DYNAMIC MODELING AND ANALYSIS OF VIBRATION EFFECTS ON PERFORMANCE IN OPTICAL SYSTEMS

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ABSTRACT

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In order to understand the effects of structurally induced line of sight (SILOS) jitter (vibration) and to predict its effects on optical system performance, a simple and practical vibratory model and software are developed by using discrete and finite element modeling techniques.

For an existing simple optical system, discrete and FE dynamic models are constructed and they are validated by modal tests for the frequency range of interest. In order to find material properties of adhesive, which is used in optical system, a simple test is constructed and these properties are found by using a single degree of freedom model. The effects of vibration on the system performance are investigated under random vibration load conditions by using the software developed. It is concluded that the analytical model suggested can successfully be used in preliminary design stage of a simple optical system when the optical housing and lens behave rigidly in the frequency range of interest. The optical performance prediction software combines the optical element tolerances and displacements in order to determine the optical performance.

Keywords: Jitter, Vibration, Optics, Finite Element, Optical Element Tolerances, Optical System Performance.

ÖZ

OPTİK SİSTEMLERİN DİNAMİK MODELLEMESİ VE TİTREŞİMİN ETKİLERİNİN ANALİZİ

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Optik sistemlerdeki titreşimin optik performans üzerindeki etkilerinin incelenmesi ve etkilerinin bulunması için basit ve pratikte kullanılabilecek bir titreşim modeli geliştirilmiştir.

Basit bir optik sistem için toplanmış parametreli ve sonlu elemanlar modelleri oluşturulmuş ve bu modeller ilgili frekans aralığında deneysel modal analiz yöntemi ile doğrulanmıştır. Optik sistem içersinde kullanılan yapıştırıcının malzeme özelliklerinin bulunabilmesi amacıyla basit bir test düzeneği kurulmuştur ve tek serbestlik dereceli sistem modeli kullanılarak bu özellikler bulunmuştur. Titreşimin optik performansa olan etkisi, rastgele yükler altında, geliştirilen yazılım kullanılarak incelenmiştir. Optik muhafazanın ve lensin rijit davrandığı frekans aralığında, geliştirilen basit analitik modelin basit optik sistemlerin ön tasarım çalışmalarında başarılı bir şekilde kullanılabileceği sonucuna varılmıştır. Geliştirilen optik performans inceleme yazılımı, optik toleranslar ile değişik yükler altında elde edilen yer değiştirmeleri birleştirerek optik performans hakkında bilgi vermektedir.

Anahtar Kelimeler: Jitter, Titreşim, Optik, Sonlu Elemanlar, Optik Eleman Toleransları, Optik Sistem Performansı.

To My Family…

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LIST OF SYMBOLS

A_1	: Peak Sine Acceleration Amplitude at Frequency f_1
A_2	: Peak Sine Acceleration Amplitude at Frequency f_2
A_3	: Peak Sine Acceleration Amplitude at Frequency f_3
A_4	: Peak Sine Acceleration Amplitude at Frequency f_4
A	: Area
C, c	: Damping
D	: Diameter
t	: Thickness
E	: Young's Modulus (Modulus of Elasticity)
E^*	: Complex Modulus Elasticity
f	: Frequency
f_n	: n-th Natural Frequency
G	: Shear Modulus
$H(\omega)$	: Frequency Response Function
h	: Height
I	: Inertia
K^*	: Complex Stiffness
K, k	: Stiffness
L	: Length
L_0	: Acceleration Spectral Density Amplitude
P	: Power Spectral Density Amplitude
G	: Output of the Power Spectral Density Analysis
R	: Radius
M	: Mass
M	: Maximum Modulus
m	: Mass
u	: Mode Shape

Tr	: Transmissibility
W_0, W_1	: Acceleration Spectral Density Amplitude
x	: Displacement
X	: Transnational Direction
Y	: Transnational Direction
Z	: Translational Direction
α	: Receptance Function
ε	: Strain
ϕ	: Rotational Direction
γ	: Shear Strain
η	: Loss Factor
ν	: Poisson Ratio
θ	: Rotational Direction
ρ	: Density
σ	: Stress
τ	: Shear Stress
ω	: Frequency
ζ	: Damping Ratio

CHAPTER 1

INTRODUCTION

1.1 Literature Survey

Opto-mechanics can be defined as a science or engineering discipline, and/or art of keeping the proper shapes and positions of the functional elements of an optical system so that the system performance needs are fulfilled [1]. Mechanical design of optical system or opto-mechanical design is very important to optical system development, where the optics and mechanics interface [2]. Purpose of the opto-mechanics and mechanical design is that optical elements should be hold in the optical system for keeping their proper positions and orientation [3].

Determination of the proper positions and orientation of optical elements is the issue of optics discipline. Proper position and orientation of optical elements depend on sensitivity and tolerance analysis of optic system. Using commercial software, such as CODE V or ZEMAX, sensitivity and tolerance analysis can be done.

Sensitivity and tolerance analysis should be conducted by considering both optical and mechanical design together. This type of analysis, which considers both optics and mechanics, is called opto-mechanical sensitivity and tolerance analysis. Magarill explained [4] that opto-mechanical analysis is the determination of the optical system performance and any deviation of the primary parameters in optical system design can cause performance degradation. Sensitivity and tolerance parameters should be determined carefully. Decenter and tilt are examples for these parameters.

The main purpose of the optical tolerancing is to find variation of optical elements and assembly parameters [4]. These parameters should fulfill the performance of the optical system and decrease the manufacturing and assembly costs.

Under environmental conditions, positions of optical elements may be changed because of loading. Loading, under environmental condition, can be vibration, shock, thermal effects, air flow or humidity.

Optical alignment is also one of the important procedures for the determination of the optical performance. When optical elements are assembled to the housing, alignment should be done, in other words, optical elements should be placed in their proper positions. Under environmental conditions, these positions may be changed or alignment in optical elements may slightly get out of order. These problems can degrade the optical performance.

Optical system performance can be degraded by excessive vibration. Vibration can cause performance degradation and failure of structure, induce a time-variant blur (for high level) and a blur in the focus (for low level) [1]. Because of these reasons, in order to prevent performance loss dynamic characteristics of an optical system should be known.

Vibration in optical systems is also called jitter [3]. Because of vibration structurally induced line of sight (SILOS) jitter occurs and this causes optical components move with respect to each other, i.e. optical elements displace their position and this cause decentration and tilt. Jitter also manifests itself as scene motion, blurring and reduction in range and recognition. System level dynamic jitter can be obtained from dynamic analysis (transient dynamic analysis, harmonic analysis or power spectral density analysis) by using image equations of motion in FE model [3].

Dynamic characteristics of an optical system depend on housing, optical element and mounting method. Housing is the covering part of an optical system and different types of optical elements can be handled by this part. Optical elements, lens, doublets, mirror, prisms, windows and beam splitters, can be classified by their functions. These optical elements are assembled into housing by using retainer, elastomer, o-ring or elastomer with retainer. Example for lens mounting can be seen in Figure 1.1.

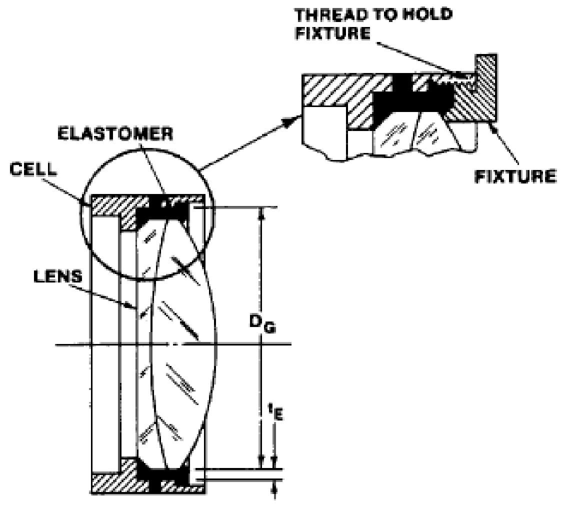


Figure 1. 1 Mounting Lens with Elastomer and Retainer [5]

According to normal gravity loading the decentration of optical elements are very small, however, this decentration may increase under shock and vibration loading. Therefore, design of elastomeric mount should be done carefully in order to avoid excessive decentration under vibratory environment [5].

Housing and lens mounts should maintain the position of the optical elements with respect to each other considering the design tolerances. Self-weight is the common load in optical systems and self-weight cause problem in alignment tolerances due to gravity. Because of this reason, housing and lens mounts should be designed in order to prevent excessive self-weight deflection.

If housing of the optical system is stiff enough, than lens mounts will be designed carefully in order to prevent any movement of the lens. Design requirements of the lens mounts are given by Jones [6] as follows; lenses should be placed within the optical tolerances and kept in proper design positions under the environmental conditions, such as vibration, temperature or humidity.

Lens mount should be designed to avoid thermal and dynamic effect without causing considerable aberrations into optics [6]. All designed mounts should be analyzed and tested under worst environmental conditions (vibration, thermal or humidity environment) for obtaining best performance of mounts.

Design of glass to metal [7] interface in optical system must be done carefully. Any temperature change causes contact stresses between the glass and the metal because of the different thermal expansion coefficient of glass and metal. Therefore, an adhesive buffer between glass and metal should be used in order to maintain the lens in the mount and provide the stress-spacing.

In order to find dynamic characteristics of an optical system, simple finite element analysis is performed in reference [8]. This analysis is done for finding suitable adhesive thickness to satisfy the proper athermal and dynamic design. Results of the dynamic analysis are given in Figure 1.2. It is stated in this study that designer should consider mount geometry, material properties of adhesives and proper optic mounting.

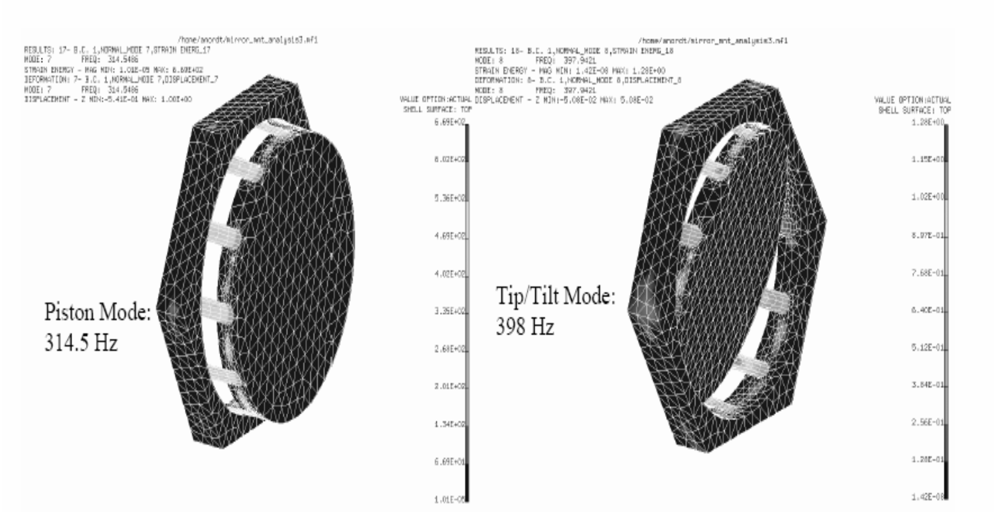


Figure 1. 2 First and Second Mode of the Mount

Dynamic characteristics of a military optical system, which are affected by housing, lens and lens mounting, should be assessed by using MIL-STD-810 or other related standards. Military optical systems should withstand worst environmental conditions. Military environmental effects are very challenging and very wearing, therefore optical system should be designed to overcome these effects. Environmental effects are very well defined in standards. One of the most popular and well-known standard is MIL-STD-810. In MIL-STD-810, all environmental specifications, such as temperature, vibration, shock or humidity etc., are given for design requirements. These specifications are prepared for worst conditions. After design is finished, product should pass all environmental tests, which are defined in MIL-STD-810. Vibration test purposes are stated in MIL-STD-810 F [9] as follows:

> *"Vibration tests are performed to:*
> *a. Develop material to function in and withstand the vibration exposures of a life cycle including synergistic effects of other environmental factors, material duty cycle, and maintenance. Combine the guidance of this method with the guidance of Part One and other methods herein to account for environmental synergism.*
> *b. Verify that materiel will function in and withstand the vibration exposures of a life cycle."*

Military optical systems are integrated to military platforms, such as helicopters, aircrafts or battle tanks. Therefore, environmental conditions are very challenging. In military standards, environmental conditions for specific applications or vehicles are defined clearly. Random vibration characteristics of propeller aircraft are as defined in MIL-STD-810F [9] and are given in Figure 1.3.

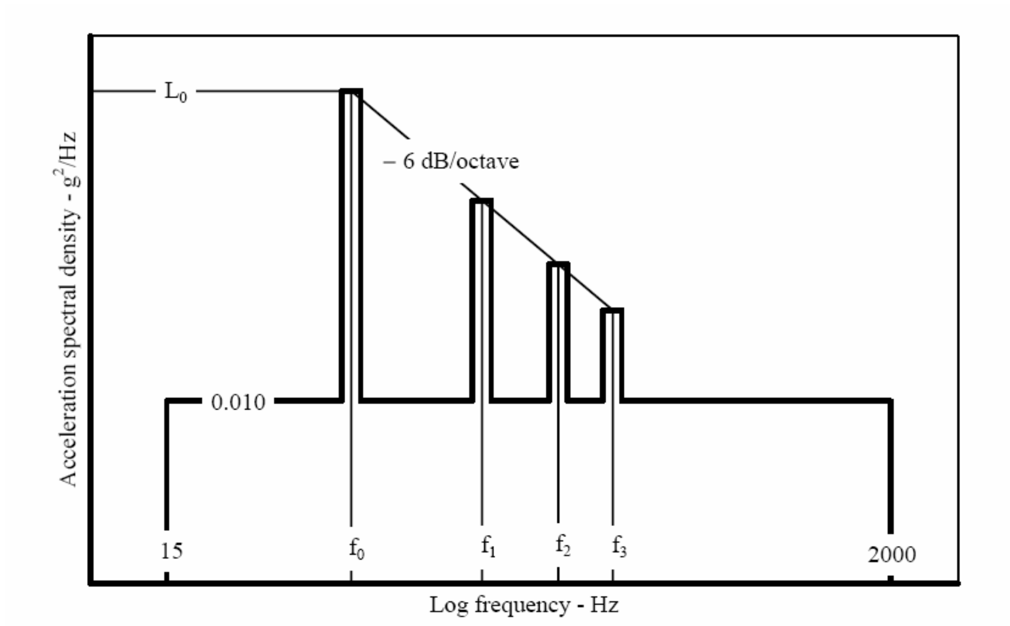


Figure 1. 3 Vibration Profile of Propeller Aircraft [9]

In order to obtain a good design for an optical system, both mechanical and optical issues should be assessed together. Mechanical parts of optical system have to be designed to prevent the adverse effects of environmental conditions and keep optical elements as stable as possible.

1.2 Objective

The objective of this thesis is to determine the dynamic characteristics of optical system by making vibration analysis for understanding the effects of vibration on optical system. For this reason, analytical and finite element models will be constructed for a simple optical system. The computational results obtained from both models will be compared and also will be verified by using experimental modal test results. After verification, random vibration analysis will be carried out by using analytical and finite element model in order to see the effect of vibration on optical system. Moreover, random vibration test will be conducted for comparing experimental results with those of the computational methods.

Two simple software are also developed in this thesis. First software is for the determination of material properties of adhesive, which is used in the optical system. The other is for the prediction of vibration effects on optical systems. These software will be used by opto-mechanical engineers during the design process. In the first step of opto-mechanical design, opto-mechanical engineer will use the simple analytical model to analyze the effect of vibration without carrying out any detailed analysis. If detailed analysis is needed, finite element model can be constructed and analyzed.

1.3 Scope of Thesis

In this thesis, dynamic model of an optical system will be constructed and optical performance of an optical system will be investigated under dynamic loading. In this aspect, the outline of the thesis is as follows:

Finite element model of an optical system will be obtained in Chapter 2 in order to understand the dynamic behavior of optical system elements. In this chapter each optical element, case and lens will be analyzed by using finite element method, and results will be validated by experimental techniques. Finally, optical system assembly will be constructed and dynamic analysis will be carried out for the whole assembly.

In Chapter 3, using the results of Chapter 2, an analytical model will be developed for a simple optical system. The material properties of adhesive, which is elastomer type material, will be also determined in this chapter. Using material properties of the case, lens and adhesive and a simple analytical model will be developed. The model will be used to find the natural frequencies of optical system and the dynamic response under random vibration loading.

The simple analytical model developed in Chapter 3 will be validated in Chapter 4 by using finite element model and experimental techniques. Firstly, a simple optical system will be modeled by using finite elements and dynamic characteristics of this system will be found. Next, the system will be produced and dynamic analysis will be carried out by experimental techniques, modal and random vibration test. Results of both finite element analysis and experiments will be compared with those obtained from analytical model.

In order to understand the effect of optical parameters under vibratory environment, tolerance analysis will be conducted in an optical design program in Chapter 5. In this tolerance analysis, decenter and tilt tolerances will be considered and tolerance limits obtained are compared with random vibration analysis results in order to see the effect of vibration on optical performance.

In Chapter 6, two simple computer programs will be developed; one is for determination of material properties of adhesives and the other is for analyzing vibratory behavior of the optical system. The latter is developed by using simple analytical model suggested in the previous chapters. First software will find material properties of the adhesives by using experimental values and a simple discrete model of the system. Second software requires basic optical system parameters, such as diameter of lens, modulus elasticity of adhesive as the input. This software calculates modal parameters of the optical system. Also under known random vibration input, response of the system can be obtained with this software. Moreover, by inserting decenter tolerance limits, optical system performance will be predicted by taking the results of random vibration analysis results into account.

Summary of the results is given in Chapter 7. Discussion of the results and the conclusions are also given in this chapter. Suggestions for a future study are also presented in this chapter.

CHAPTER 2

FINITE ELEMENT MODEL OF THE OPTICAL SYSTEM

In order to determine the vibration effects on an optical system, dynamic characteristics of the optical system should be investigated. Modal parameters of the system should be found over the interested frequency range and by using these parameters displacement of the system under vibration should be predicted.

For finding dynamic characteristics of an optical system, finite element method can be used. In this thesis, ANSYS is used for solving the finite element model of the optical system. ANSYS has two versions, one is ANSYS Classic and another is ANSYS Workbench. Both have pre and post processing modules. Moreover, both can be connected directly to CAD programs, such as Pro/ENGINEER, in other words; designed optical system can be read by ANSYS from Pro/ENGINEER and also if something is changed in CAD model, finite element model can easily be updated.

In order to obtain reliable results from finite element program, one should know the material properties of components, boundary conditions and loads. In military optical systems, the only difference from civil optical systems is that military optical systems should have enough strength to survive under harsh environmental conditions, i.e. materials and boundary conditions of military optical systems may be the same as civil optical system, however loads in military optical systems are different from those in civil optical systems. For this reason, military optical systems should be analyzed by considering military standards.

Military optical system should withstand harsh environmental conditions. They should be designed by considering military standards, customer needs and environmental conditions can be determined by experimental tests. Usually, military standards simulate the worst environmental conditions. If an optical system designed with respect to military standards, it already satisfies the customer needs and environmental conditions determined by experimental tests. As explained in Chapter 1, most popular military standard is MIL-STD-810F [9] and it explains the environmental conditions and test specifications. In MIL-STD-810F [9] dynamic effect, acceleration, vibration and shock environment are specified. Given load conditions in military standards can be used and dynamic analysis can be conducted in finite element program.

In this study, the finite element model is constructed for a specific military optical system. This optical system consists of an afocal, where all optical elements and detectors are mounted, and lenses that are fixed to the afocal by using elastomeric adhesive.

First of all, each part of the optical system is analyzed separately and modal parameters are found for free-free boundary conditions. After finding modal parameters, dynamic characteristic of each part of optical system is investigated over the interested frequency range, which is given in MIL-STD-810F [9]. As explained, there are different vibration profiles for different platforms. After determination of dynamic characteristic of each part of optical system, their behavior in frequency range of interest can be understood.

In MIL-STD-810F [9], vibration environment of different helicopters is defined [9] in “TABLE 514.5C-IV” and “FIGURE 514.5C-10”, which are given here in Figure 2.1 and 2.2**.** In these figures, general vibration levels and frequency ranges are given and also for specific helicopter, rotor frequency, blade passage, 1st and 2nd harmonic frequencies are specified.

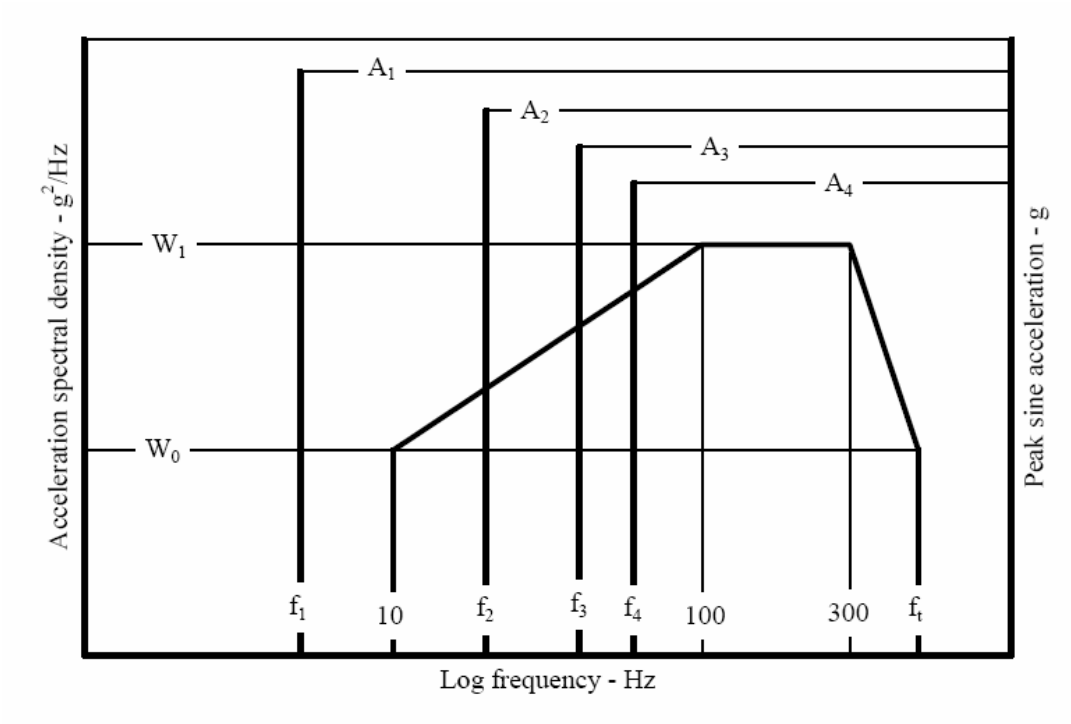


Figure 2. 1 General Vibration Profile of Helicopter [9]

For AH-1 (Cobra) helicopter, rotor frequency f_1 is 5.4 Hz and number of blades is 2 (Figure 2.2). Moreover f_t, which defines the frequency range of interest, is 10-500 Hz, W_0, minimum random vibration level, is 0.001 g^2/Hz and W_1, maximum random vibration level, is 0.01 g^2/Hz. Using rotor frequency and number of blade, blade passage, 1^{st} and 2^{nd} harmonic frequencies can be calculated by using Equation 2.1, 2.2 and 2.3 [9]. For these frequencies, sine peak acceleration, A_1, A_2, A_3 and A_4, can be found by using "TABLE 514.5C-IV" [9].

$$\text{Blade Passage} = n \times f_1 \tag{2.1}$$

$$1^{st}\ \text{Harmonic} = 2 \times n \times f_1 \tag{2.2}$$

$$2^{nd}\ \text{Harmonic} = 3 \times n \times f_1 \tag{2.3}$$

MATERIEL LOCATION	RANDOM LEVELS	SOURCE FREQUENCY (f_x) RANGE (Hz)	PEAK ACCELERATION (A) at f_x (GRAVITY UNITS (g))
General	$W_0 = 0.0010\ g^2/Hz$ $W_1 = 0.010\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.10 \times f_x$ 2.50 $6.50 - 0.10 \times f_x$ 1.50
Instrument Panel	$W_0 = 0.0010\ g^2/Hz$ $W_1 = 0.010\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.070 \times f_x$ 1.750 $4.550 - 0.070 \times f_x$ 1.050
External Stores	$W_0 = 0.0020\ g^2/Hz$ $W_1 = 0.020\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.150 \times f_x$ 3.750 $9.750 - 0.150 \times f_x$ 2.250
On/Near Drive System Elements	$W_0 = 0.0020\ g^2/Hz$ $W_1 = 0.020\ g^2/Hz$ $f_t = 2000$ Hz	5 to 50 50 to 2000	$0.10 \times f_x$ $5.0 + 0.010 \times f_x$

Main or Tail Rotor Frequencies (Hz) Determine 1P and 1T from Specific Helicopter or from Table (below).			Drive Train Component Rotation Frequency (Hz) Determine 1S from Specific Helicopter and Component.	
$f_1 = 1P$	$f = 1T$	fundamental	$f = 1S$	fundamental
$f = n \times 1P$	$f = m \times 1T$	blade passage	$f = 2 \times 1S$	1st harmonic
$f = 2 \times n \times 1P$	$f = 2 \times m \times 1T$	1st harmonic	$f = 3 \times 1S$	2nd harmonic
$f = 3 \times n \times 1P$	$f = 3 \times m \times 1T$	2nd harmonic	$f = 4 \times 1S$	3rd harmonic

Helicopter	MAIN ROTOR		TAIL ROTOR	
	Rotation Speed 1P (Hz)	Number of Blades n	Rotation Speed 1T (Hz)	Number of Blades m
AH-1	5.40	2	27.7	2
AH-6J	7.80	5	47.5	2
AH-64(early)	4.82	4	23.4	4
AH-64(late)	4.86	4	23.6	4
CH-47D	3.75	3	2 main rotors and no tail rotor	
MH-6H	7.80	5	47.5	2
OH-6A	8.10	4	51.8	2
OH-58A/C	5.90	2	43.8	2
OH-58D	6.60	4	39.7	2
UH-1	5.40	2	27.7	2
UH-60	4.30	4	19.8	4

Figure 2. 2 Helicopter Vibration Exposure [9]

For the example system, AH-1 helicopter (Cobra), Figure 2.3, is chosen as an airborne platform and its vibration profile is used to see the dynamic characteristics each part of optical system. Finally, dynamic analysis is done for whole optical assembly.

Figure 2. 3 AH-1 Cobra Helicopter

2.1 Finite Element Analysis of Afocal and Validation of the Results

As explained above, dynamic characteristics of the optical system are analyzed component by component. Therefore, outer part of the optical system, housing or afocal, is modeled using finite element method. Using the finite element model, modal analysis is done and modal parameters, especially natural frequencies of afocal are found. Finally, it is investigated to see whether afocal is affected by vibration level of AH-1 helicopter.

Geometry of afocal, which is the main part of the optical system, is very complex. Different types of optical element, different type of lenses, beam splitters or detectors exist in the afocal; therefore it can only be constructed in an advanced computer aided design program, Pro/ENGINEER. Geometry of afocal can be seen in Figures 2.4 and 2.5.

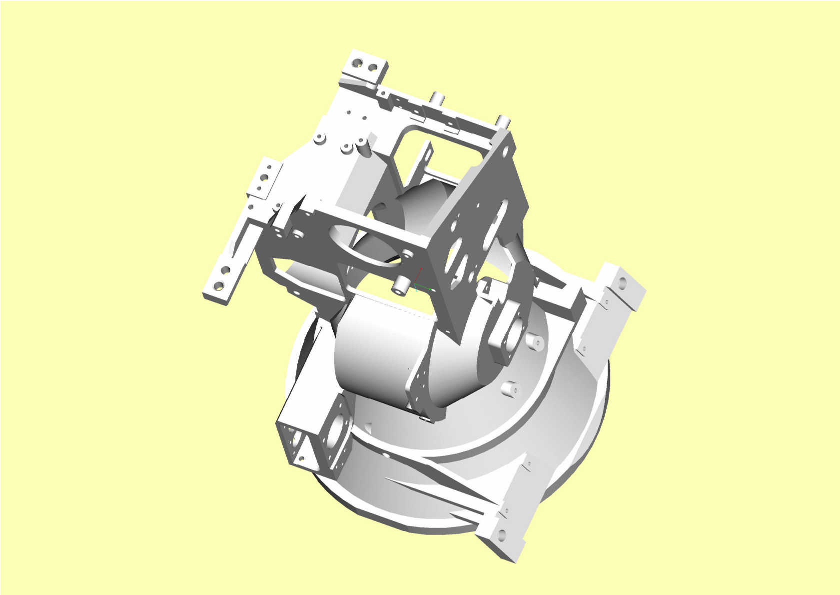


Figure 2. 4 Geometry of Afocal – View 1

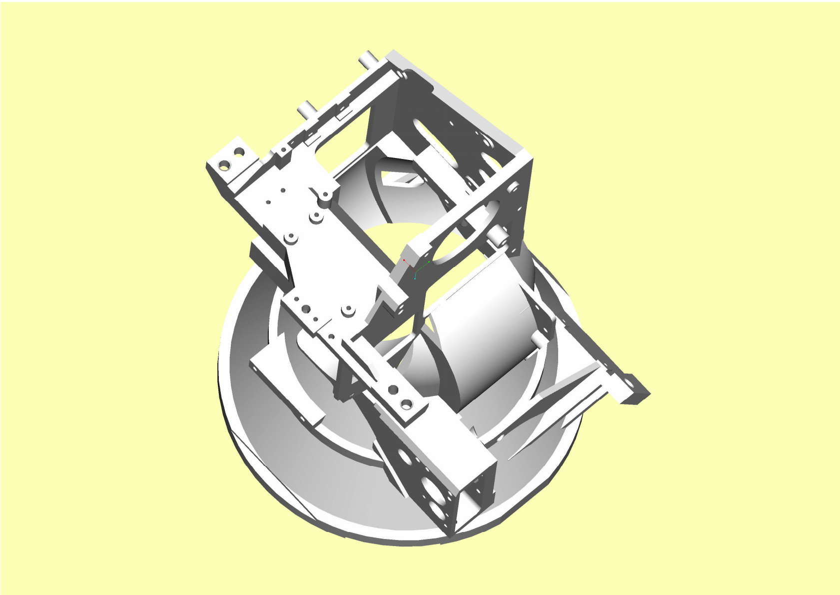


Figure 2. 5 Geometry of Afocal – View 2

This geometry is automatically imported to the finite element program ANSYS Workbench. In ANSYS Workbench, geometry of afocal is modeled by nodes and elements. Also material property of the afocal can be inserted into the program. Finally, modal analysis is conducted for free-free boundary conditions.

In ANSYS Workbench, default element type for solid bodies is SOLID 186 [10], which is higher order element and contains 20 nodes. Mesh model of the afocal can be seen in Figure 2.6 and this model contains 80600 element and 148250 nodes.

The afocal is made of aluminum. Material properties of aluminum are used from material library of ANSYS Workbench. After determination of the material properties of the afocal, modal analysis is done for free-free boundary conditions and first 6 modes are found.

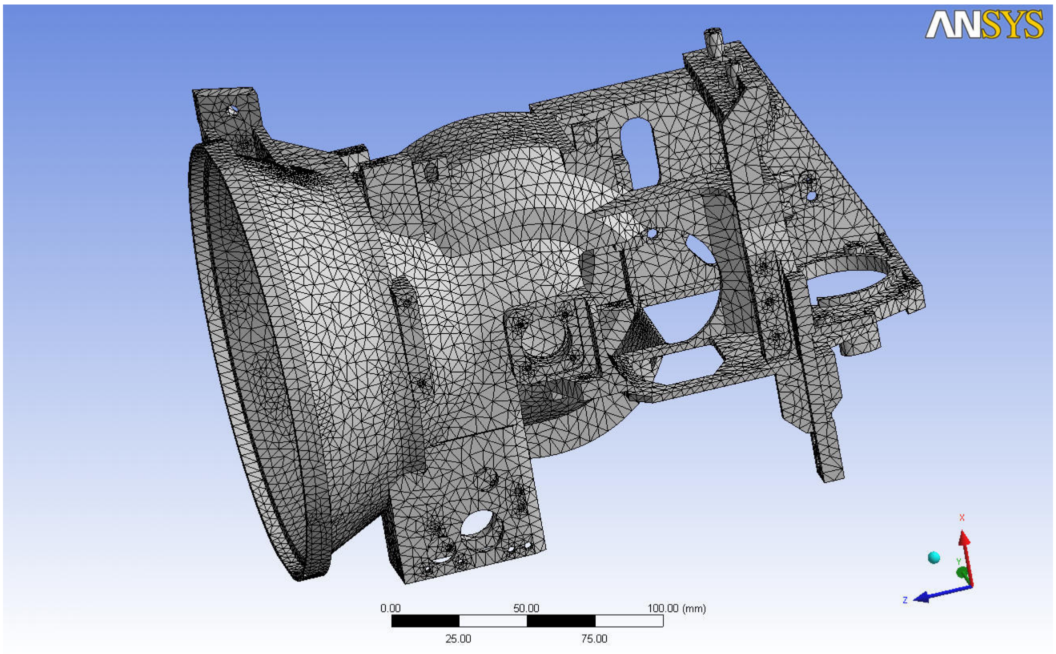


Figure 2. 6 Mesh of Afocal

As can be seen in Table 2.1, first natural frequency of the afocal is 830 Hz. These results are also validated by modal tests. In modal test, Bruel & Kjaer system is used. In order to obtain experimental results fast, impact testing is carried out.

Table 2. 1 Natural Frequencies of Afocal

ANSYS Results	
Mode	Frequency (Hz)
1	830,24
2	986,81
3	1193,97
4	1247,60
5	1287,30
6	1418,57

The modal test requires several tools, such as a force transducer, an accelerometer, data acquisition system and signal processing software. The test object can be excited by using a hammer or a shaker can be used. Hammer is the simplest excitation technique because, it is easy to use. The instruments used and the properties of transducers can be seen in Table 2.2 and 2.3.

Table 2. 2 Instrumentation and Software

Instrumentation and Software	
Accelerometer	Bruel & Kjaer 4507 biax
Impact Hummer	Bruel & Kjaer 8200+2646
Analyzer	Pulse Front-End 3560C
Software	Pulse 11.0

Table 2. 3 Transducer Properties

Transducer Type	Nom. Sensitivity	External Gain	Input Sensitivity
Force	1 mV/N	1 V/V	1 mV/N
Accelerometer	10 mV/m/s^2	1 V/V	10 mV/m/s^2

As can be seen in Figure 2.7, in test one fixed accelerometer is used and hammer is roved between ten points. Parameters used in the signal analysis configuration for the modal test can be seen in Figure 2.8. By PULSE 11 [11], time data can be performed in order to obtain auto-spectrums, cross-spectrums, FRFs and coherences.

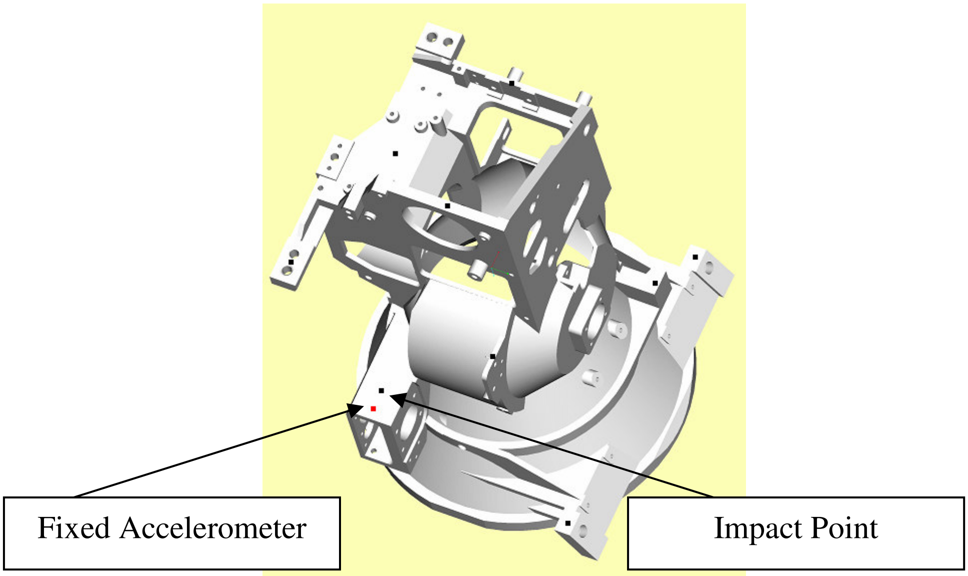


Figure 2. 7 Test Configuration

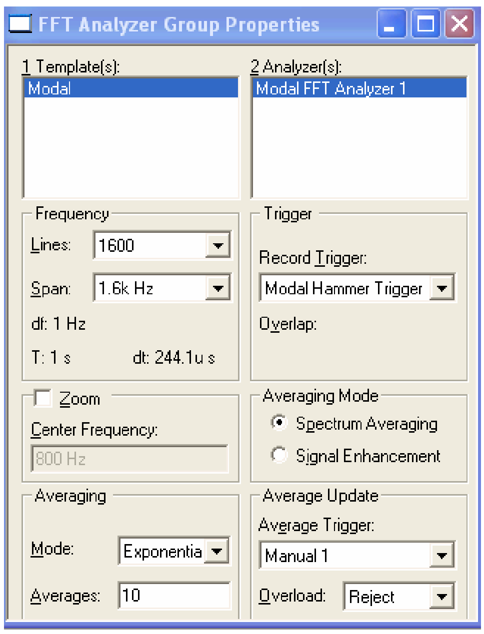


Figure 2. 8 FFT Properties

Finally, by using FRF curves and ME'scopeVES Version 4 [12], modal parameters can be found. These results are presented in Table 2.4. As can be seen in Table 2.5; error in natural frequencies obtained from FE analysis is less than approximately 1.1%. It can be say that results for each analysis are very accurate, therefore it can be said that finite element model is validated.

Table 2. 4 Experimental Results

Experimental Result		
Shape	Frequency (Hz)	Damping (%)
1	828	0.105
2	976	0.167
3	1190	0.273
4	1250	0.114
5	1300	0.172
6	1430	0.109

Table 2. 5 Comparison of the ANSYS and Experimental Results

	Experimental Results	**ANSYS Results**	**Error of Frequency**
Shape	Frequency (Hz)	Frequency (Hz)	(%)
1	828	830.24	- 0.27
2	976	986.81	- 1.1
3	1190	1193.97	- 0.33
4	1250	1247.60	+ 0.19
5	1300	1287.30	+ 0.99
6	1430	1418.57	+ 0.81

For AH-1 Helicopter, interested frequency range is 10-500 Hz; therefore afocal is rigid enough in the frequency range. Therefore, dynamic characteristics of the optical system will not be affected from the dynamics of afocal in AH-1 Helicopter and afocal can be taken as rigid body in the analyses.

2.2 Finite Element Analysis of Lens and Validation of the Results

Another critical part of optical system is the group of optical elements, which consists of lenses, detectors, mirrors and beam splitters. These optical systems play important role in determination of the dynamic characteristics. In order to see the effect of optical elements, one of the optical elements, namely a lens is taken as case study and it is modeled in finite element program, and dynamic characteristics of lens are investigated.

As can be seen in Figure 2.9, lens geometry looks like a circular plate; however there are cavities on both upper and lower surfaces. The lens analyzed here was made of germanium with the following material properties:

Table 2. 6 Material Properties of the Lens

Modulus of Elasticity, E	130 GPa
Density, ρ	5323 kg/m3
Poisson ratio, ν	0.28-0.32

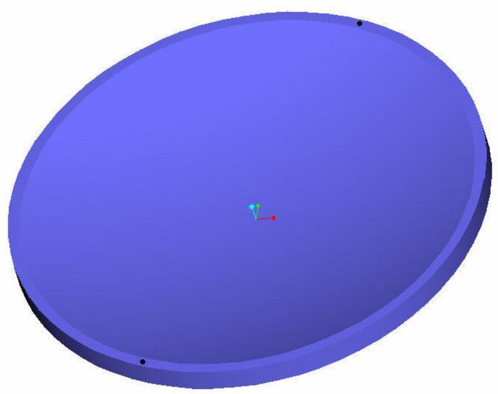


Figure 2. 9 Geometry of the Lens

Finite element model of the lens is constructed in ANSYS Workbench. It contains 59635 nodes and 38915 SOLID 186 elements as shown in Figure 2.10.

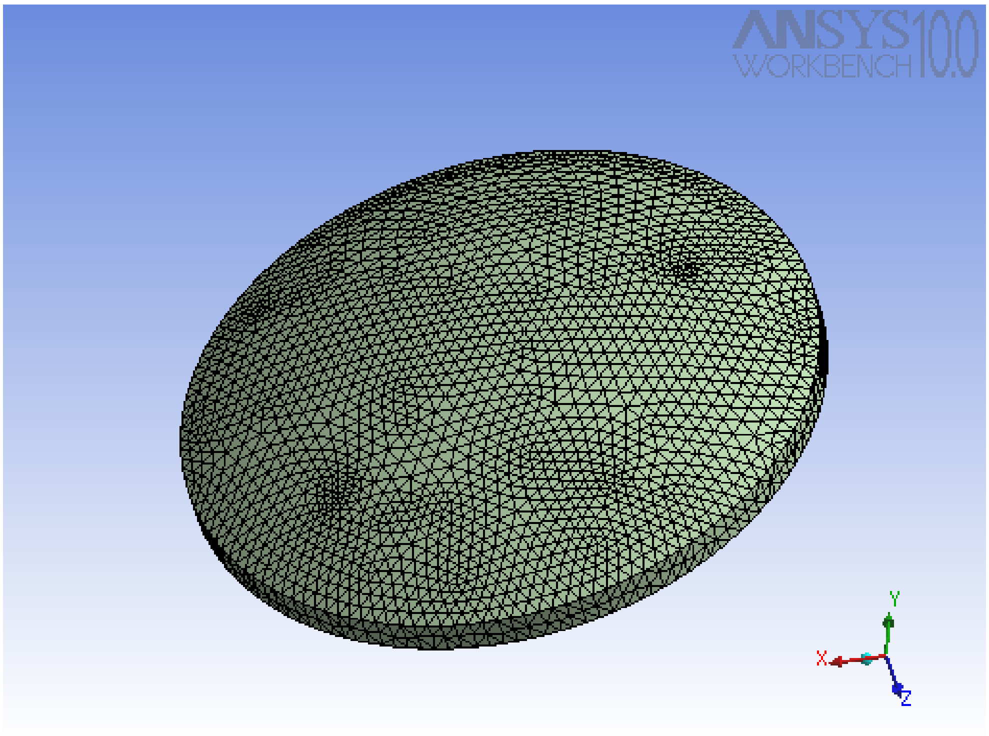


Figure 2. 10 Mesh of the Lens

After modeling the lens in the finite element program, modal analysis is carried out for free-free boundary conditions and modal frequencies of the first 3 modes are found (Table 2.7).

Table 2. 7 Natural Frequencies of the Lens

ANSYS Results	
Mode	Frequency (Hz)
1	3250
2	6500
3	6800

First three natural frequencies of the lens can be seen in Table 2.7. In order to verify the FE results, three natural frequencies are also obtained by experimental methods. The procedure explained in previous section is used for finding the natural frequencies. The instruments used and the properties of transducers are the same as in Table 2.2 and 2.3.

The comparison of the natural frequencies of the lens calculated from finite element model and those obtained experimentally are given in Table 2.8 along with percentage error in FEA results.

Table 2. 8 Comparison of the ANSYS and Experimental Results for Lens

Mode	Experimental Results (Hz)	FEA Results (Hz)	Error (%)
$\mathbf{f_1}$	3250	3292	- 1.3
$\mathbf{f_2}$	6500	6672	- 2.6
$\mathbf{f_3}$	6800	6818	- 0.3

As can be seen in Table 2.8, natural frequencies of the lens are much higher than 500 Hz, which is the upper limit of the frequency range of interest for the AH-1 Helicopter. Therefore it can be said that lens can be taken as rigid in the frequency range of 10-500 Hz.

2.3 Finite Element Model of the Optical System

Finite element models of the case, which is the afocal, and lens are obtained in previous sections and it is seen that they can be assumed to be rigid in the AH-1 helicopter excitation frequency range of 10-500 Hz. The first natural frequency of the afocal is 828 Hz and the first natural frequency of the lens is found to be 3250 Hz (see Tables 2.5 and 2.8).

Lenses are mounted to the afocal by using adhesives. One of the advantages of using adhesive is athermalization. Athermalization is the compensation of the thermal changes between two materials [1]. For example, thermal expansion coefficients of aluminum and germanium are different and due to temperature change, the characteristic lengths become different. If no interface material is used between the case and the lens, any high temperature change causes high stress between the lens and the case. Because of this reason, there maybe stress formation and crack propagation. In order to make athermalized design, an adhesive should be used as an interface material.

Adhesives are soft materials and their stiffness is much lower compared to those of aluminum, steel or germanium. Due to low stiffness, dynamic characteristic of the optical system is affected by adhesives, which is an elastomer type material. In order to see the effect of the elastomer, complete model of the optical system is obtained by using finite element program.

The optical system assembly is constructed in Pro/ENGINEER and it is imported directly to ANSYS Workbench. After importing, ANSYS Workbench automatically detects contacts between individual parts. Therefore, it finds the contact between the elastomer and the afocal, also between the lens and the elastomer in the optical assembly. These contacts are selected as bounded contacts, in other words, there is no separation between these parts.

The optical system assembly can be seen in Figure 2.11 and this system consists of the afocal, the lens and the elastomer. In this model only the first lens of the optical system is used because this lens is the powerful optical element and its mass is the highest in the optical system. Figure 2.12 shows the use of elastomer in optical system more clearly.

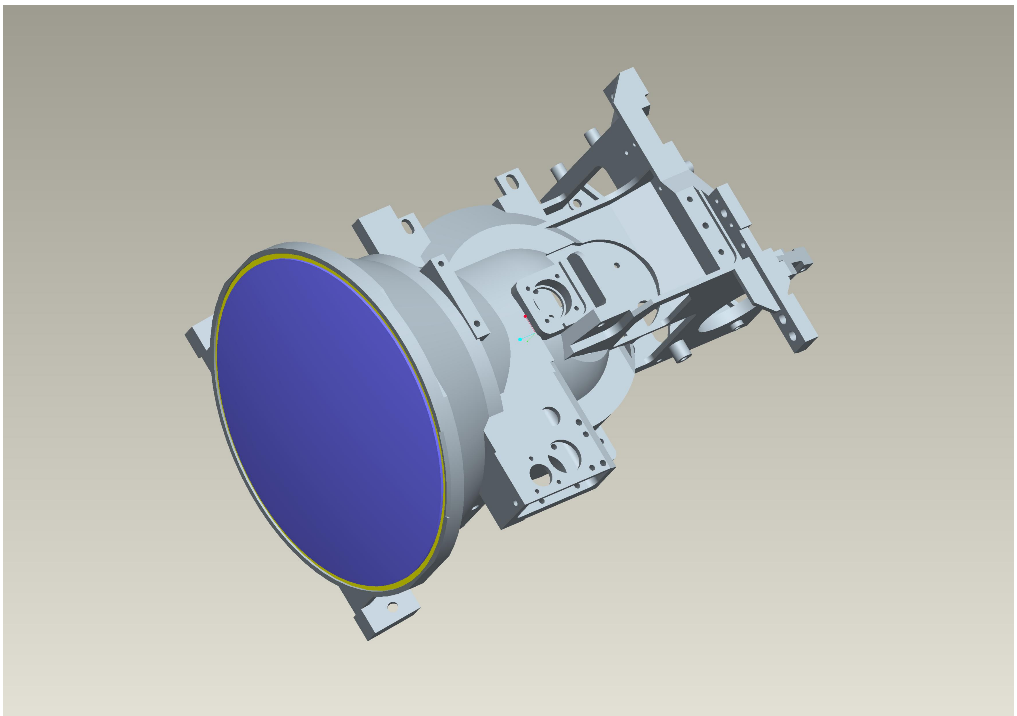


Figure 2. 11 Optical System Assembly

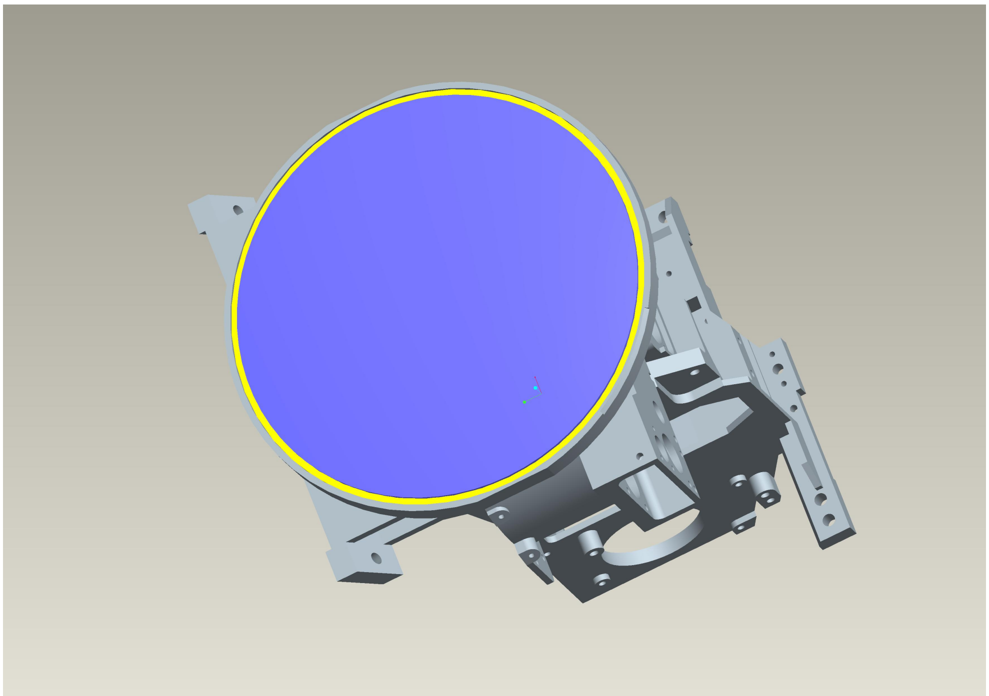


Figure 2. 12 Use of Elastomer in Optical System

By using ANSYS Workbench, material properties of each part can be defined. The afocal is made of aluminum, the lens is made up germanium and elastomer is SYLGARD 577 [13]. Material properties of SYLGARD 577 that are needed for such an dynamic analysis are taken from report, which is prepared in ASELSAN:

Table 2. 9 Material Properties of the Elastomer (SYLGARD 577)

Modulus of Elasticity, E	3.5 MPa
Density, ρ	1300 kg/m3
Poisson ratio, ν	0.4-0.49

Using automatic mesh utilities of ANSYS Workbench, mesh can be constructed by SOLID 186 elements. FE model of the optical system contains 145224 nodes and 59413 elements, as shown in Figure 2.13. In order to observe the effect of the adhesive, adhesive is finely meshed, as shown in Figure 2.14.

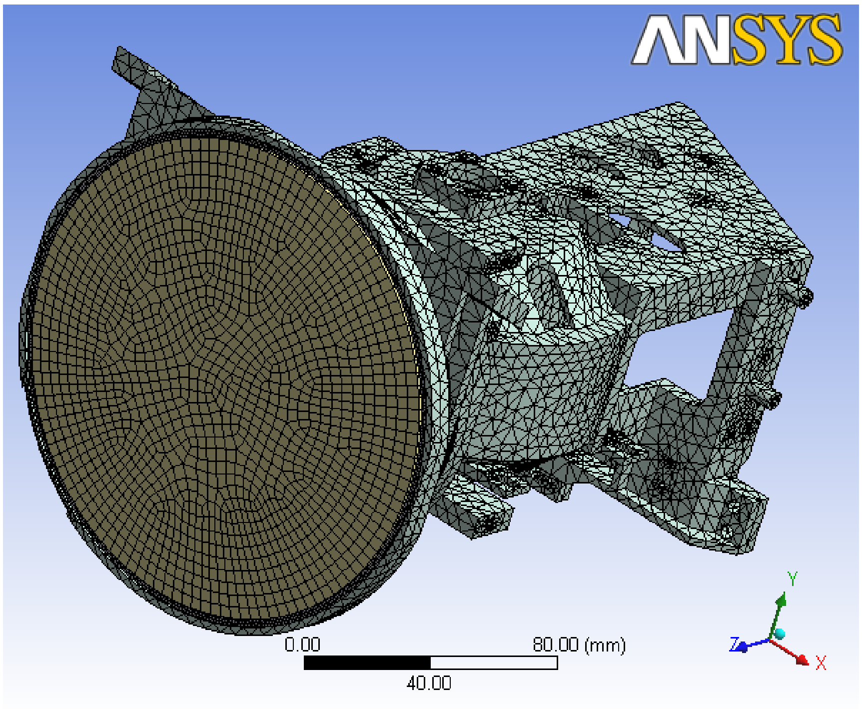


Figure 2. 13 Mesh of Optical System

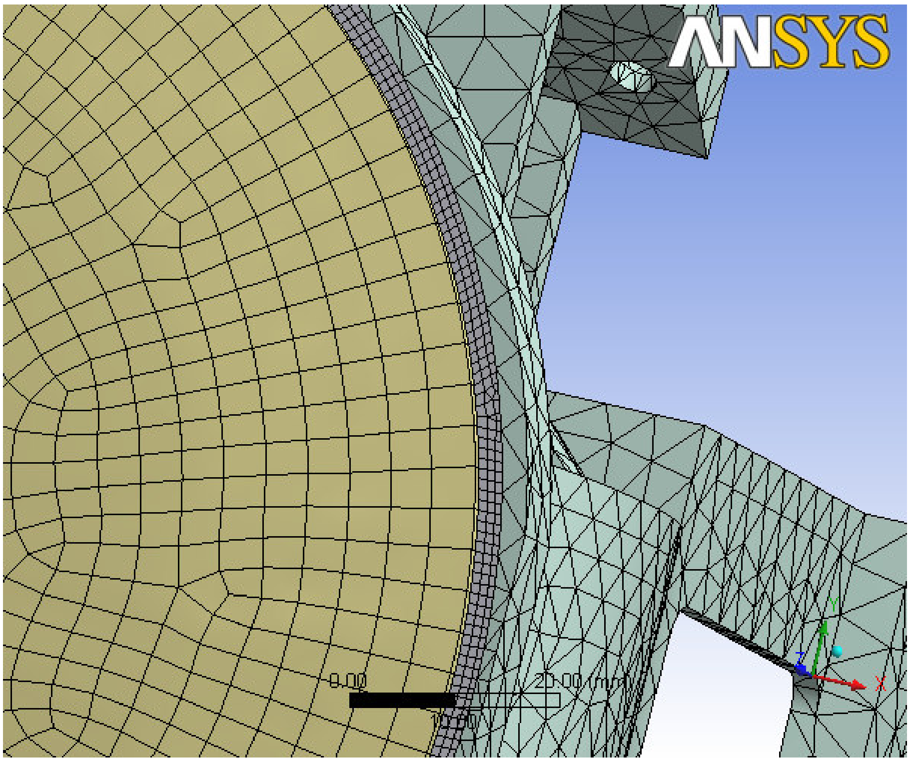


Figure 2. 14 Mesh of Elastomer

Finally, modal analysis is carried out by using ANSYS Workbench for free-free boundary conditions and the first three modes are given in Table 2.10. The mode shapes of the first three modes are given in Figures 2.15, 2.16 and 2.17.

Table 2. 10 Natural Frequencies of the Optical System

ANSYS Results	
Mode	Frequency (Hz)
1	224
2	295
3	295

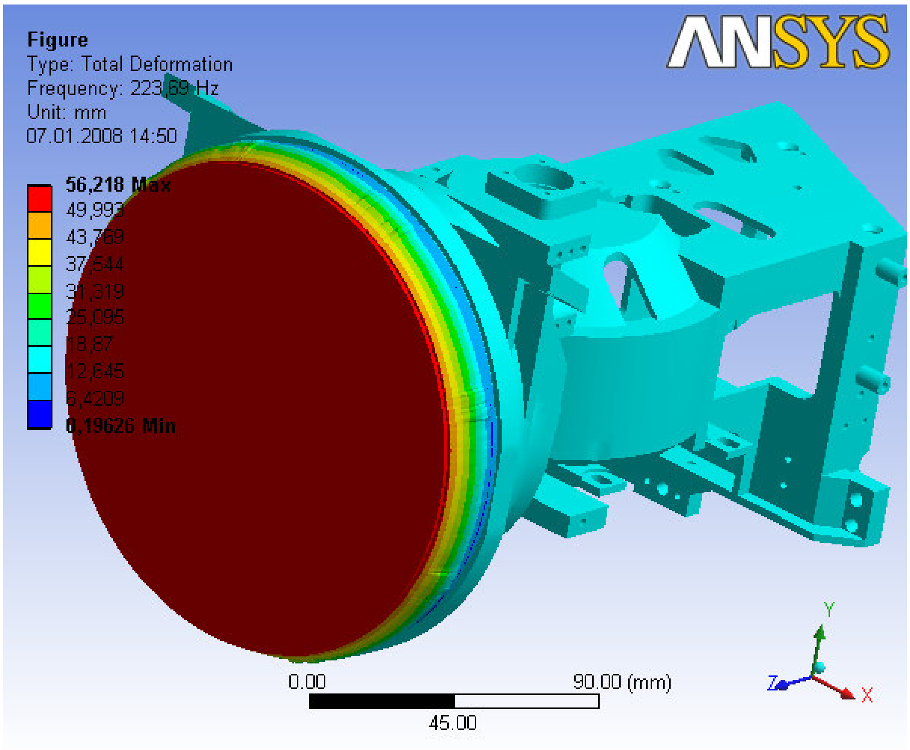


Figure 2. 15 First Mode Shape of Optical System

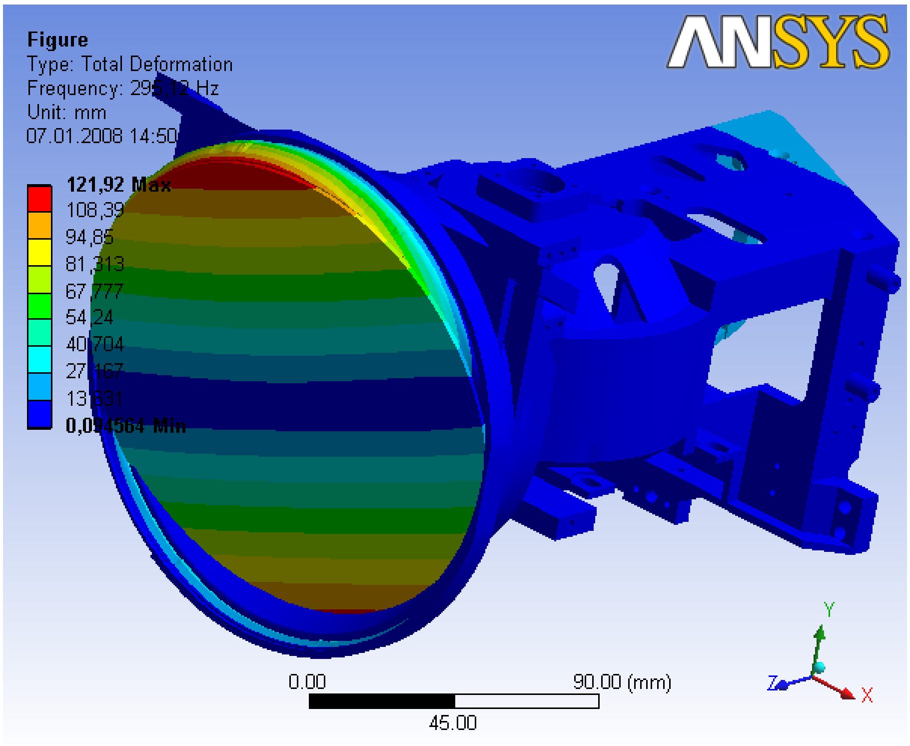


Figure 2. 16 Second Mode Shape of Optical System

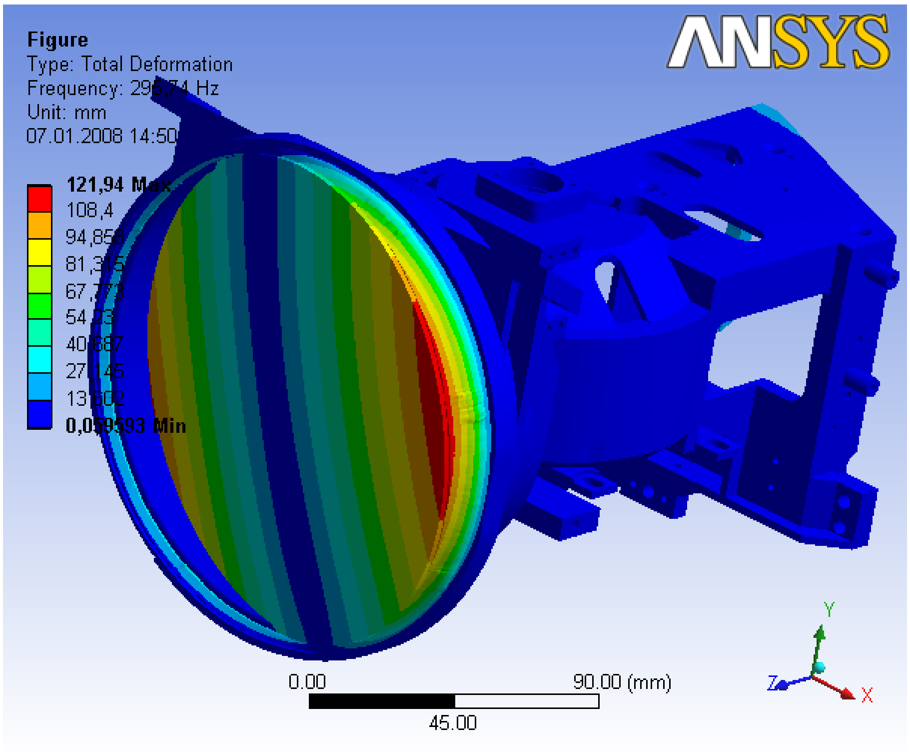


Figure 2. 17 Third Mode Shape of Optical System

As one can see from Figure 2.15, 2.16 and 2.17, all these modes are related to relative motion of lens due to the flexibility of the elastomer. These modes are in the frequency range of interest for AH-1 Helicopter (10-500 Hz), therefore elastomer should be modeled carefully and material properties of it have to be determined accurately. Because of the harsh environmental conditions in military applications, these decenter and tip / tilt modes cause performance loss in optical systems. As explained in previous sections, afocal and lens behave as rigid bodies in 10-500 Hz frequency range; therefore dynamic characteristics of optical system are affected only by the flexibility of elastomer and mass properties of the lens.

CHAPTER 3

ANALYTICAL MODEL OF A SIMPLE OPTICAL SYSTEM

In Chapter 2, finite element model is constructed and dynamic characteristic of each part and assembly are obtained. As seen before, the afocal and the lens behave as rigid body in frequency of range interest, 10-500 Hz, since their first natural frequencies are far above 500 Hz. It is also seen that the dynamic characteristics of the optical system are affected by the material properties of adhesive in 10-500 Hz. Therefore, a simple analytical model can be constructed for optical systems by taking housing and lens as rigid body, such that adhesives are modeled in the simple analytical model. In order to develop an analytical model, considering adhesive only, adhesive geometry and material properties should be known. For finding material properties of adhesive, known test techniques, which can be found in literature, could be used.

Then in order to construct an analytical model; stiffness of adhesive is calculated by using geometry and material properties. After that, lens is taken as mass, because it is assumed as rigid. Then, discrete model can be constructed by mass and stiffness. This discrete model can give decenter and tip and tilt mode of lens. Finally, using obtained model parameters, random vibration output of the optical system, under dynamic loading, can be calculated mathematically.

3.1 Determination of Material Properties of Adhesive

Lenses are generally bonded to optical housing by using adhesives. In order to determine dynamic characteristic of the optical system, all components should be modeled properly, therefore material properties and geometries of components must be known.

Sylgard 577 is one of the most commonly used potting material in the optical systems. It is stated [14] that when it is properly prepared, it could have high strength in the harsh environment factors, such as when exposed to vibration and shock. Moreover, this adhesive can work properly in a temperature range of -49 °C to 200 °C [13], and that temperature range is enough for military applications [9]. For Sylgard 577, some mechanical material properties are defined in reference [13]. These properties are tensile strength, elongation percentage and density, however these are not enough to carry out vibration analysis; therefore modulus of elasticity and damping characteristic of the material as function of frequency should also be known.

In order to find modulus of elasticity and loss factor of an adhesive, there are several methods in literature. One of the methods is *time response method* [15]. Material properties of an adhesive can be found by using Figure 3.1 and the relations given below [15].

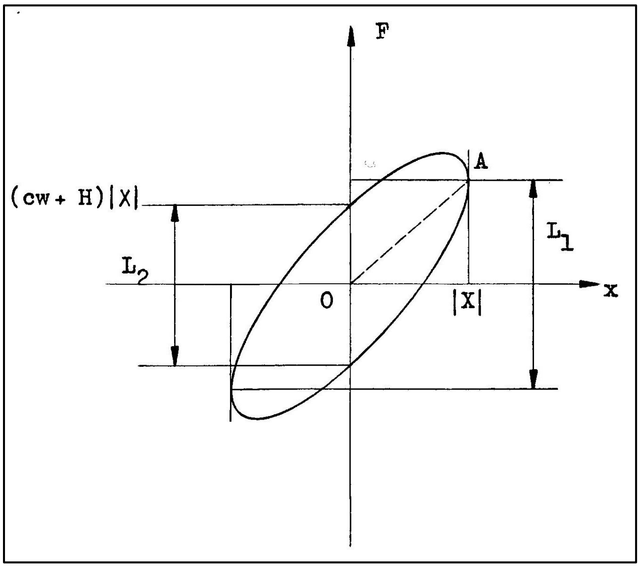


Figure 3. 1 Force, F, versus Displacement, x, Graph [15]

$$k_e = \frac{AO}{|OX|} \tag{3.1}$$

$$E_e = k_e \frac{L_e}{A_e} \tag{3.2}$$

$$\eta_e = \frac{L_2}{L_1} \tag{3.3}$$

where:

- E_e = Modulus of Elasticity of Adhesive
- k_e = Stiffness of Specimen
- L_e = Length of Specimen
- A_e = Area of Specimen
- L_1 = Difference between Force at Maximum Positive Displacement and Force at Maximum Negative Displacement
- L_2 = Difference between Maximum Negative and Positive Force at Zero Displacement
- η_e = Loss Factor

Another method is *single degree of freedom* (s.d.o.f) method [16]. By using this method, the sample can be excited in extension or in shear. For this purposes proper test setup can be constructed to obtain the parameters, modulus of elasticity and loss factor.

3.1.1 Test Setup

For single degree of freedom method [16], the test setup, shown in Figure 3.2 and 3.3, is constructed. Bruel & Kjaer instruments and PULSE software are used for data collection in this test setup (Table 3.1).

Table 3.1 Instrumentation and Software

Instrumentation and Software	
Accelerometer	Bruel & Kjaer 4507 biax
Impact Hummer	Bruel & Kjaer 8200+2646
Analyzer	Pulse Front-End 3560C
Software	Pulse 11.0

Figure 3. 2 Test Setup – View 1

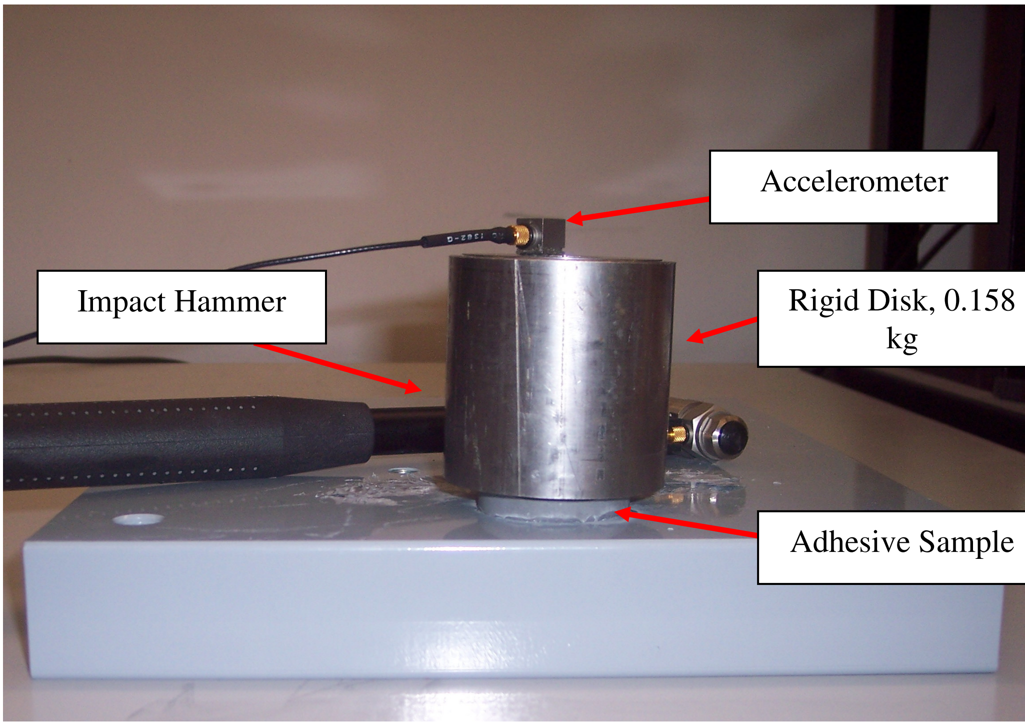


Figure 3. 3 Test Setup –View 2

As seen in Figure 3.2 and 3.3, test setup contains impact hammer, accelerometer and a rigid disk. Geometry of adhesive sample is given in Figure 3.4.

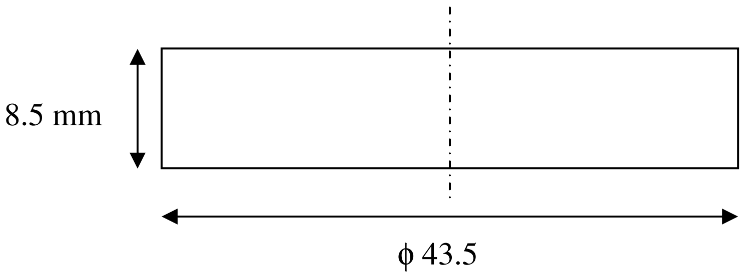


Figure 3. 4 Geometry of Adhesive Sample

During the test, input force was applied to the rigid disk and adhesive sample by using impact hammer. Frequency response of the system can be obtained by using the signals from the hammer and accelerometer. Using the FRF and the analytical model for the schematic view of the test setup, Figure 3.5, complex modulus of adhesive and loss factor can be found [16] as follows:

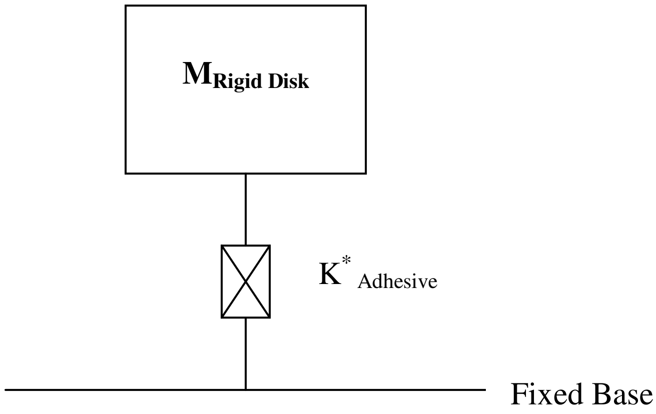


Figure 3. 5 Schematic View of the Test Setup

$$K^*_{elastomer} = K^*_{elastomer}(1 + i\eta_{elastomer}) \tag{3.4}$$

$$E^* = K^*\left(\frac{L}{A}\right) \tag{3.5}$$

$$E^*_{elastomer} = E^*_{elastomer}(1 + i\eta_{elastomer}) \tag{3.6}$$

$$K^* = \frac{\left(1 + M_{rigiddisk}\,\omega H(\omega)\right)}{H(\omega)} \tag{3.7}$$

where $K^*_{adhesive}$ is complex stiffness of the adhesive, η is loss factor of adhesive, $E^*_{adhesive}$ is the complex modulus of elasticity of adhesive, L is length of adhesive, A is area of adhesive, $M_{rigiddisk}$ is mass of rigid disk and H(ω) is FRF of rigid disk and adhesive combination.

First of all, complex stiffness of the system can be obtained from Equation 3.7 [16]. In this equation, H(ω) can be found using by PULSE 11.0. Then, complex stiffness of the system can be converted complex modulus of elasticity of adhesive by using

Equation 3.5 [16]. Finally, the real part of E^* is taken as young modulus of adhesive and the ratio of the complex part of E^* to real part is taken as loss factor of adhesive.

It is stated that [17] Poisson ratios of elastomer type adhesives varies between 0.45-0.5 and therefore, the responses of these materials are affected by geometry, and the load versus stress relations changes by aspect ratio, D/t. Because of this reason, Poisson ratio plays important role in stress relations for high D/t ratios. If Equation 3.8 [17] is analyzed with given Figure 3.6, it is seen that principles stresses are calculated by considering the Poisson ratio.

$$\begin{Bmatrix} \sigma_{11} \\ \sigma_{22} \\ \sigma_{33} \\ \tau_{12} \\ \tau_{23} \\ \tau_{31} \end{Bmatrix} = \begin{bmatrix} \frac{(1-\nu)E}{(1+\nu)(1-2\nu)} & \frac{\nu E}{(1+\nu)(1-2\nu)} & \frac{\nu E}{(1+\nu)(1-2\nu)} & 0 & 0 & 0 \\ \frac{\nu E}{(1+\nu)(1-2\nu)} & \frac{(1-\nu)E}{(1+\nu)(1-2\nu)} & \frac{\nu E}{(1+\nu)(1-2\nu)} & 0 & 0 & 0 \\ \frac{\nu E}{(1+\nu)(1-2\nu)} & \frac{\nu E}{(1+\nu)(1-2\nu)} & \frac{(1-\nu)E}{(1+\nu)(1-2\nu)} & 0 & 0 & 0 \\ 0 & 0 & 0 & G & 0 & 0 \\ 0 & 0 & 0 & 0 & G & 0 \\ 0 & 0 & 0 & 0 & 0 & G \end{bmatrix} \begin{Bmatrix} \varepsilon_{11} \\ \varepsilon_{22} \\ \varepsilon_{33} \\ \gamma_{12} \\ \gamma_{23} \\ \gamma_{31} \end{Bmatrix} \tag{3.8}$$

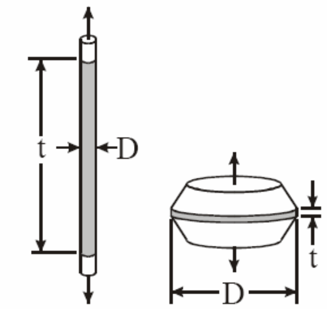


Figure 3. 6 Two Extreme Tension Test Samples [17]

In Equation 3.8, when $\sigma_{33}/\varepsilon_{33}$ is taken into account, stress is determined not only by modulus of elasticity but also by Poisson ratio, Equation 3.9.

$$\frac{\sigma}{\varepsilon} = \frac{(1-\nu)E}{(1+\nu)(1-2\nu)} = mE = M \tag{3.9}$$

M is the maximum modulus [17] and it can be determined by given Figure 3.7 and Equation 3.9. In Figure 3.7, Nu is used for Poisson ratio. It is seem from Figure 3.7 that the modulus of elasticity is increased with increasing D/t. In our system, D/t ratio of test specimen is 5, therefore, the calculated modulus of elasticity should be divided by 2, which is found from Figure 3.7.

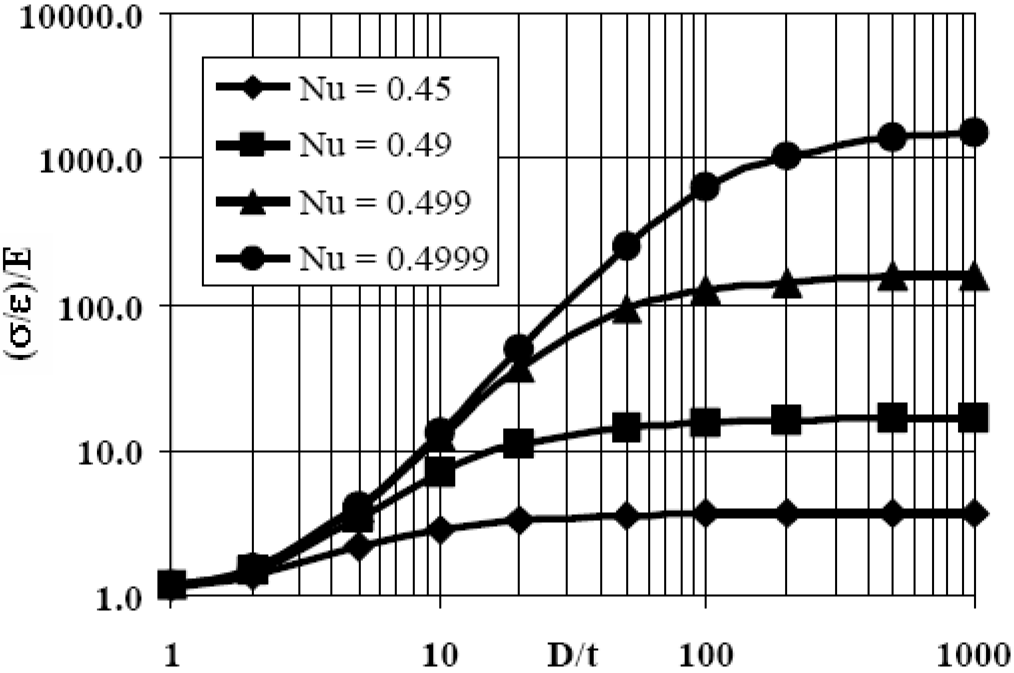


Figure 3. 7 Variation of Modulus of Elasticity with respect to D/t Ratio [17]

3.1.2 Test Results

The system is excited by hammer over the frequency range 0-1600 Hz. Using PULSE 11.0 software, $H(\omega)$ and coherence between accelerometer data and force data can be obtained. They are given in Figure 3.8 and 3.9, respectively.

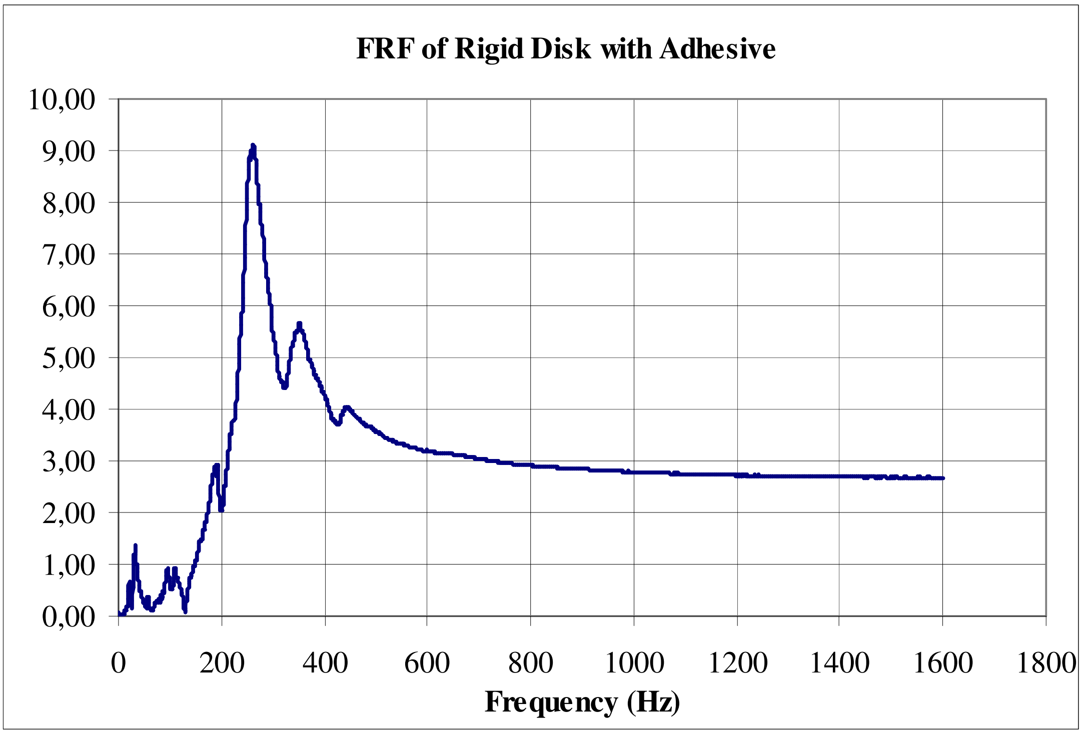


Figure 3. 8 FRF of Rigid Disk with Adhesive

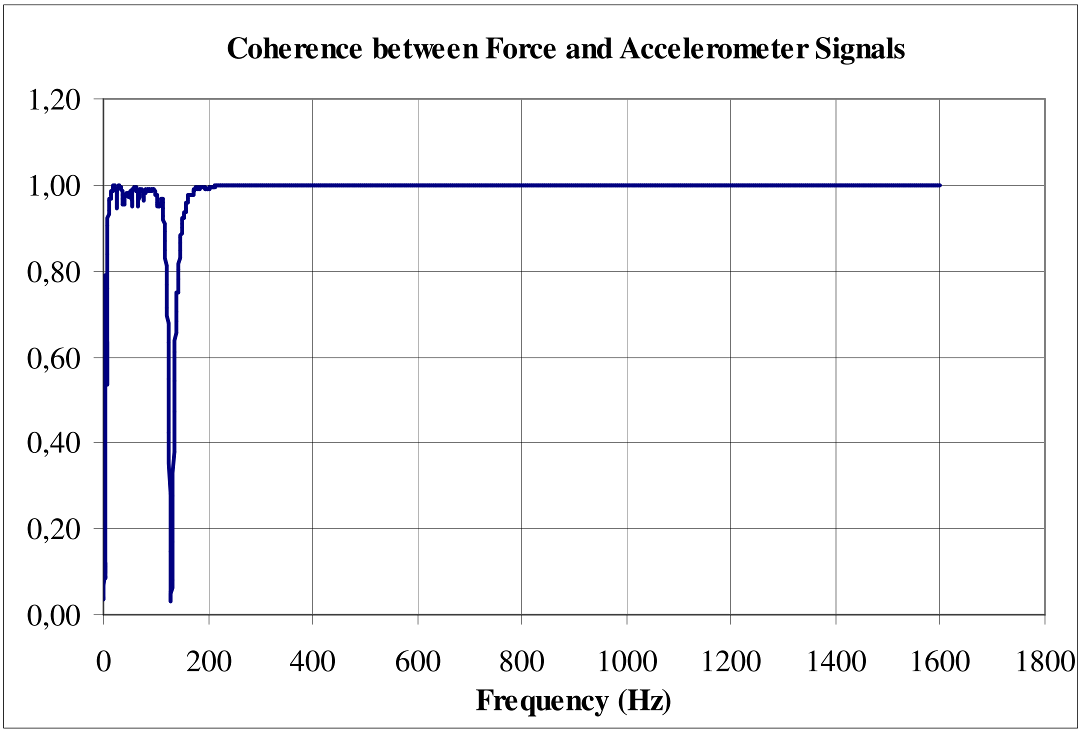


Figure 3. 9 Coherence between Force and Accelerometer Signals

As can be seen in the FRF of the system, there are 6 natural frequencies between 0-600. As can be seen in Figure 3.9, coherence is very close to 1 [18], and it can be said that measurement quality is good enough.

Complex modulus of elasticity of adhesive can be found by using Equation 3.4 to 3.7. Then the real part of complex modulus of elasticity is plotted as modulus of elasticity of adhesive and also ratio of imaginary part to real part of complex modulus of elasticity is plotted as loss factor of adhesive. As explained in section 3.2.1, modulus of elasticity found by test should be divided into 2, since the aspect ratio of the test specimen is approximately 5 (Figure 3.7 shows that when D/t is 5 and Poisson ratio is 0.45, m becomes 2). Modulus of elasticity and loss factor graphs are given in Figure 3.10 and 3.11.

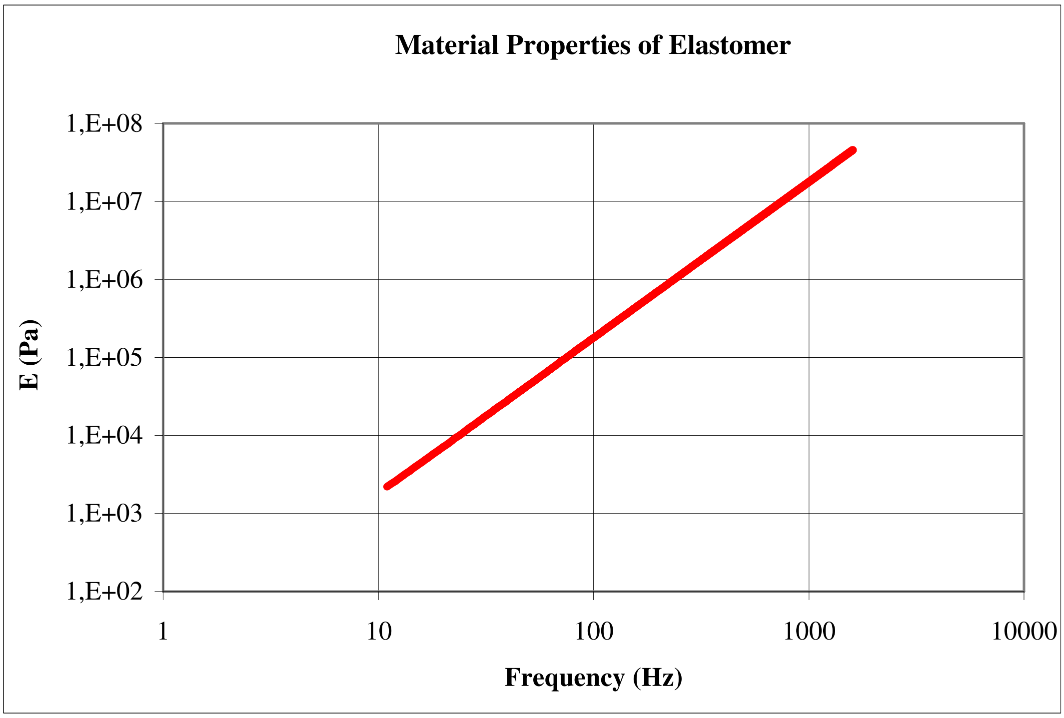


Figure 3. 10 Modulus Elasticity of Adhesive

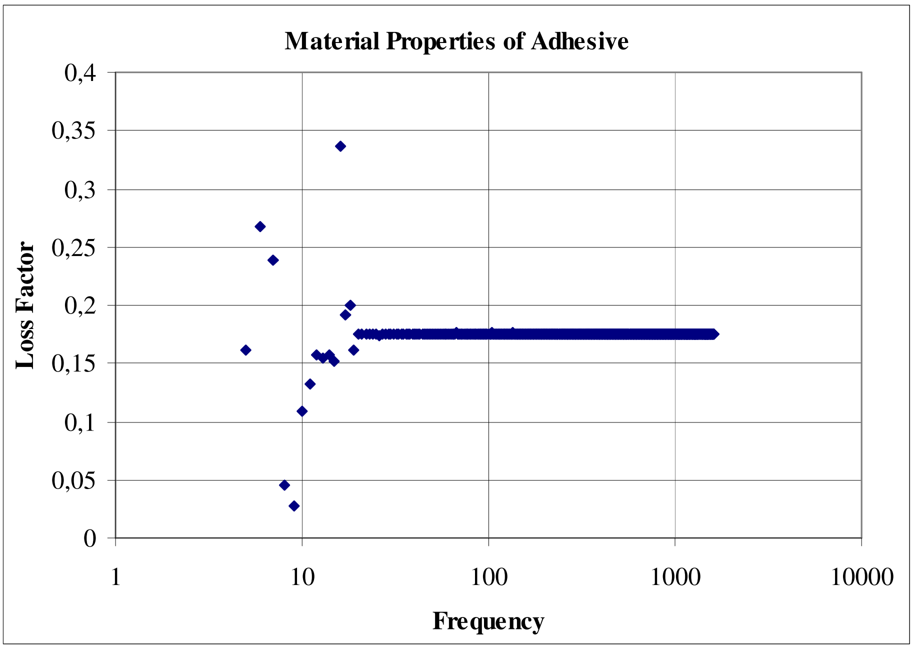


Figure 3. 11 Loss Factor of Adhesive

Using curve fitting techniques, trendline for modulus of elasticity of adhesive could be found in log-log scale. Trendline and relation between the modulus of elasticity and frequency can be seen in Figure 3.12.

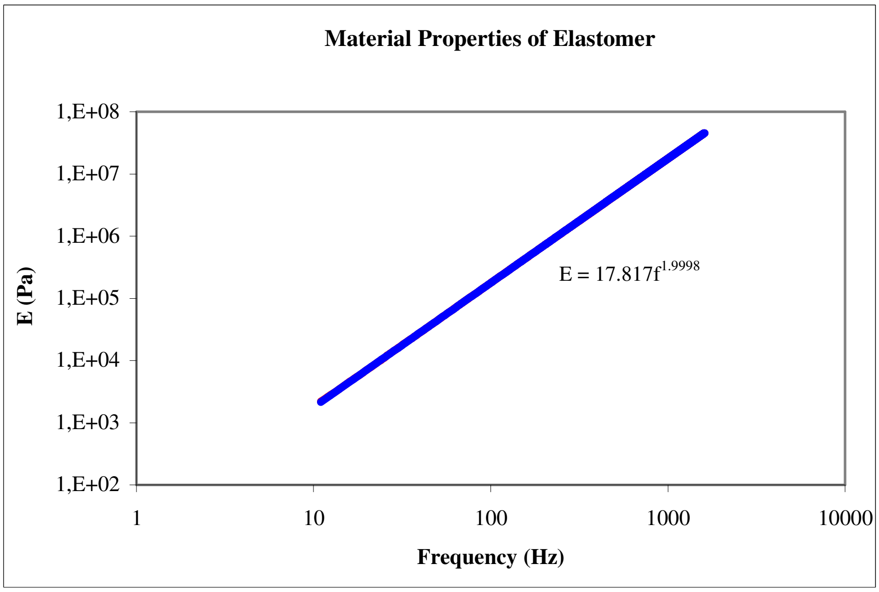


Figure 3. 12 Trendline (Blue Line) of Young Modulus of Adhesive

The test results show that, modulus elasticity of the adhesive is linear in log-log scale. Therefore, the equation obtained for modulus of elasticity can be used in analytical model of dynamic characteristics of optical system. However, in finite model of the optical system, average value of modulus of elasticity should be used in interested frequency range, since modal analysis is done only for constant material properties in finite element program.

3.2 Construction of Analytical Model

After obtaining the material properties of the adhesive, analytical model of a simple optical system can be created. As explained before, in this analytical model, housing and lens are taken as rigid and adhesive is modeled as a flexible element with damping.

Opto-mechanical designers can use analytical model in preliminary design work for predicting the dynamic behavior of optical system. Therefore the model should have enough representation of the real optical model. Moreover, if detailed analysis is required, finite element model can be constructed for finding dynamic characteristic of optical system. Geometrical model of the optical system can be seen in Figure 3.13. In this model, case and lens are modeled as rigid bodies and only adhesive is modeled as a flexible element.

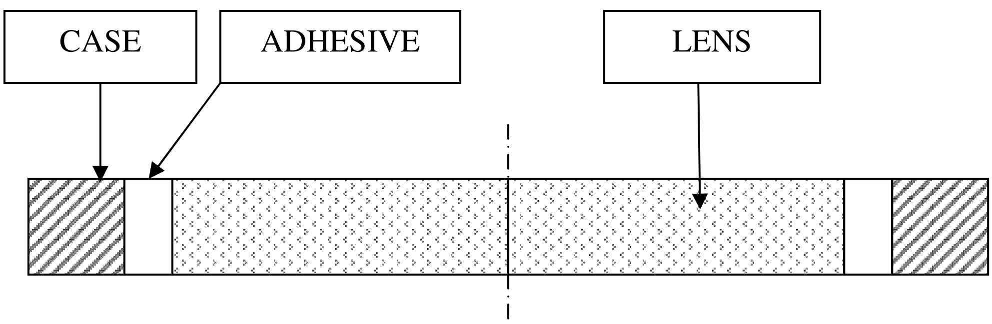


Figure 3. 13 Geometrical Model of Optical System

3.2.1 Stiffness of the Adhesive

As you can see from Figure 3.14, when force F is applied on lens, adhesive deflects in Z direction. In this movement, shear effects are important, therefore for creating a mathematical model, only shear forces and deflection are considered. Material properties of adhesive are obtained in section 3.1. Shear modulus of adhesive, G, [19] can be found as:

$$G = \frac{E}{2(1+\nu)} \qquad (3.10)$$

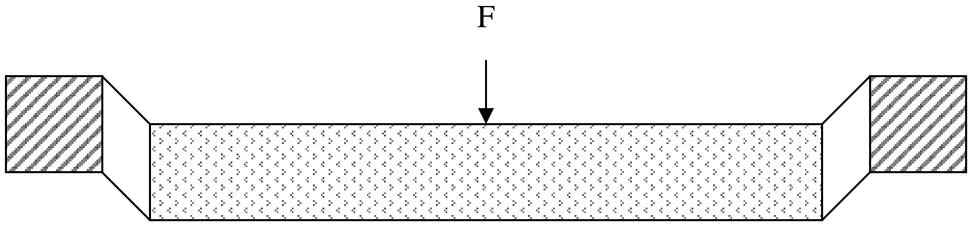


Figure 3. 14 Deformed Shape of Adhesive

In order to find stiffness, first of all shear force should be calculated. For obtaining shear force, shear area can be calculated by using outer diameter or inner diameter of adhesive, however, in order get approximate results, effective shear area can be calculated by d_{mean}, which is the average of outer and inner diameter of adhesive. Shear area can be found as:

$$A_s = \pi d_{mean} h \qquad (3.11)$$

$$d_{mean} = \frac{d_i + d_o}{2} \qquad (3.12)$$

where h is the height of adhesive, d_i is the inner diameter of adhesive, d_o is the outer diameter of adhesive, d_{mean} is the mean diameter of adhesive and A_s is the effective shear area.

After finding shear area, effective shear stress can be calculated as follows [19]:

$$\tau = \frac{F_s}{A_s} \tag{3.13}$$

$$\tau = G\gamma \tag{3.14}$$

where F_s is the shear force, A_s is the shear area, G is the shear modulus and γ is the shear deflection.

Then the shear force can be written in terms of shear area, shear modulus and shear deflection as:

$$F_s = GA_s\gamma \tag{3.15}$$

Shear deflection can be eliminated from Equation 3.15 by writing shear deflection as function of x, deflection, and t, thickness of adhesive. If force is applied on lens, adhesive is deformed as shown in Figure 3.15.

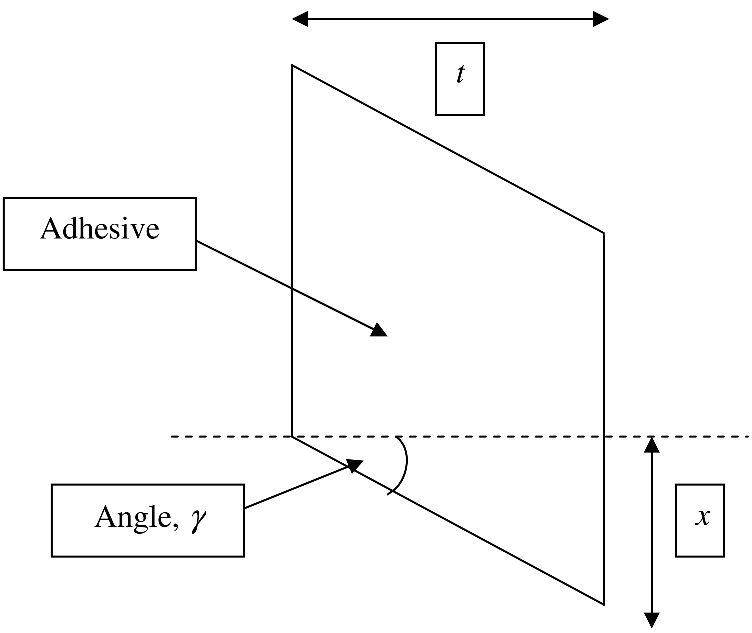


Figure 3. 15 Deflection in Adhesive

Using Figure 3.15, shear defection can be obtained as:

$$\tan(\gamma) = \frac{x}{t} \tag{3.16}$$

For a small angle, tangent of γ can be approximated γ as:

$$\tan(\gamma) = \gamma \tag{3.17}$$

$$x = t\gamma \tag{3.18}$$

Using Equation 3.15 and 3.18, shear force can be written as:

$$F_s = \frac{A_s G x}{t} \tag{3.19}$$

When force is applied on the spring, force-deflection relation will be:

$$F = kx \tag{3.20}$$

Then, shear stiffness of the adhesive is obtained by using Equations 3.19 and 3.20 as:

$$k = \frac{A_s G}{t} \tag{3.21}$$

3.2.2 Natural Frequency of Optical System

Natural frequencies of a simple optical system can be found by using stiffness of adhesive, which is given by Equation 3.21, and mass of lens. First of all, discrete model of optical system should be constructed. In this discrete model, stiffness of the adhesive can be distributed over the lens in order to obtain decenter and tip and tilt modes, Figure 3.16.

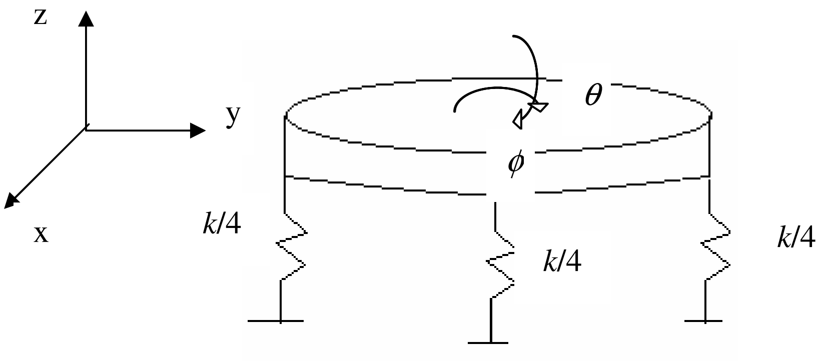


Figure 3. 16 Discrete Model of a Simple Optical System

In order to simulate decenter and tip and tilt modes, three coordinates, z, θ and ϕ, can be used. After that, Equation of motion in these three directions can be written as follows:

$$m\ddot{z} + kz = 0 \tag{3.22}$$

$$I_{\theta}\ddot{\theta} + \frac{R^2}{2}\theta = 0 \tag{3.23}$$

$$I_{\phi}\ddot{\phi} + \frac{R^2}{2}\phi = 0 \tag{3.24}$$

where;

m = mass of the lens

k = stiffness of the lens

I_θ = inertia of lens in θ coordinates around x axis

I_ϕ = inertia of lens in ϕ coordinates around y axis

R = radius of lens

Equations 3.22, 3.23 and 3.24 can be written in matrix form as;

$$\begin{bmatrix} m & 0 & 0 \\ 0 & I_{\theta} & 0 \\ 0 & 0 & I_{\phi} \end{bmatrix} \begin{matrix} \ddot{z} \\ \ddot{\theta} \\ \ddot{\phi} \end{matrix} + \begin{bmatrix} k & 0 & 0 \\ 0 & \frac{R^2}{2}k & 0 \\ 0 & 0 & \frac{R^2}{2}k \end{bmatrix} \begin{matrix} z \\ \theta \\ \phi \end{matrix} = 0 \tag{3.25}$$

Mass and stiffness matrices of the optical system are:

$$M = \begin{bmatrix} m & 0 & 0 \\ 0 & I_\theta & 0 \\ 0 & 0 & I_\phi \end{bmatrix} \tag{3.26}$$

$$K = \begin{bmatrix} m & 0 & 0 \\ 0 & I_\theta & 0 \\ 0 & 0 & I_\phi \end{bmatrix} \tag{3.27}$$

If Eigenvalue problem can be solved for Equation 3.25, natural frequencies and mode shapes can be obtained. In order to solve Eigenvalue problem, MATHCAD can be used and natural frequencies of simple optical system can be calculated as follows:

$$E = KM^{-1} \tag{3.28}$$

$$c = sort\left(eigenvals\left(E\right)\right) \tag{3.29}$$

$$u = eigenvec\left(E\right) \tag{3.30}$$

$$f_1 = \frac{\sqrt{c_{0,0}}}{2\pi} \tag{3.31}$$

$$f_2 = \frac{\sqrt{c_{1,0}}}{2\pi} \tag{3.32}$$

$$f_3 = \frac{\sqrt{c_{2,0}}}{2\pi} \tag{3.33}$$

3.2.3 Random Vibration Analysis of Optical System

Random vibration analysis is used mostly for military application. As known, military systems are integrated to platforms, such as helicopters, aircrafts or tanks. After integration, military systems are exposed to environmental conditions and one of the most important environmental conditions is vibration. In order to find response of the system under vibration environment, random vibration analysis should be conducted.

When a single spring-mass system, Figure 3.17, is excited by random vibration, the system responds by vibrating mainly at its resonant frequency.

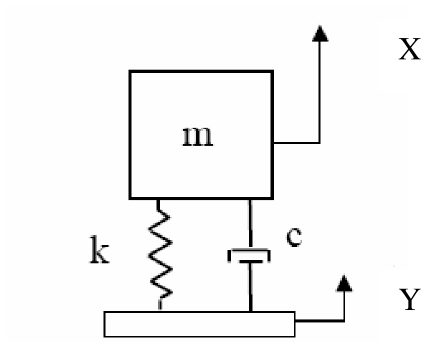


Figure 3. 17 Single spring-mass system

It is known [20] that if a random vibration input is given to the system, the mean square acceleration response of the mass can be calculated by using the area under the power spectral density (PSD) of the response vs. frequency. This area can be found by taking the integral of PSD (P_{out}) from f_1 to f_2 as follows:

$$area = G_{out}^{\;2} = \int_{f_1}^{f_2} P_{out} df \qquad (3.34)$$

The response (or output) PSD, P_{out}, is required to obtain the response of the mass to the random vibration input. The output PSD, obtained from the input PSD, P, can be calculated by using the relation:

$$P_{out} = Tr^2 P \qquad (3.35)$$

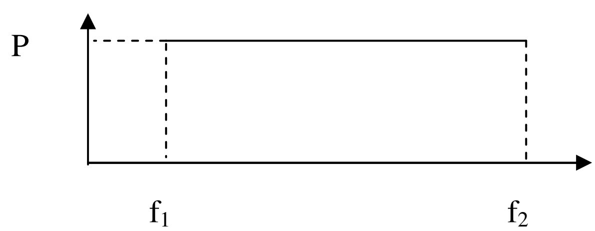


Figure 3. 18 Power Spectral Density – PSD - Graph

where Tr is transmissibility of the single degree of freedom system and it can be written as:

$$Tr(f)=\sqrt{\frac{1+(2\frac{f}{f_n}\zeta)^2}{(1-(\frac{f}{f_n})^2)^2+(2\frac{f}{f_n}\zeta)^2}} \tag{3.36}$$

where: f_n is natural frequency of the system and ζ is damping ratio.

Then, by using Equation 3.34 to 3.36, G_{out} can be obtained as:

$$G_{out}(f)=\sqrt{\int_0^\infty \frac{\left(1+(2\frac{f}{f_n}\zeta)^2\right)P}{\left((1-(\frac{f}{f_n})^2)^2+(2\frac{f}{f_n}\zeta)^2\right)}df} \tag{3.37}$$

The above equation will be the mean square acceleration response of the mass to the random vibration input.

As explained in previous part, first three modes of the optical system are decenter, tip and tilt. Tip and tilt modes are two multi modes and also uncoupled modes. By making single degree of freedom approximation for the optical system, transmissibility equation for each mode can be written as:

$$Tr_1=\alpha(f)_{1,1}\left(K_{1,1}+ifC_{1,1}\right) \tag{3.38}$$

$$Tr_2=\alpha(f)_{2,2}\left(K_{2,2}+ifC_{2,2}\right) \tag{3.39}$$

$$Tr_3=\alpha(f)_{3,3}\left(K_{3,3}+ifC_{3,3}\right) \tag{3.40}$$

Tr_n is transmissibility for n-th mode (n = 1, 2 and 3), and $\alpha(f)_{n,n}$ is the n-th diagonal element of the frequency response function given by:

$$\alpha(f)=\left(K-f^2M-ifC\right)^{-1} \tag{3.41}$$

$K_{n,n}$ is the n-th diagonal element of the stiffness matrix, which is given by Equation 3.27. $C_{n,n}$, can be calculated as a function of damping ratio of the adhesive, is also the n-th diagonal element of the damping matrix given by:

$$C = \begin{bmatrix} C_{1,1} & 0 & 0 \\ 0 & C_{2,2} & 0 \\ 0 & 0 & C_{3,3} \end{bmatrix} \tag{3.42}$$

where;

$$C_{1,1} = 2\zeta\sqrt{km} \tag{3.43}$$

$$C_{2,2} = 2\zeta\sqrt{\frac{R^2}{2}kI_\theta} \tag{3.44}$$

$$C_{3,3} = 2\zeta\sqrt{\frac{R^2}{2}kI_\phi} \tag{3.45}$$

ζ = modal damping ratio of the adhesive

3.3 Example for Analytical Model of a Simple Optical System

In order to test the analytical model, a simple optical system can be constructed. This system contains case, lens and adhesive. CAD model of this optical system can be seen in Figure 3.19.

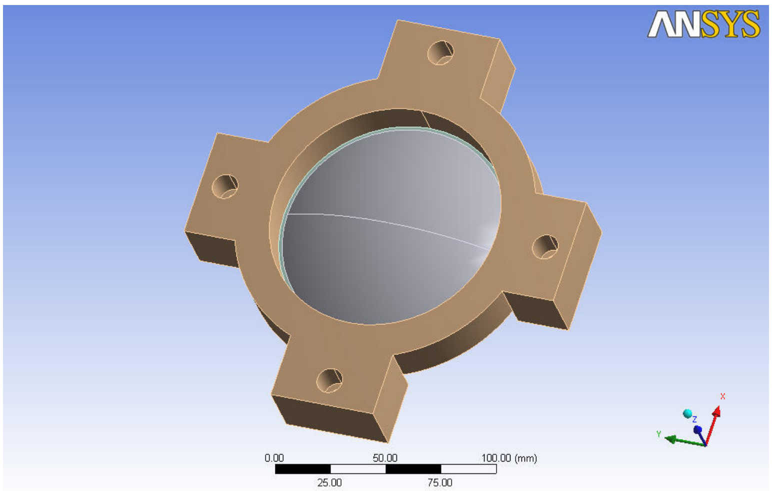


Figure 3. 19 A Simple Optical System

Material properties of adhesive are given in part 3.1, these values which is obtained from test, can be directly used. As it is known, modulus of elasticity of adhesive depends on frequency and as explained part 3.1.1, diameter to height ratio of ring bond is very high; therefore, Poisson ratio effect should be considered. As can be seen in Table 3.2, outer diameter to height ratio is 54 and Poisson ratio is 0.45; therefore, by using Figure 3.7, m can be found as 4. Because of this reason, modulus of elasticity, which is obtained from material test of elastomer, should be multiplied by 4.

$$E = 71.268 f^{1.9998} \tag{3.46}$$

Then the shear modulus of elasticity can be written in terms of modulus of elasticity by using Equation 3.10:

$$G = \frac{71.268 f^{1.9998}}{2(1+\nu)} \tag{3.47}$$

Finally, stiffness of adhesive can be found from Equation 3.21 and 3.47 as follows:

$$k = \frac{A_s}{t}\left(\frac{71.268 f^{1.9998}}{2(1+\nu)}\right) \tag{3.48}$$

It can be seen in Equation 3.48 that the stiffness k depends on frequency; therefore it is difficult to solve the Eigenvalue problem for unknown frequency. Because of this reason, in order to find decenter mode, simple mass and spring model is used for predicting the first natural frequency of the optical system:

$$f_{decenter} = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{3.49}$$

Using stiffness of adhesive, $f_{decenter}$ can be solved from:

$$f_{decenter} = \frac{1}{2\pi}\sqrt{\frac{71.268 A_s f_{decenter}^{\ \ 1.9998}}{2t(1+\nu)m}} \tag{3.50}$$

Geometrical properties of a simple optical system are given in Table 3.2.

Table 3.2 Geometrical Properties of a Simple Optical System

Geometrical Properties of a Simple Optical System	
Height of the adhesive, h	0.002 m
Inner diameter of the adhesive, d_i	0.104636 m
Outer diameter of the adhesive, d_o	0.108 m
Mean diameter of the adhesive, d_m	0.10632 m
Thickness of the adhesive, t	0.00168 m
Shear area of the adhesive, A_s	6.68e-4 m^4
Poisson ratio of adhesive, υ	0.45
Mass of the lens, m	0.225 kg
Inertia of the lens in θ coordinates, I_θ	1.3554e-4 m^4
Inertia of lens in ϕ coordinates, I_ϕ	1.3554e-4 m^4

First natural frequency of a simple optical system can be predicted as 183 Hz by using given properties. Then, using this frequency, modulus of elasticity of adhesive can be found as 2.3634 MPa. Finally, stiffness of the adhesive can be calculated by using the Equation 3.48, and then natural frequencies of a simple optical system can be found by using this stiffness value. The calculated natural frequencies of optical system are given in Table 3.3.

Table 3.3 Natural Frequencies of a Simple Optical System

Analytical Results of a Simple Optical System	
Natural Frequency, f_1	191 Hz
Natural Frequency, f_2	292 Hz
Natural Frequency, f_3	292 Hz

After finding natural frequencies, random vibration analysis can be performed for a simple optical system. As these three natural frequencies are uncoupled, for each mode, single degree of freedom model can be used for random vibration analysis by using Equation 3.37.

In part 3.1, material properties of adhesive are obtained. Loss factor η is found as 0.175, this is structural damping and it can be calculated by using frequency response function of material. In frequency response function, structural damping can be obtained by using half power points ratio [21]. Also it is known that, structural damping is twice of the modal damping ratio. In random vibration analysis viscous modal damping ratio is used.

$$\eta = \frac{\omega_2 - \omega_1}{\omega_n} \tag{3.51}$$

$$\zeta = \frac{\omega_2 - \omega_1}{2\omega_n} \tag{3.52}$$

Using Equation 3.51 and 3.52, modal damping ratio can be found as 0.0875. During the random vibration analysis, dynamic input, which is the PSD input of AH-1 Helicopter, is given in each direction, z, θ and ϕ, therefore; random vibration output of the optical system, G_{out} can be found by Equation 3.37 for each mode. Results are given in Table 3.4.

Table 3.4 Random Vibration Output of a Simple Optical System

Random Vibration Analysis Results of a Simple Optical System	
G_{out}^1 for Natural Frequency, f_1	41 m/s^2
G_{out}^2 for Natural Frequency, f_2	51 m/s^2
G_{out}^3 for Natural Frequency, f_3	51 m/s^2

CHAPTER 4

VALIDATION OF ANALYTICAL MODEL WITH FINITE ELEMENT MODEL AND EXPERIMENTAL TECHNIQUES

In order to test the accuracy of analytical model, finite element model and experimental techniques can be used. First of all, analytical results can be obtained for known material properties and geometries of each optical system components. After that, finite element model can be constructed in ANSYS Workbench by using the same model and output of the dynamic analysis is compared with analytical model results. Moreover, prototype of this optical model is produced and hammer test is carried out for finding modal parameters, natural frequency, damping of each modes and mode shapes.

As explained in Chapter 2, case and lens are rigid in interested frequency range, 10-500 Hz, and also it is seen that elastomer is a very soft material and therefore the first three modes of the optical system are due to the flexibility of elastomer.

The analytical model constructed can be used in preliminary identification of dynamic characteristic in opto-mechanical system design. If detailed analysis is required, finite element techniques should be used. In this chapter the analytical model is verified by using finite element analysis and also making an experimental study. It is known that material properties of elastomer is found by experimental methods and these properties will also be confirmed if analytical model, finite element model and experimental method give close results.

4.1 Finite Element Model of a Simple Optical System

CAD model of a simple optical system is constructed in Pro/ENGINEER and as explained before it is directly imported to ANSYS Workbench by means of direct connection between ANSYS Workbench and Pro/ENGINEER. Geometrical properties of a simple optical system are given in Table 4.1 and CAD model of this simple optical system is shown in Figure 4.1.

Table 4. 1 Geometrical of a Simple Optical System

Geometrical Properties of a Simple Optical System	
Height of the Elastomer, SYLGARD 577	2 mm
Inner diameter of the Elastomer, SYLGARD 577	104.636 mm
Outer diameter of the Elastomer, SYLGARD 577	108 mm
Diameter of Lens	104.636 mm
Inner Diameter of Case Diameter of Lens	108 mm

The case is made of aluminum, the lens is made of germanium and the material properties of elastomer are given section 3.1. Moduli of elasticity, densities and Poisson ratios are given in Table 4.2 for each part of the optical system.

Table 4. 2 Material Properties of Each Part of a Simple Optical System

	Modulus of Elasticity	**Density**	**Poisson Ratio**
Case, Aluminum	71000 MPa	2770 kg/m^3	0.33
Lens, Germanium	130000 MPa	5323 kg/m^3	0.3
SYLGARD 577, Elastomer	2.3634 MPa	1300 kg/m^3	0.45

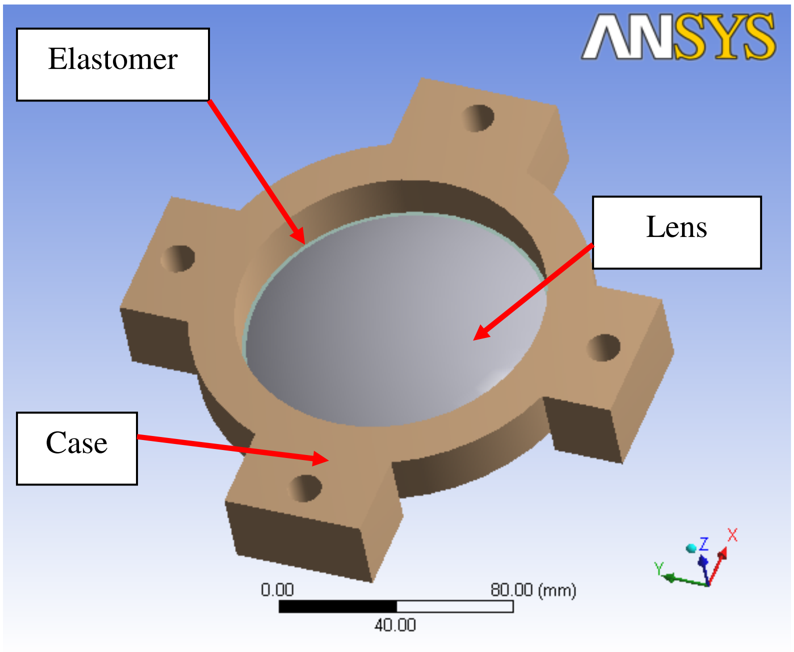


Figure 4. 1 CAD Model of a Simple Optical System

After importing a simple optical system to ANSYS Workbench, given material properties are defined and finite element model is obtained. As in Chapter 2, SOLID 186 [10] elements can be used and the finite element model of this optical system has 77723 nodes and 16582 elements, Figure 4.2 and 4.3. Finally, modal analysis and random vibration analysis are conducted in ANSYS Workbench.

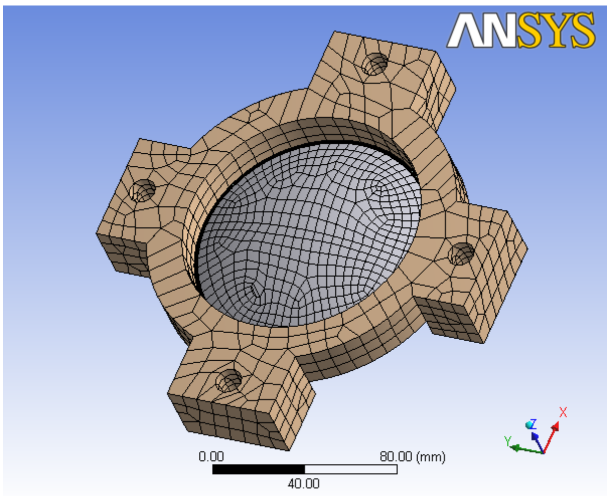


Figure 4. 2 Mesh of a Simple Optical System

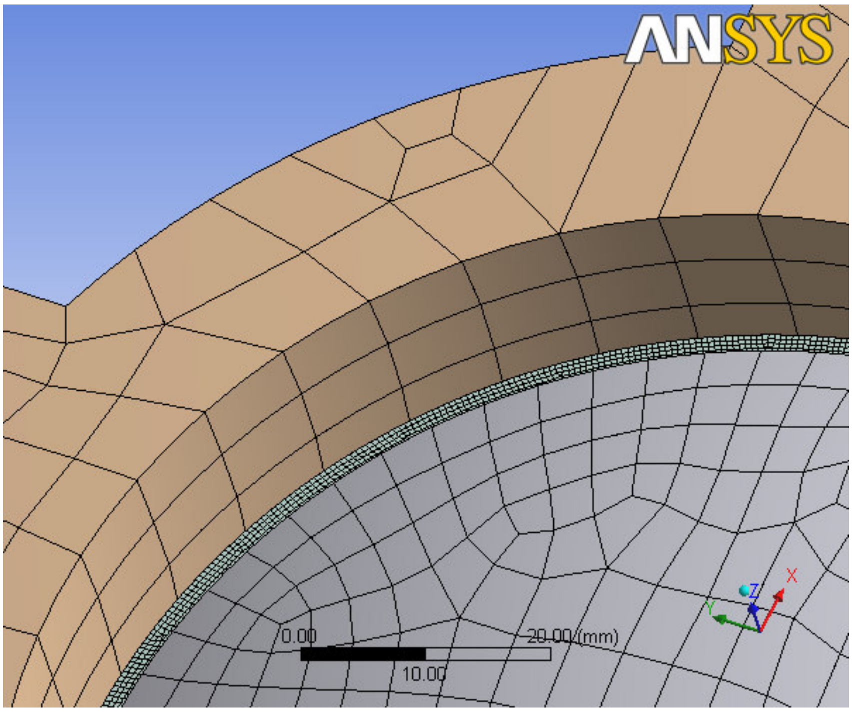


Figure 4. 3 Mesh of Elastomer

In modal analysis, first three modes are found by giving fixed boundary conditions to holes, Figure 4.1. Results of the modal analysis are given in Table 4.3.

Table 4. 3 Results of Modal Analysis

Mode No	**Natural Frequency, f_n (Hz)**
1	169
2	243
3	243

It can be seen in Table 4.3 that the first mode is at 169 Hz. This is the decenter mode, other two modes are multiple modes, tip and tilt modes, and their value is 243 Hz. Mode shapes are given in Figure 4.4, 4.5 and 4.6.

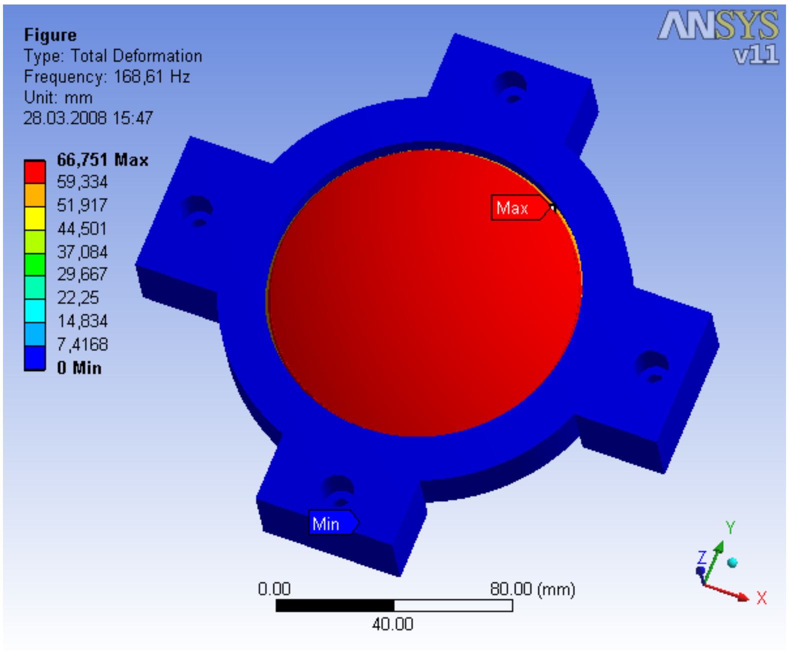


Figure 4. 4 First Mode Shape (Decenter), 169 Hz

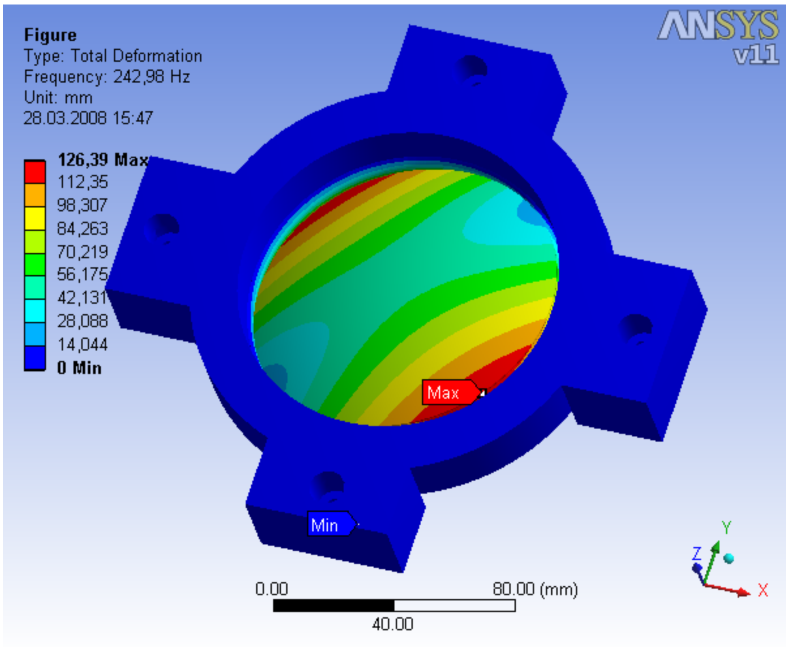


Figure 4. 5 Second Mode Shape (Tip and Tilt), 243 Hz

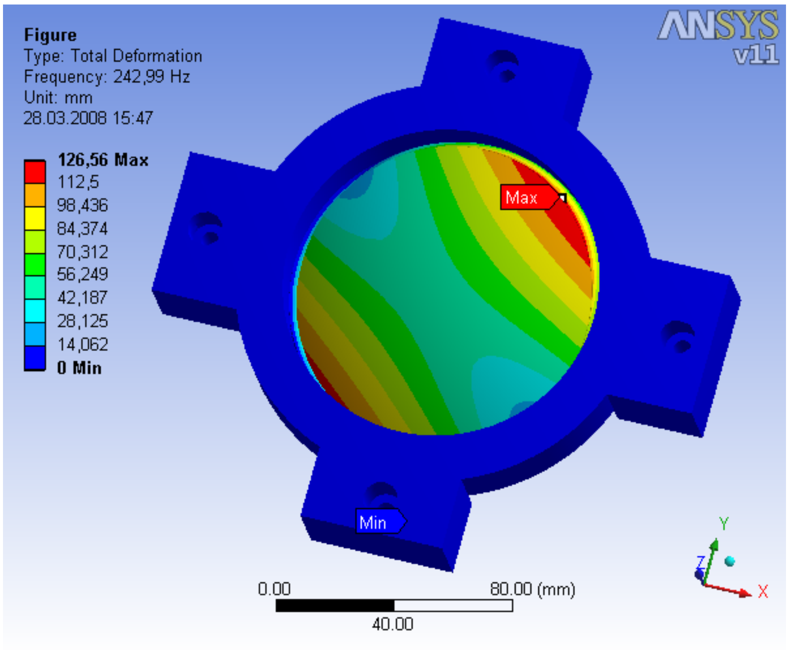


Figure 4. 6 Third Mode Shape (Tip and Tilt), 243 Hz

After completing the modal analysis, random vibration analysis is carried out. In random vibration analysis, directional accelerations can be obtained by using known random input. As explained in Chapter 2, if this optical system is integrated to AH-1 helicopter, frequency range of interest is 10-500 Hz and random vibration level W_0 is 0.001 g^2/Hz and W_1 is 0.01 g^2/Hz, as shown in Figure 4.7.

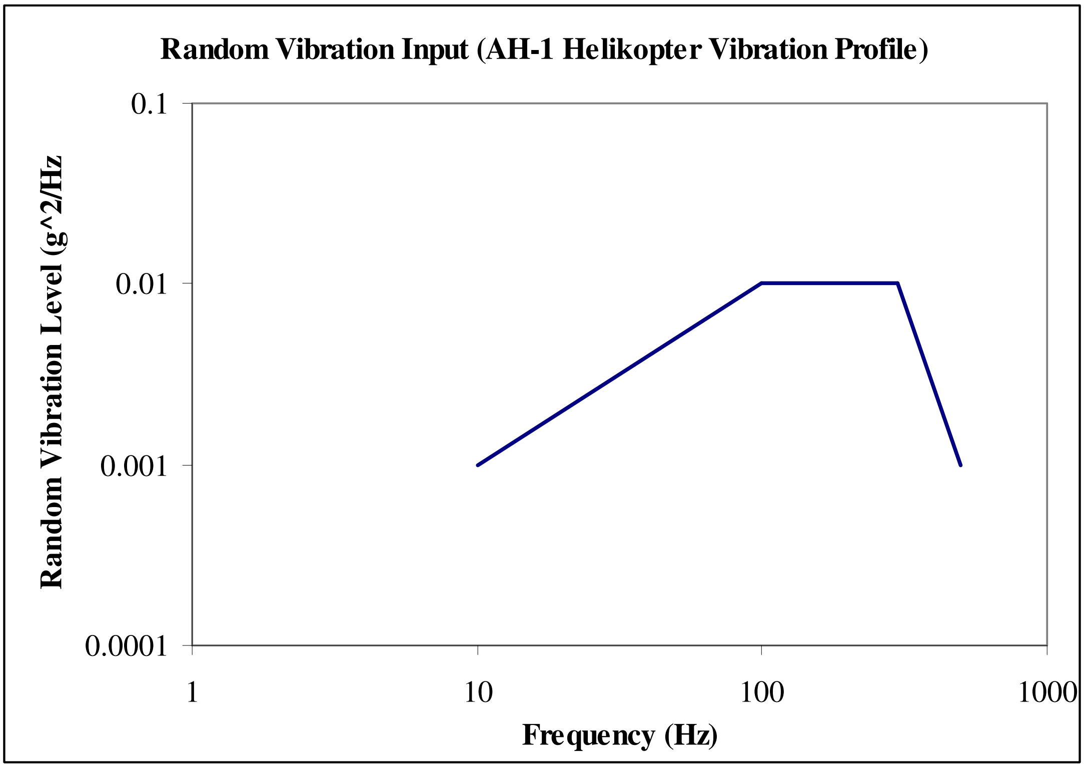


Figure 4. 7 Random Vibration Input (AH-1 Helicopter Vibration Profile) [9]

This random vibration input is applied to the system in each axis (X-Y-Z), than output (directional acceleration) is obtained for X-Y-Z axis. Moreover, damping value should be defined in ANSYS Workbench. Damping value of elastomer is found in Chapter 3, η = 0.175 and it is loss factor, however in ANSYS Workbench, modal damping ratio is used. In Chapter 3, it is explained that modal damping ratio, ζ is half of the loss factor, η; therefore, ζ will be 0.0875. Results of random vibration analysis are given in Table 4.4.

Table 4. 4 Results of Random Vibration Analysis

	Random Vibration in X Direction	Random Vibration in Y Direction	Random Vibration in Z Direction
X (m/s^2)	42	5	7
Y (m/s^2)	5	42	7
Z (m/s^2)	50	50	38

As one can see from the results of random vibration analysis, first three modes are in the interested frequency range, directional acceleration in each axis is close to each other. Especially, for random input in Z direction, directional acceleration output in all direction is high due to fact that first three mode shapes are dominant in Z direction. ANSYS Workbench results for input in each direction are given in Figure 4.8, 4.9 and 4.10.

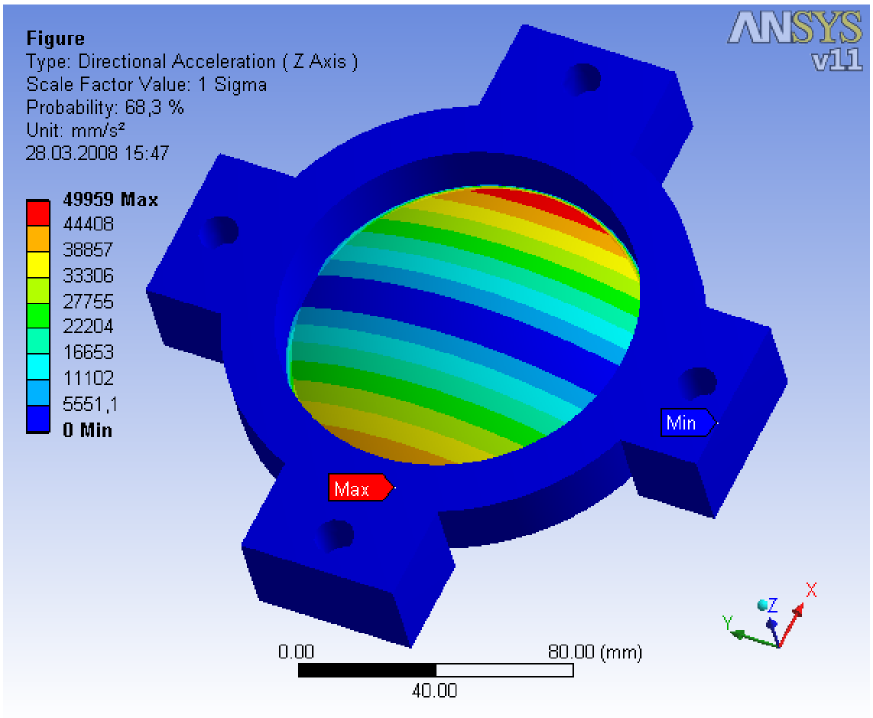


Figure 4. 8 Directional Acceleration in Z Direction for Random Input in X Direction

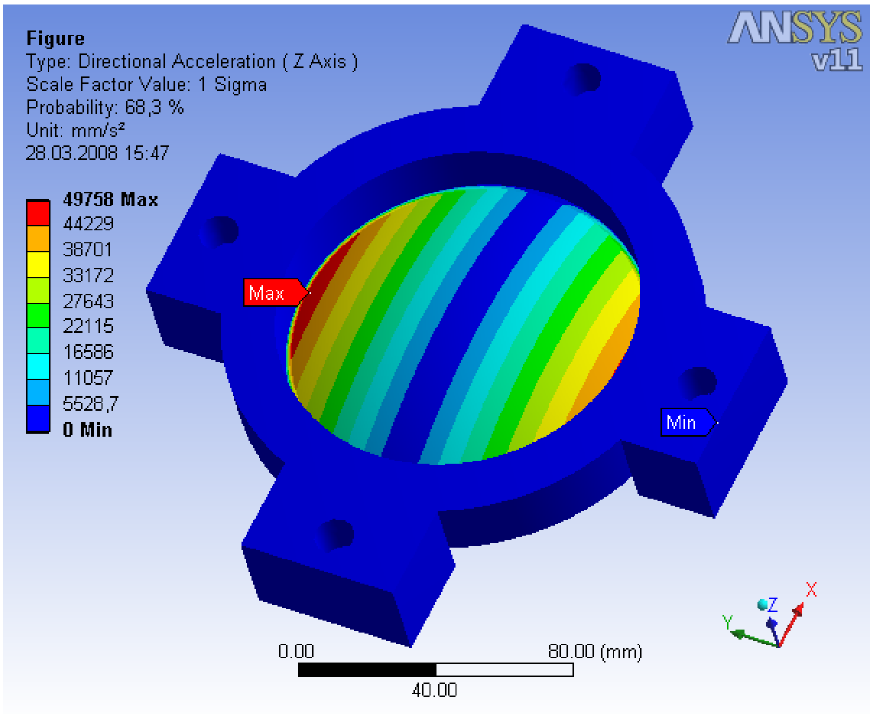


Figure 4. 9 Directional Acceleration in Z Direction for Random Input in Y Direction

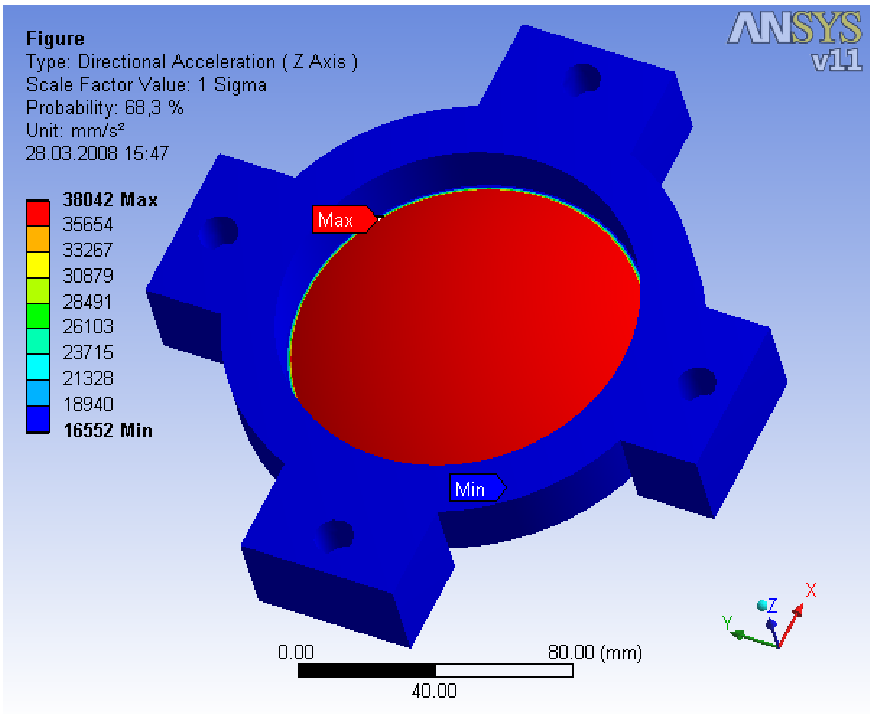


Figure 4. 10 Directional Acceleration in Z Direction for Random Input in Z Direction

4.2 Experimental Results

Modal and random vibration analyses are also carried out by using experimental techniques. By using experimental modal analysis techniques, natural frequencies of a simple optical system can be obtained. Also modal analysis gives mode shape and damping of the system. Thus damping values of the simple optical system considered can be compared with experiment results. After modal test, random vibration tests can be done by using shaker table in X-Y-Z axis by applying AH-1 Helicopter vibration profile. In experimental random vibration analysis, it is not possible to give input in θ and ϕ direction, since shaker tables can only give input in X, Y and Z directions. Also it is not possible obtain analytical results in X and Y directions in analytical modeling. Therefore, only the results obtained from the analytical model in Z direction are compared in random vibration analysis with those obtained by finite element model and experimental study. However, the results of the finite element analysis are compared with experimental results in X and Y direction as well.

For experiments, optical system used in both analytical model and finite model is manufactured (it is shown in Figure 4.11). Case is produced from aluminum, lens is made of germanium and SYLGARD 577 is used as elastomer, adhesive.

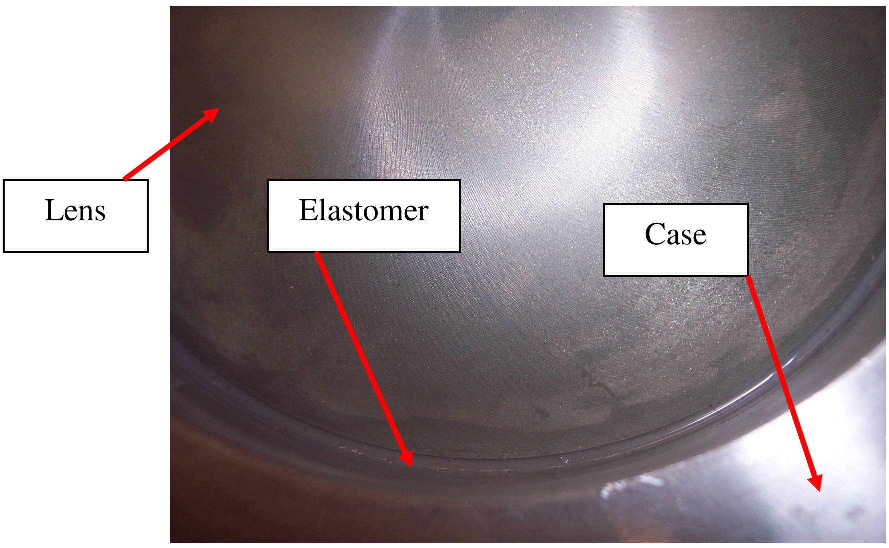


Figure 4. 11 Simple Optical System

Natural frequencies of the optical system can be obtained by hammer test since hammer excitation is enough to see natural frequencies and also it is easy and fast. In hammer test, one bi-axis accelerometer is used and this accelerometer is roved through 5 points. Instruments and transducer properties are given in Table 4.5 and 4.6.

Table 4. 5 Instrumentation and Software

Instrumentation and Software	
Accelerometer	Bruel & Kjaer 4508 B biax
Impact Hummer	Bruel & Kjaer 8200+2646
Analyzer	Pulse Front-End 3560C
Software	Pulse 11.0

Table 4. 6 Transducer Properties

Transducer Type	**Nom. Sensitivity**	**External Gain**	**Input Sensitivity**
Force	1 mV/N	1 V/V	1 mV/N
Accelerometer	10 mV/m/s^2	1 V/V	10 mV/m/s^2

Test setup for hammer test is shown in Figure 4.12. In this test setup, optical system is fixed by its holes, as can be seen in Figure 4.12, to optical table. Measurement points are shown in Figure 4.13

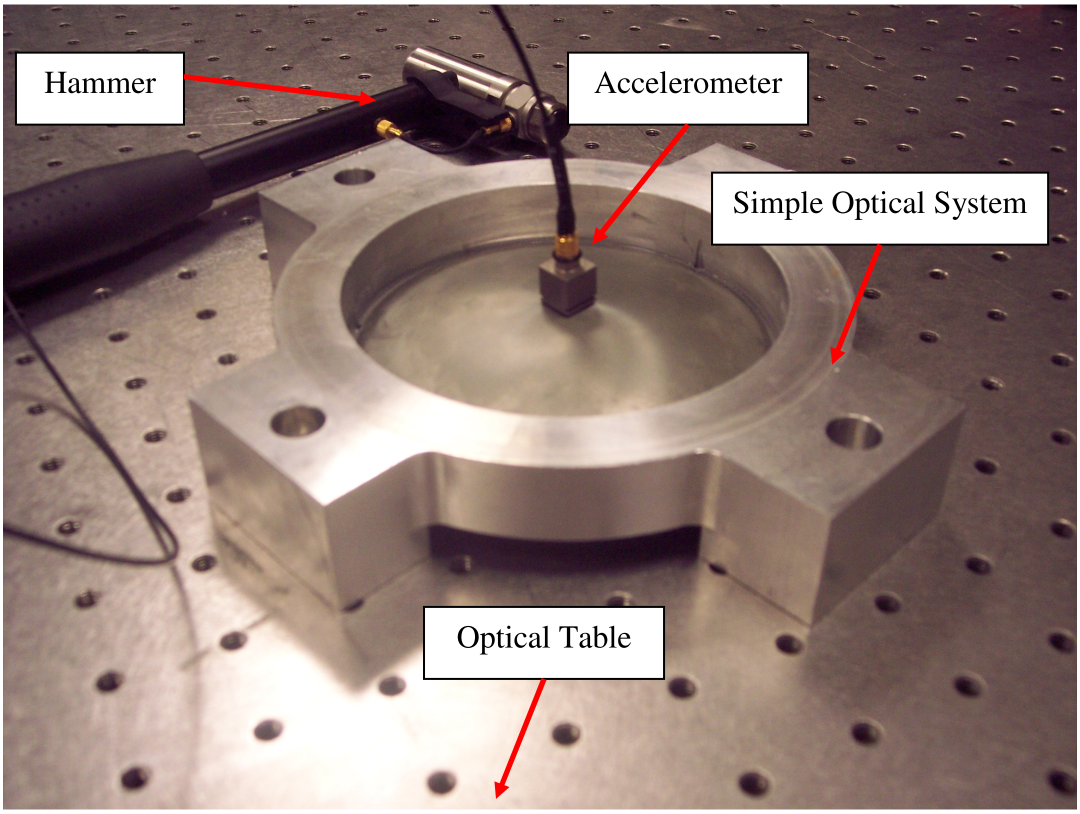


Figure 4. 12 Test Setup for Hammer Test

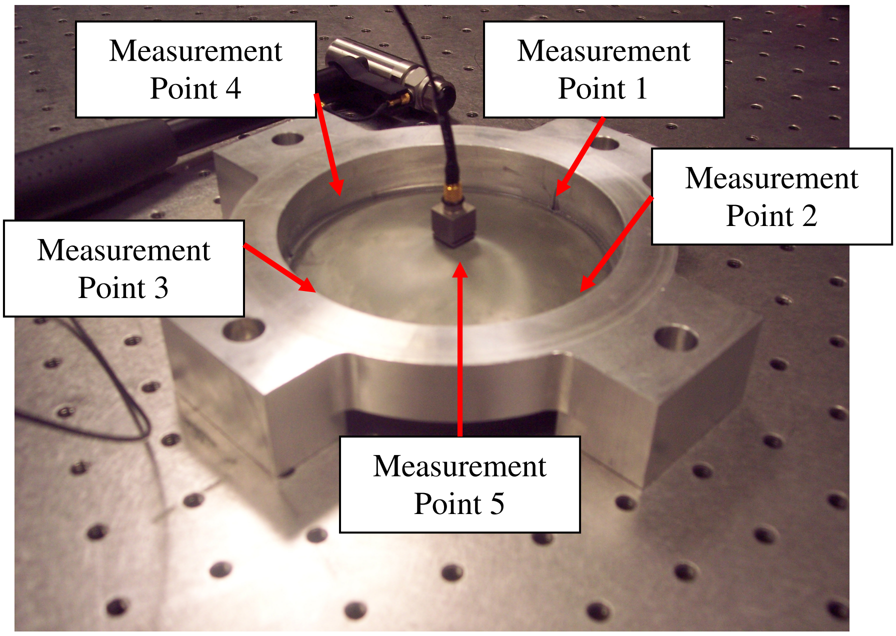


Figure 4. 13 Measurement Points

Modal test is conducted between 0-400 Hz and data is collected at 5 points. By PULSE 11.0 software, time data is used in order to obtain auto-spectrum, cross-spectrum, coherence and FRFs. FRF and coherence at measurement point 3 are given in Figures 4.14 and 4.15.

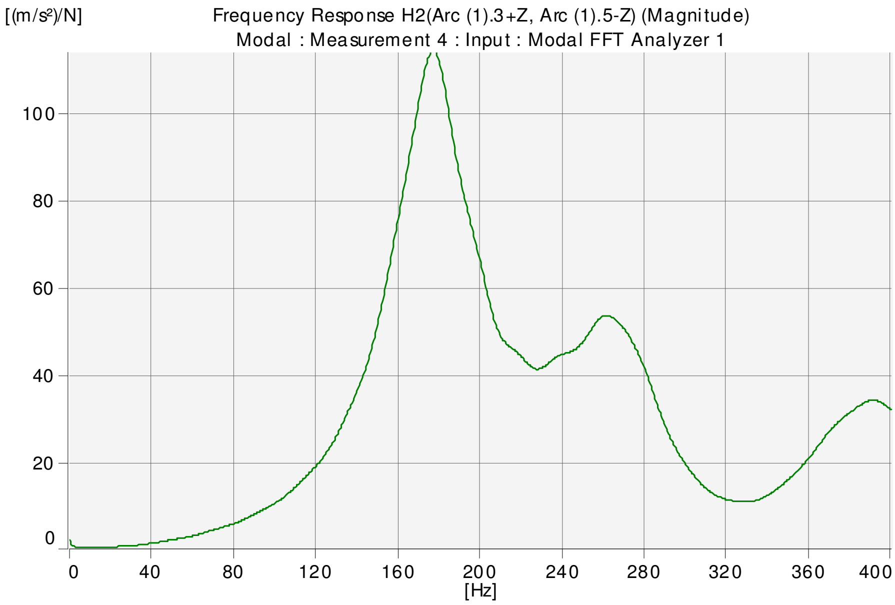


Figure 4. 14 FRF at Measurement Point 3

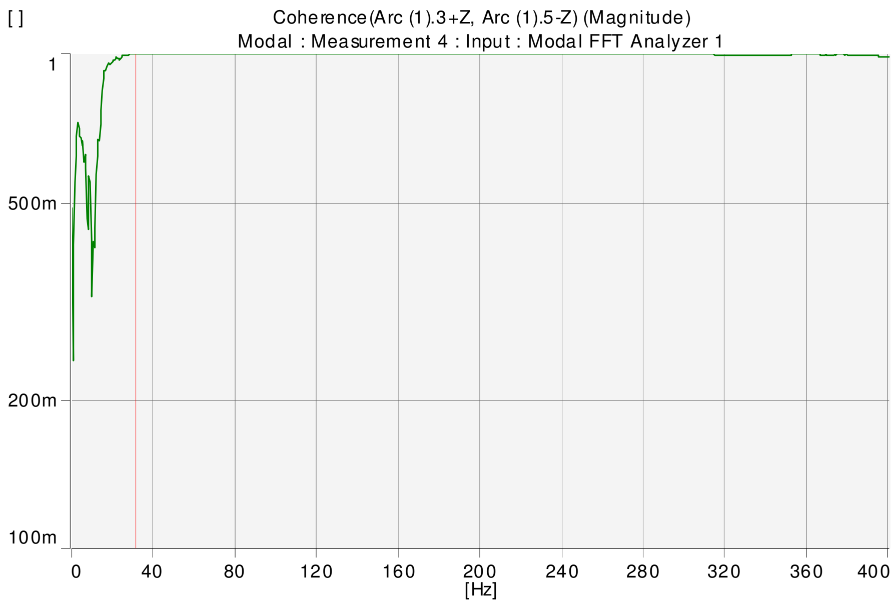


Figure 4. 15 Coherence at Measurement Point 3

Finally, by using obtained FRFs and ME'Scope 4.0, modal parameters of a simple optical system can be found. Natural frequencies and damping values are given in Table 4.7. It can be seen in Figure 4.14, that mode 3 can not be identified because at this frequency, 261.9 Hz, there are two modes and they are multiple modes. In order to identify multiple modes and coupled modes, Multi Input Multi Output (MIMO) and Single Input Multi Output (SIMO) analysis should be done. At least two shakers In MIMO test and at least two reference accelerometers in SIMO test should be used.

Table 4. 7 Natural Frequency and Damping Values

Mode No	Frequency, f_n (Hz)	Damping, ζ (%)
1	176.9	8.814
2	261.9	7.995

Moreover, mode shape of a simple optical system can be found by representative geometry in PULSE 11.0 and ME'Scope 4.0 and obtained FRFs. Mode shape at 1^{st} and 2^{nd} modes are given Figure 4.16 and 4.17, respectively.

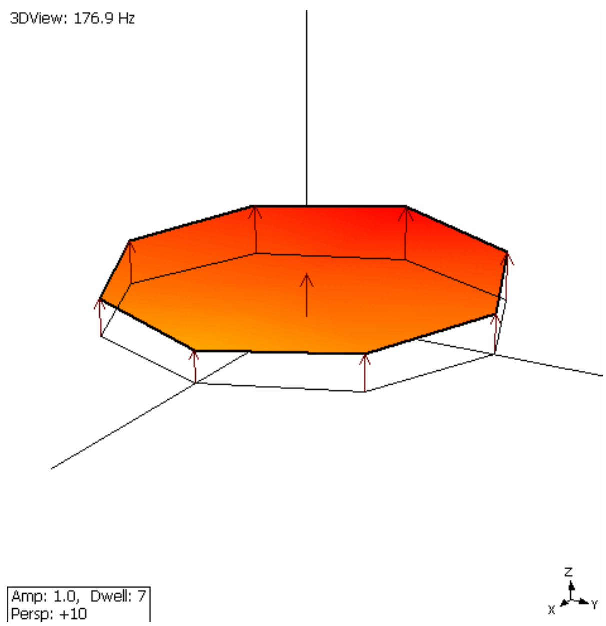


Figure 4. 16 Mode Shape at 1^{st} Mode

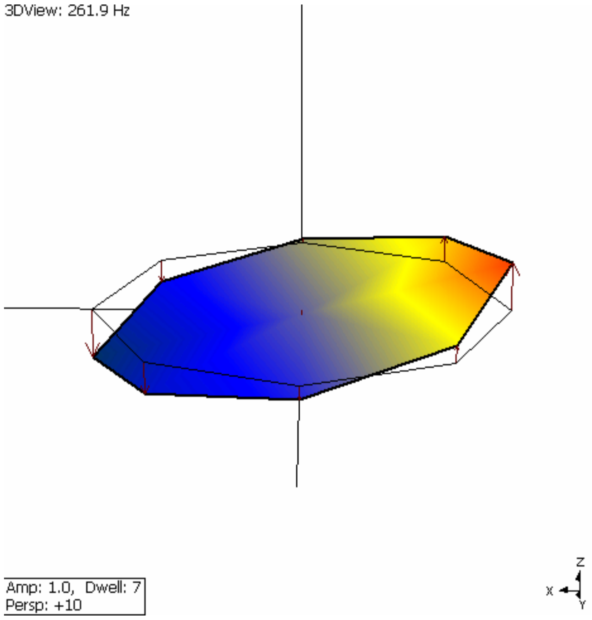


Figure 4. 17 Mode Shape at 2^{nd} Mode

After modal test, random vibration test can be done. In order to carry out random vibration test, shaker table must be used in X-Y-Z axis. For each axis, AH-1 Helicopter vibration profile given in Figure 4.7, is applied and directional accelerations are measured.

In random vibration test, two three-axis accelerometers are used in order to collect data in each axis, X-Y-Z. Test setup is shown in Figures 4.18 and 4.19. There is no need to test system in Y axis, because optical system is symmetric. As can be seen in Figures 4.18 and 4.19, there are two different shaker tables, one is used for X axis excitation and other one is used for Z axis excitation.

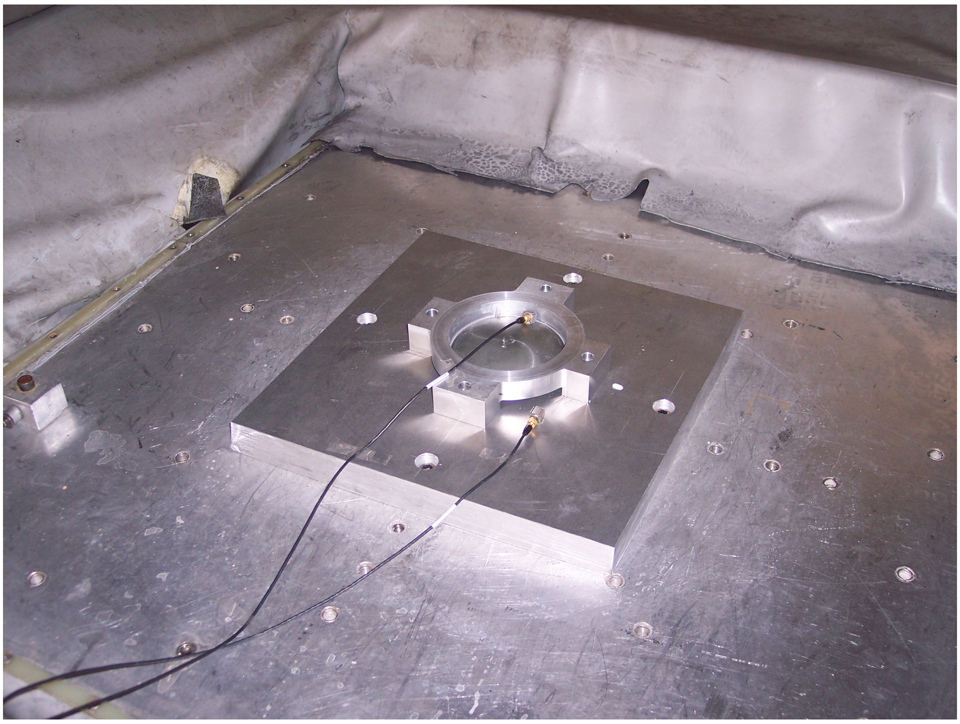


Figure 4. 18 Random Vibration Test Setup for Exciting the System in X Direction

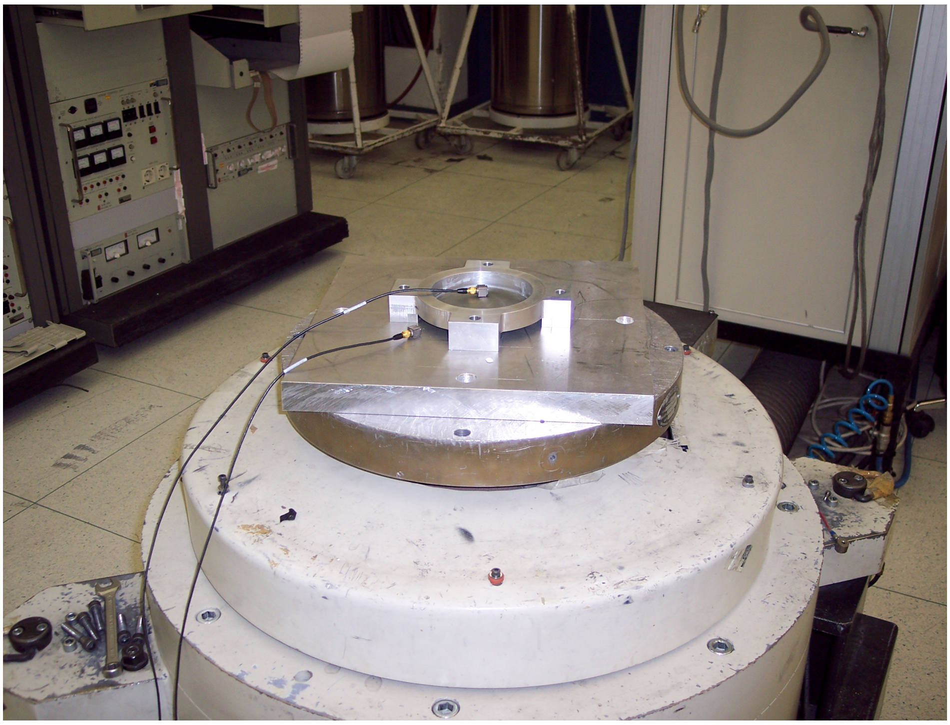


Figure 4. 19 Random Vibration Test Setup for Exciting the System in Z Direction

Using the analyzer and the software (Table 4.5), acceleration data can be collected from two triaxial accelerometers. One accelerometer is used for controlling the vibration table output and the other accelerometer is used to collect to vibration data from lens. During the test, auto-spectrum of acceleration data can be obtained in real time. Collected vibration data from reference, which is used for validation of given input to the shaker table, for both axes, X and Z, are given in Figures 4.20 and 21. As seen in Figure 4.20-21, there is no problem in vibration table for giving correct helicopter vibration profile. Exact vibration profile of AH-1 helicopter and vibration profile, obtained from reference accelerometer, are almost the same.

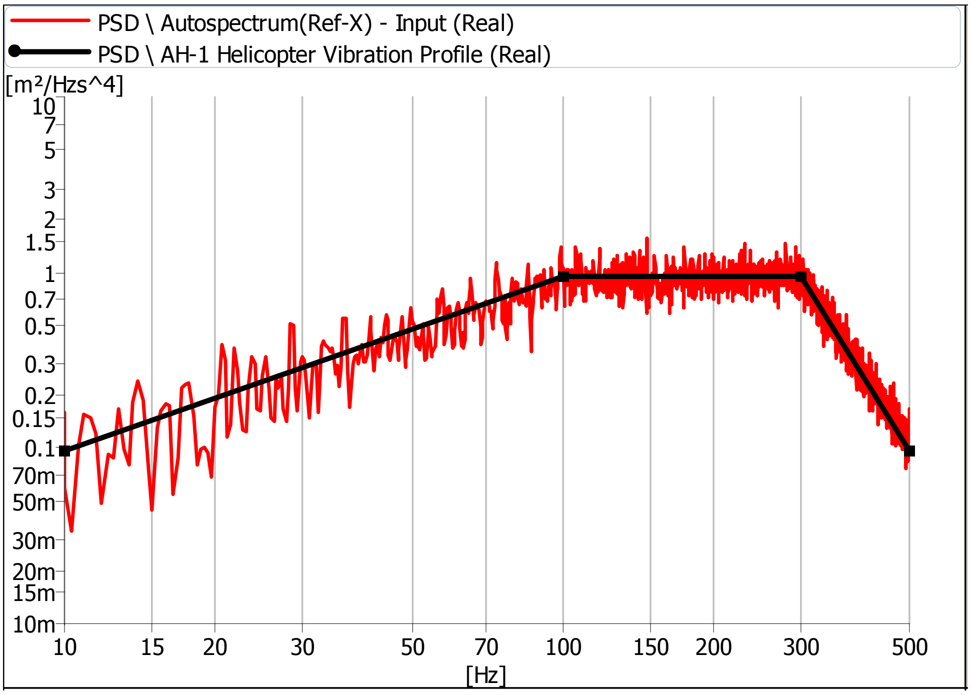


Figure 4. 20 PSD of Reference Accelerometer in X Directions

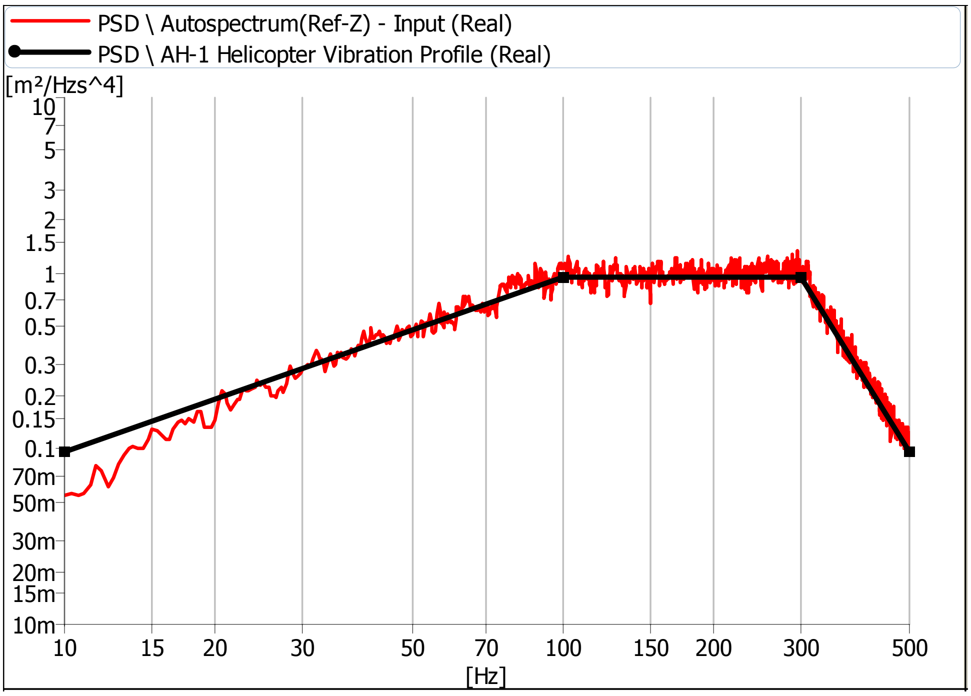


Figure 4. 21 PSD of Reference Accelerometer in Z Directions

By using second accelerometer, vibration data is obtained from lens, given in Figures 4.22 and 23. Natural frequencies of the simple optical system can be easily seen in figures. Total acceleration rms (m/s^2) is calculated for both X and Z axis by using software, PULSE 11.0 and they can be read directly from figures. Only these to results are shown in order to give an example.

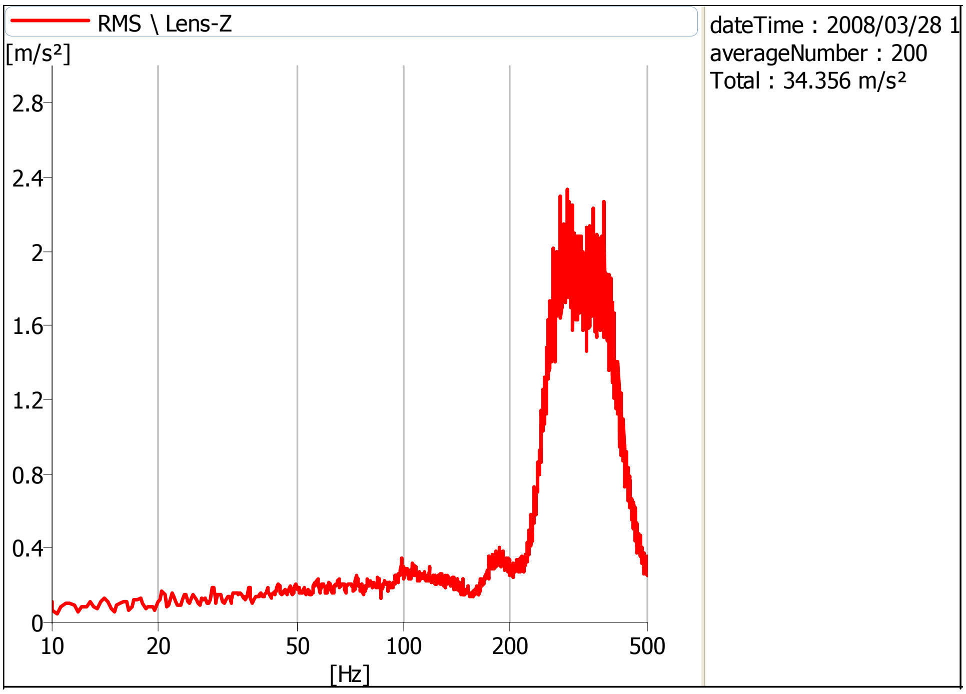


Figure 4. 22 RMS Acceleration Measured on Lens in Z Directions (Input in X Direction)

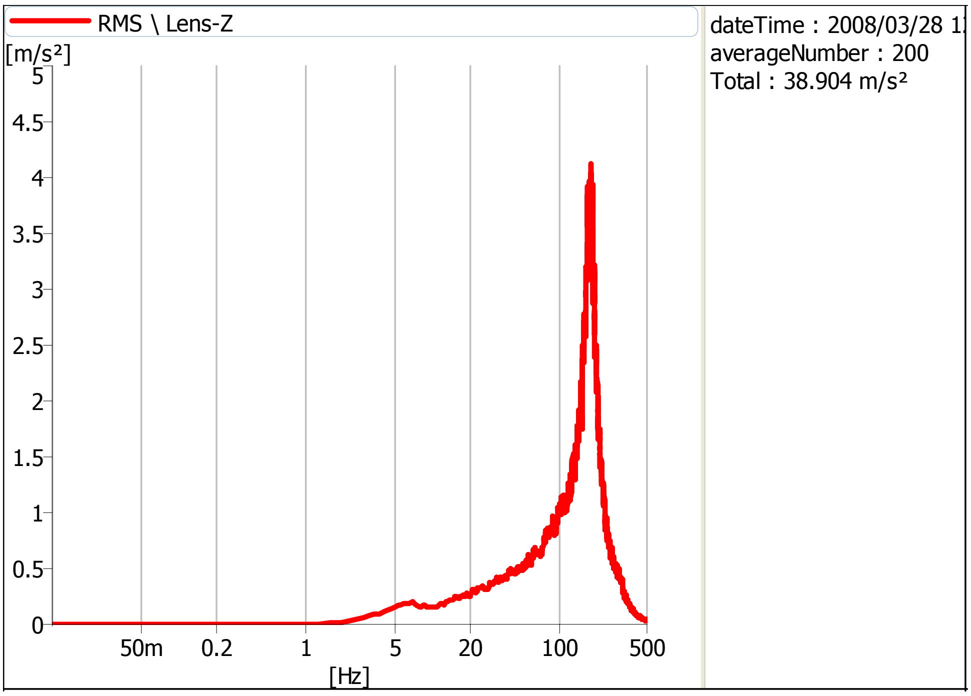


Figure 4. 23 RMS Acceleration Measured on Lens in Z Directions (Input in Z Direction)

4.3 Comparison of Results

Results, which are obtained from analytical model, finite element model and experiments, are compared in this section in order to validate analytical model and material properties of the elastomer.

In analytical model, stiffness of elastomer is obtained by shear forces and this stiffness is used to find natural frequencies of a simple optical system. Also material properties (Young Modulus and loss factor) of elastomer are obtained from experiment, as explained in section 3.1. Young modulus is used to obtain stiffness of elastomer and loss factor used to obtain random vibration output of a simple optical system.

Modal and random vibration analysis are carried out by finite element software, natural frequencies and random vibration output of the system are found by ANSYS Workbench by using material properties given in section 3.1.

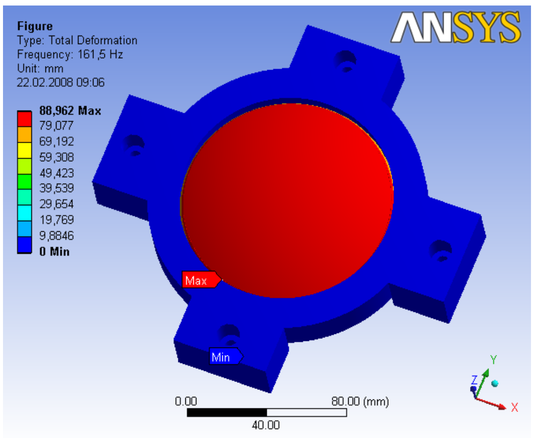
Finally, natural frequencies, damping and mode shapes are found from experimental study. Using the same system, random vibration test is done and output of the system is obtained. In order to verify three results, all results are compared. Natural frequencies, which are found by all three methods, are given in Table 4.8 where percentage differences from experiment results are shown in parenthesis.

Table 4. 8 Comparison of Natural Frequencies

Mode No	Analytical Results (Hz)	Finite Element Results (Hz)	Experimental Results (Hz)
1	191 (- 7.9 %)	169 (+ 4.5 %)	177
2	292 (- 11.4 %)	243 (+ 7.3 %)	262
3	292 (- 11.4 %)	243 (+ 7.3 %)	262

As seen in Table 4.8, all three results are close to each other. In analytical model shear force is used to find stiffness of the elastomer and when thickness of the elastomer is increased also normal force affect the stiffness of the elastomer; therefore results of analytical model differ from both finite element and experimental results. Effect of the elastomer thickness is presented in discussion and conclusion chapter.

The comparison of mode shapes obtained from the finite element analysis and experimental studies are given in Figure 4.24 and 4.25 for first and second modes, respectively. Mode shape 3 is not given since 2nd and 3rd are symmetric. It is seen in these figures that the first and second mode shapes are the same in both finite element analysis and experimental modal analysis.



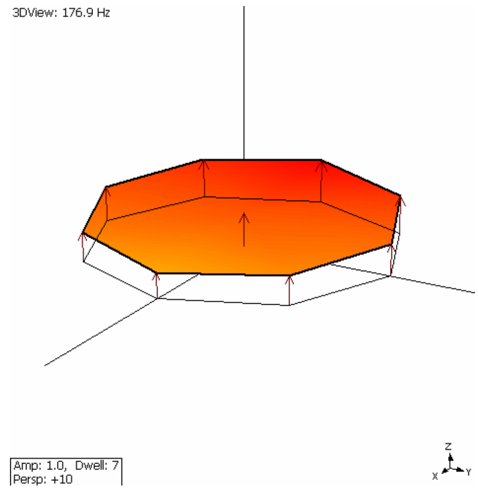


Figure 4. 24 Comparison of 1st Mode Shape of a Simple Optical System

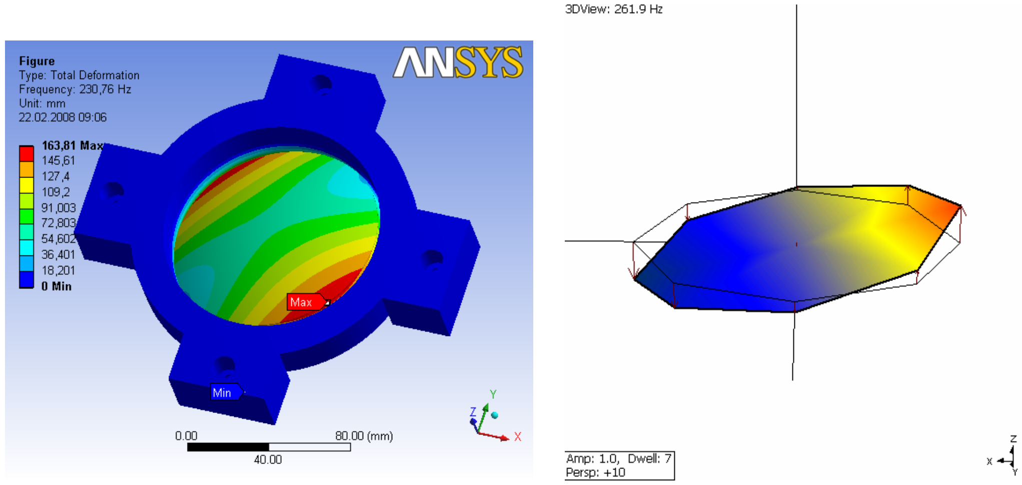


Figure 4. 25 Comparison of 2nd Mode Shape of a Simple Optical System

Damping values, which are obtained from material test and modal test, are compared in Table 4.9. In material test, loss factor of the elastomer is found and in modal test viscous damping of the elastomer is obtained, as explained in Chapter 3. It is seen form damping values that damping values found from both material test and modal test are close to each other.

Table 4. 9 Comparison of Damping Values

Mode No	Material Test Damping Value	Modal Test Damping Value	Difference (%)
1	0.0875	0.0881	0.7
2	0.0875	0.0799	9

Finally, random vibration analysis results are compared. As it can be recalled, the principles coordinates are chosen as z, θ and ϕ for analytical model in order to obtain natural frequencies and random vibration output, however principles coordinates are x, y and z in the finite element model and in experiments (Figure 4.26). It is not possible to give input in direction θ and ϕ in both finite element analysis and experiment, therefore, only output in z direction are compared for analytical model, finite element and experiments, which are given in Table 4.10 (percentage difference from experiment results are shown in parenthesis).

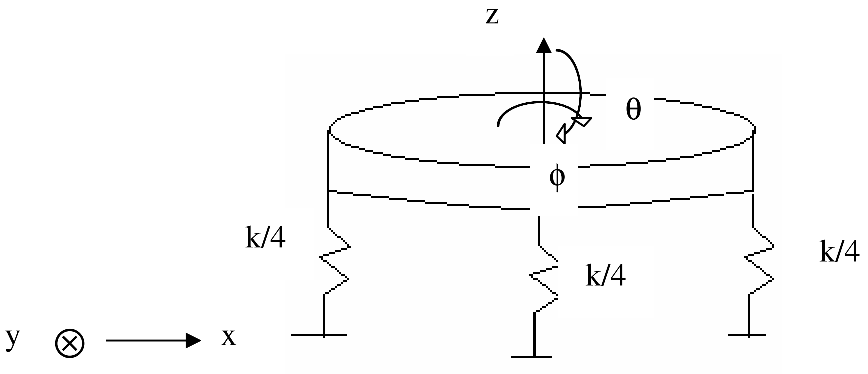


Figure 4. 26 Discrete Model of a Simple Optical System

Table 4. 10 Comparison of Random Vibration Analysis Results in Z Direction

	Analytical Results (m/s^2)	**Finite Elem. Results (m/s^2)**	**Experiment Results (m/s^2)**
Input Z – Output Z	41 (- 5.4 %)	38 (+ 2.3 %)	38.9

Random vibration analysis results of finite element and experiments can be compared for x and y directions as well, as shown in Table 4.11 (percentage difference from experiment results are shown in parenthesis). As seen in Figure 4.18, because of accelerometer size, it is not possible to place it to tip position of lens. Accelerometer can be placed to lens a little bit far point from tip of the lens, therefore results should be compared by taking account this situation. If Figure 4.8 is investigated, finite element results in Z direction is 38 m/s^2 (76% of 50 m/s^2) for approximate position of accelerometer in experiments, when PSD is applied in X direction. It is known that, the simple optical system is symmetric, therefore; there is no need to compare results in y direction, i.e. results in y direction are the same as results in x direction due to symmetricitiy.

Table 4. 11 Comparison of Random Vibration Analysis Results

	Finite Elem. Results (m/s^2)	Experiment Results (m/s^2)
Input Z – Output X	7 (+ 5.4 %)	7.4
Input X – Output Y	5 (- 6.4 %)	4.7
Input X – Output Z	38 (- 10 %)	34.4

As can be seen in Table 4.10 and 4.11, results for input Z – output Z is very close to each other. Moreover, results of input Z – output X, input X – output Y and input X – output Z is in acceptable error range.

Finally, it can be said that, simple analytical and finite element models are validated by experiments. There are differences in results of modal and random vibration analysis, which are not exceeding 11.4 % in natural frequencies and 10 % in random vibration responses. It can be concluded that these differences are acceptable and the analytical model developed can be used in preliminary dynamic analysis.

CHAPTER 5

EFFECT OF VIBRATION ON OPTICAL PARAMETERS

Military optical system should be designed in order to withstand the harsh environmental conditions. These environmental conditions, which include temperature, humidity, vibration and shock etc., cause structural damage or play important role on the performance of optical system. Because of this reason, when designing an optical system, all environmental effects should be considered and taken into account.

The scope of this thesis is to determine the effect of vibration on optical system performance; therefore, optical element’s movements should be found under vibration.

During optical system design, all optical parameters, such as wave-front residual, thickness and radius of optical elements, index, V-number, homogeneity, decenter, tilt, sphericity and irregularity, are taken into account [22]. These parameters should be within the optical tolerances limit. Except decenter and tilt, all other parameters are related to geometry and material properties of optical elements, where as decenter and tilt should are related with the movement of optical elements.

If the tolerances of optical elements are out of the limits, aberration can be occurred in optical system. These aberrations cause problems in image formation, therefore tolerance analysis should be done carefully and the changes in optical elements parameters due to environmental effects have to be found.

Also, there are other parameters [23], which affect the optical performance. One of these parameters is acuity and it can be used for the measurement of optical performance. In order to understand acuity, imprecise terms (sharpness, resolution and depth of focus etc.) and precise terms (modulation transfer function (MTF), encircled energy and point spread function (PSF) etc.) can be investigated.

5.1 Aberrations

Aberrations can be defined [24] as image deformation because of change in small-angle (paraxial, which is called also optical axis) imaging. It is seen in Figure 5.1 [24], after passing the optical elements, ray from object point intersect the image surface and path of the ray can described by s and θ. For every intersection points, polynomial equations for $x^{'}$ and $y^{'}$ can be written and constants of these equations can be used for determination of aberrations.

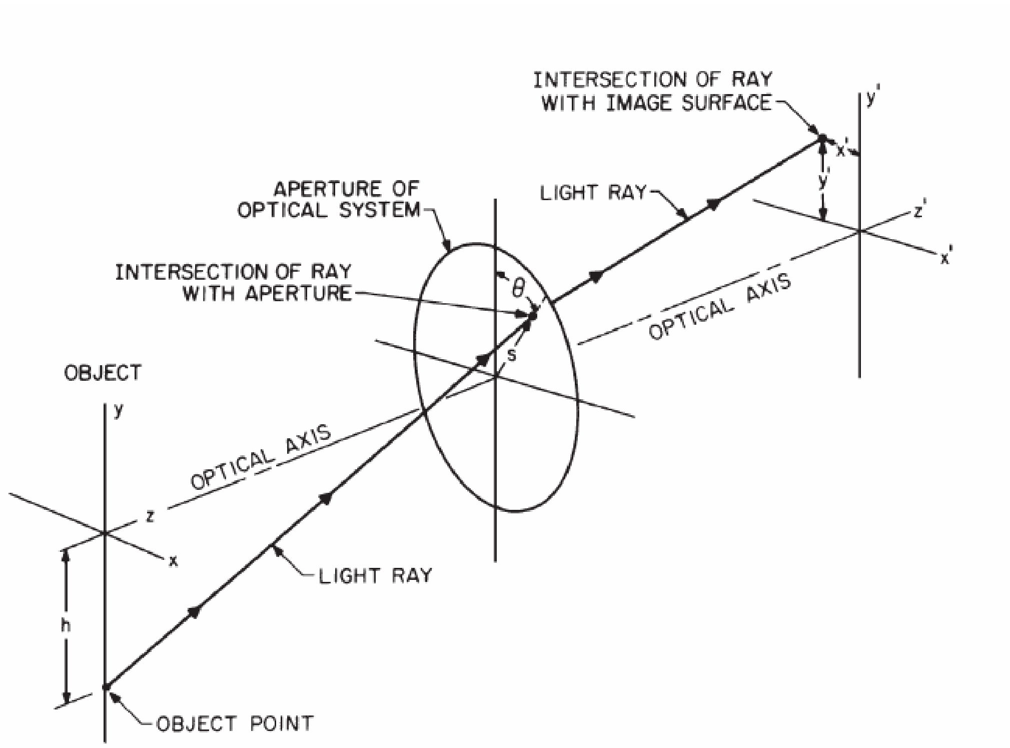


Figure 5. 1 Image Formation in an Optical System [24]

Also, using these constants, aberration curves can be drawn and aberration curves show the degree of aberration in optical system, i.e. lens performance can be understood from aberrations curves [22]. Moreover, different type of aberrations, such as spherical, comma, astigmatism, chromatic etc., can be evaluated from aberration curves.

Types of the aberrations curves are ray errors and optical path difference (OPD) [22]. Ray errors are commonly used for determining aberrations. Also ray errors can be divided into two types; one is transverse ray plots and another is longitudinal (field) plots.

Transverse ray plots can be generated by passing a bunch of rays through the optical elements [22]. Example of transverse ray plots is shown in Figure 5.2. In Figure 5.2, after passing the lens, ray crosses the optical axis at three different points. "Plane a" shows the minimum ray errors, because “Plane a” close the horizontal axis of transverse ray plot and "Plane c" indicates the spherical aberration.

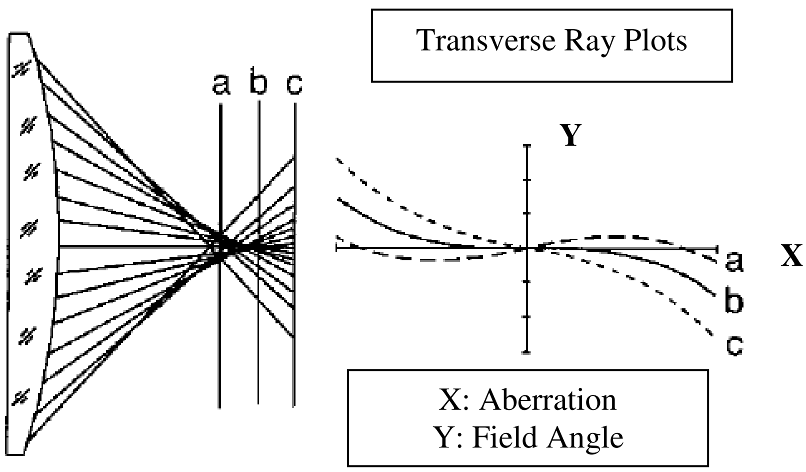


Figure 5. 2 Transverse Ray Plots [22]

Another example for transverse ray plots is given in Figure 5.3 and it shows the spherical, coma and combined aberrations.

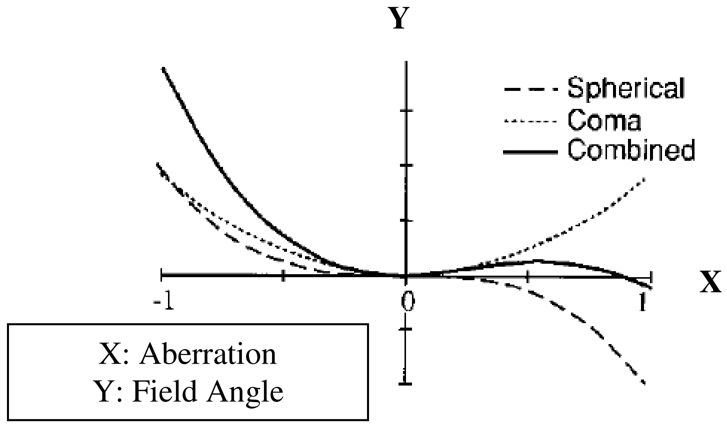


Figure 5. 3 Transverse Ray Plots for Combined Aberrations [22]

Field (longitudinal) plots are another type of ray error plots. When these plots are created, field angle is plotted in vertical axis and aberrations are shown in horizontal axis. These flied plots are usually used for indicating distortion, field curvature lateral color and different wave length of ray. Example for distortion is shown in Figure 5.4.

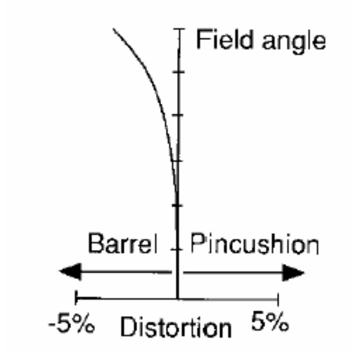


Figure 5. 4 Field Plots for Distortion [22]

5.2 Measure of Optical Performance

As explained before, there are other parameters for the determination of optical performance. One of the most common is acuity and it is the measure of optical performance. Acuity contains imprecise and precise terms. Imprecise terms can be sharpness, definition, resolution, diffraction-limited and depth of focus. Point spread function, modulation transfer function, root-mean-square optical path difference, Strehl ratio and encircled energy can be used for precise terms.

In this thesis, modulation transfer function (MTF) is used for predicting the optical performance, as MTF is one of the easy and quick methods to determine the optical performance. MTF can be defined [23] as plotting the image contrast as a function of spatial frequency. Because objects and images are interpolated in frequency domain, it is useful to analyze in whole range of linear-system. Moreover, transfer function method gives opportunity to facilitate in analyzing complex optical problems. MTF can be calculated by optical design program or can be measured from real system by optical measurement devices.

Basis of tolerance process can be done by deciding minimum level of satisfactory image quality [22]. Minimum level of acceptable image quality is shown by desired level of contrast at a specific spatial frequency as shown by the modulation transfer function.

By looking MTF graph optical performance and aberrations can be understood for different tolerances [22]. In this graph, ξ is spatial frequency and ξ_{cutoff} is image space cut-off frequency, these two parameters are used to calculate the MTF. If MTF is linear along the ξ/ξ_{cutoff} axis, then it can be said that there is no problem in the optical system performance. Moreover, if decenter values exceed the tolerance limit, defocus can be seen in optical system and this can be understood from MTF graph, Figure 5.5. In Figure 5.5, there are different MTF curves and A curve shows the optical system in focus and others show the defocus because A curve is linear along the ξ/ξ_{cutoff} axis.

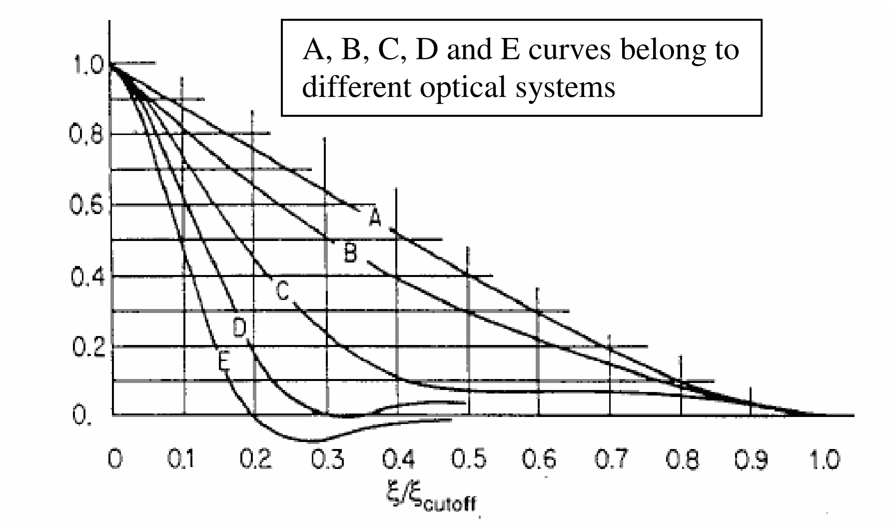


Figure 5. 5 MTF Graph for Focus Determination [22]

Another example for MTF graph is given in Figure 5.6. This figure shows the Diffraction MTF for system with third-order spherical aberration.

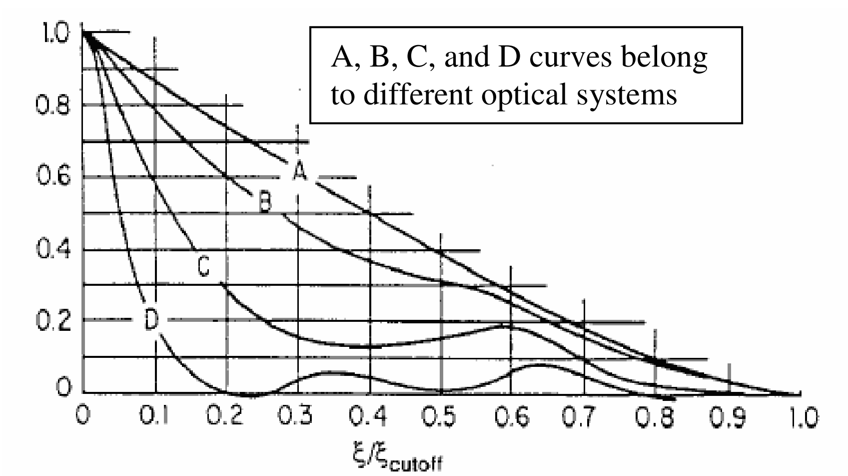


Figure 5. 6 MTF Graph for Spherical Aberration Determination [22]

5.2 Tolerance Analysis in Optical System

Tolerance analysis should be done in an optical system in order to determine the effects of the optical parameters on performance. If these parameters exceed the tolerance limits, optical performance degrades and problem can be seen in image formation. Aberrations curves and MTF show whether the optical parameters are in tolerance limits or not. Also, using aberration curves and MTF, optical performance can be determined.

There are different ways to do optical tolerance analysis. Most common and quick method is using optical design software. ZEMAX and CODEV are the most popular optical design software, and both of them are used in ASELSAN. In this thesis, ZEMAX is used to do the tolerance analysis to see the effect of vibration on optical system.

There are different tolerance types [25], these are tolerances on single surfaces (radius, thickness, index etc.), tolerances on elements (wedge etc.), tolerances on components (decenter, tilt etc.) and tolerances on polarization properties (reterdance, faraday rotation, etc.), In this thesis, only decenter and tilt tolerances, i.e. tolerances on components, are considered, other tolerances, wave-front residual, radius and thickness of lens and homogeneity, are not taken into account because they are not related to optical element's movement, they are associated with geometry and material properties of optical elements.

In Figure 5.7, schematic view of tilt and decenter of an optical element can be seen [25]. Decenter can be seen in both X and Y directions. During the tolerance analysis, for decenter tolerances, optical components position is changed in X and Y direction, and also for tilt tolerances, optical element is rotated with respect to Z axis. After that, MTF curves are analyzed and so required decenter and tilt tolerances are determined. Finally, displacement (decenter) and rotations (tilt) of optical elements are calculated under the vibration and these values are checked whether they are in tolerance limits or not, by doing this procedure, optical performance degration under the vibration is investigated.

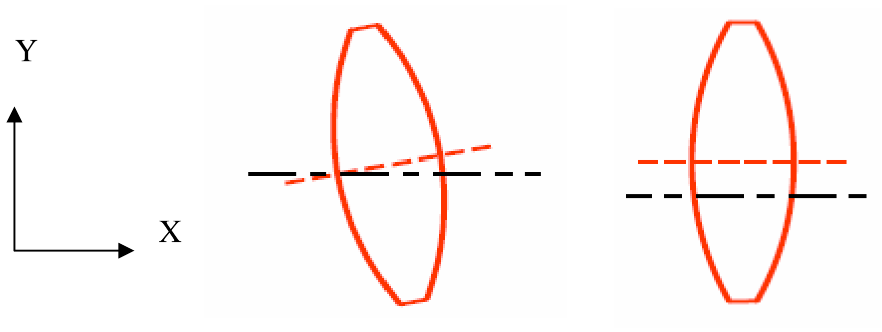


Figure 5. 7 Tilt and Decenter of Optical Element [25]

5.3 Tolerance Analysis of a Simple Optical System

In order to do tolerance analysis of a simple optical system, which is constructed in Chapter 2, ZEMAX [26] software is used. Optical element layout is given in Figure 5.8. In this layout only 1st lens are considered and effect of movement of this lens are investigated on optical performance.

In ZEMAX, different MTF values are calculated for different component tolerances at different spatial frequency in cycle per mm. Nominal MTF value is 0.5899 at spatial frequency of 20 cycles per mm, Figure 5.9.

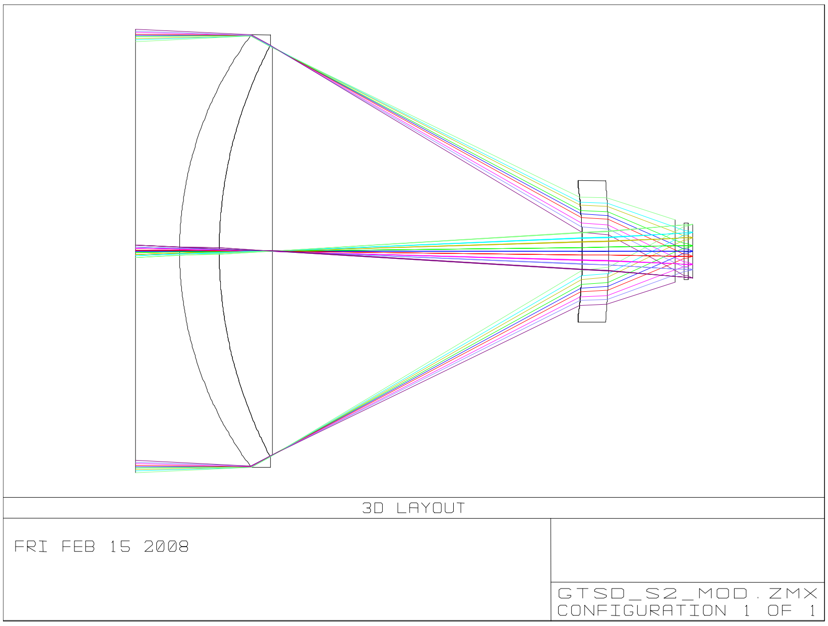


Figure 5. 8 Optical Element Layout

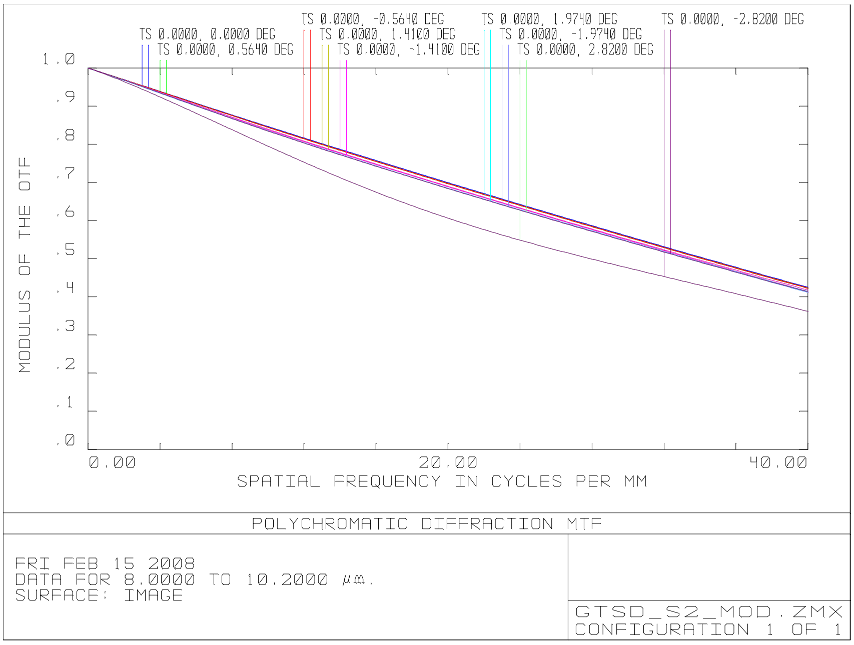


Figure 5. 9 MTF Graph of a Simple Optical System

Lens is moved in Z direction (decenter) from +/- 0.01 to +/- 0.06 mm and rotated with respect to X and Y direction (tilt) from +/- 0.01 to +/- 0.06 degree in ZEMAX, then MTF values for these movements are calculated, Table 5.1. Finally, estimated MTF values and nominal MTF values are compared in Table 5.1.

Table 5. 1 MTF Values for Different Decenter and Tilt Values

Decenter Z (mm)	**Tilt X (degree)**	**Tilt Y (degree)**	**Estimated MTF**	**Nominal MTF**	**Error in MTF (%)**
+/- 0.01	+/- 0.01	+/- 0.01	0.5805	0.5899	1.59
+/- 0.015	+/- 0.015	+/- 0.015	0.5858	0.5899	0.70
+/- 0.03	+/- 0.03	+/- 0.03	0.5737	0.5899	2.74
+/- 0.031	+/- 0.031	+/- 0.031	0.5727	0.5899	2.92
+/- 0.032	+/- 0.032	+/- 0.032	0.5716	0.5899	3.10
+/- 0.033	+/- 0.033	+/- 0.033	0.5704	0.5899	3.30
+/- 0.034	+/- 0.034	+/- 0.034	0.5693	0.5899	3.49
+/- 0.035	+/- 0.035	+/- 0.035	0.5681	0.5899	3.69
+/- 0.04	+/- 0.04	+/- 0.04	0.5617	0.5899	4.77
+/- 0.05	+/- 0.05	+/- 0.05	0.5469	0.5899	7.28
+/- 0.06	+/- 0.06	+/- 0.06	0.5297	0.5899	10.20

As can be seen in Table 5.1, tolerance analysis is a statistical process and for different tolerance values MTF values change, therefore; error in MTF should be decided and for a rule of thumb, change in MTF should not exceed 1 % in order to obtain good optical performance. It is seen in Table 5.1, that the difference between nominal and estimated MTF values is 0.7 % for 0.015 mm decenter and 0.015 degree tilt and these decenter and tilt values are acceptable for obtaining good optical performance. By considering random vibration analysis output for a simple optical system and values in Table 5.1, optical system performance can determined.

In previous chapters, dynamic model of a simple optical system is constructed by both analytical and finite element model and these are validated by experiments. In order to find whether optical performance is degraded or not, finite element results are used, since finite element analysis can give both decenter and tilt values of the lens. The results of random vibration analysis, which is carried out by using input as AH-1 helicopter vibration profile, are given in Table 5.2.

Table 5. 2 Finite Element Results of Random Vibration Analysis

	Tilt X (degree)	Tilt Y (degree)	Decenter Z (mm)
Random Vibration Analysis Results	0.0117	0.01177	0.0337

If Table 5.1 and 5.2 are studied together, estimated MTF exceeds the 1 % for 0.01° tilt and 0.03 mm decenter. As can be seen in results of both random vibration analysis and optical tolerance analysis, optical performance are degraded under the specified vibration environment.

CHAPTER 6

COMPUTER PROGRAM

In this thesis, a simple computer program is developed in order to find optical performance under vibration. As explained in previous chapters, optical system is analyzed and it is seen that basic optical system components, case and lens, are rigid for interested frequency range, 10-500 Hz. Also it is found that dynamic characteristic of optical system is affected by elastomer, which connects lens to case, for 10-500 Hz frequency range. After that, a simple optical model is constructed and analytical model is prepared for this simple optical system by considering elastomer flexibility. This analytical model can be used to calculate the first three natural frequencies of the optical system and also to find output of optical system under random vibration input. Results of the analytical model are compared with those of the finite element model and experimental study and it is seen that results of these three analyses are close to each other. Therefore, the analytical model can be used for preliminary dynamic analysis of the optical system.

The lens moves with respect to the optical axis under vibration and these movements are decenter and tilt due to the first three modes of the optical system. During the optical system design, tolerances of optical components are determined for optimum image formation. If these tolerances are exceeded, optical system performance is degraded considerably. It is explained that decenter and tilt tolerances may be exceeded because of optical component movement under vibration. First, decenter and tilt of optical elements under vibration should be calculated, than these values should be compared with decenter and tilt tolerances in order to predict optical performance under vibration.

For preliminary design stage, two computer programs are developed. One is for determination of material properties of elastomer, another is for optical system performance prediction. These two programs are very simple and they give information about material properties of elastomer and effect of dynamic characteristics of optical system on optical performance for the first stage of design of optical system. Computer programs are created in MATLAB [27] and they have simple user interfaces.

6.1 Computer Program for Determination of Material Properties of Elastomer

In Chapter 3, material properties of elastomer are found and elastic modulus and loss factor of elastomer are determined [16]. The method employed [16] can be used for different type of elastomers; therefore, a simple computer program is prepared. In this computer program, test results are used for finding elastic modulus and loss factor of elastomer. SI unit system is used in this program. First of all, test results have to be imported from vibration analysis program, PULSE. Imported file format is text and using values, which are in text file, "Microsoft Excel Sheet" can be created. File names for imaginary and real part of measured Frequency Response Function – FRF for rigid mass with elastomer, shown in Figures 3.2 and 3.3, are "imaginary.xls" and "real.xls". These two files should be in working, current, directory of MATLAB. Using user interface of computer program, mass of rigid disk and geometric values, length and diameter, of elastomer can be entered to the program. After this process, when "Plot Material Properties of Elastomer" is clicked, imaginary.xls" and "real.xls" files are read by MATLAB and elastic modulus and loss factor of elastomer are plotted in simple user interface. Moreover, this computer program finds elastic modulus and loss factor of elastomer for simple optical system. First, 1st natural frequency of the optical system is predicted by using single degree of freedom model and than, using this frequency, elastic modulus and loss factor of elastomer are calculated. Example is given in Figure 6.1 for SYLGARD 577 elastomer test setup.

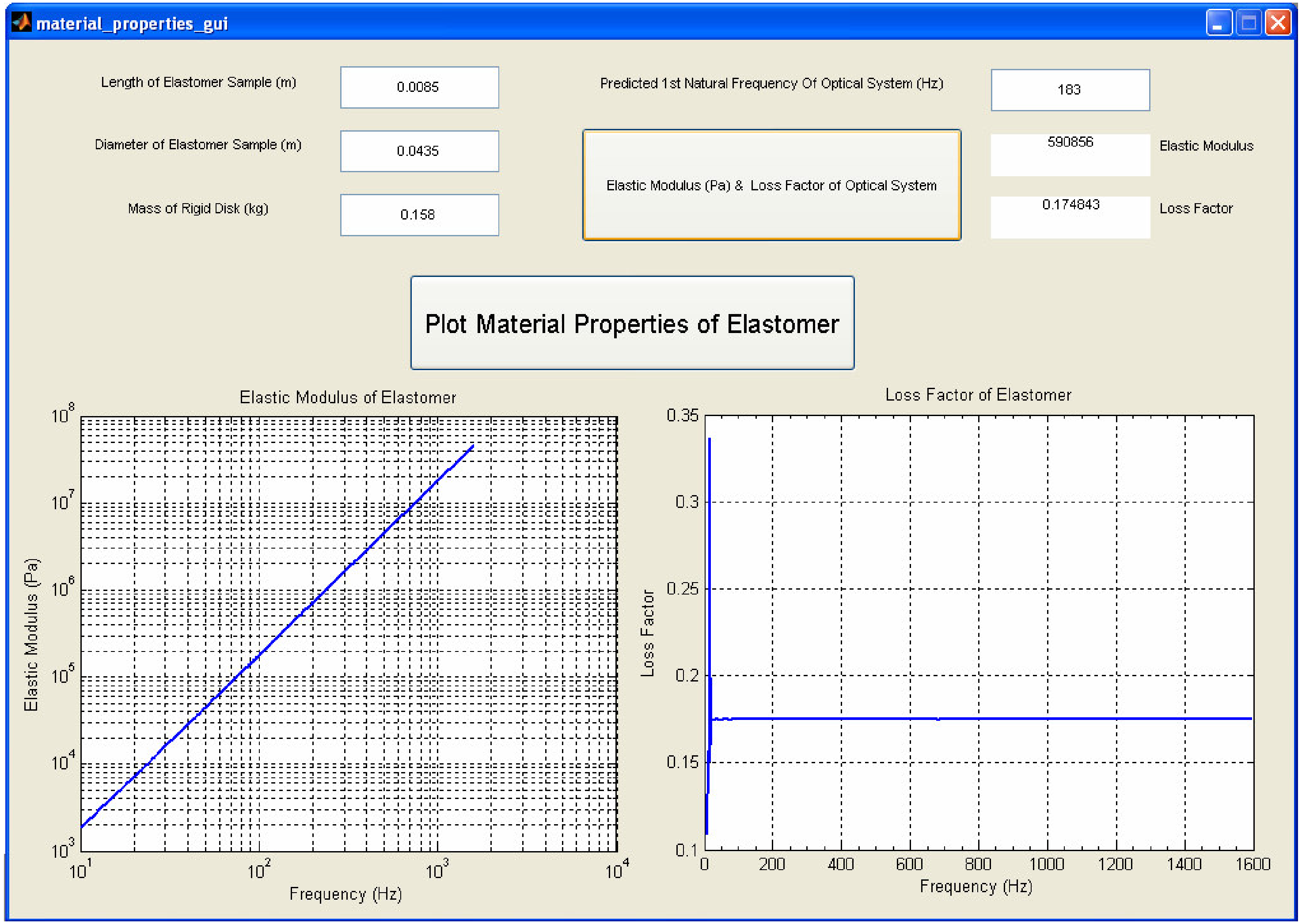


Figure 6. 1 Example for User Interface of Computer Program for Determination of Material Properties of Elastomer

6.2 Computer Program for Determination of Optical Performance under Vibration

The second program is used for the preliminary analysis of dynamic characteristic of an optical system. It also determines the effect of vibration on optical system.

By using mathematical model developed, which is presented in Chapter 3, a simple computer program is created. SI unit system is used in this program. This computer program first finds the stiffness of elastomer by using geometric and material properties found by the previous computer program (Figure 6.2).

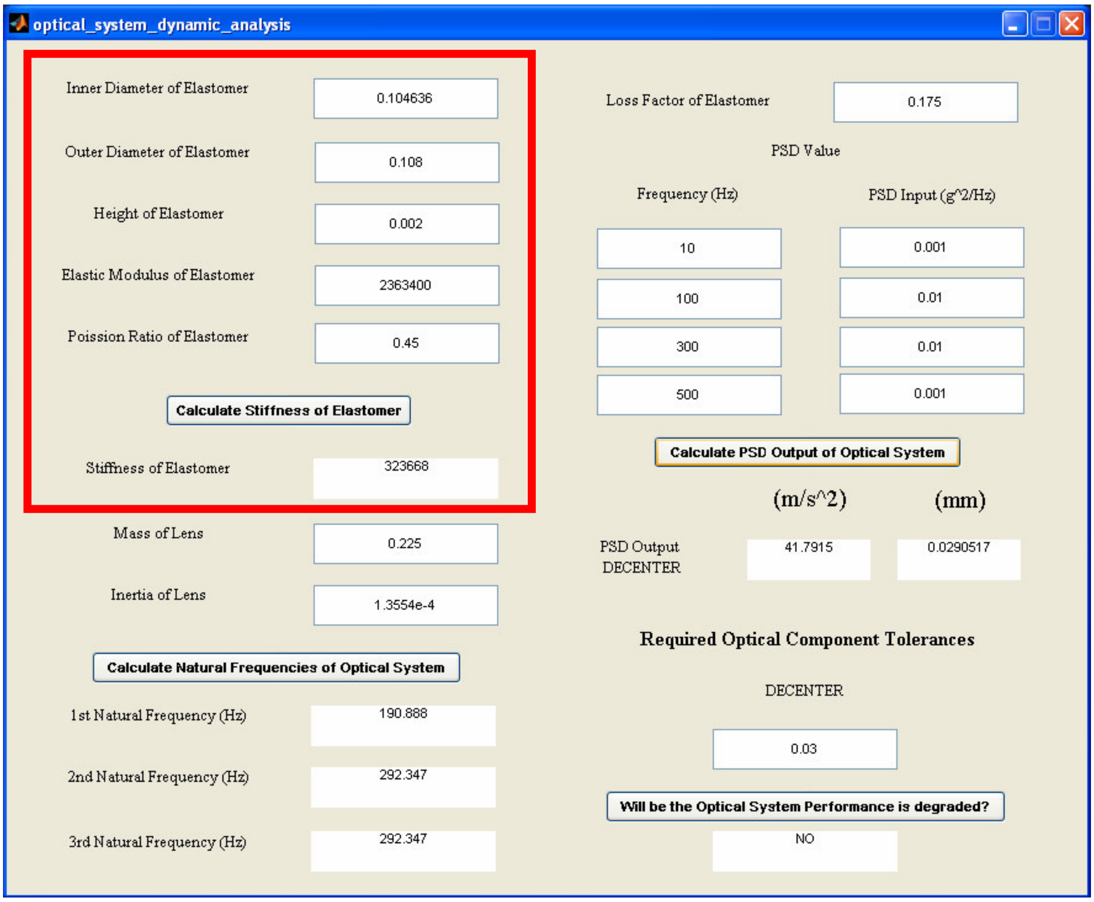


Figure 6. 2 Calculation of Stiffness of Elastomer

After, stiffness of elastomer is found; natural frequencies of optical system are calculated by using mass and inertia of lens (Figure 6.3).

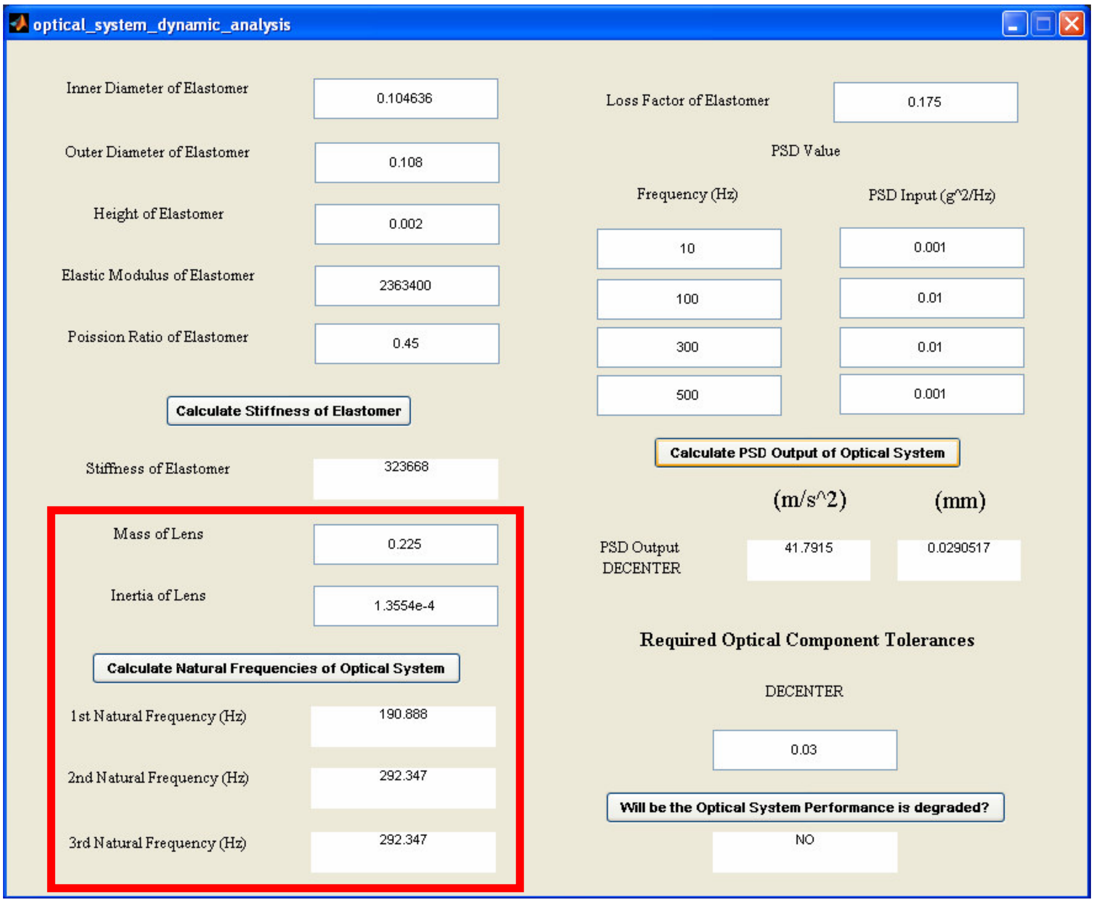


Figure 6. 3 Calculation of Natural Frequencies of Optical System

Finally, this program calculates the output power spectral density of the optical system, by using loss factor of elastomer and input vibration level, PSD value (Figure 6.4). Using output of this analysis, optical performance under vibration can be predicted and the results can be compared with optical element tolerances of decenter.

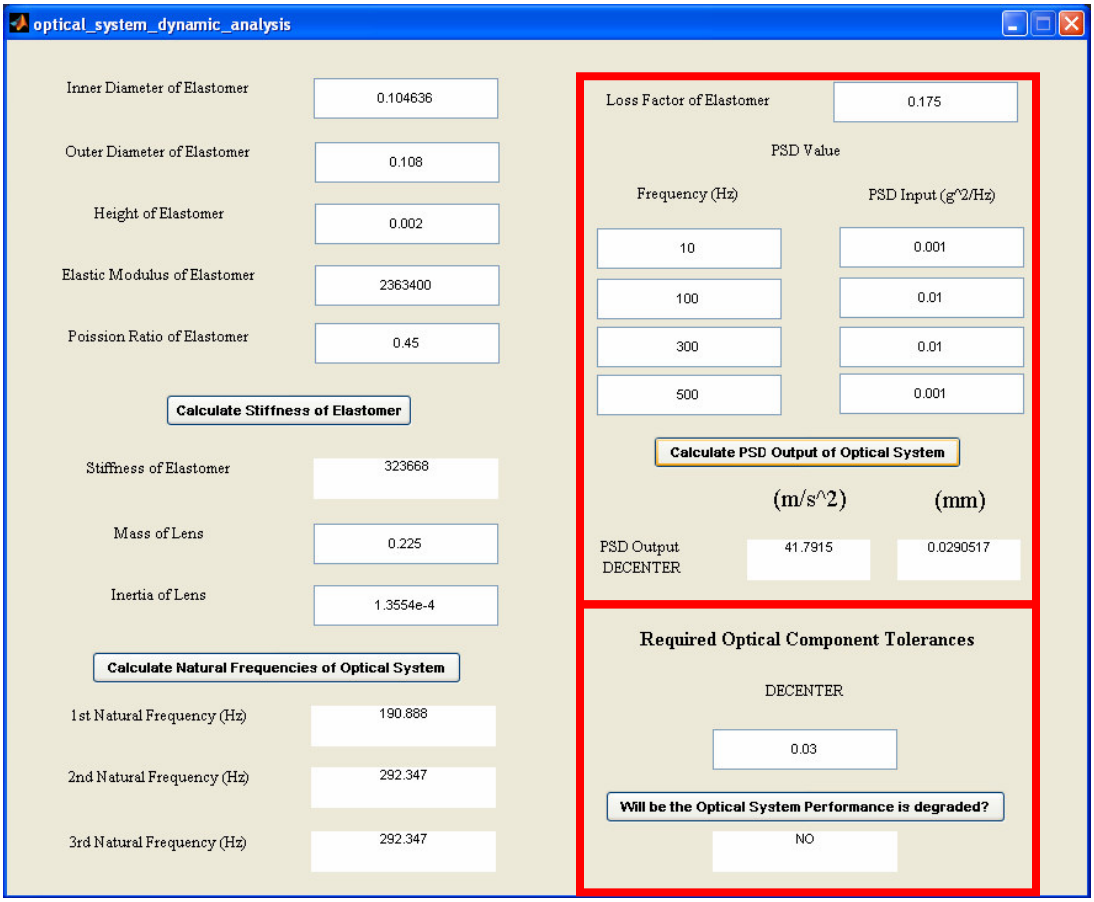


Figure 6. 4 Calculation of PSD Output of the Simple Optical System

A typical output of the program is depicted in Figure 6.5.

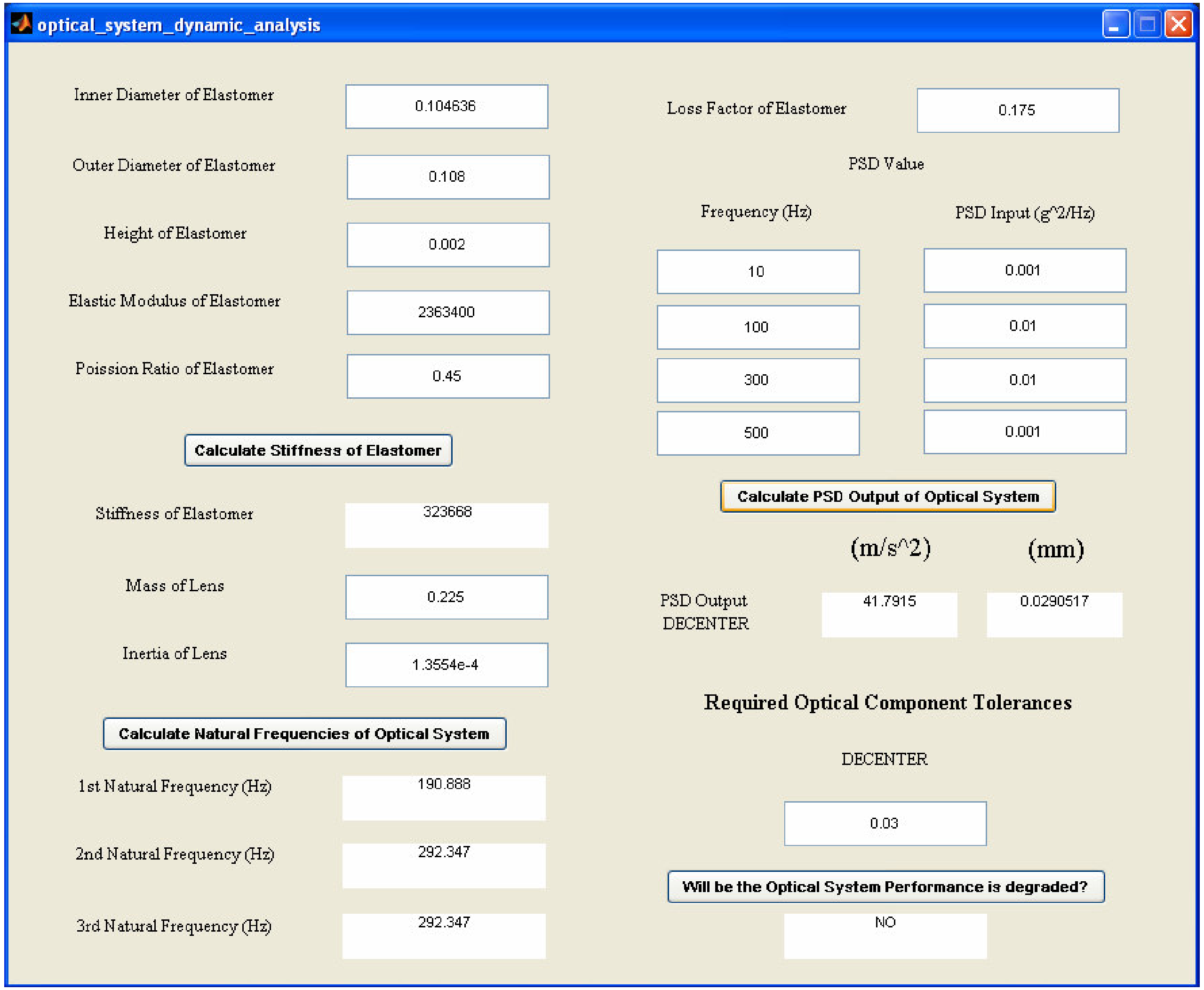


Figure 6. 5 Example for Analytical Model of a Simple Optical System

These two computer programs can be used for preliminary analysis by opto-mechanical design engineer in ASELSAN. User manuals of both programs are given in APPENDIX A.

First computer program can read and execute the test data, which is collected from test setup for determination of material properties of elastomer, after that it finds the material characteristic of elastomer. This program also calculates the modulus of elasticity of elastomer, which is used for a simple optical system, by predicting first natural frequency of optical system.

For the first stage of design of optical system, second computer program can be used and it can give ideas about dynamic characteristics of the optical system. Moreover, this computer program can predict the optical system performance under vibration. As explained before, optical element tolerance, which is decenter, is used and output of PSD analysis is checked whether optical element displacements is in the tolerance range or not. This procedure can be done automatically by the computer program.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Overview of Results and Conclusion

Dynamic characteristics of optical systems and vibration effects on their performance are investigated in this thesis. In order to understand the dynamic characteristic of an optical system, a simple analytical model and a finite element model are constructed and these models are validated by experimental techniques. Modal characteristics of an optical system and response of optical system under dynamic loading conditions can be found by using the simple analytical model developed. Also, by using this analytical model, a simple computer program, with a user friendly interface is developed in MATLAB for preliminary or quick opto-mechanical checks.

Military optical systems are integrated to military platforms, such as aircraft, battle tank, ship or helicopter, and they work under harsh environmental conditions. Therefore, they should be designed accordingly and also are qualified by considering military standards. The environmental conditions for military applications, such as temperature, humidity, vibration and shock etc., are given in MIL-STD-810.

Generally, in military standards, environmental conditions are given for most extreme cases. Therefore, if a platform, where an optical device is integrated, is decided, it is a better practice to collect the real environmental data from platform by experimental techniques. By using different types of sensors, vibration, temperature, humidity and shock data can be obtained. After processing this data, environmental conditions of the platform can be determined more accurately.

In this thesis, dynamic characteristic of an optical system are investigated for AH-1 helicopter vibration characteristic. First of all, dynamic characteristics of the optical system components are analyzed separately by using finite element models. In these finite element models, mesh size is determined by trial procedure not to affect the results of the finite element analysis and the trial procedure is not presented in this thesis. Natural frequencies of the afocal and the lens are found in free-free conditions. Then, finite element analysis results are validated by experimental modal analysis techniques. The finite element and experimental results are found to be very close to each other. First natural frequency of afocal (830 Hz) and lens (3292 Hz) are higher than the frequency range of interest for AH-1 helicopter, which is between 10-500 Hz. Then by using adhesive between lens and afocal, a dynamic model for an optical system assembly is constructed and dynamic analysis is carried out for the assembly. In this assembly, the first lens of optical system is used, because it is the biggest and the most powerful lens in the optical system. After dynamic analysis, first three natural frequencies of the optical system are found and it is observed that these natural frequencies are in the frequency range of interest. After observing that the case and the lens behave as rigid bodies, it is decided to construct a simple analytical model by considering the only adhesive flexibility.

The finite element results show that natural frequencies of the afocal and the lens used in this study are out of the frequency range of interest, however, in general, natural frequencies of the case and / or lens may be in the frequency range of interest because of geometry and material properties, i.e., natural frequencies of these parts may be lower than our example system due to the flexibility of the afocal or size of the lens. Moreover, the frequency range of interest may be high enough to cover the natural frequencies of the case and lens (For instance, frequency range may be 0-2000 Hz, unlike the frequency range of interest in our example). If case or lens can not be taken as rigid and mode shapes affect the lens movement, then a detailed finite element analysis should be carried out, or the analytical model results have to be verified by experiments, since the analytical model developed is valid only for systems where the case and lens can be taken as rigid.

The material properties of adhesive are determined at room temperature (25 °C) by using single degree of freedom system [16]; however, it is known that material properties of adhesive may change with changing temperatures due to visco-elastic material behavior. Therefore, experiments for the determination of material properties of adhesive should be conducted at different temperature values.

Experimentally measured material properties of adhesive are corrected by considering Poisson's ratio effects. It is explained that a correction is required in this experimental method, and the correction affects the modulus of elasticity dramatically. Moreover, in analytical model, shear modulus is calculated by using modulus of elasticity and Poisson's ratio. Therefore another correction factor is selected for the Young modulus of adhesive, which is used in optical system as thin hollow cylinder geometry. In Chapter 3, it is explained that because of the aspect ratio of test specimen and adhesive geometry in the simple optical system, there is no difference in correction values obtained from Figure 3.7, for Poisson's ratio between 0.45-0.50. Therefore, Poisson's ratio is assumed to be 0.45 in both the analytical and finite element models. Poisson's ratio of adhesive can not be determined by the single degree of freedom test method. In order to obtain more accurate results, Poisson's ratio of the adhesive should be determined by different experiments. Detailed information on these experiments can be found in references [28] and [29].

Using shear stress modeling, stiffness of adhesive is obtained. By using a three degrees of freedom model, where case and lens are taken as rigid, first three natural frequencies of the optical system are found, and then dynamic response of the optical system is obtained under random vibration loading. This simple analytical model is compared with finite element model and these discrete and finite element models are validated by experimental techniques. As it is seen in the results, finite element model gives more accurate results (maximum difference from experimental results is 7.3 %) than simple analytical model (maximum difference from experimental results is 11.4 %) as expected. Only shear effects are considered in the analytical model; if normal stresses are included in the model, analytical results would be more accurate.

However, the accuracy of the simple analytical model is still very good considering that even the finite element model yields 7.3 % error.

Sensitivity of the analytical and finite element model results to thickness variation of the adhesive can also be analyzed. For the different thickness values, the first natural frequency of the optical system is obtained by analytical and finite element models and these results are compared in Table 7.1. When thickness of adhesive increases, the difference between analytical and finite element results increase as can be seen in Table 7.1 and Figure 7.1. The reason is the fact that only shear stress effects are considered in the analytical model, and when thickness increases, normal stresses become important. Therefore, the simple analytical model may be improved by considering both normal and shear effects; however, this is not so easy for this circular geometry.

Table 7. 1 Thickness Variation of Adhesive

			1st Natural Frequency	
Outer Diameter (m)	Inner Diameter (m)	Thickness (m)	Analytical Result (Hz)	Finite Element Results (Hz)
0.106	0.104636	0.001364	298	285
0.107	0.104636	0.002364	227	209
0.108	0.104636	0.003364	191	162
0.109	0.104636	0.004364	168	141
0.110	0.104636	0.005364	152	121

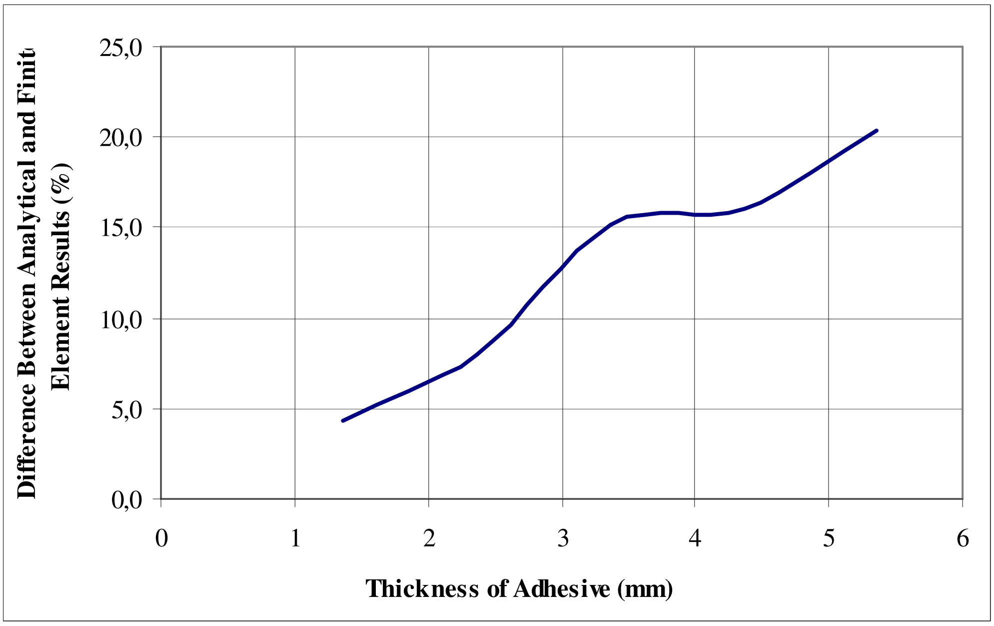


Figure 7. 1 Difference between Analytical and Finite Element Results

Optical parameters are analyzed in order to see the effect of dynamic characteristics of optical system on optical performance. There are different types of optical system parameters, some of these are related to dynamics of optical system, some of them are not. For example, index and V-number do not affects the optical performance under dynamic loading. In this thesis, only decenter and tilt tolerances are considered, because lens can move the direction of decenter and tilt in its housing under dynamic loading and this causes optical system performance degration. The optical tolerance analysis is done by only considering tolerance of decenter and tilt for a simple optical system, which consists only one lens. Tolerance analysis is statistical processes and it should be done by using different values of decenter and tilt, however, dynamic output of optical system is known and selected decenter and tilt values are chosen close to this outputs. If optical tolerance analysis is done before dynamic analysis, it is better to use statistical methods in order to obtain optimum tolerance values and this can be done by optical design programs.

Two computer programs are developed; one for the determination of material properties of adhesive and one for the dynamic characteristics of optical system. These programs are very simple and user friendly, and they can be used by design engineers easily. These two programs are written in MATLAB. First user interface is for finding material properties of an adhesive. It reads the experimental results in Excel file format and displays the modulus elasticity and loss factor of the adhesive. Second program is used to obtain the dynamic characteristics of an optical system once the geometrical and material properties of the optical system are given.

As the ultimate goal in this thesis is to determine the dynamic characteristics of an optical system and to see the effects of these dynamic characteristics on optical system performance, firstly the optical system and its components are analyzed in finite element program. When it is observed that dynamic characteristic of the optical system are affected only by adhesive flexibility, in order to find material properties of an adhesive, a simple test setup is designed. Then the effect of the aspect ratio of adhesive is analyzed to obtain more accurate results. In the simple analytical model suggested these material properties are used.

For the first stage of optical system design, determination of design parameters (considering dynamic characteristics of the optical system) can easily done by the analytical model developed. As can be seen in the results, analytical model gives good approximation for dynamic characteristics of the optical system. Therefore it will be useful for the preliminary analysis. Moreover, this analytical model is implemented on a computer program with a simple user interface, which is developed in MATLAB, to be easily used by design engineers.

7.2 Recommendations

In this thesis a simple analytical model is developed in order to find dynamic characteristics of an optical system. This analytical model is compared with finite element model and these two models are validated by experimental techniques.

Optical performance degration can be predicted under dynamic loading from the dynamic characteristics of optical system.

During the determination of the material properties of the adhesive, single degree of freedom modeling is used. If Figure 3.8 is investigated carefully, it is seen that, the system, rigid disk with adhesive, does not behave as a single degree of freedom model. Therefore, approximation to the material properties of the adhesive may not be good enough. In order to increase the accuracy of the results, accelerometer, which is used in this test, may be placed at the exact center point of the rigid disk and the rigid disk should be placed concentrically on the adhesive sample.

As it is explained in the literature, characteristics of elastomer type of materials change with temperature; therefore, elastomer material properties should be determined at different temperatures. In military standards, operational temperature range of the military devices is defined. For the extreme values of this range, the lowest and highest temperatures, the material properties of the adhesive may be found in order to understand the behavior of the adhesive. At these temperature values, the stiffness and damping of the adhesive may decrease which increases the decenter and tilt values under vibratory environment and may affect the degration of the optical performance. Moreover, Poisson's ratio is assumed to be 0.45; however Poisson's ratio of elastomer may vary between 0.45-0.5, so accurate values of Poisson's ratio should be determined experimentally in order to obtain more accurate analytical model. Furthermore, normal stress effects should be included into the analytical model in addition to the shear stress effects. By improving the analytical model in this way, difference between analytical and experimental results can be further reduced.

In this thesis, only random vibration input is considered in obtaining dynamic output of the optical system. Optical systems are also exposed to shock loadings in military environmental conditions. Therefore, analytical model may be improved by considering different dynamic loadings.

It is important that one should be careful in using the analytical model suggested. This analytical model is developed for an optical system where the case and the lens can be taken rigid compared to the rest of the system. If the stiffness of the case and the lens are high enough, then the analytical model developed can be used. If there is doubt about the relative flexibility of the case and lens, simple finite element analysis may be done for individual parts of the optical system: case, lens etc. Finite element analysis in ANSYS Workbench for individual parts are very easy and fast, but it could be difficult and complicated to make a similar analysis for the assembly itself due to the selection of contacts between individual parts. After this analysis, if the first natural frequencies of lens and case are not in the frequency range of interest, the simple analytical model suggested can easily be used for the system. If natural frequencies of the case is in the frequency range of interest but the mode shapes do not affect the lens movement, then again the simple analytical model can still be used for this system. If both lens and case are not rigid in the frequency range of interest and their mode shapes affect the optical characteristics, detailed analysis should be done by using finite element modeling, and this analysis has to be validated by experiments.

After material test, modulus of elasticity of the adhesive is predicted by using single degree freedom model of the simple optical system. The value of the young modulus is underestimated due to this reason. Therefore, by improving this prediction, deviation between experimental and theoretical results may be reduced. Moreover, in experimental modal analysis for the simple optical system, impact hammer and one bi-ax accelerometer are used. Mass of the accelerometer is 4.8 gram and mass of the lens is 225 gram. Considering the mass loading effect of accelerometer, experimental results may be corrected by removing the mass effect of accelerometer in experimental modal analysis. The difference between the theoretical and experimental results may also be reduced further by adding the mass of accelerometer to the analytical and finite element model.

During tolerance analysis, optical parameters should be decided by considering the dynamic characteristics of the optical system. For the simple optical system, it is seen that optical performance is degraded under the AH-1 Helicopter vibration profile. By increasing the stiffness of the adhesive, the natural frequencies of the simple optical system may be shifted out of the frequency range of interest. Decenter and tilt values can also be decreased by increasing the damping of the adhesive. The optical performance of the simple optical system may be improved by changing these two properties of the adhesive. Moreover, different type of lens mounts, such as elastomer with retainer or o-ring with retainer type, may be used to prevent the decenter and tilt motion of the lens. In this thesis, only decenter and tilt tolerances are considered due to the mode shapes of the simple optical system. For instance, if there is any deformation on the lens because of the first natural frequency, which may be in the frequency range of interest, surface tolerances should also be considered in tolerance analysis.

The two simple computer programs constructed in MATLAB may be written in Visual Basic in order to execute these programs without using MATLAB. Then these programs can be used even if MATLAB is not installed on the computer. Moreover, the visuality of these programs can be improved in Visual Basic.

REFERENCES

[1] Ed. A. Ahmad, Optomechanical Engineering Handbook, CRC Pres LLC, 1999.

[2] D. Anıl, Opto-Mechanical Design of Lens Mount, M.Sc. Thesis, Middle East Technical University, Department of Mechanical Engineering, Ankara, Türkiye, 2000.

[3] V. L. Genberg, K. B. Doyle, G. J. Michels, Sigmadyne, Inc., Optical Performance as a Function of Dynamic Mechanical Loading, SPIE-5178-4, 2003.

[4] S. Magarill, U. S. Precision Lens, Inc., Optomechanical Sensitivity and Tolerancing, 1999.

[5] P. R. Yoder, Jr., Mounting Lenses in Optical Instruments, SPIE Optical Engineering Press, 1995.

[6] G. E. Jones, S. Davall & Sons Ltd., High Performance Lens Mounting, SPIE Vol.0073 (1975), p. 9-17.

[7] T. S. Mast, P. I. Choi, D. Cowley, S. M. Faber, E. James, and A. Shambrook, University of California, Elastomeric Lens Mounts, SPIE Vol.3355 (June 1998), p. 144-151.

[8] P. Mammini, A. Nordt, D. Stubbs, Lockheed Martin Space Systems, B. Holmes, Jet Propulsion Laboratory, Sensitivity Evaluation of Mounting Optics Using Elastomer and Bipod Flexures, SPIE Vol. 5176 (2003), p. 26-35.

[9] Department of Defense Test Method Standard for Environmental Engineering Considerations and Laboratory Tests, MIL-STD-810F, Department of Defense, USA, 2000.

[10] ANSYS Classic 11.0 and ANSYS Workbench 11.0 Help Manual.

[11] PULSE 11.0 Help Manual.

[12] ME'Scope 4.0 Help Manual.

[13] Sylgard 577 Information about Dow Corning brand Adhesive/Sealants, Dow Corning, USA, 2006.

[14] Sylgard 577 Information about Electrical/Electronic Materials Dow Corning, USA, 2006.

[15] T. C. Bakiler, 1984, Experimental Determination of Complex Modulus of Elasticity of Viscoelastic Materials, M.Sc. Thesis, Middle East Technical University, Department of Mechanical Engineering, Ankara, Türkiye.

[16] G. O. Özgen, Design and Development of a Complex Shear Modulus Measurement Setup for Viscoelastic Materials, SAE 2005 Transactions Journal of Passenger Cars - Mechanical Systems (2006) 2638-2647.

[17] G. J. Michels, V. L. Genberg, Sigmadyne, Inc., Keith B. Doyle, Optical Research Associates, Finite Element Modeling of Nearly Incompressible Bonds, Proc. SPIE Vol. 4771, p. 287-295.

[18] D.E. Newland, An Introduction to Random Vibrations, Spectral & Wavelet Analysis, Longman, 1993.

[19] F. B. Beer, E. R. Johnston, Jr., Mechanics of Materials, McGraw-Hill Inc., 1992.

[20] D. S. Steinberg, Vibration Analysis for Electronic Equipment, John Wiley & Sons Inc., 1988.

[21] S. Gade, H. Herlufsen and H. Konstantin-Hansen, Bruel & Kjaer, How to Determine the Modal Parameters of Simple Structures, Application Notes 99/05.

[22] M. Bass, Editor in Chief, Handbook of Optics, Volume I, Fundamentals, Techniques, and Design, McGraw-Hill Inc., 1995.

[23] Lecture Notes, Optical Performance, PHYS 4220 Optical System Design Georgia Tech School of Physics, Spring 2004.

[24] W. J. Smith, Modern Optical Engineering, The Design of Optical Systems, McGraw-Hill Inc., 2000.

[25] Lecture Notes, Tolerancing in CODE V, Selected Topics from Introduction and Advanced Training & Other Materials, Optical Research Associates.

[26] ZEMAX – Engineering Edition – Optical Design Help Manual.

[27] MATLAB R2007b Help Manual.

[28] C. E. Stromeyer, Experimental Determination of Poisson Ratio, Proceedings of the Royal Society of London, Vol. 55, (1894), p. 373-383.

[29] H. W. Smith and R. E. Chapel, Poisson's ratio determined with strain rosettes, Experimental Mechanics, Vol. 9 Number 3, 2006, p. 140-141.

APPENDIX A

USER MANUAL

In this appendix, user manuals of two software developed are given. First program is for determination of material properties of adhesives and second program is used for finding the dynamic characteristics of an optical system.

A.1 Computer Program for Determination of Material Properties of Adhesives

The computer program is written by using MATLAB functions and user interface tool. General view of the computer program can be seen in Figure A.1.

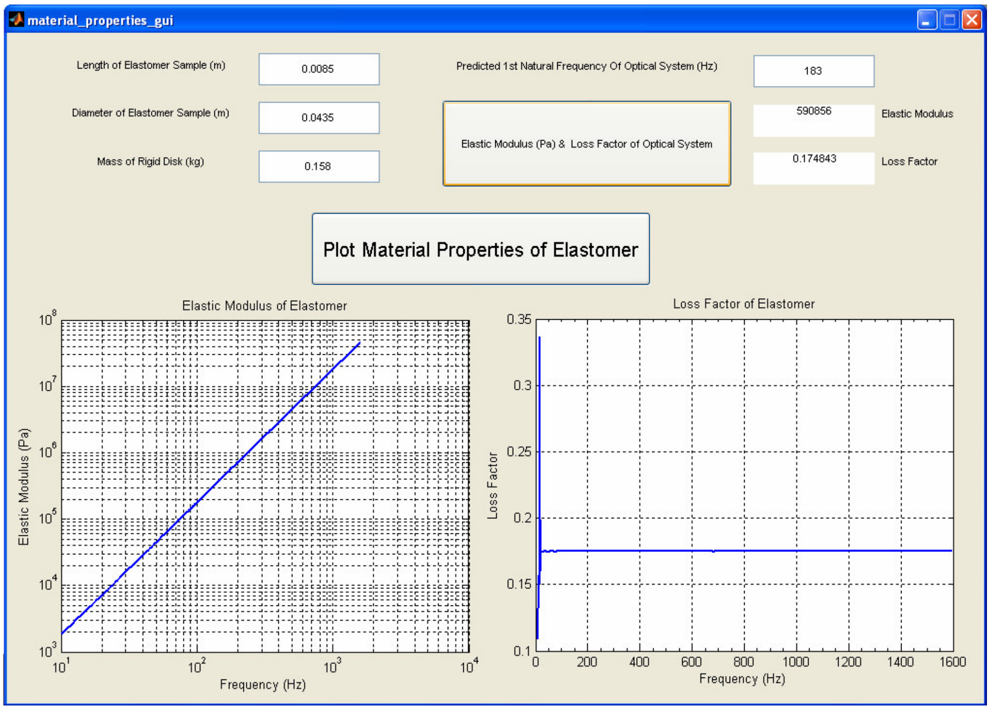


Figure A. 1 General View of User Interface of Computer Program

First of all, test results should be prepared in EXCEL format, which is explained in part 3.1, and this EXCEL file has to be in the working directory of MATLAB.

This software can be divided into three parts. Geometrical properties of elastomer sample and mass of the rigid disk are given in the first part, Figure A.2.

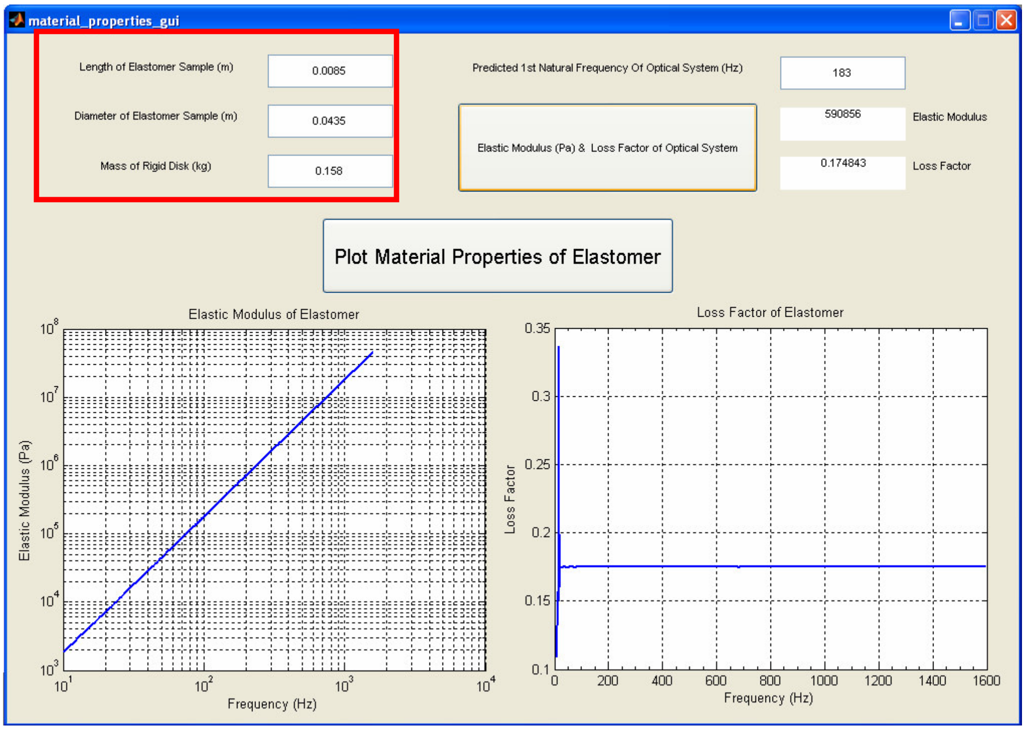


Figure A. 2 Geometrical Properties of Elastomer Sample and Mass of Rigid disk

In the second part of software, material properties of elastomer are calculated and plotted. Material properties of elastomer will be plotted as shown in Figure A.3.

Using these plots, mathematical model of elastic modulus can be found, and then using this mathematical model, the first natural frequency of the optical system can be predicted. After prediction of the first natural frequency of the optical system, elastic modulus and loss factor of elastomer can be found (Figure A.4).

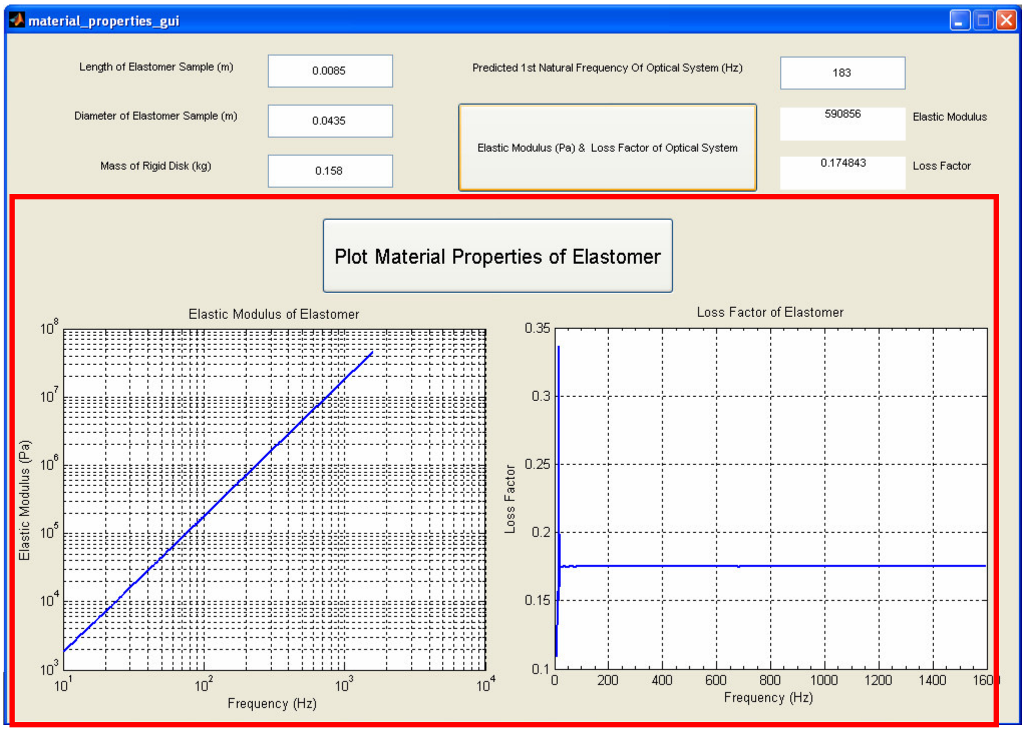


Figure A. 3 Plot of Material Properties of Elastomer

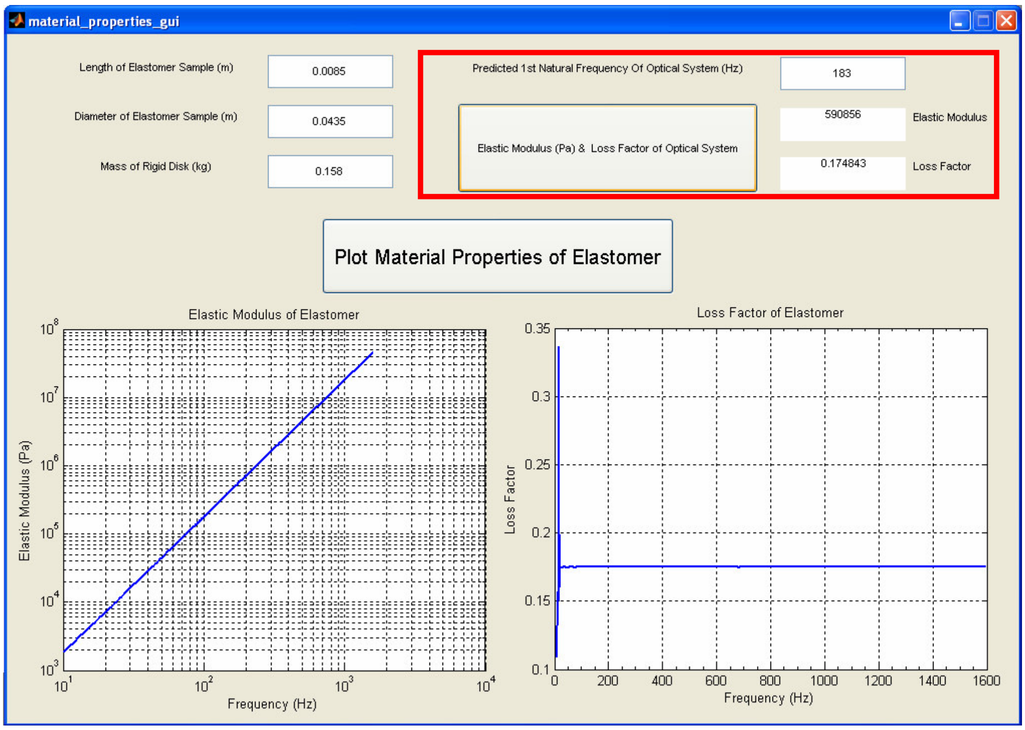


Figure A. 4 Elastic Modulus and Loss Factor of Elastomer

A.2 Computer Program for Determination of Dynamic Characteristics of Optical System

Dynamic characteristics of optical system can be found by using this computer program which is also developed in MATLAB. The analytical formulation given in Chapter 3 is used in this computer program. User interface of this program is shown in Figure A.5.

This computer program can be divided into four parts. In the first part of the computer program stiffness of elastomer is calculated. For this calculation, geometrical parameters, inner and outer diameter and height of elastomer are required. The material properties calculated in the previous computer program are also used in the computer program. An example of an output is shown in Figure A.6.

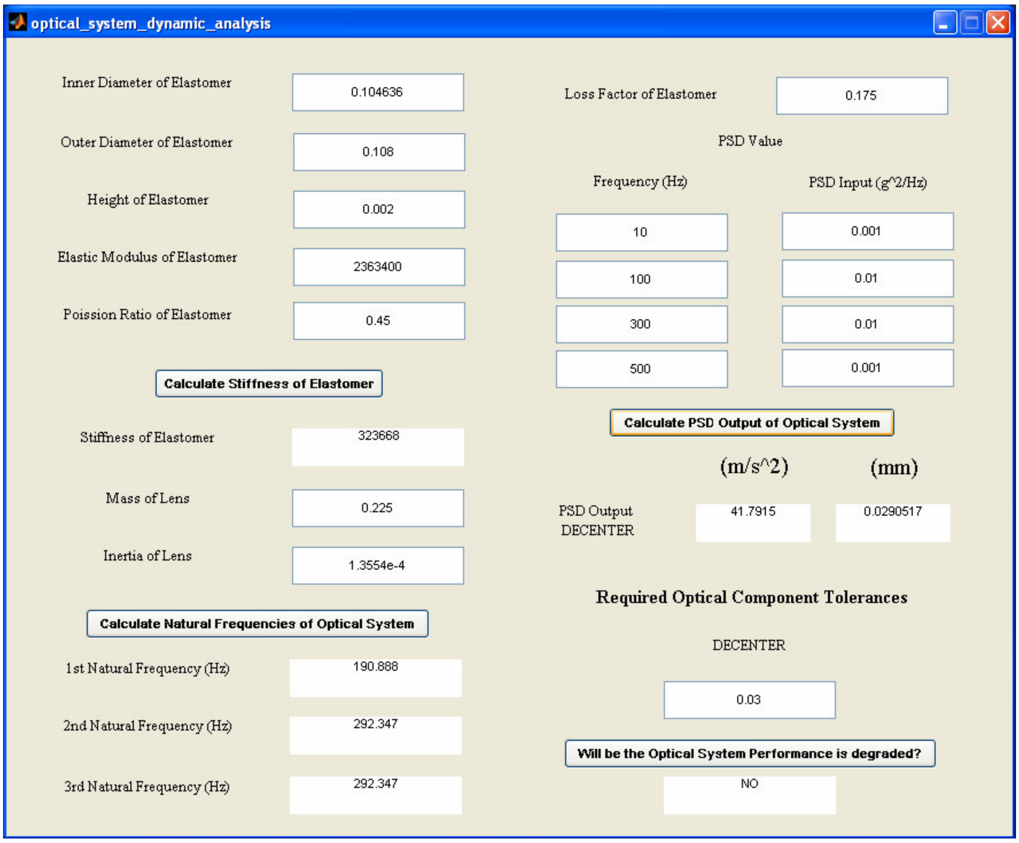


Figure A. 5 User Interface of Computer Program for Determination of Dynamic Characteristics of Optical System

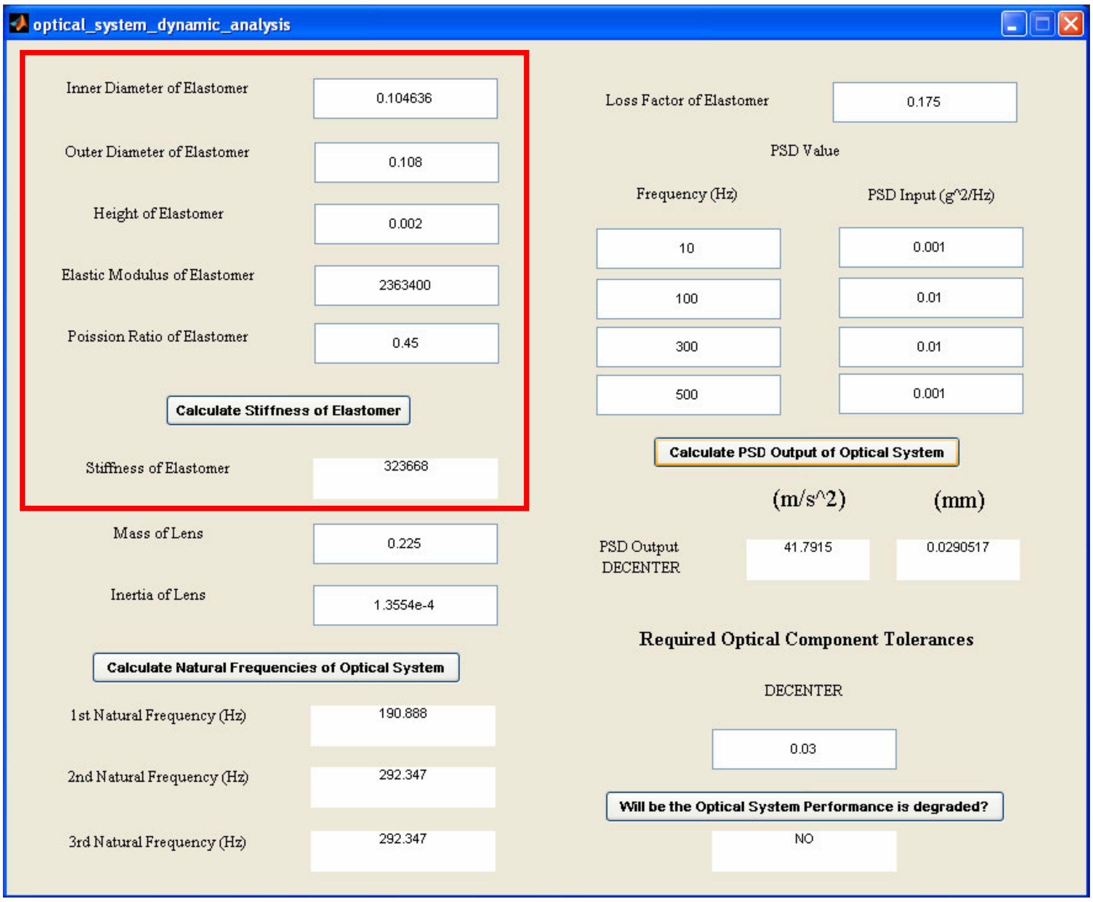


Figure A. 6 Calculation of Stiffness of Elastomer

In the second part of the computer program, by inserting mass and inertia of lens first three natural frequencies of the optical system can be calculated (Figure A.7).

Using loss factor of elastomer and PSD of the vibratory input, which is specified by military standards, random vibration analysis results can be obtained (Figure A.8).

Finally, optical system performance degration can be checked by inserting decenter and tilt tolerances, which are found from optical system design program (Figure A.9).

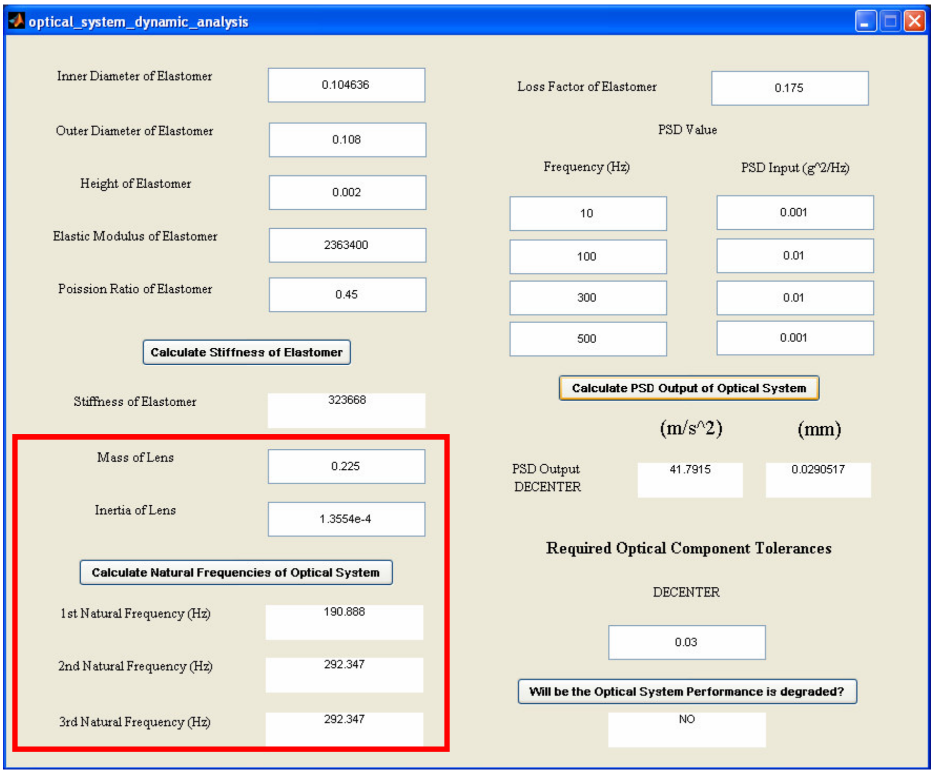


Figure A. 7 Calculation of First Three Natural Frequencies of the Optical System

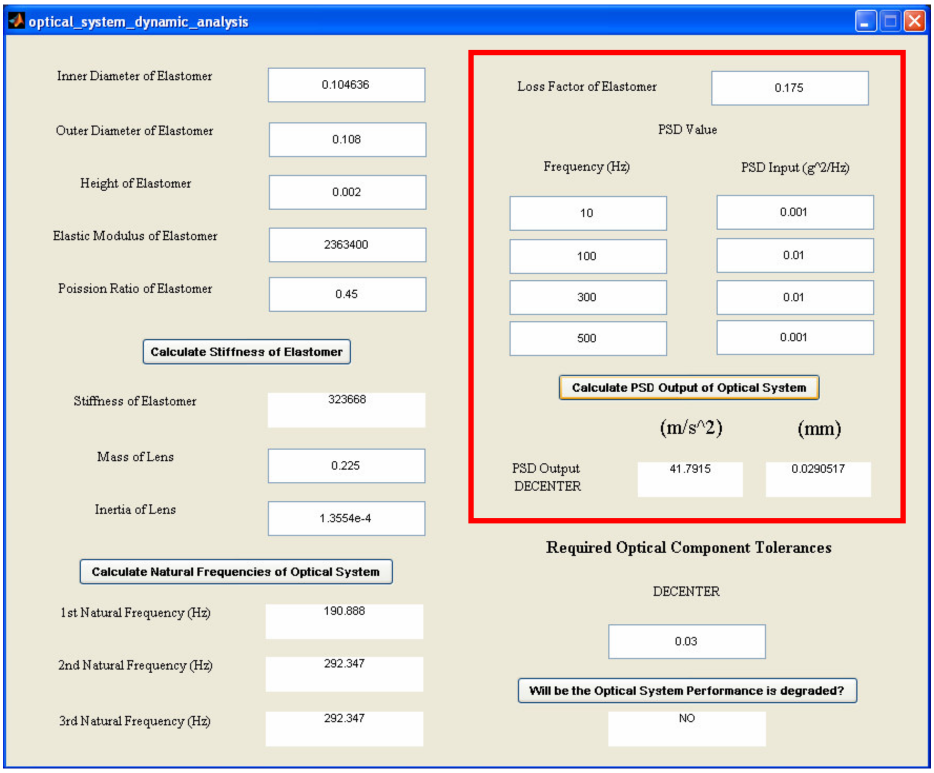


Figure A. 8 Calculation of Random Vibration Analysis Results

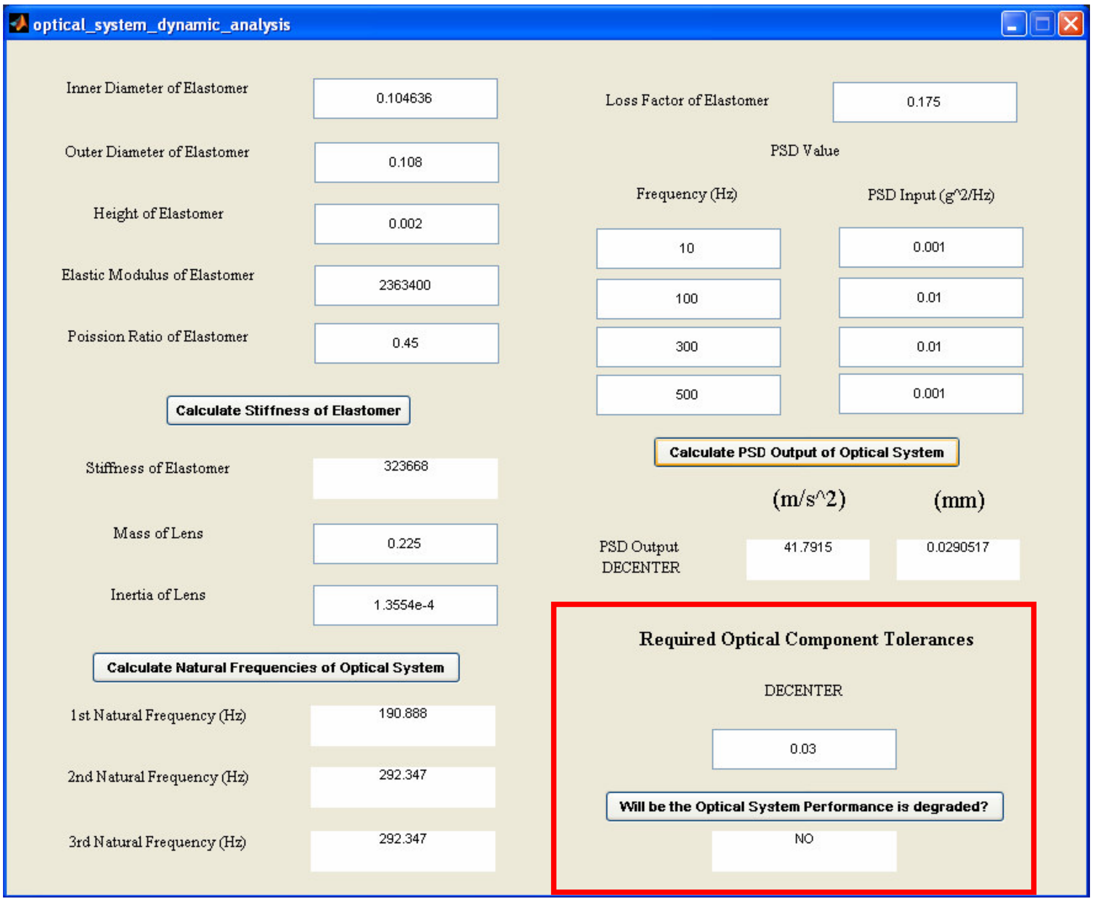


Figure A. 9 Optical Performance Checking