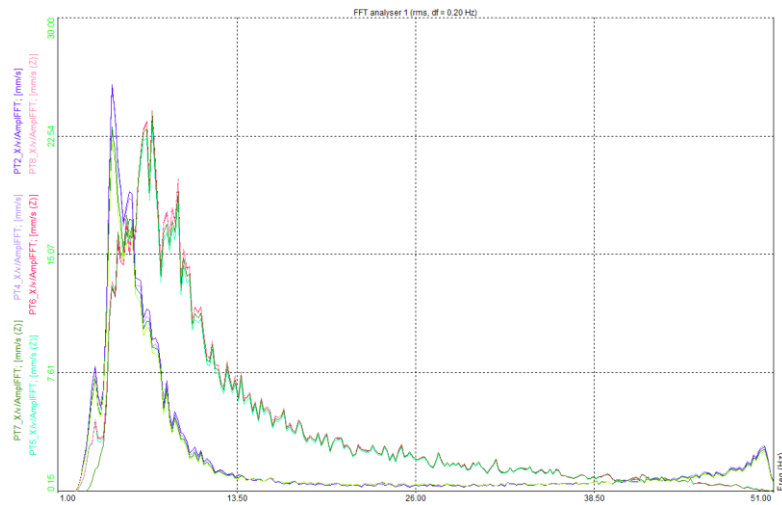
## 75kg Transversal – Softer Elements



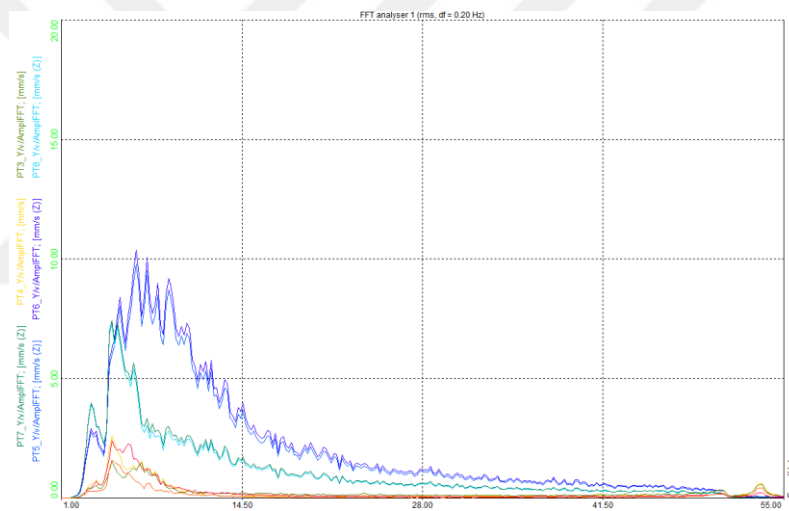
## 75kg Vertical – Softer Elements



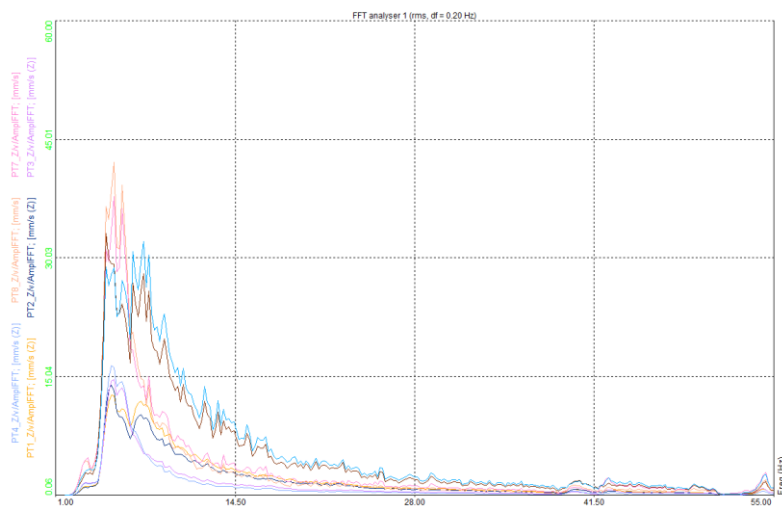
## 90kg Longitudinal – Softer Elements



## 90kg Transversal – Softer Elements

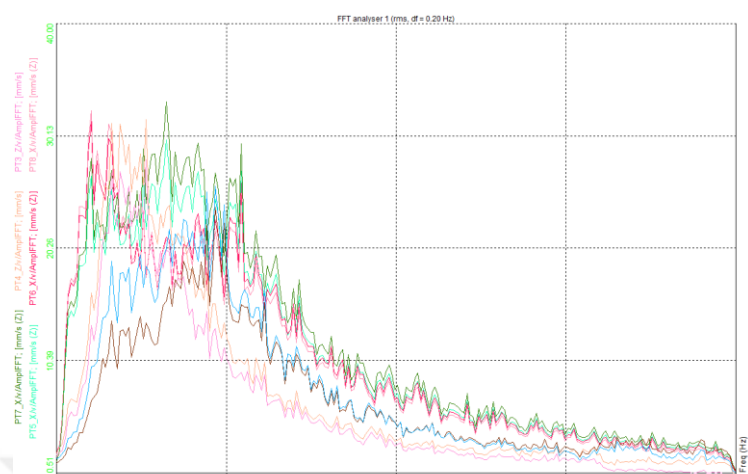


## 90kg Vertical – Softer Elements

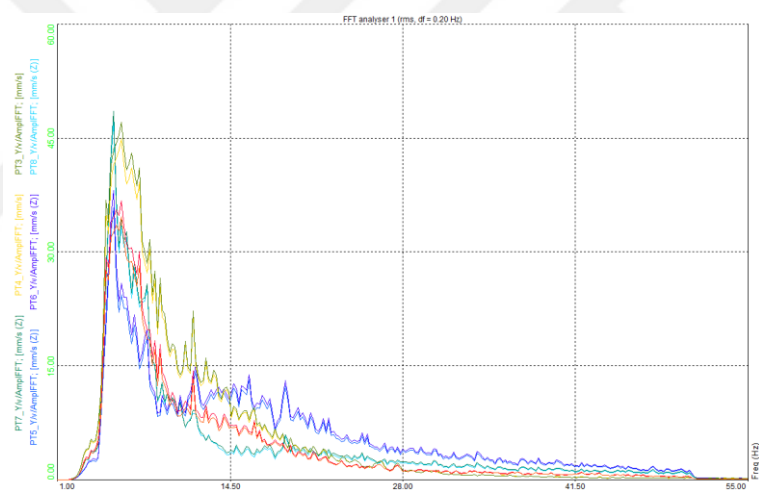


## Electrical Cabinet Random Vibration Measurements for Stiffer Elements

### 90kg Longitudinal – Stiffer Elements



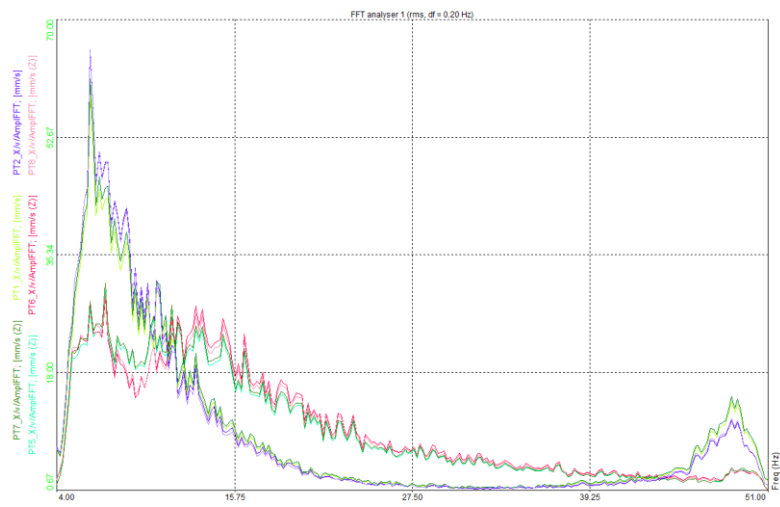
### 90kg Transversal – Stiffer Elements



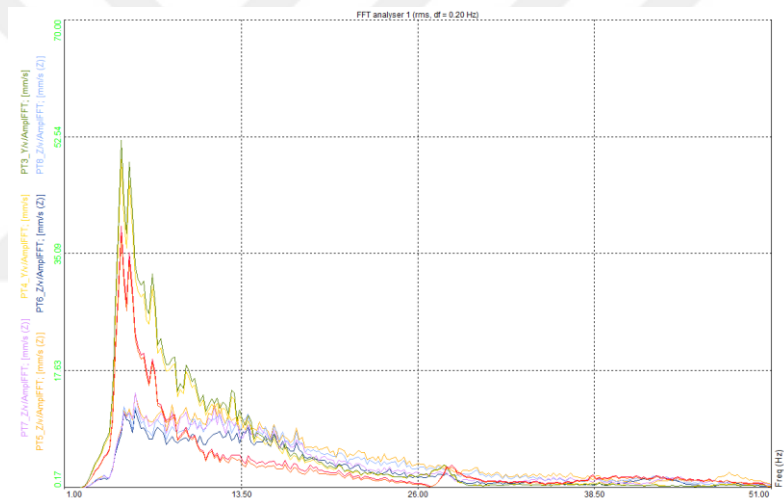
### 90kg Vertical – Stiffer Elements



## 120kg Longitudinal – Stiffer Elements



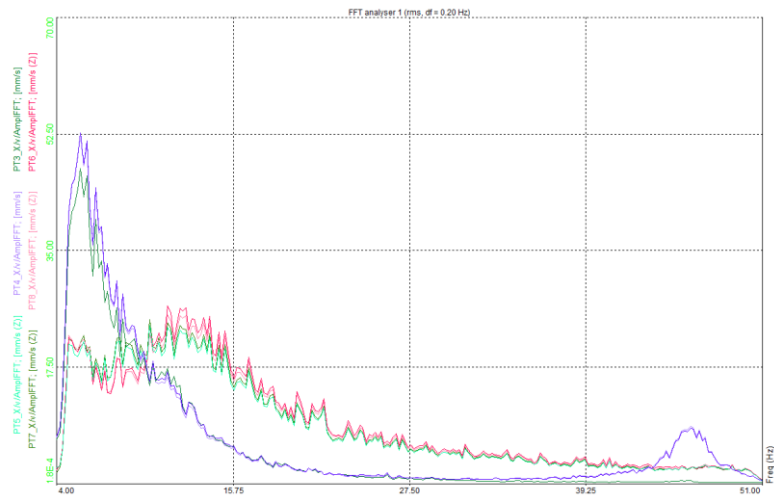
## 120kg Transversal – Stiffer Elements



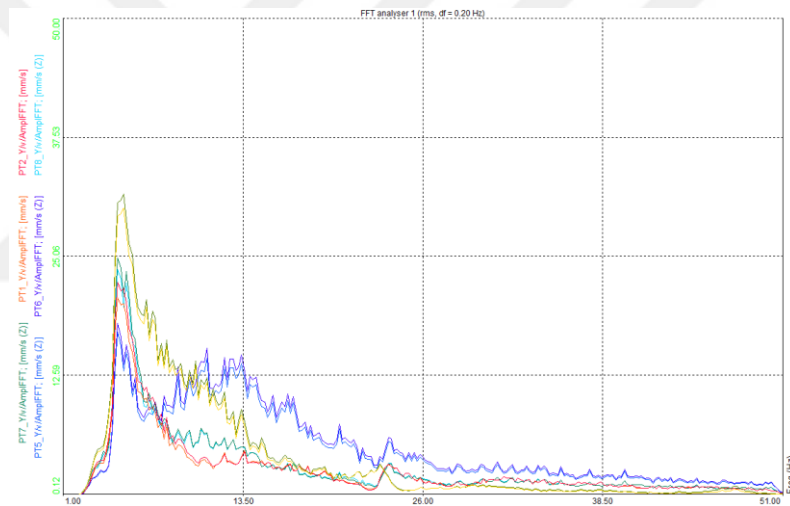
## 120kg Vertical – Stiffer Elements



## 150kg Longitudinal – Stiffer Elements



## 150kg Transversal – Stiffer Elements



## 150kg Vertical – Stiffer Elements

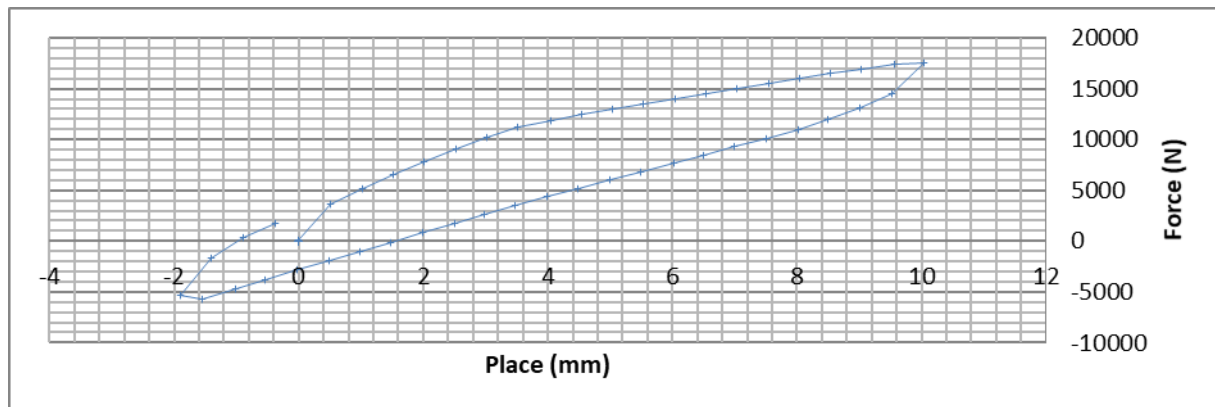


## Appendix D

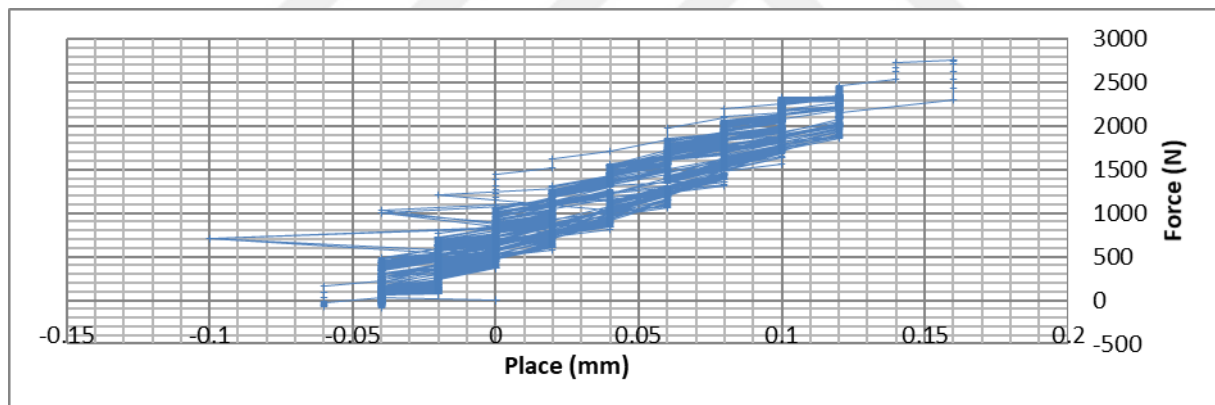
### Stiffness Test Results for Engine Mount Wire Rope Elements

#### Vertical Direction Stiffness Test Results

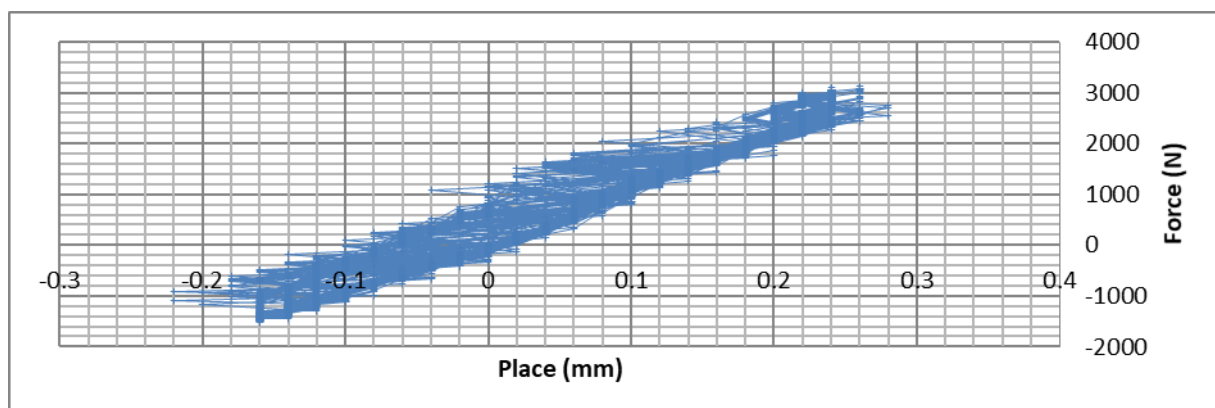
Static Condition – Vertical



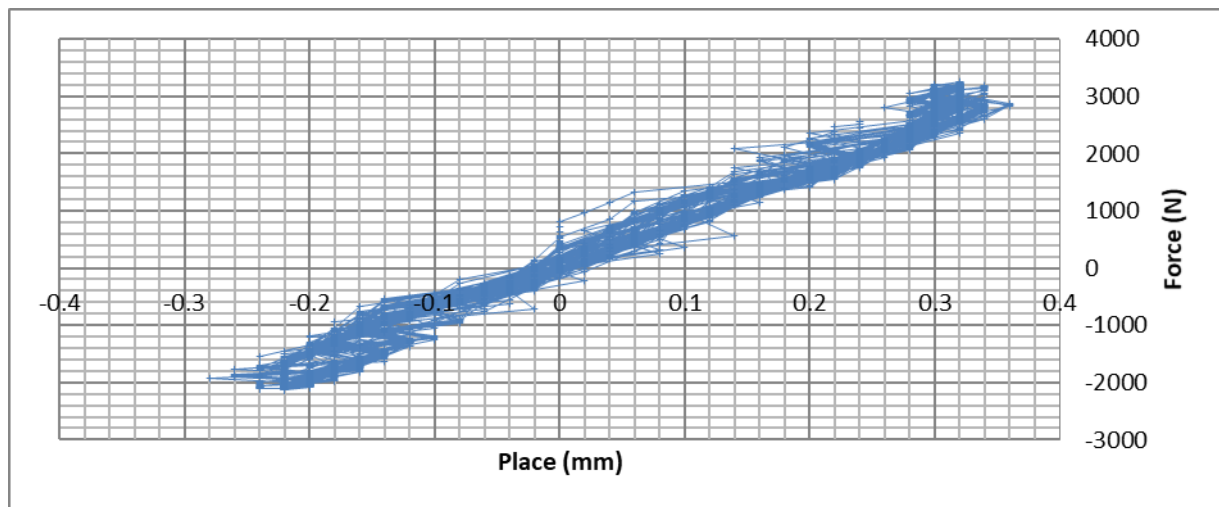
Dynamic Condition – 0.2mm – Vertical



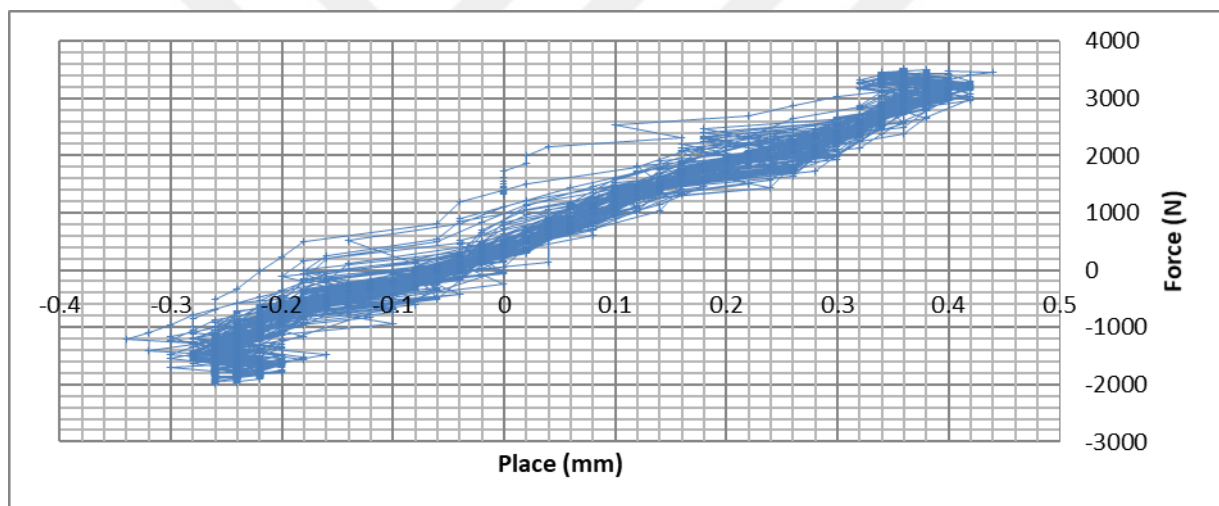
Dynamic Condition – 0.3mm – Vertical



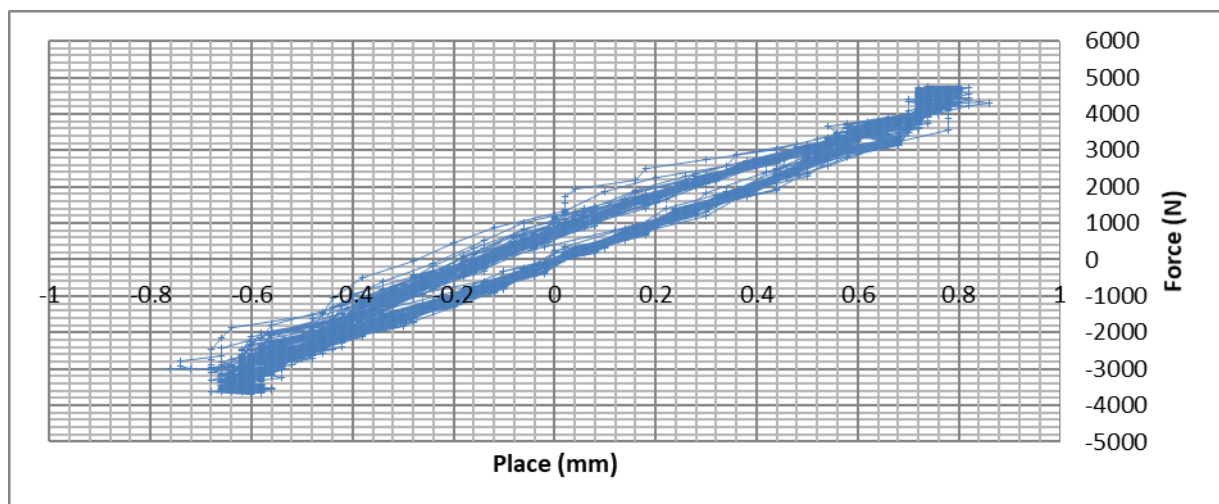
Dynamic Condition – 0.4mm – Vertical



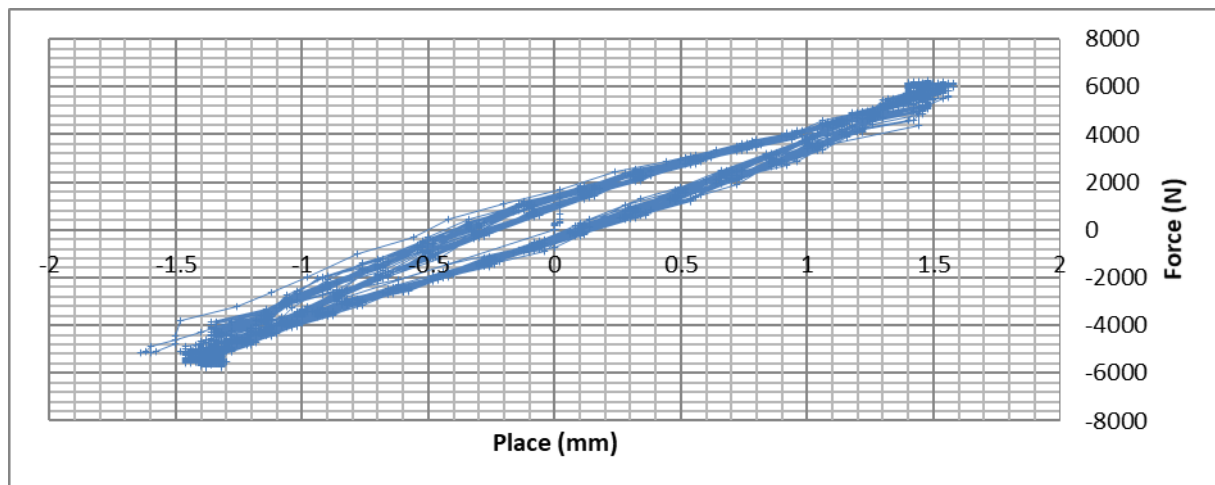
Dynamic Condition – 0.5mm – Vertical



Dynamic Condition – 1mm – Vertical

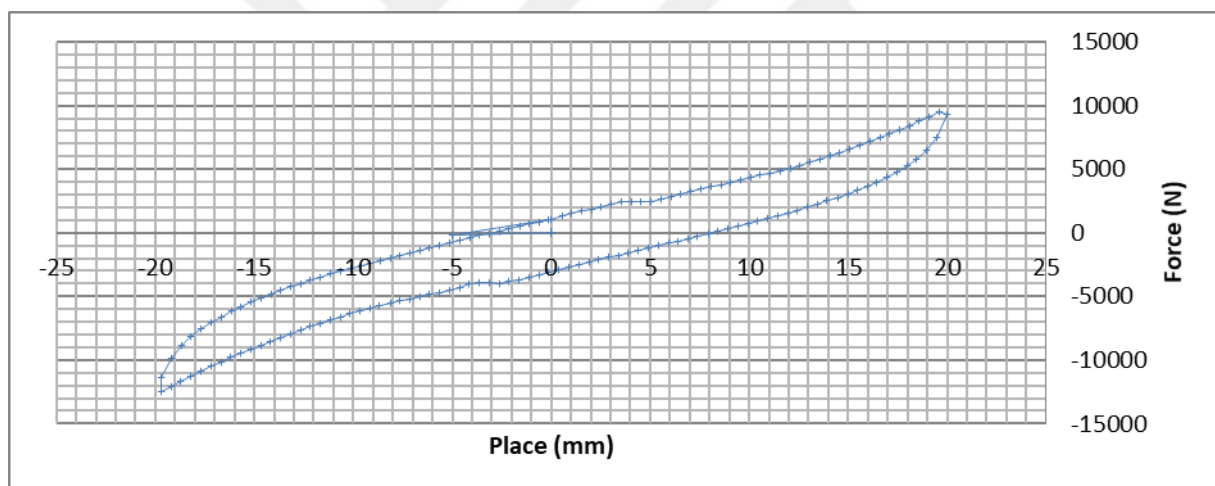


## Dynamic Condition – 2mm – Vertical

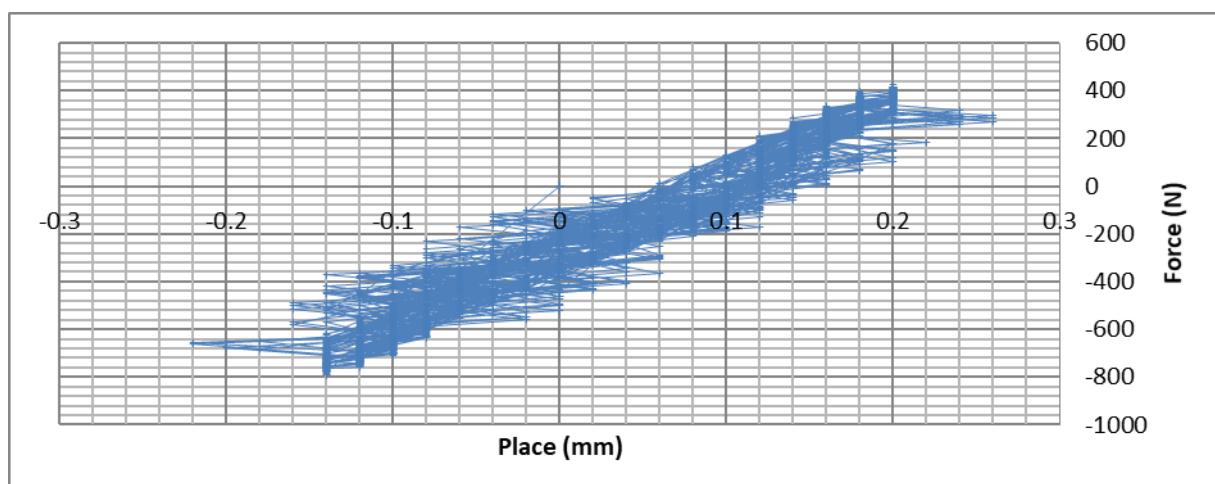


## Longitudinal Direction Stiffness Test Results

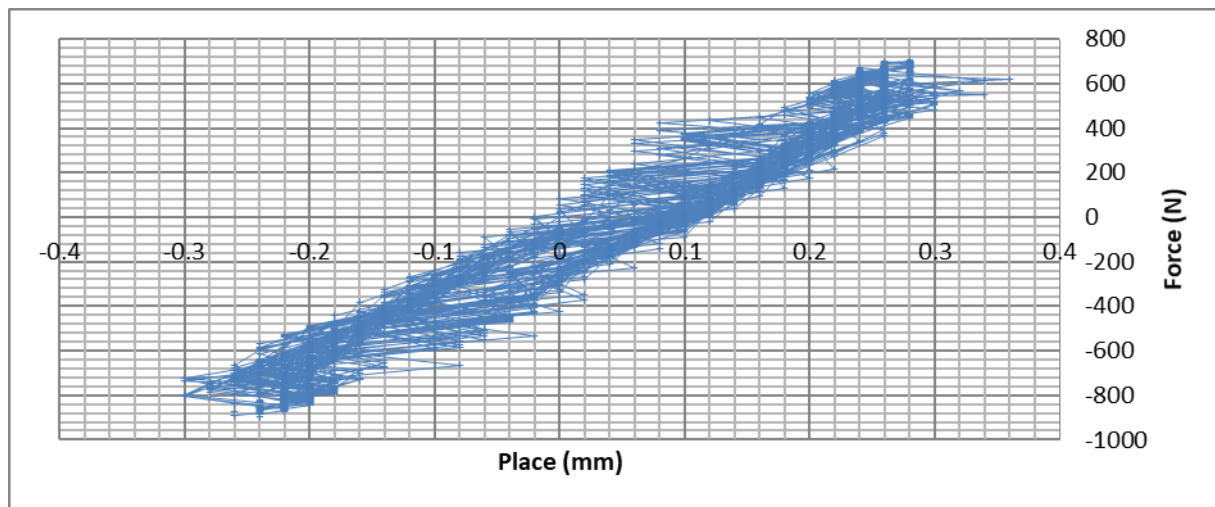
### Static Condition – Longitudinal



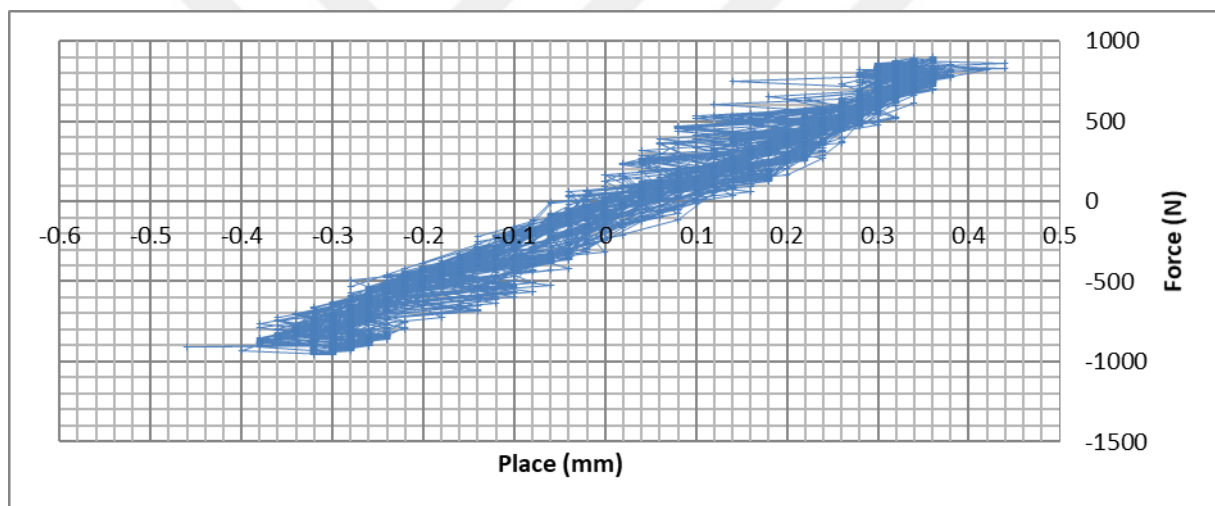
### Dynamic Condition – 0.2mm – Longitudinal



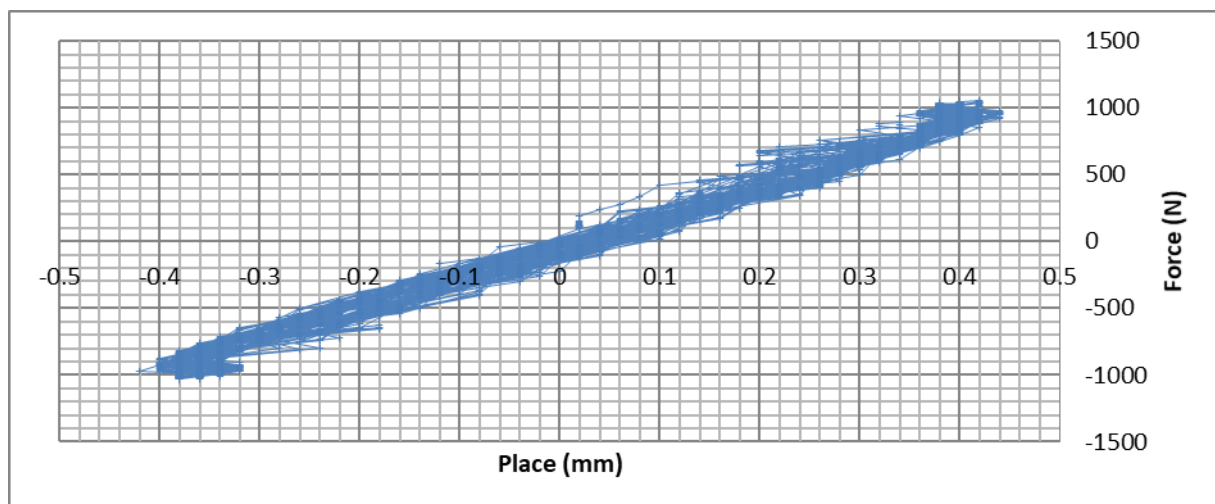
Dynamic Condition – 0.3mm – Longitudinal



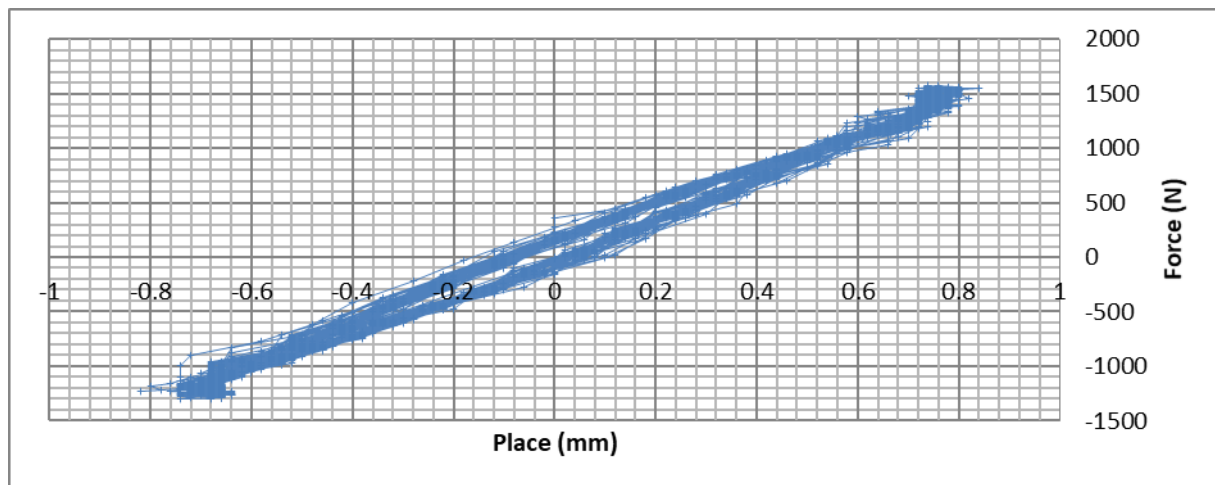
Dynamic Condition – 0.4mm – Longitudinal



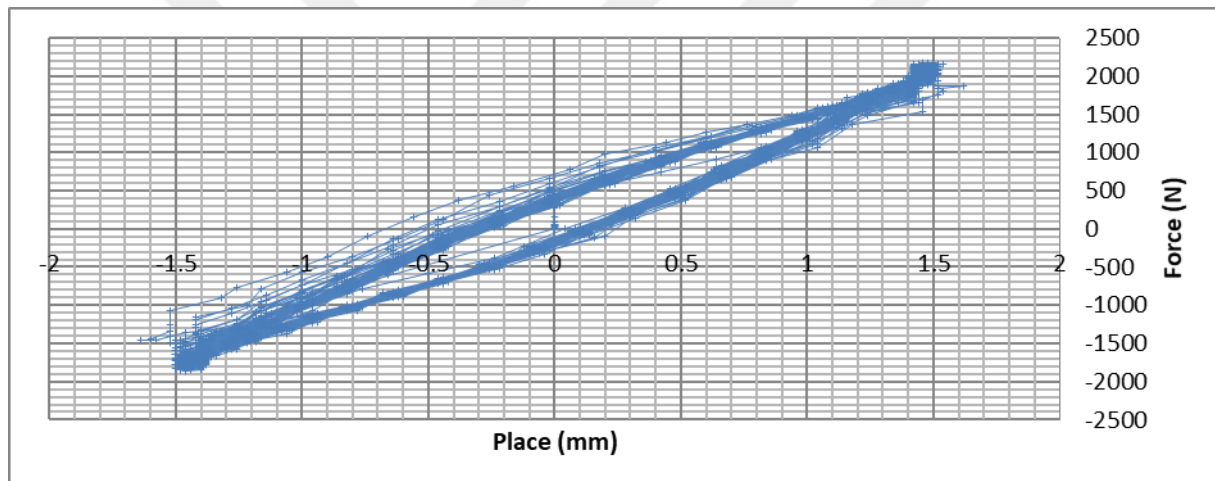
Dynamic Condition – 0.5mm – Longitudinal



### Dynamic Condition – 1mm – Longitudinal

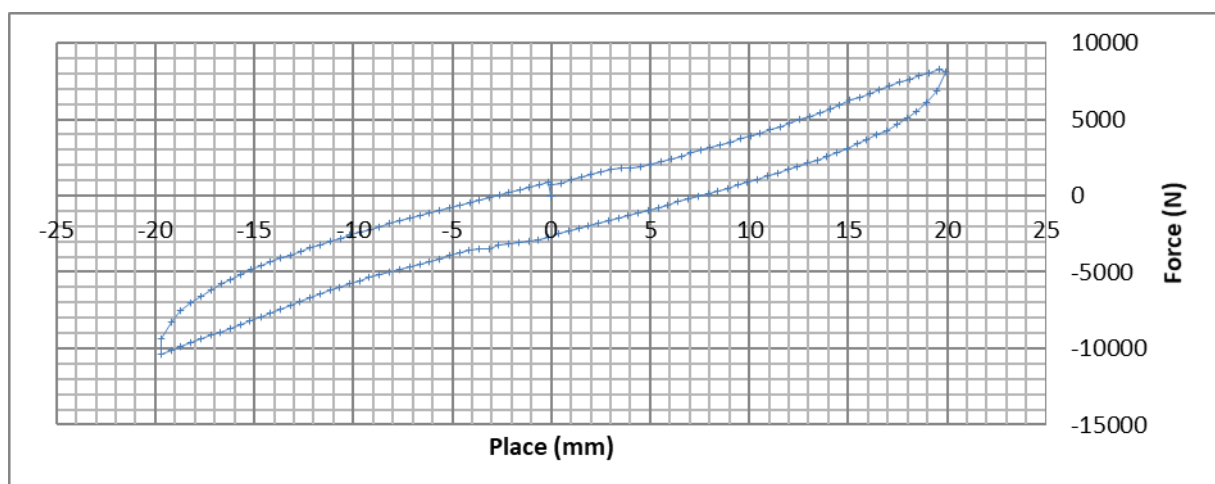


### Dynamic Condition – 2mm – Longitudinal

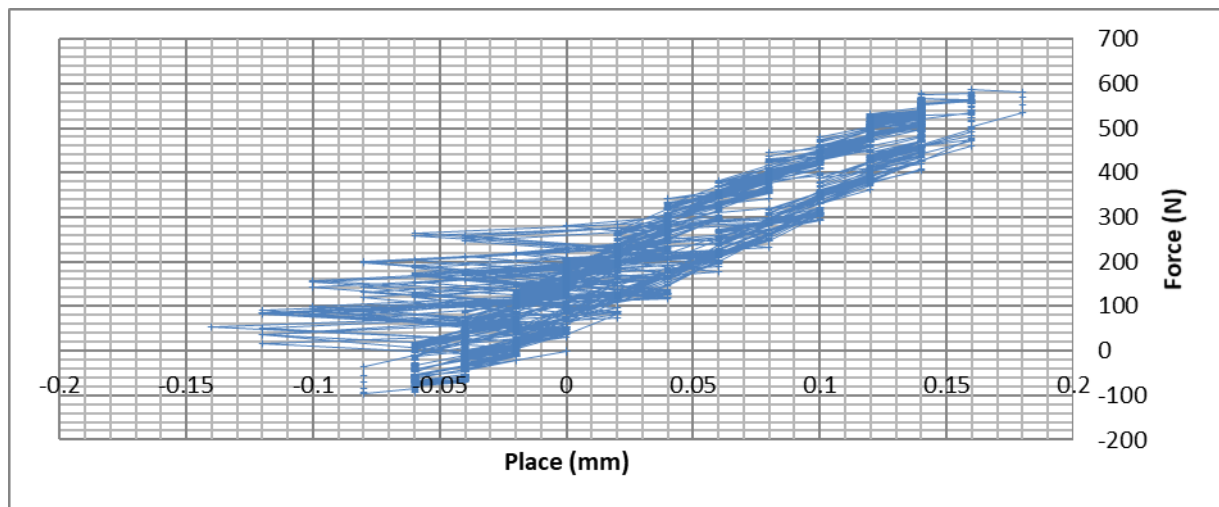


## Transversal Direction Stiffness Test Results

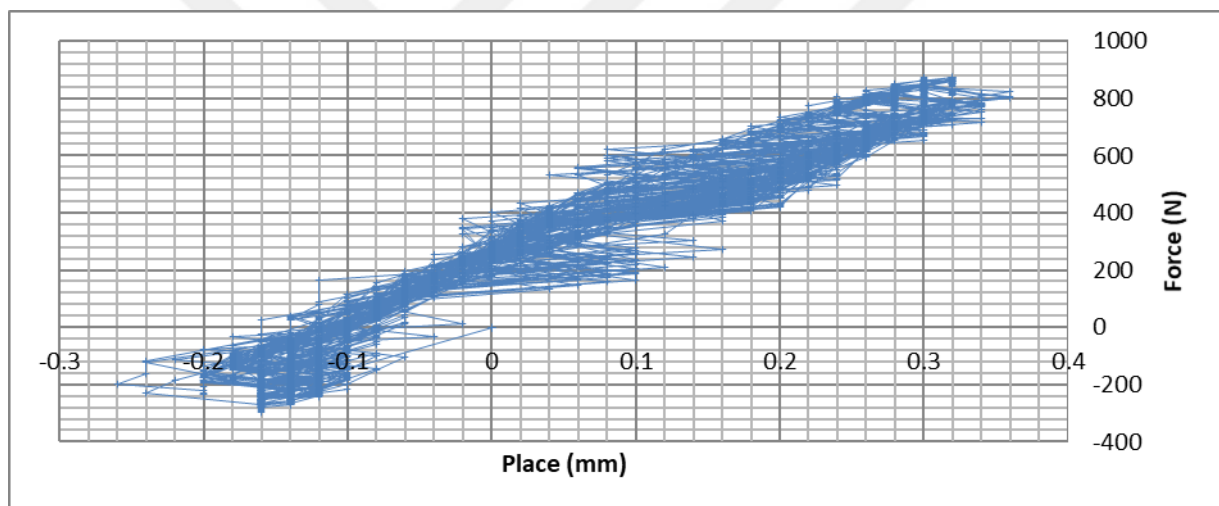
### Static Condition – Transversal



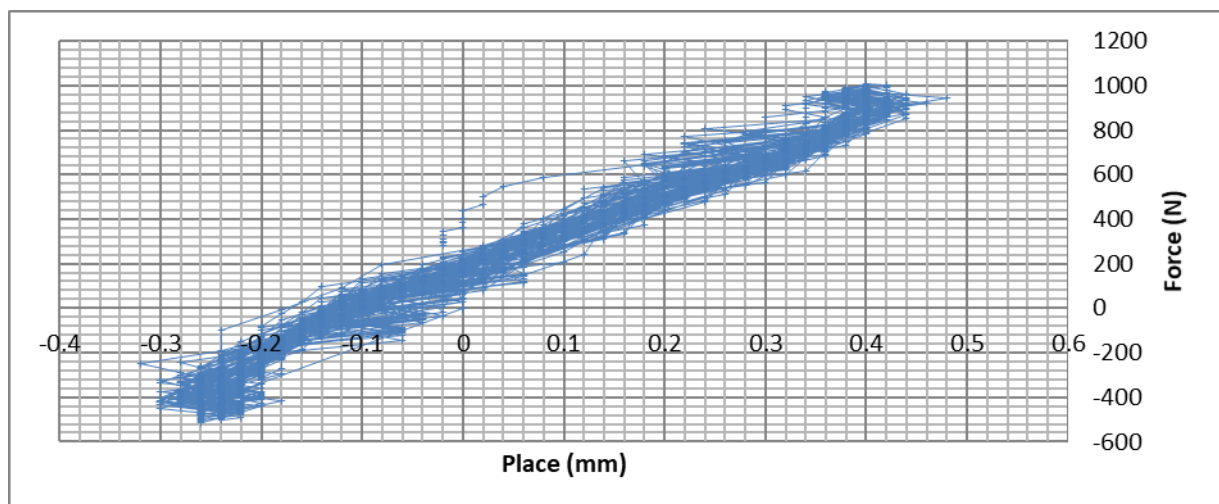
Dynamic Condition – 0.2mm – Transversal



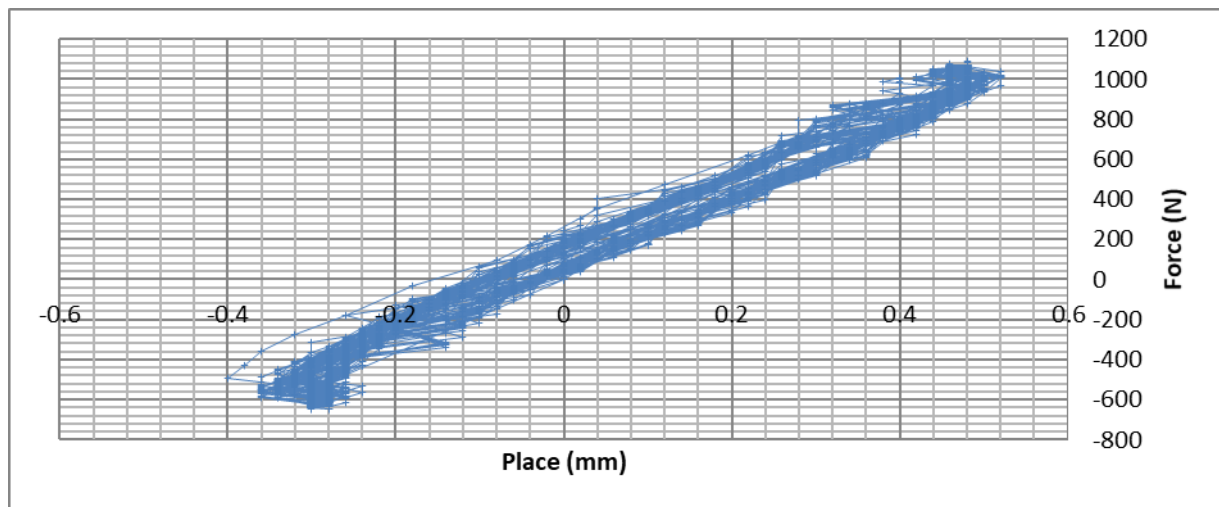
Dynamic Condition – 0.3mm – Transversal



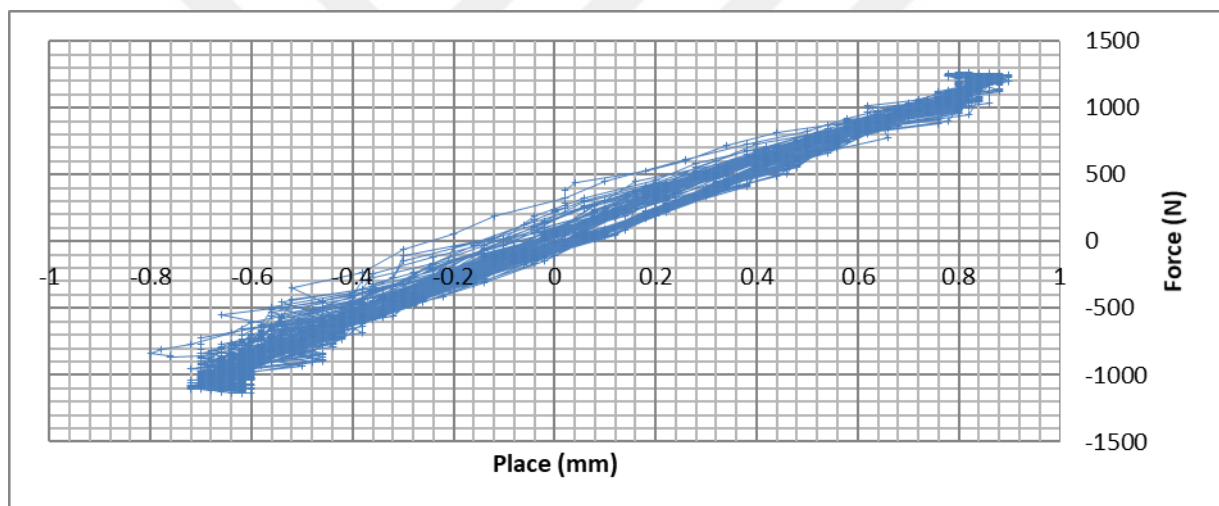
Dynamic Condition – 0.4mm – Transversal



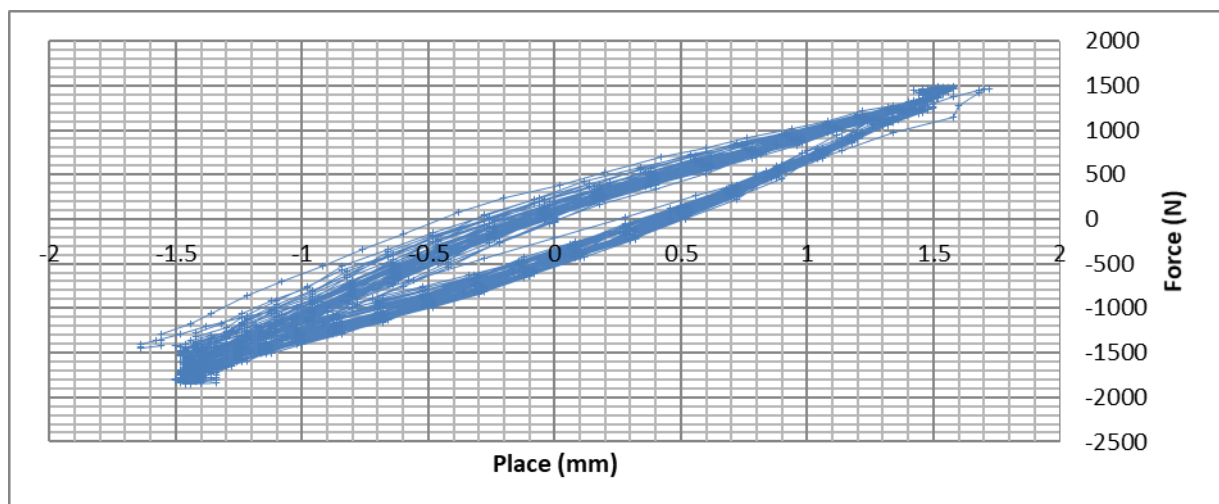
Dynamic Condition – 0.5mm – Transversal



Dynamic Condition – 1mm – Transversal



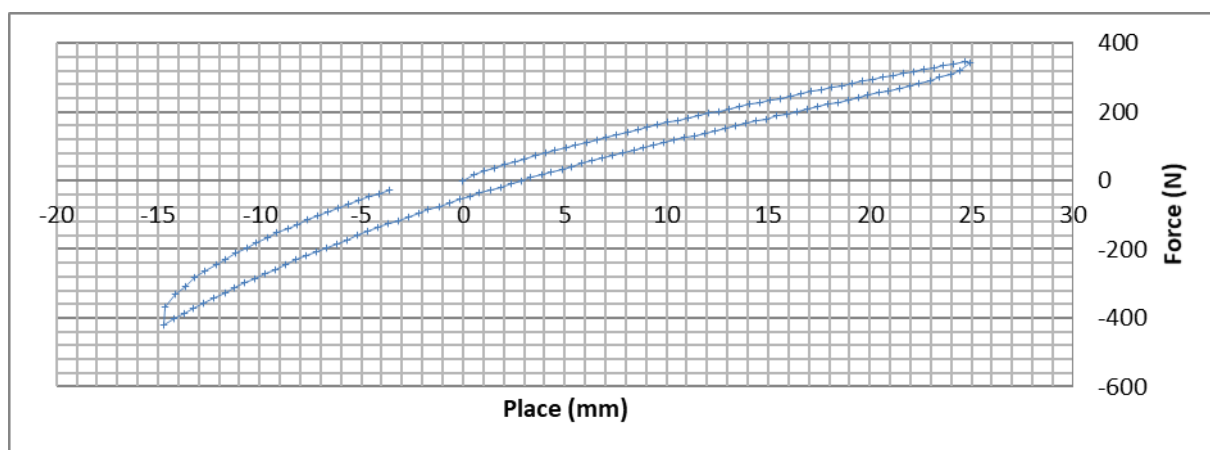
Dynamic Condition – 2mm – Transversal



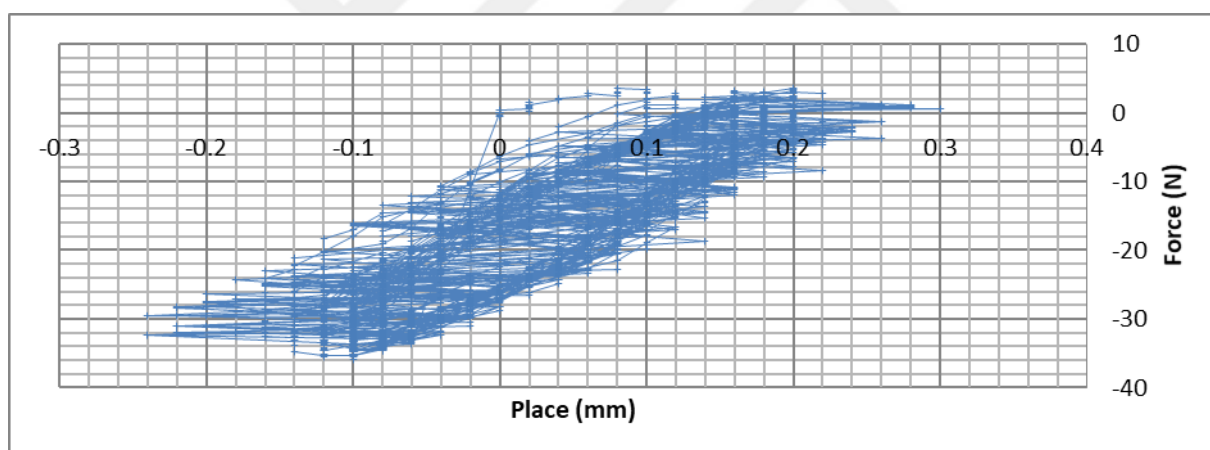
## Stiffness Test Results for Electric Cabinet Wire Rope Elements

### Softer Element – Vertical Direction Stiffness Test Results

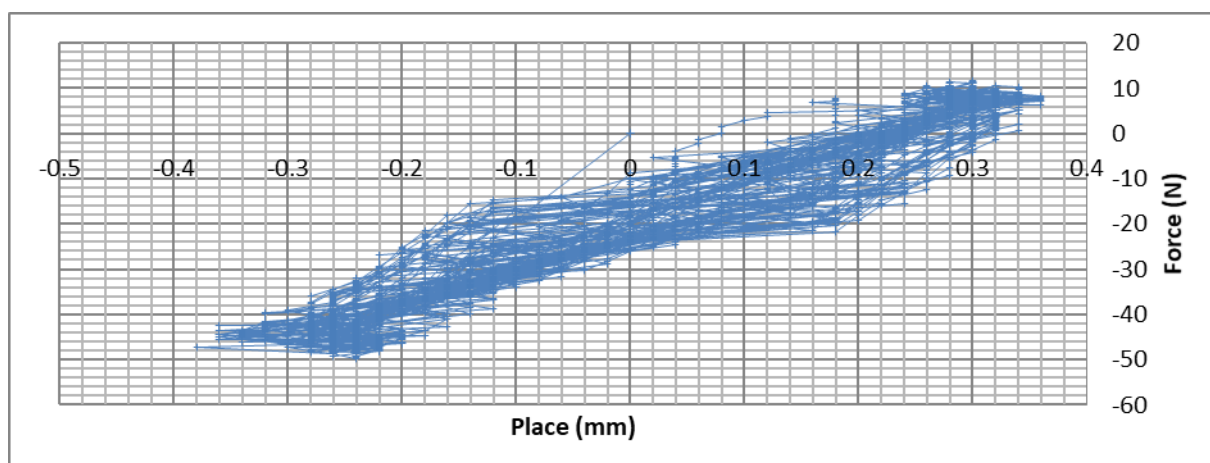
Softer Element – Static Condition – Vertical



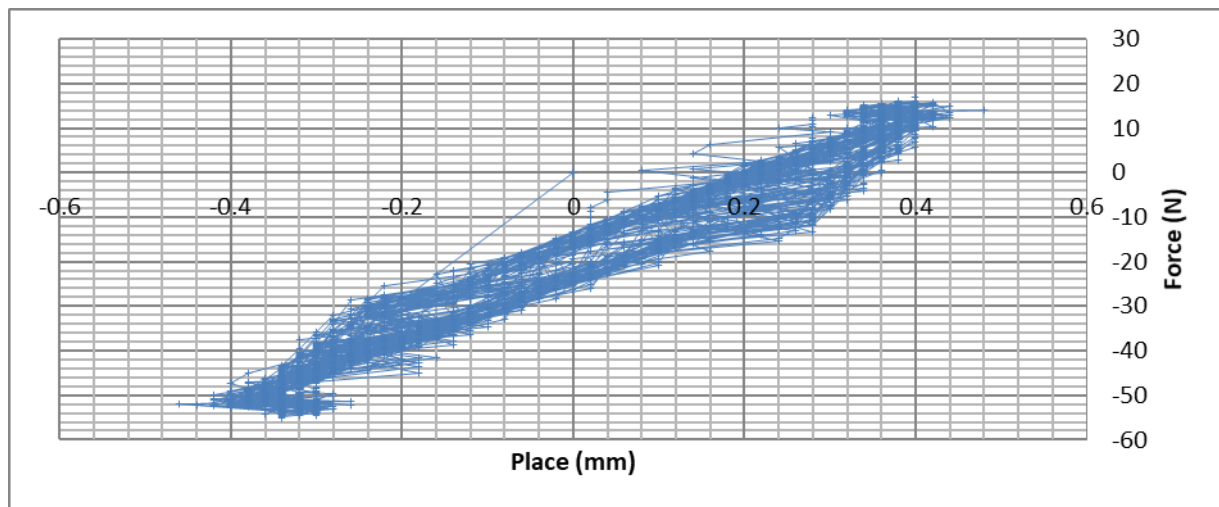
Softer Element – Dynamic Condition – 0.2mm – Vertical



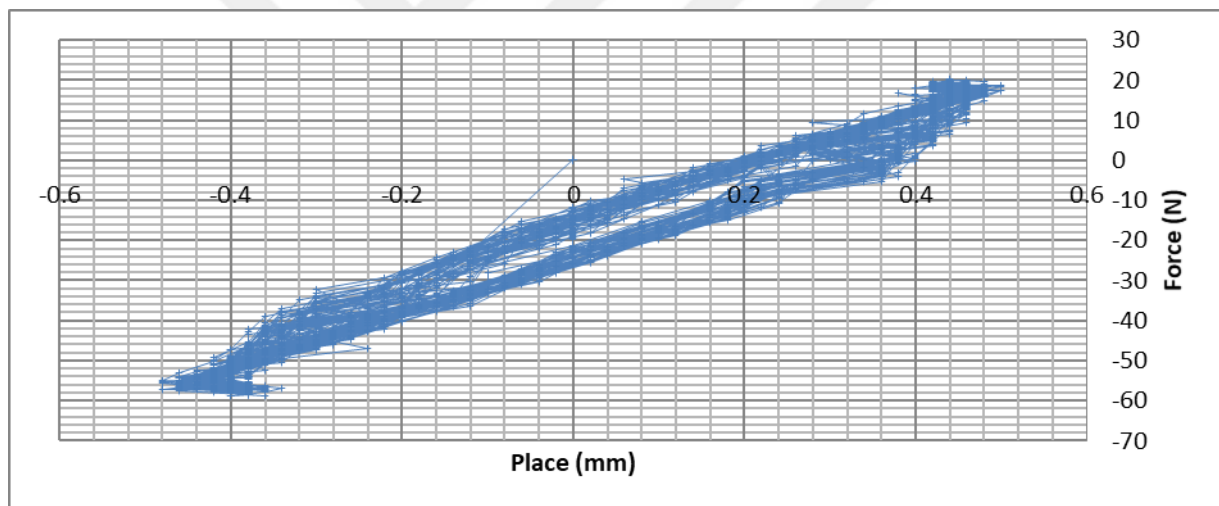
Softer Element – Dynamic Condition – 0.3mm – Vertical



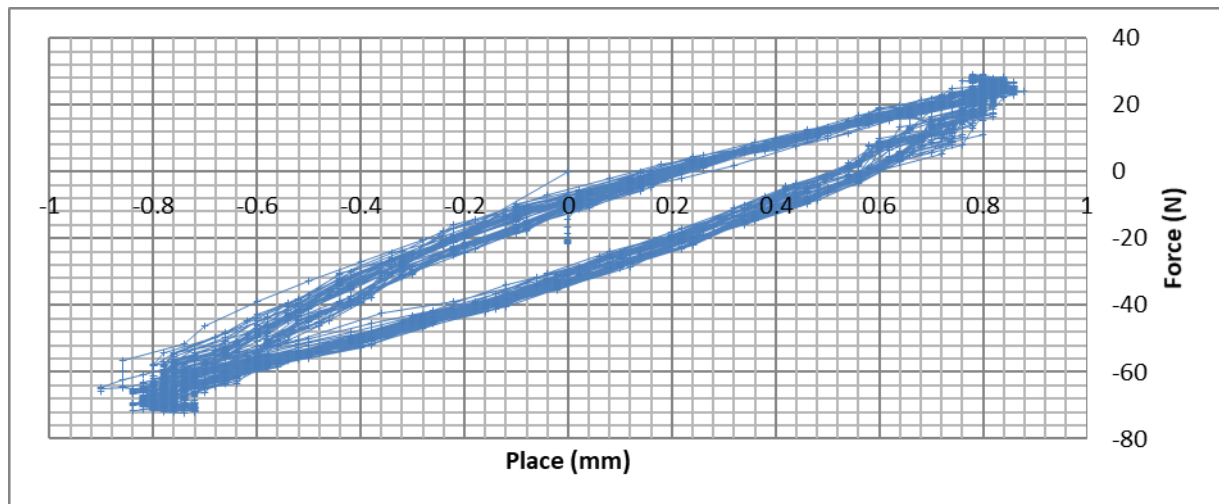
Softer Element – Dynamic Condition – 0.4mm – Vertical



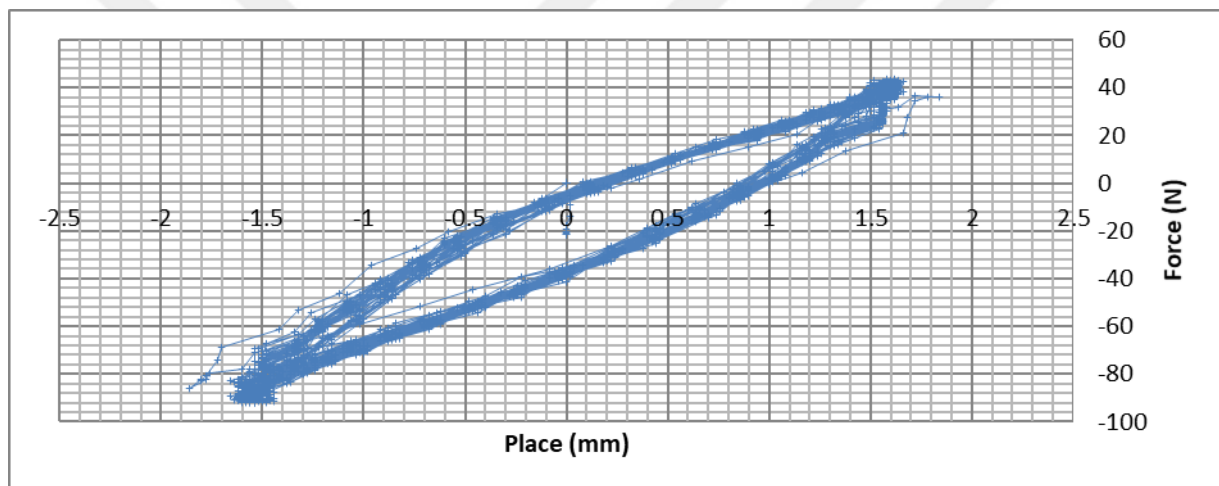
Softer Element – Dynamic Condition – 0.5mm – Vertical



Softer Element – Dynamic Condition – 1mm – Vertical

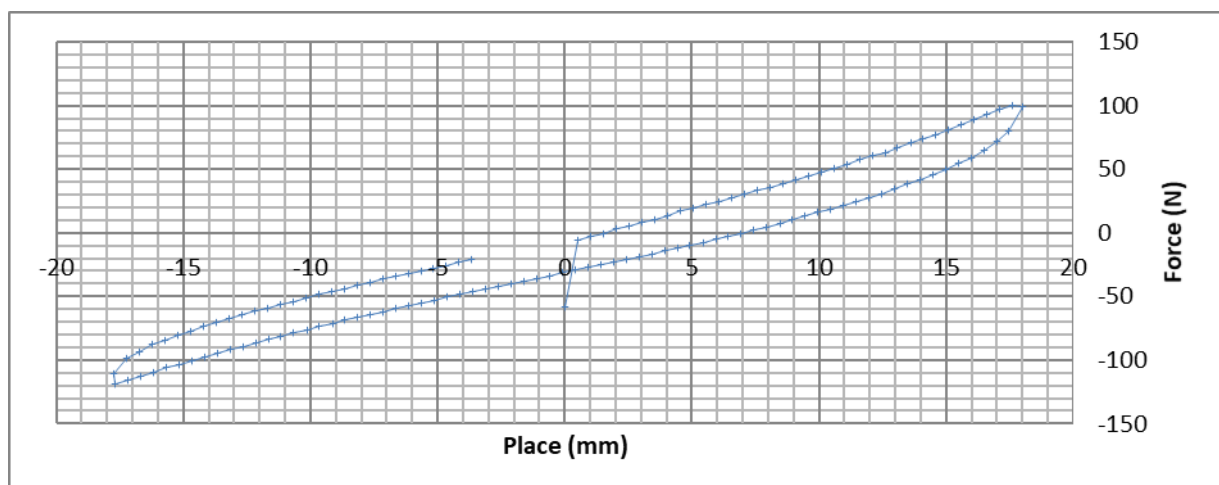


Softer Element – Dynamic Condition – 2mm – Vertical

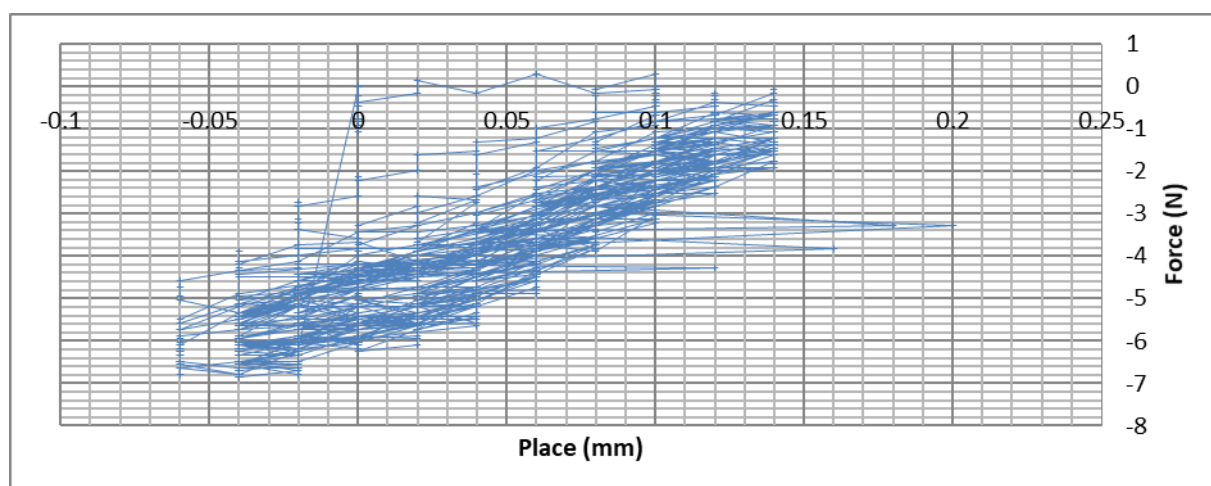


### Softer Element – Longitudinal Direction Stiffness Test Results

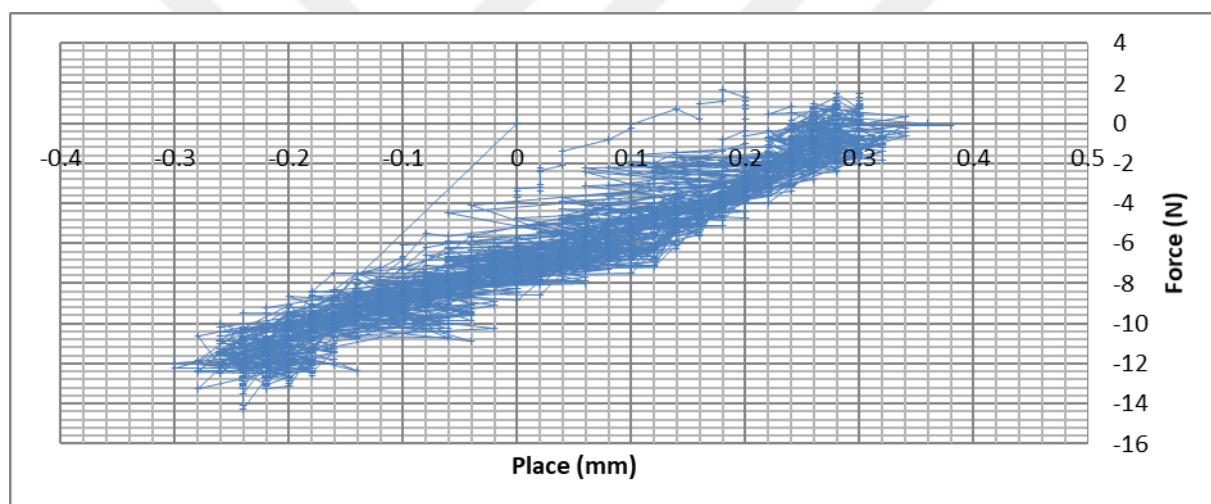
Softer Element – Static Condition – Longitudinal



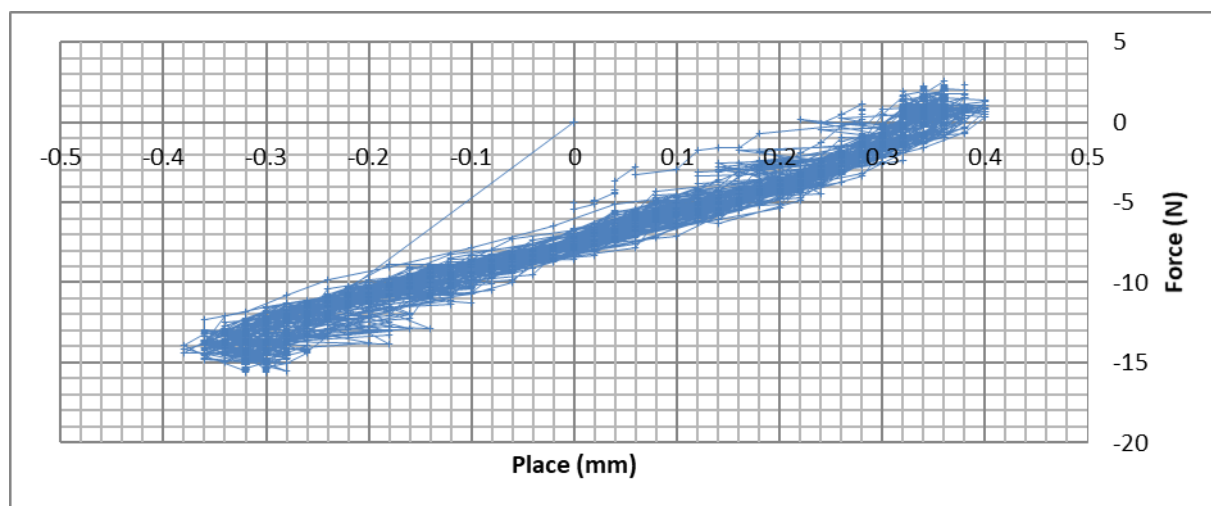
Softer Element – Dynamic Condition – 0.2mm – Longitudinal



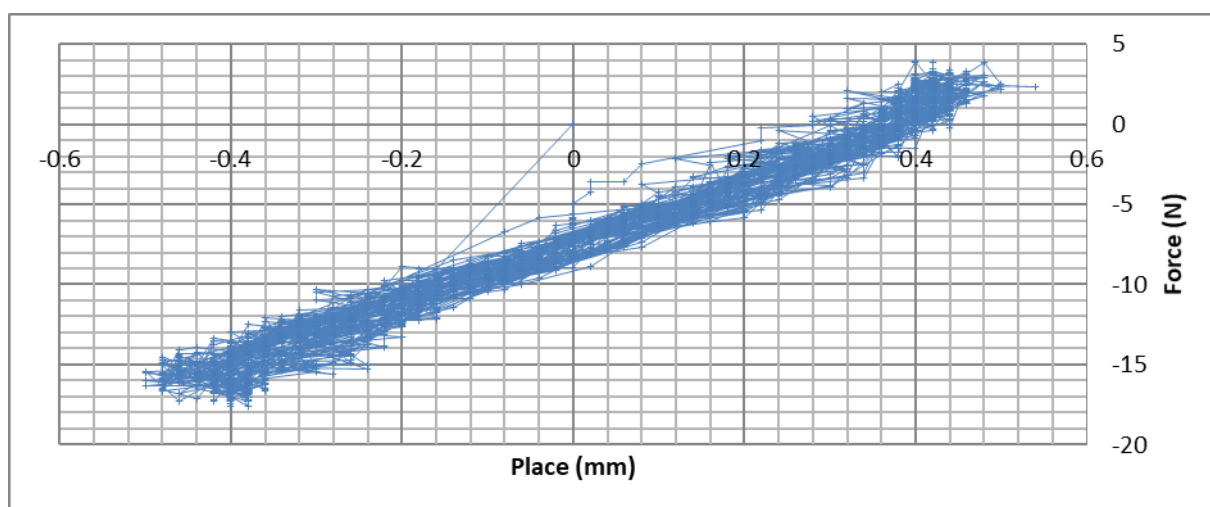
Softer Element – Dynamic Condition – 0.3mm – Longitudinal



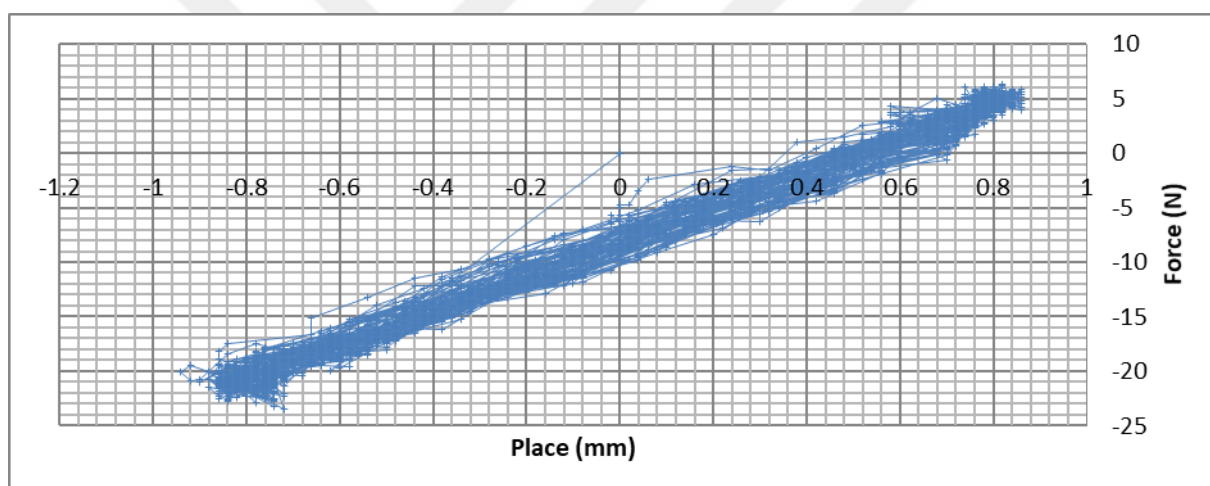
Softer Element – Dynamic Condition – 0.4mm – Longitudinal



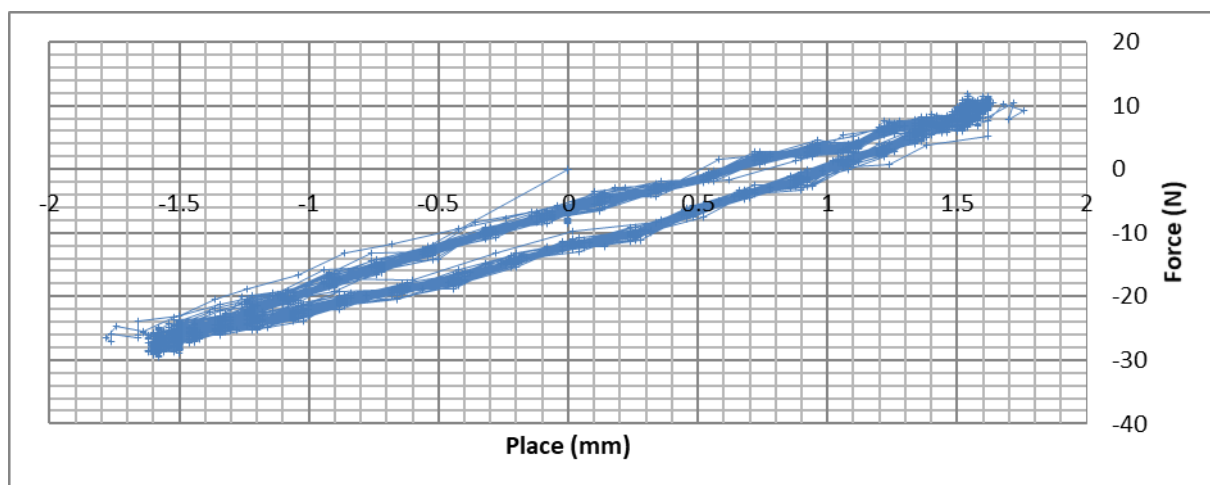
Softer Element – Dynamic Condition – 0.5mm – Longitudinal



Softer Element – Dynamic Condition – 1mm – Longitudinal

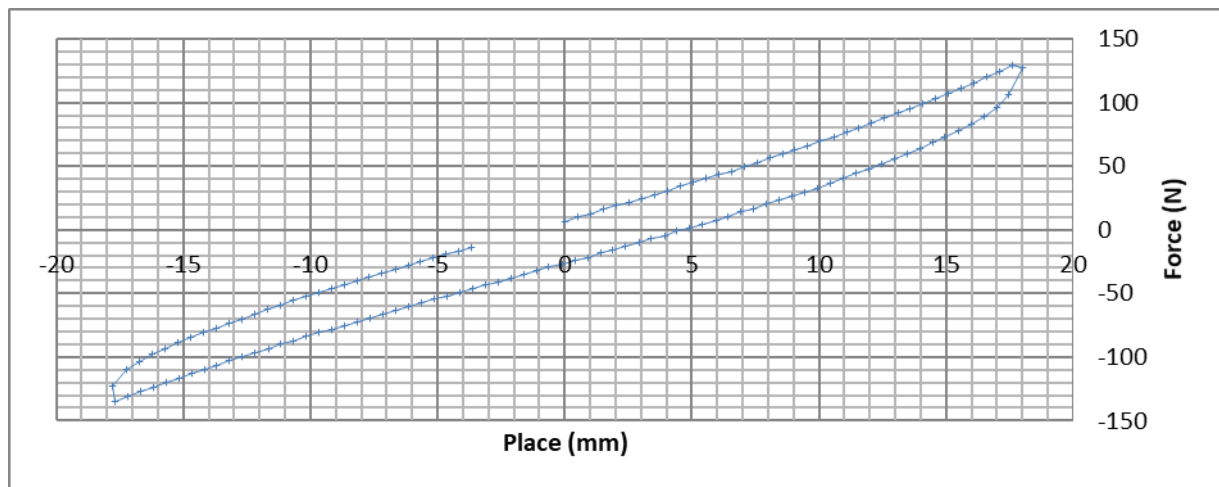


Softer Element – Dynamic Condition – 2mm – Longitudinal

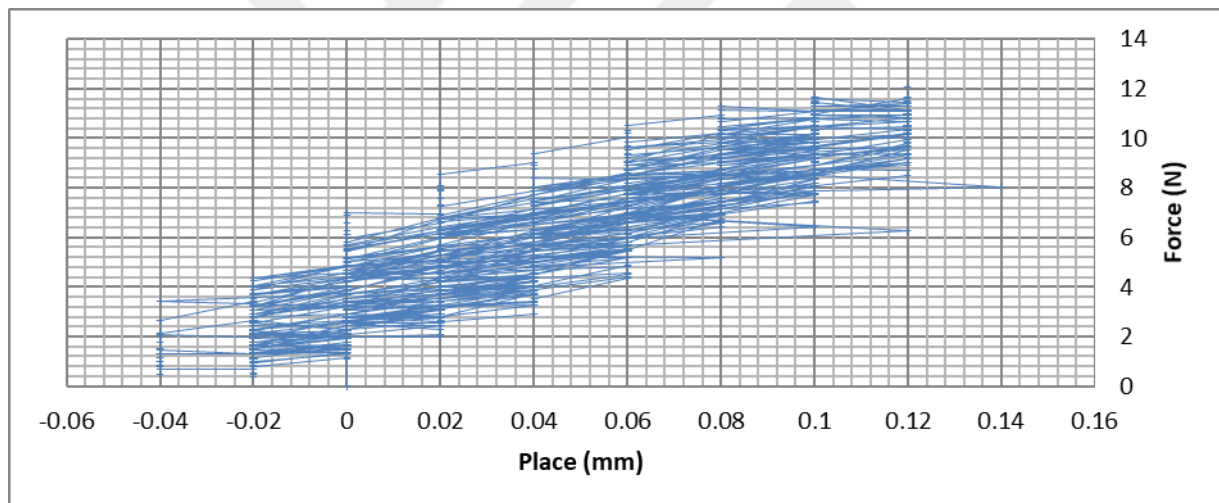


## Softer Element – Transversal Direction Stiffness Test Results

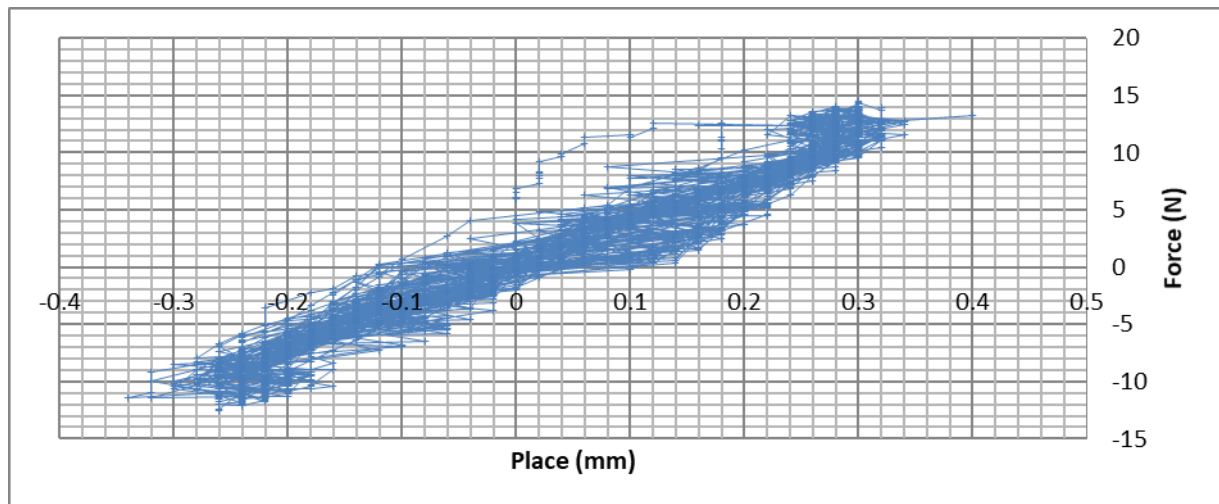
Softer Element – Static Condition – Transversal



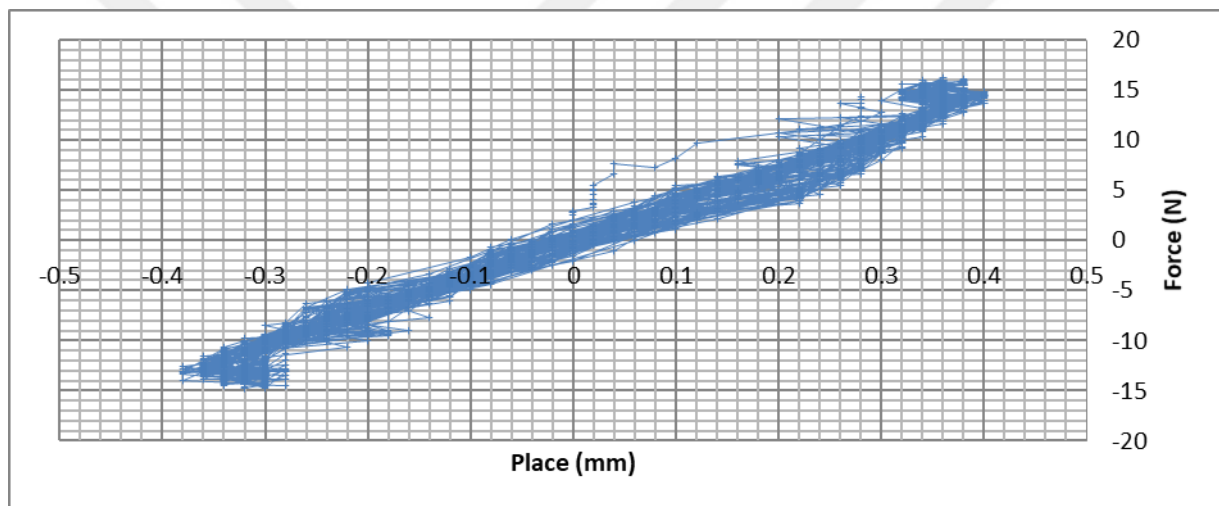
Softer Element – Dynamic Condition – 0.2mm – Transversal



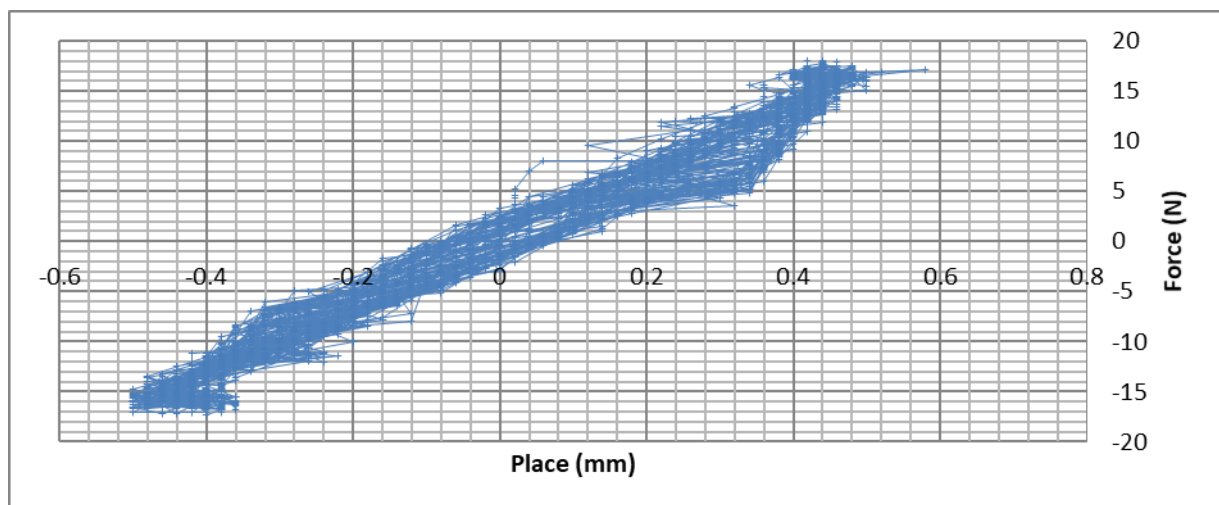
Softer Element – Dynamic Condition – 0.3mm – Transversal



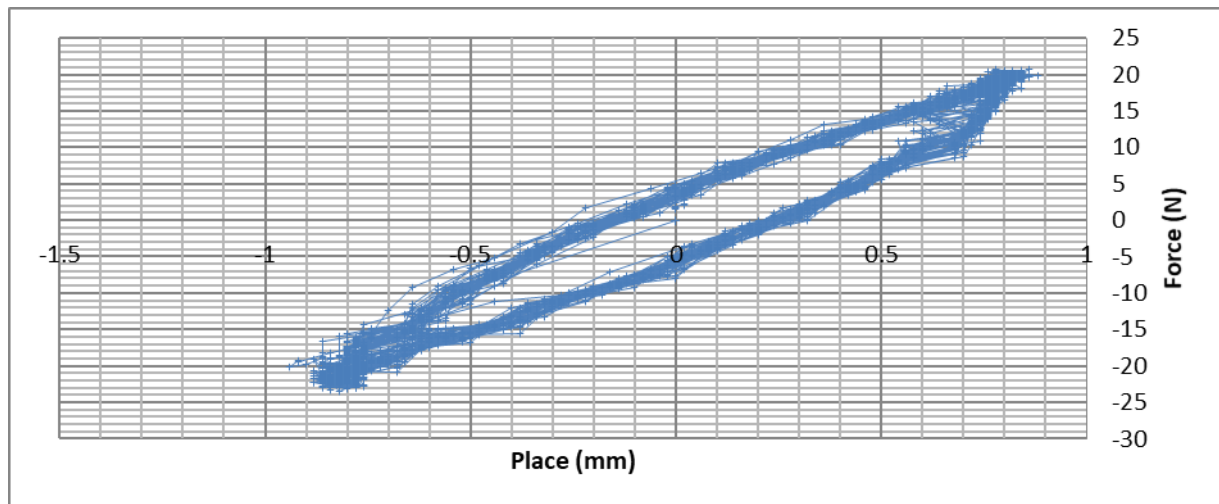
Softer Element – Dynamic Condition – 0.4mm – Transversal



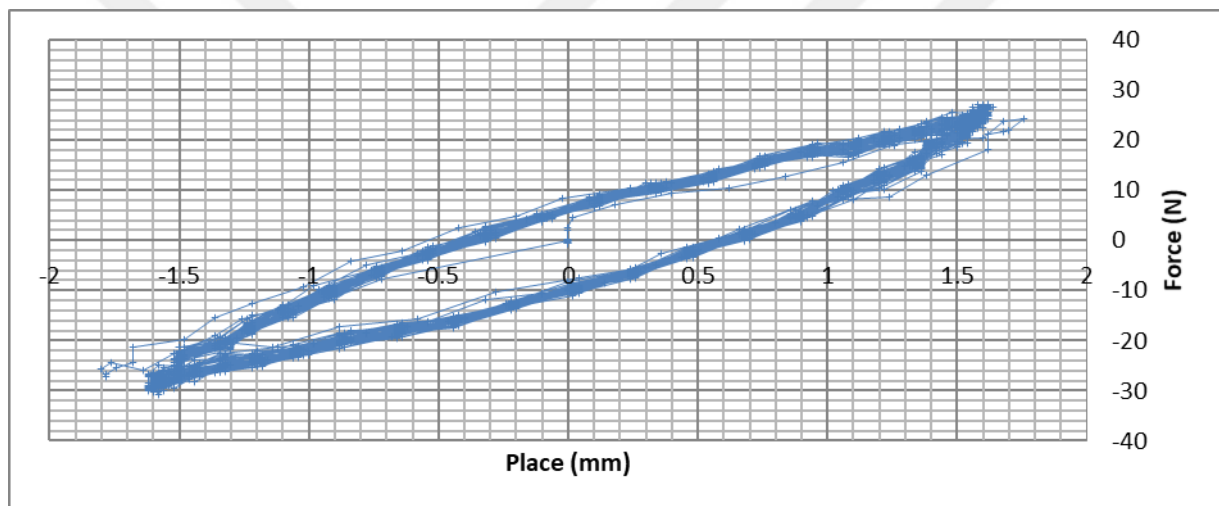
Softer Element – Dynamic Condition – 0.5mm – Transversal



Softer Element – Dynamic Condition – 1mm – Transversal

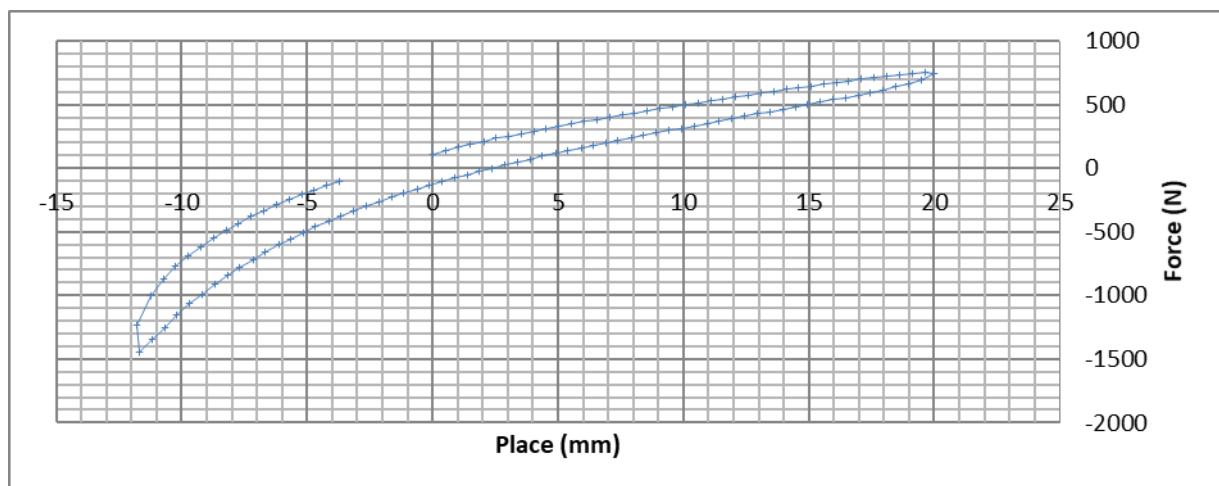


Softer Element – Dynamic Condition – 2mm – Transversal

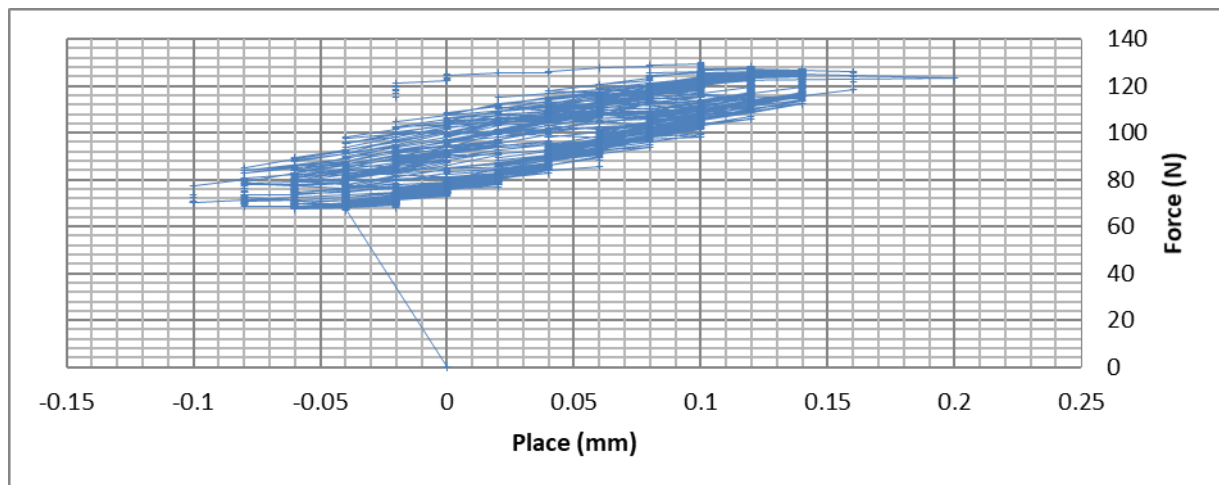


Stiffer Element – Vertical Direction Stiffness Test Results

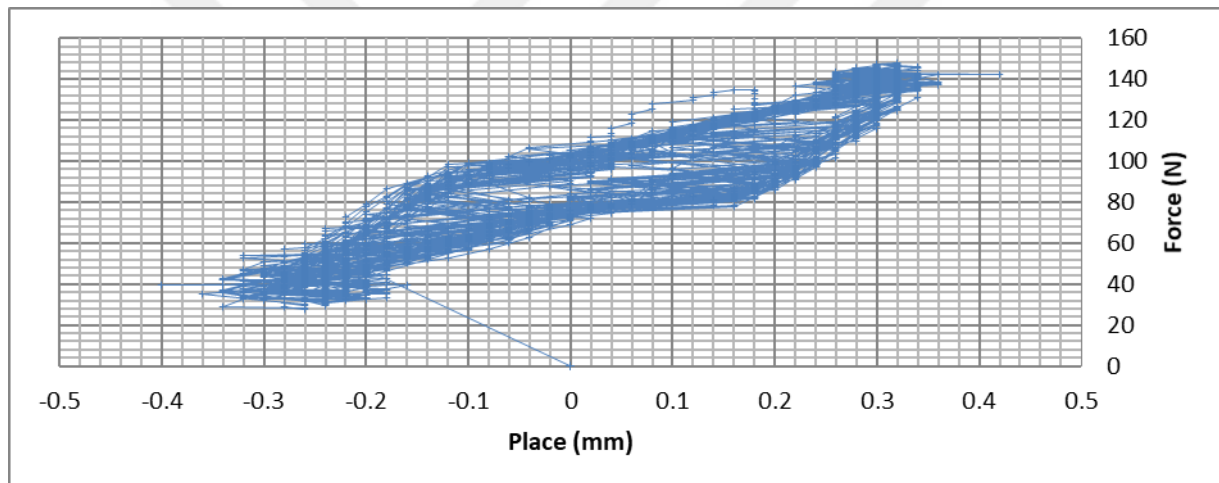
Stiffer Element – Static Condition – Vertical



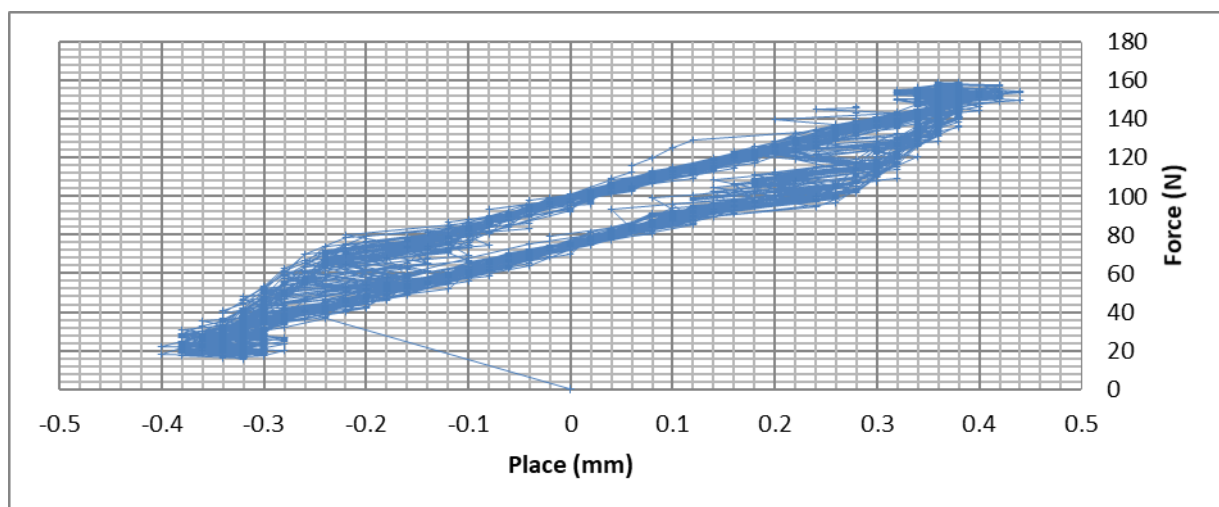
Stiffer Element – Dynamic Condition – 0.2mm – Vertical



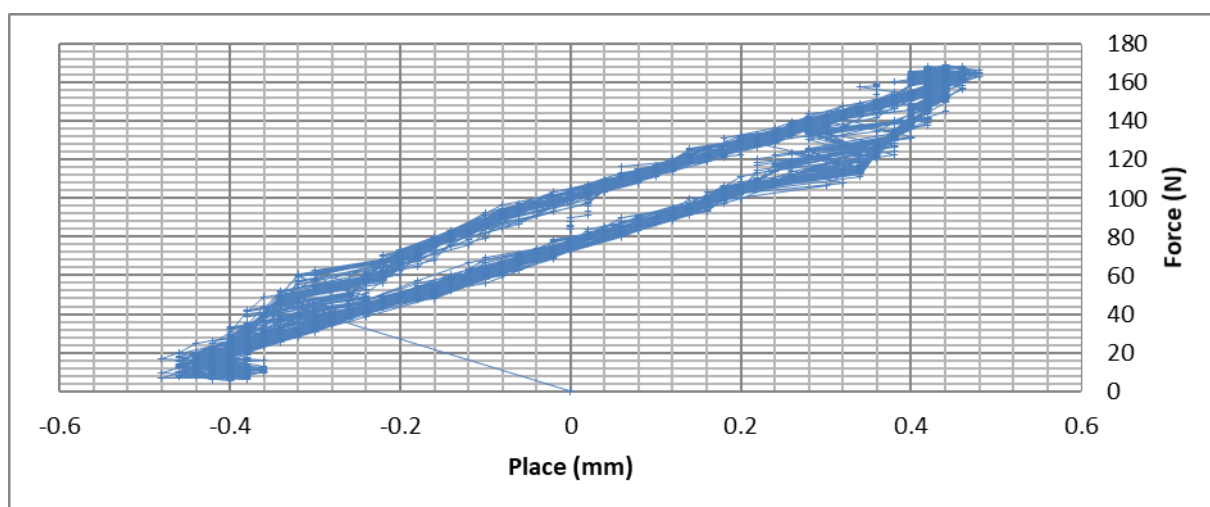
Stiffer Element – Dynamic Condition – 0.3mm – Vertical



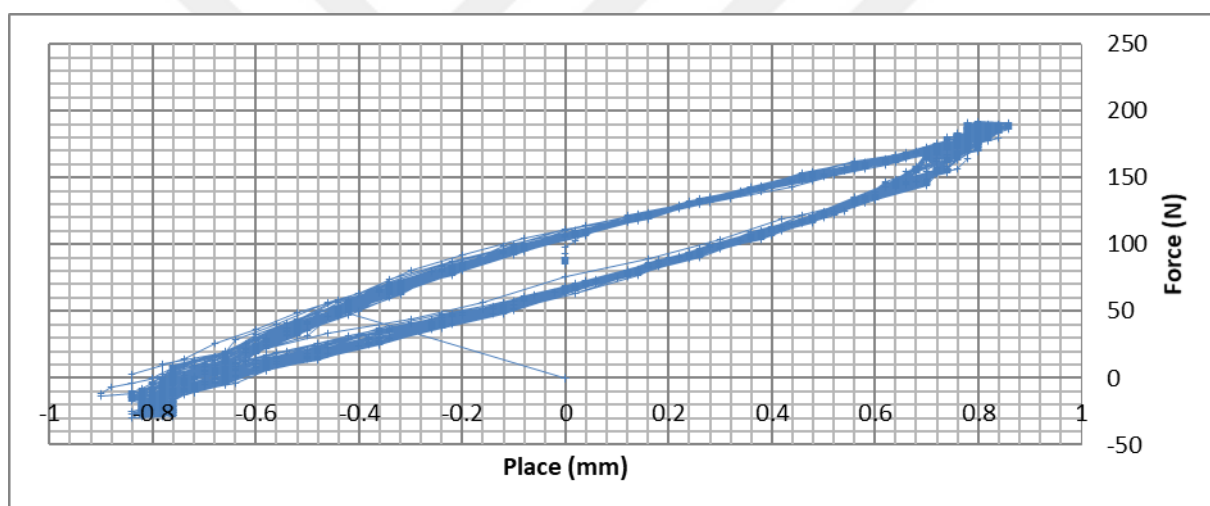
Stiffer Element – Dynamic Condition – 0.4mm – Vertical



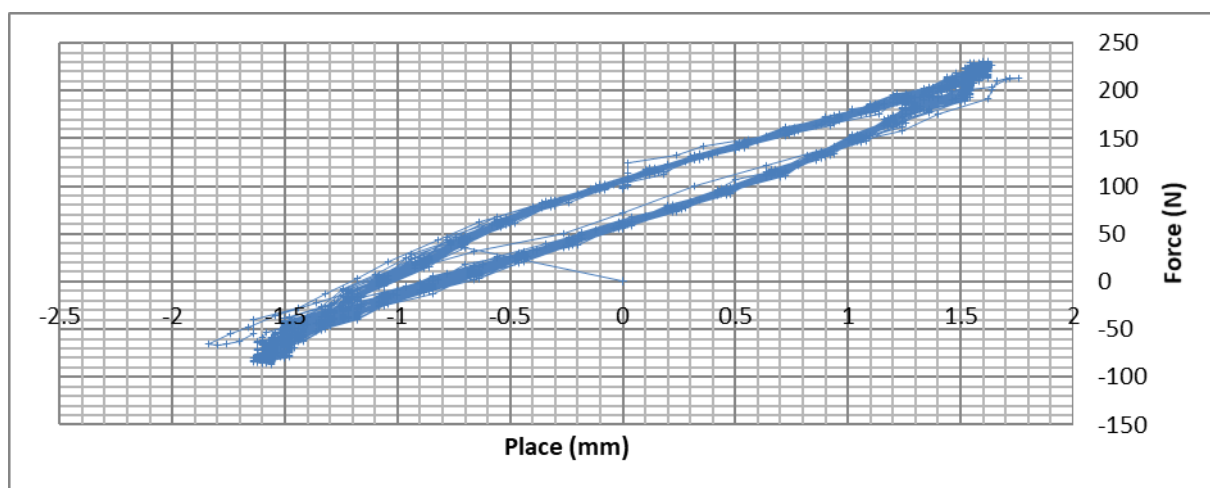
Stiffer Element – Dynamic Condition – 0.5mm – Vertical



Stiffer Element – Dynamic Condition – 1mm – Vertical

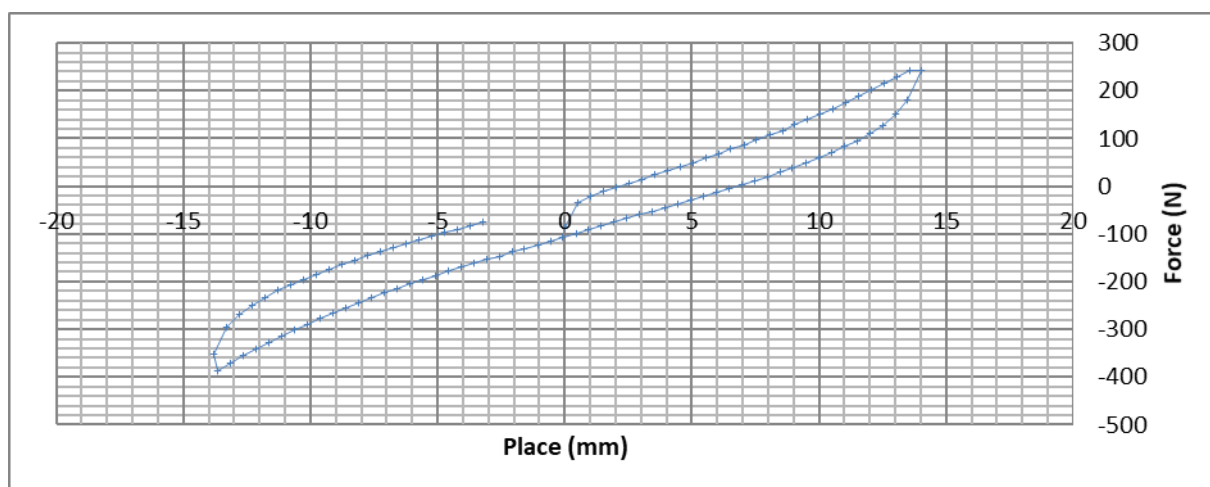


Stiffer Element – Dynamic Condition – 2mm – Vertical

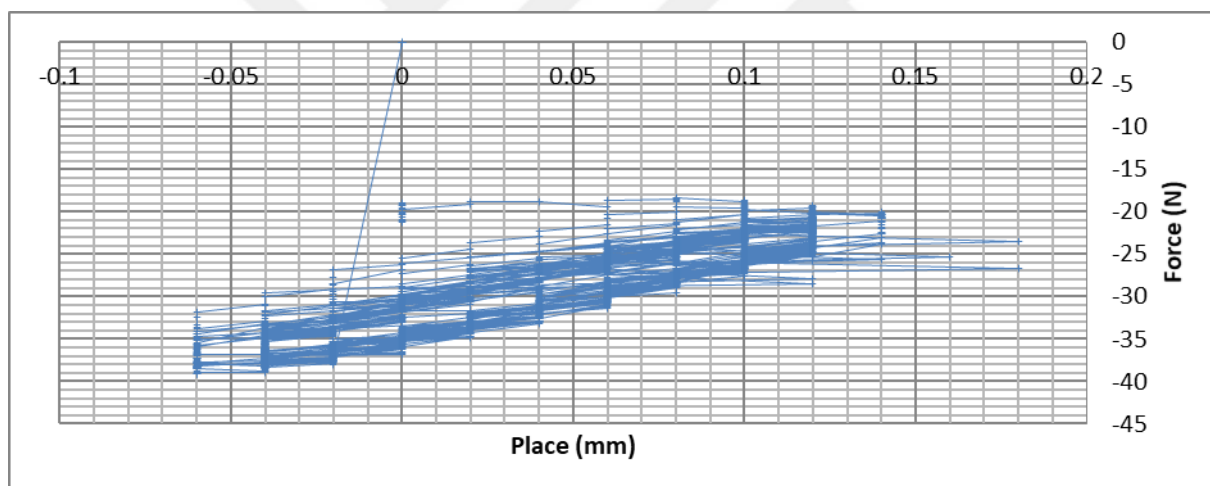


## Stiffer Element – Longitudinal Direction Stiffness Test Results

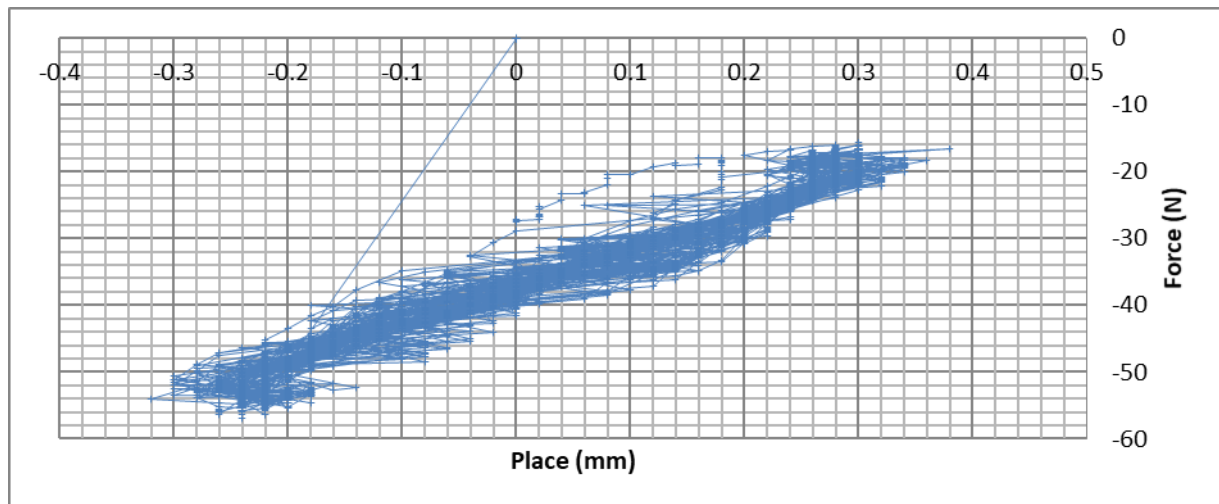
Stiffer Element – Static Condition – Longitudinal



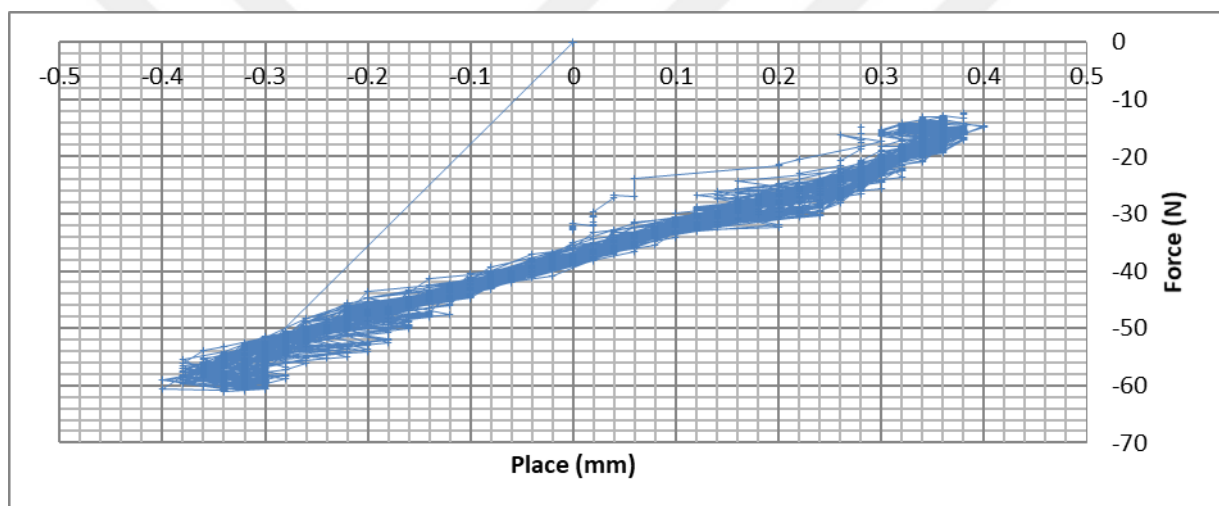
Stiffer Element – Dynamic Condition – 0.2mm – Longitudinal



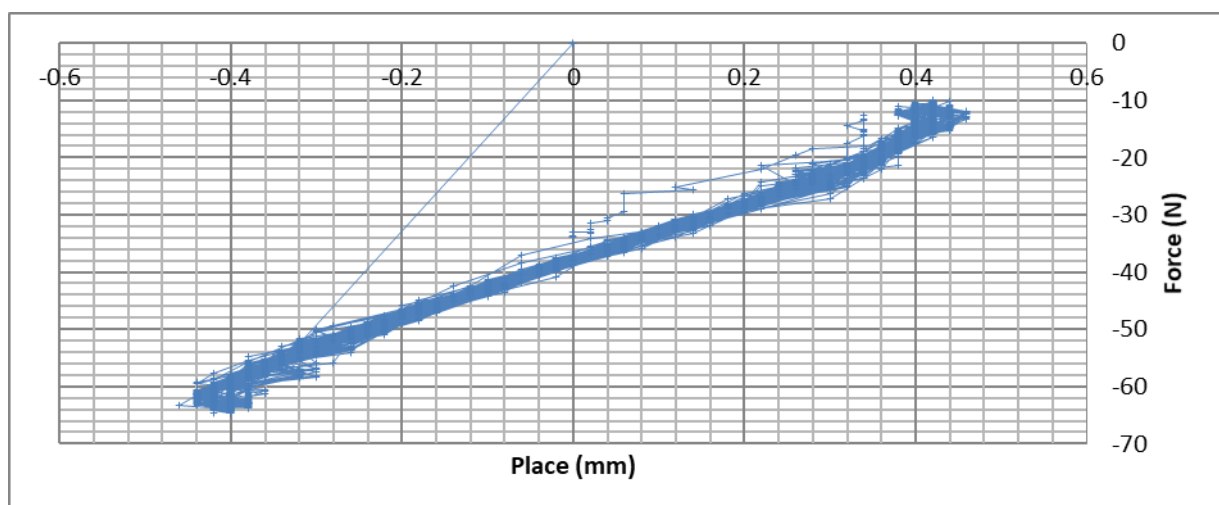
Stiffer Element – Dynamic Condition – 0.3mm – Longitudinal



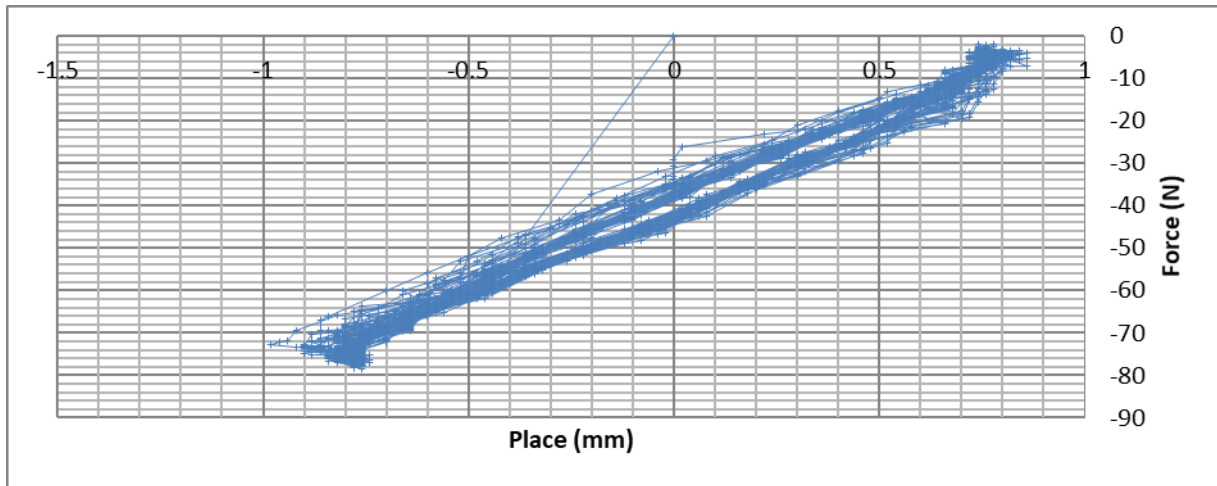
Stiffer Element – Dynamic Condition – 0.4mm – Longitudinal



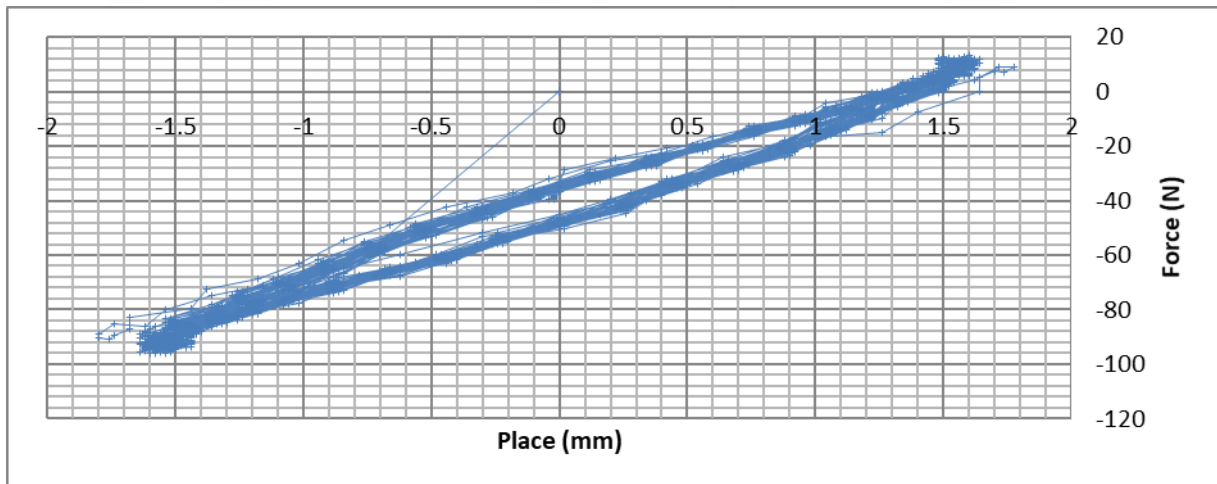
Stiffer Element – Dynamic Condition – 0.5mm – Longitudinal



Stiffer Element – Dynamic Condition – 1mm – Longitudinal

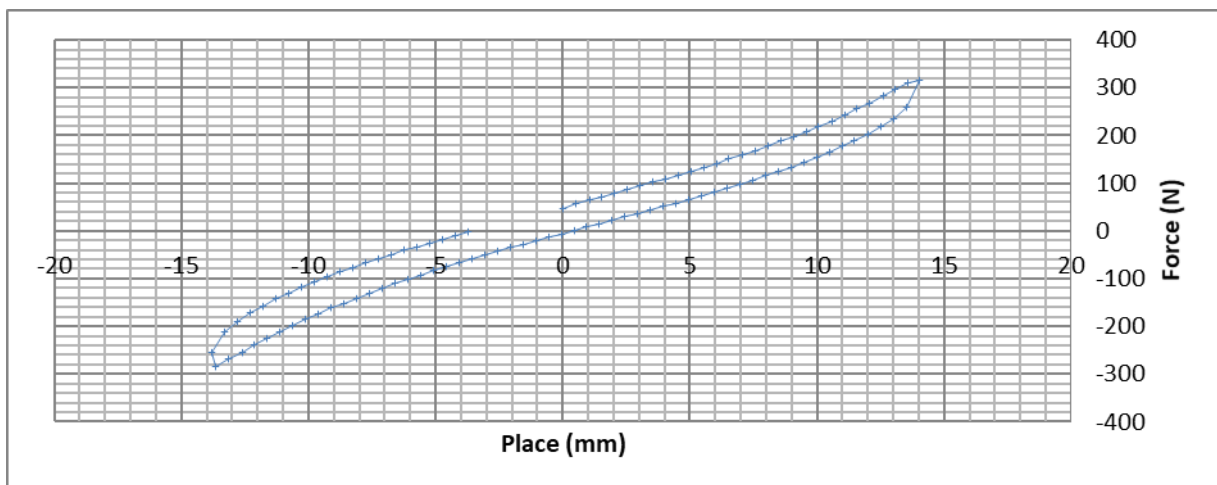


Stiffer Element – Dynamic Condition – 2mm – Longitudinal

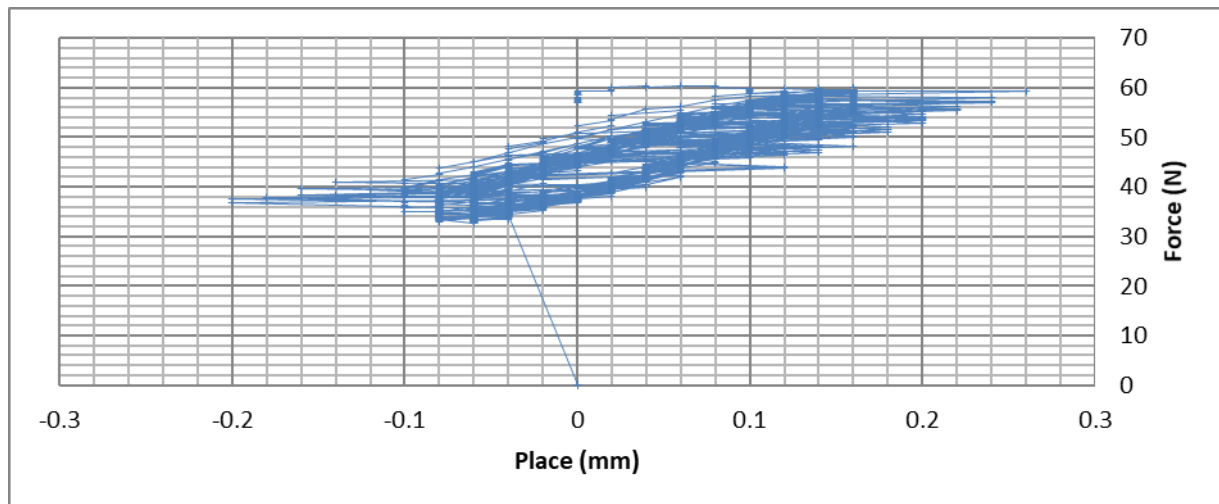


### Stiffer Element – Transversal Direction Stiffness Test Results

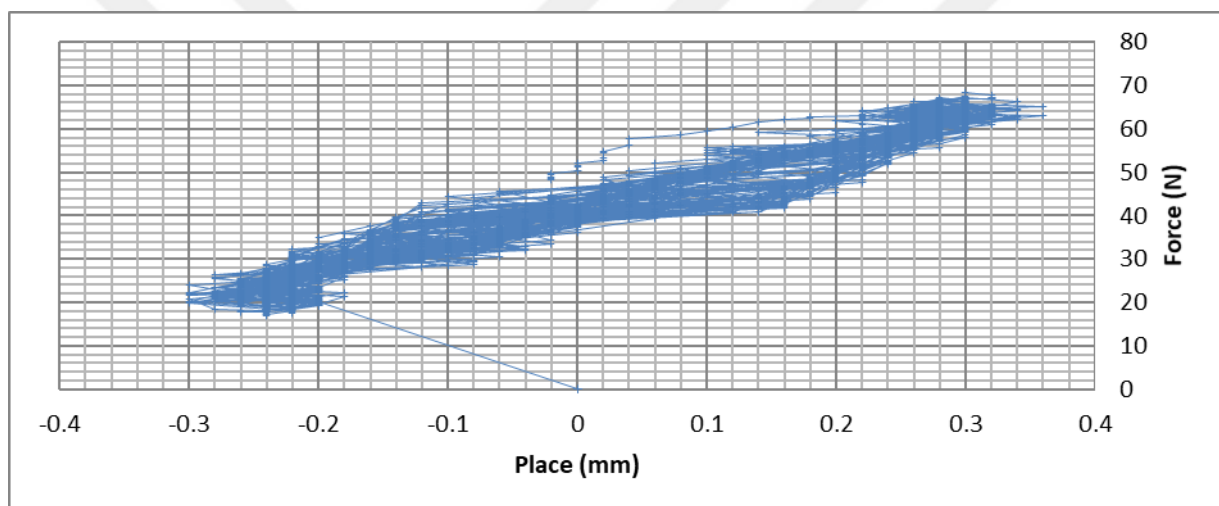
Stiffer Element – Static Condition – Transversal



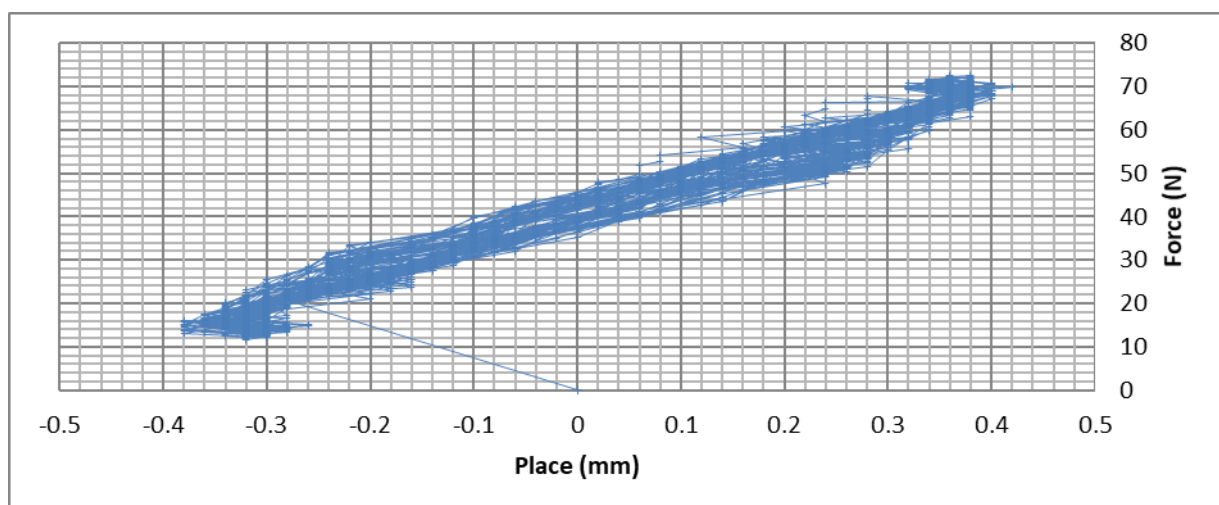
Stiffer Element – Dynamic Condition – 0.2mm – Transversal



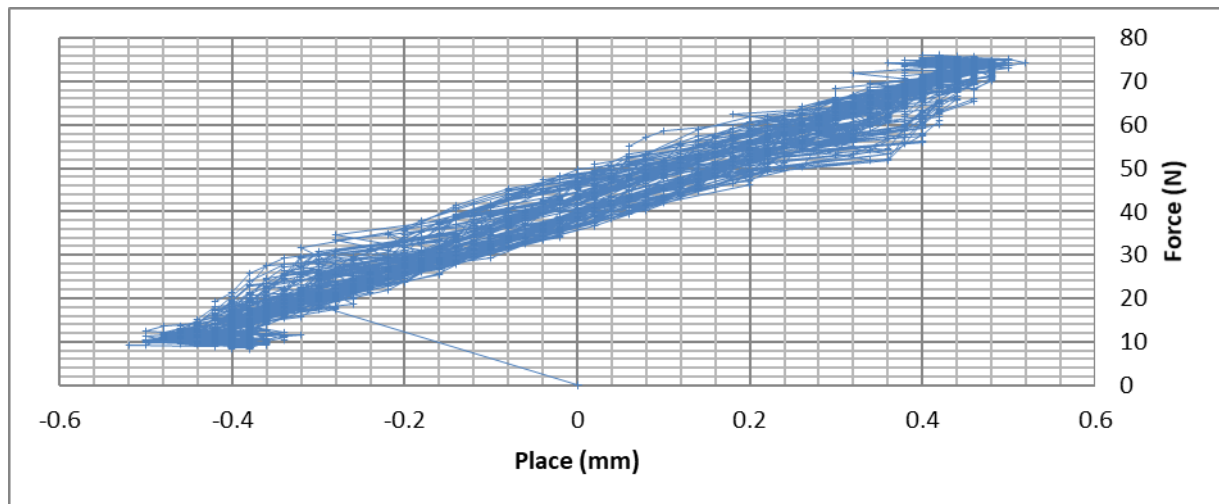
Stiffer Element – Dynamic Condition – 0.3mm – Transversal



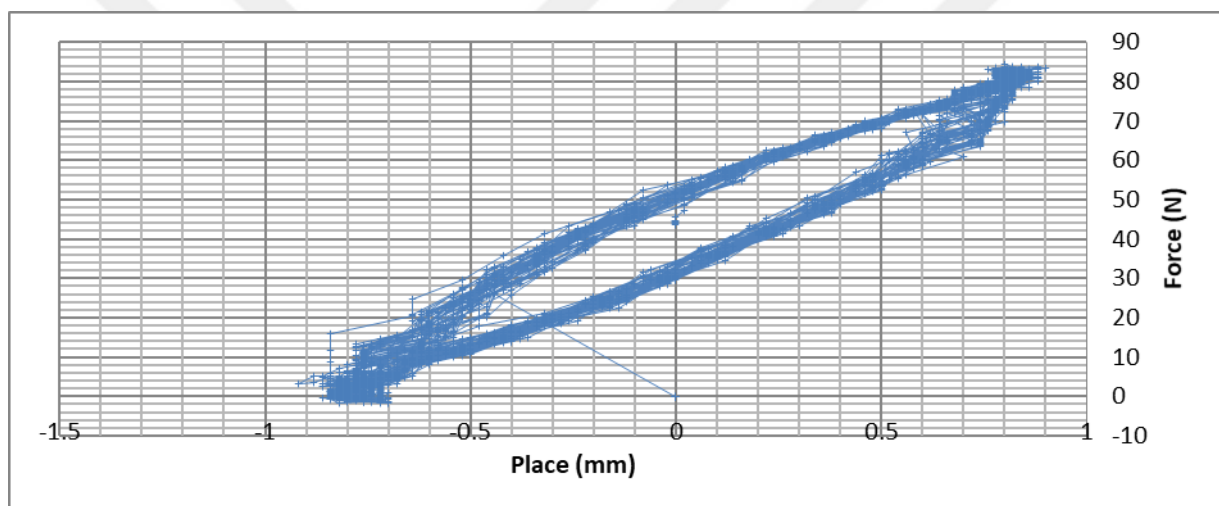
Stiffer Element – Dynamic Condition – 0.4mm – Transversal



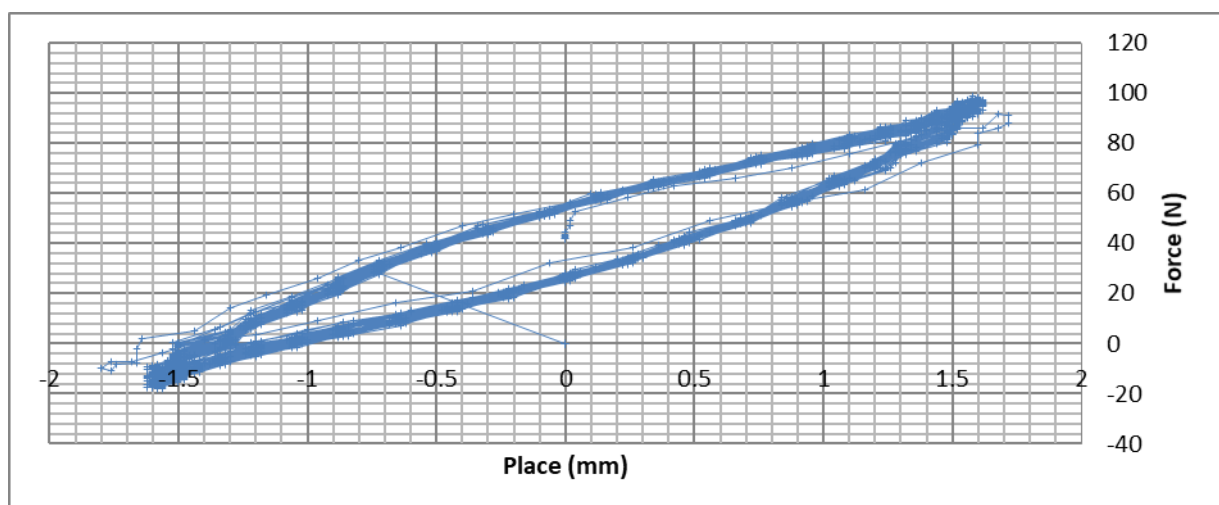
Stiffer Element – Dynamic Condition – 0.5mm – Transversal



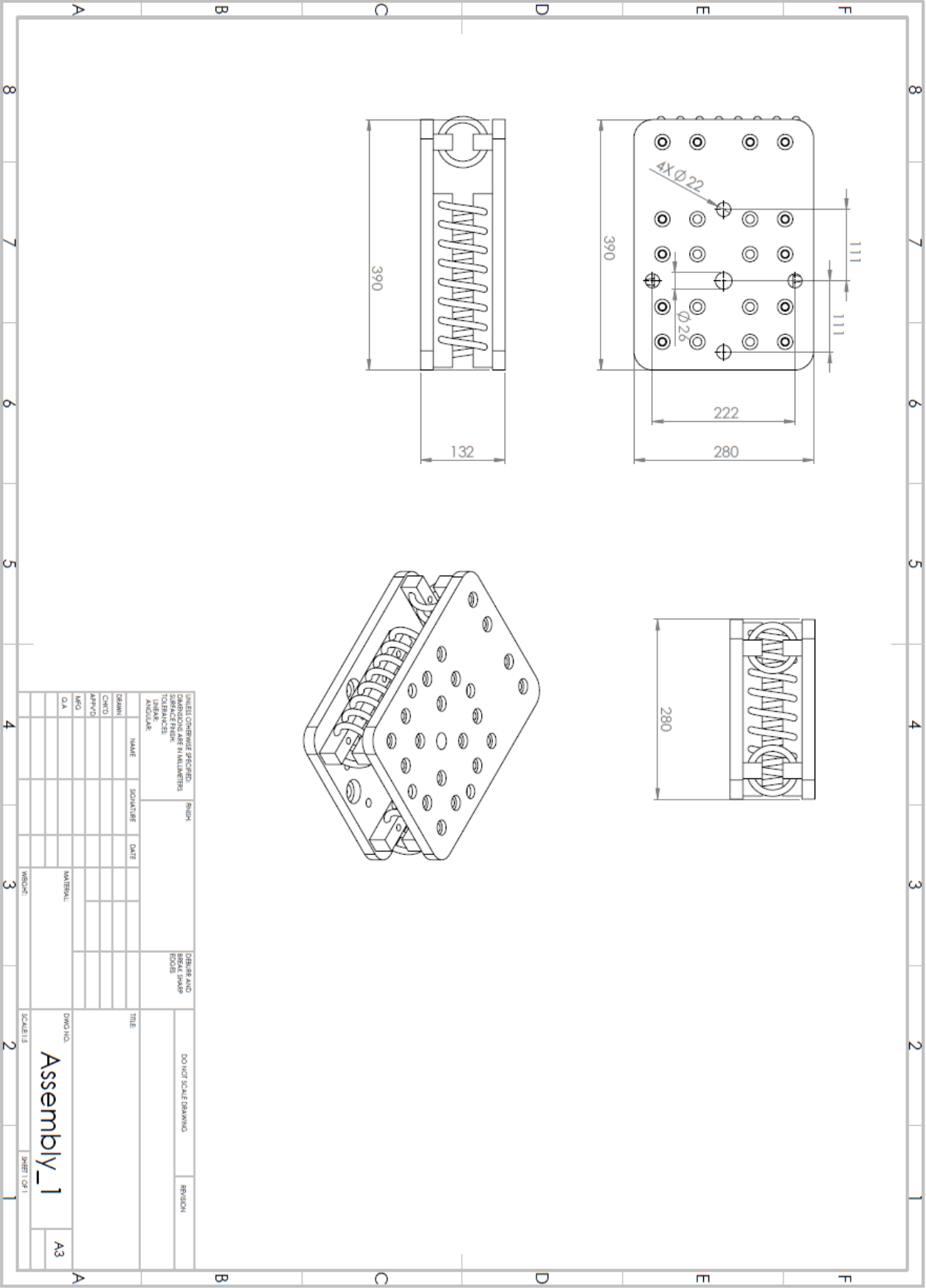
Stiffer Element – Dynamic Condition – 1mm – Transversal



Stiffer Element – Dynamic Condition – 2mm – Transversal



## Appendix E



# CURRICILUM VITAE

Tuncer Evci

E-mail: [tuncerevci@gmail.com](mailto:tuncerevci@gmail.com)

## WORK EXPERIENCE

- 02.2019-Present      **Vibrol Oy, Tuusula**  
**Master Thesis Worker**
- Wire Rope Engine Mount Design for Wärtsilä Diesel Engines
  - Wire Rope Electric Cabinet Mount Design for Wärtsilä Diesel Engines
  - Design of Vibrational Experimental Set-ups
  - Operational Modal Analysis
  - Design of Required Test Parts
- 01.2017-11.2018      **Ford Motor Company, R&D Center, Istanbul**  
**Powertrain Noise & Vibration Test Engineer**
- Powertrain NVH Test Responsible for Ford Passenger Cars, Light and Heavy-Duty Commercial Vehicles
  - Conduct NVH Tests in USA, Germany And Belgium
  - Modal Testing and Analysis
  - Operational Modal Analysis
  - Transfer Path Analysis
  - Powertrain Vibration & Acoustic Analysis
  - Torsional Vibration Measurements and Analysis
  - Subjective Vehicle Evaluations
- 09.2012- 01.2017      **Ford Motor Company, R&D Center, Istanbul**  
**Mechanical Design Engineer**
- Design of Engine and Powertrain Parts
  - Good Knowledge in Different Manufacturing Processes (Casting, Welding, Machining, Sheet Metal, Plastic Injection Moulding)
  - Design of Engine Cooling Systems
  - Design of LPG Conversion of Heavy-Duty Diesel Engine
  - Preparation Design Guidelines
  - Preparing Technical Drawings
  - Geometric Dimensioning & Tolerancing
  - Compliance with Design Guidelines
- 08.2011 - 09.2011      **Borçelik – Arcelor Mittal, Bursa**  
**Intern Engineer**
- Assisted manufacturing engineers with major and minor project details
  - Learned new skills quickly in a high-stakes atmosphere
- 06.2010 - 07.2010      **Yeşilyurt Iron and Steel Management, Samsun**  
**Intern Engineer**
- Worked as an intern in the area of the rolling mills and maintenance and repair in the iron and steel factory
  - Assisted manufacturing engineers with major and minor project details

## EDUCATION INFORMATION

|                   |                                                                                      |
|-------------------|--------------------------------------------------------------------------------------|
| 09.2006 – 08.2012 | Middle East Technical University, Ankara<br>Bachelor's level, Mechanical Engineering |
| 09.2014 – Present | Marmara University, Istanbul<br>Msc, Mechanical Engineering                          |

## LANGUAGE

|         |                                          |
|---------|------------------------------------------|
| English | Advanced in reading writing and speaking |
|---------|------------------------------------------|

## COMPUTER SKILLS

|                     |              |
|---------------------|--------------|
| Catia               | Advanced     |
| Teamcenter          | Advanced     |
| LMS Test.Lab        | Advanced     |
| MS Office           | Advanced     |
| Squadiga            | Advanced     |
| Head Visor          | Advanced     |
| Samtec & DET        | Advanced     |
| Dewetron & Dewesoft | Advanced     |
| SolidWorks          | Intermediate |
| Artemis             | Intermediate |
| Mathcad             | Intermediate |
| Matlab              | Intermediate |