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Vibration Analysis of a Generator Anti-Vibration Rubber Mounts

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Abstract: The aim of this study is to measure the vibration and the stresses developed in the anti-vibration rubber mounts of a generator resulted from vibration of a generator at duty cycle. This is done by measuring the acceleration of the engine case using accelerometer; piezoelectric sensor. The stresses were calculated numerically by finite element analysis using ANSYS 15.0 software package.

Keywords: anti-vibration rubber mount, stress analysis, generator.

1. Introduction

Most of the Iraqi families use small generators to produce electrical power for their domestic purposes. In general, all types of generators consist of two parts, eelectrical part which is responsible for transforming the mechanical power to electricity and a mechanical part which is an IC engine and responsible for burning fuel to produce sufficient energy to rotate a shaft which in turn rotates the electrical part. During their work engines are usually accompanied by a great deal of vibrations and usually manufacturers tend to get rid of them by placing the whole engine on an anti-vibration rubber mounts which is shown in Figure (1). So, due to the dynamic load of the continuous vibrations these mounts fail after a period of time, consequently, there is a need to study the stresses induced at these rubber parts. Finite element analysis (FEA) was used for the analysis process which is a reliable and efficient analytical method. The rubber mounts used for this study is V-shaped generator mounts which is shown in figure (2) because this type is popular in local markets.

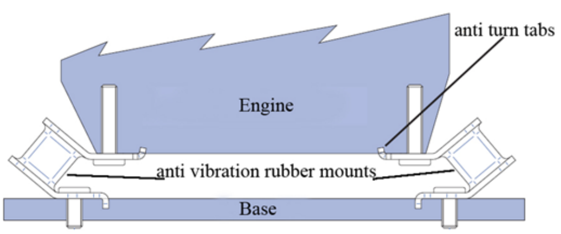

Figure 1. Assembly of the generator and anti-vibration rubber mounts

Figure 2. V-shaped anit-vibration rubber mounts

2. Experimental Work

In order to analyze the problem rubber properties must be measured in the v-shaped generator mount so as to substitute them in the FEA. Rubber substance in the mounts carries a generator of a mass (m) and absorbs engine vibration that is transformed to the base frame so it can be represented as v-shaped generator mount as mass-spring- damper system as shown in Figure (3). The system includes free vibration, the stiffness coefficient K and damping coefficient C for the rubber pieces empirically calculated [1].

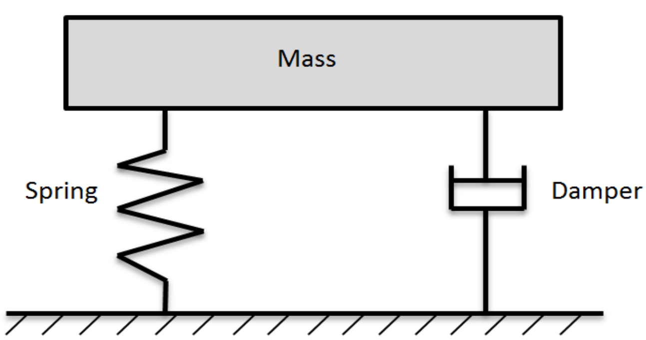

Figure 3. Mass-Spring- Damper system representation

3. Measuring Stiffness Coefficient

The rubber material was well executed from the v-shaped generator mount and tested by compression test to estimate the stiffness from the following equation

$k=F/X$ ……… (1)

Where k is the stiffness coefficient, F is applied load in N and X is the deformation in m. The compression device is illustrated in Figure (4) which is computer recorded data.

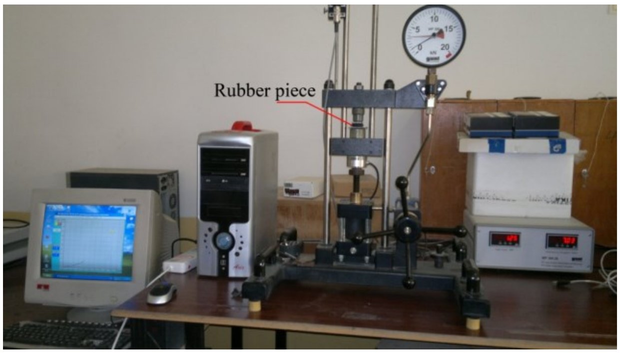

Figure 4. compression test device

k was calculated at elongation equal to 0.1 mm when the applied force is 0.06 KN. Stiffness was found equal to 600000 N/m.

4. Piezoelectric Accelerometer

The acceleration sensor used in this study is Accelerometer Type B & K 4371 shown in Figure (5). It measures acceleration in one dimension. The data measured by this sensor in millivolt. To make data in m/sec2 it must be multiplied by 10.2

Figure 5. B & K 4371 Accelerometer

5. Damping Coefficient Measurement

Damping coefficient C is calculated from the following equation:

$$C/M=2\zeta w_n \quad \ldots\ldots\ldots\ldots (2)$$

Where M is mass and ζ is damping ratio and w_n is the natural frequency. The parameters in equation (2) were estimated by the experiment shown in Figure (6). This experiment models the system shown in Figure (3) where the rubber material in the v-shaped generator mounts was released and joined together with a mass block of 1 Kg. An acceleration sensor is attached to the mass. The whole test instrument are shown in Figures (6) and (7).

Figure 6. Mass-spring-damper system modeling

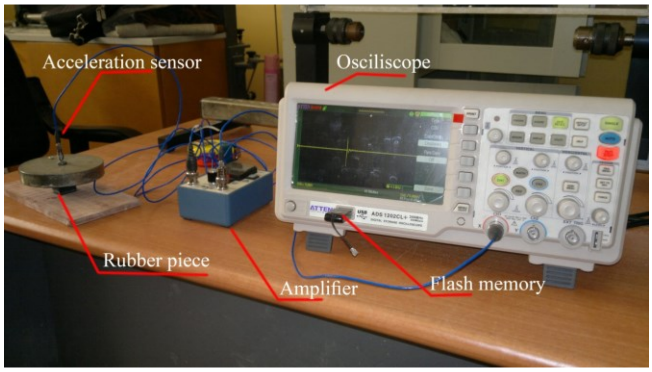

Figure 7. Damping ratio measurement system

Firstly, the damping ratio has to be calculated. The signal measured by this experiment, which is acceleration data, was noisy and therefor it was processed by Sigview software package to get rid of noise then multiplied by the correction factor of the piezoelectric accelerometer which is 10.2, the final signal shown in Figure (8)

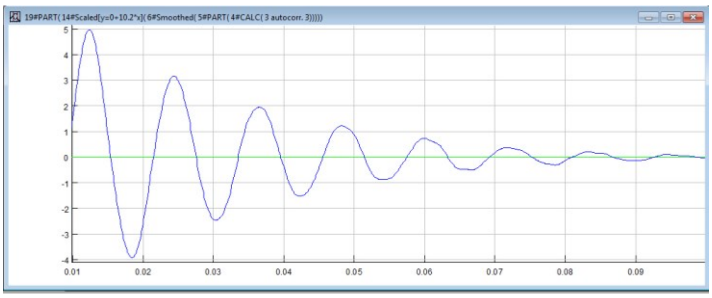

Figure 8. Acceleration vs Time

Acceleration curve then is integrated twice to produce velocity and displacement curves as shown in Figures (9) and (10)

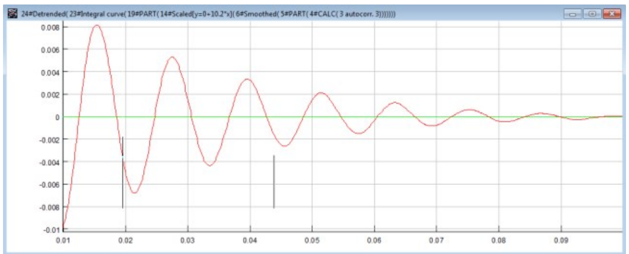

Figure 9. Velocity vs Time

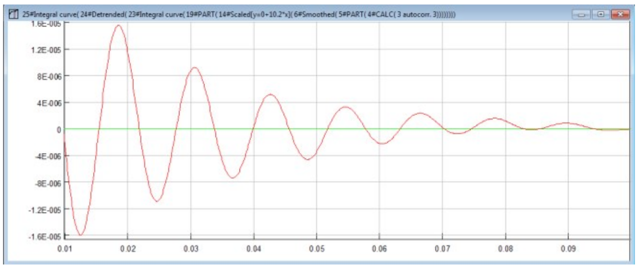

Figure 10. Displacement vs time

It was found by Sigview package that the first peak value in Figure (10), is $X_1 = 1.5587*10^{-5}$ and the second peak value is the curve $X_2 = 9.2479*10^{-6}$ so the damping ratio ζ is calculated from this equation

$$\Delta = ln\frac{X_1}{X_2} = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \quad \ldots\ldots\ldots\ldots..(3)$$

Where Δ is the logarithmic decrement. By substituting X_1 and X_2 values in (3) and solving for ζ, it was found that the damping ratio is equal to $\zeta = 0.462$, a second parameter is needed in equation the natural frequency which can be calculated from the following equation:

$$w_n = \sqrt{\frac{k}{M}} \quad \ldots\ldots\ldots\ldots\ldots\ldots.(4).$$

This leads to w_n= 774.59 Hz

So by substituting values of ζ and w_n in equation (2) the damping coefficient C can be calculated that it equal to C =715.73 N.S/m.

6. Initial Load Mastering

During operation of the generator the acceleration of the mechanical and electrical parts was measured to determine the force acting upon the v-shaped generator mounts. The mechanical and electrical parts weighted by a balance to estimate the mass of the mechanical and electrical parts are assumed to be one block. The mass was found to be equal to m = 32Kg

Thus the force acting upon v-shaped mounts can calculated from the equation:

$$F=m*a \ldots\ldots\ldots..(6),$$

where F is exciting force on the generator rubber mounts. Acceleration of the mechanical and electrical parts was measured in x,y and z axis by the accelerometer as illustrated in Figure (11).

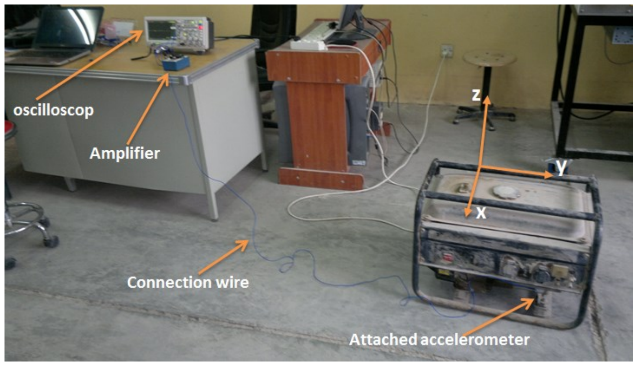

Figure 11. Measuring acceleration of the engine mass

The data gained by the piezoelectric sensor in millivolt is firstly multiplied by 10.2 to be in m/sec^2 by substituting it with the mass value of the mechanical and electrical parts of the generator in equation (6), so as to obtain the force excitation in a specific direction. Figures (12), (13) and (14) show the acceleration in x, y and z directions in the time domain.

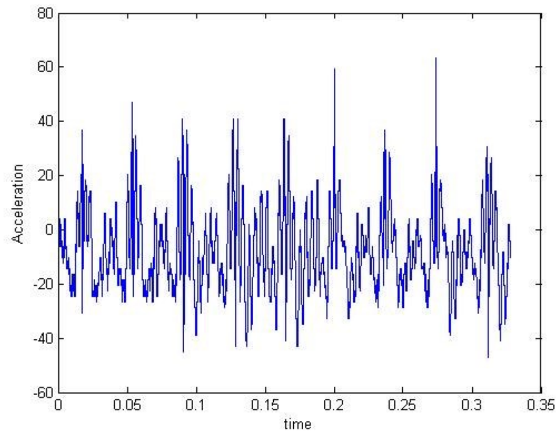

Figure 12. Acceleration in x-direction

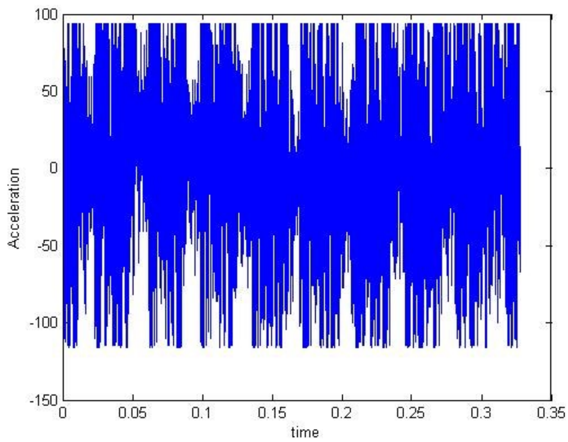

Figure 13. Acceleration in y-direction

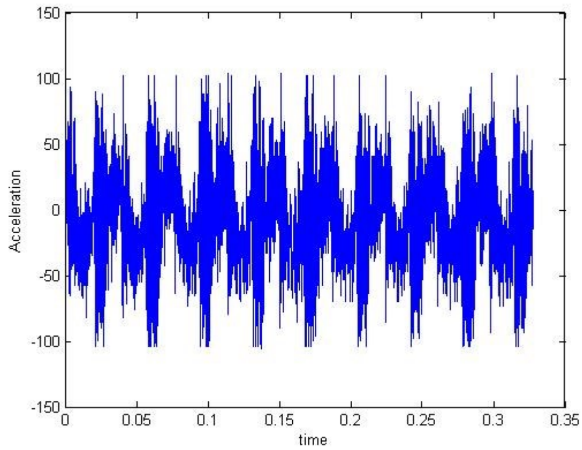

Figure 14. Acceleration in z-direction

The rubber properties and dynamic loads exerted on the mounts will be used as inputs for the FEA in ANSYS 15.0.

Transient response analysis in ANSYS needs force data input as a function of frequency not a time. So it should transform the measured acceleration from time domain to frequency domain, therefor, MATLAB program was built for this purpose and Fast Fourier Transform (FFT) [2] was used for the transformation process.

The resulting transformations to frequency domain are shown in Figures (15), (16) and (17) and frequencies data up to 2000 Hz were used to estimate remote forces in x, y and z directions.

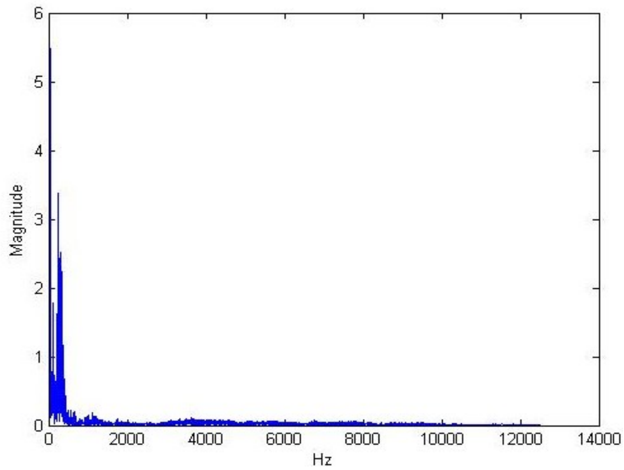

Figure 15. Acceleration in x-direction against frequency

7. Modeling the generator

Generator with a block of mass lied on a four v-shaped anti-vibration rubber mounts was modeled by Design modeler which is a package supports of 2D and 3D drawings to ANSYS solvers. All dimensions taken from real specimens (except the mass block which is assumed as a box of a mass) equal to the mass in real world of the entire block of the mechanical and electrical parts in the generator of type Astra shown in Figure (18).

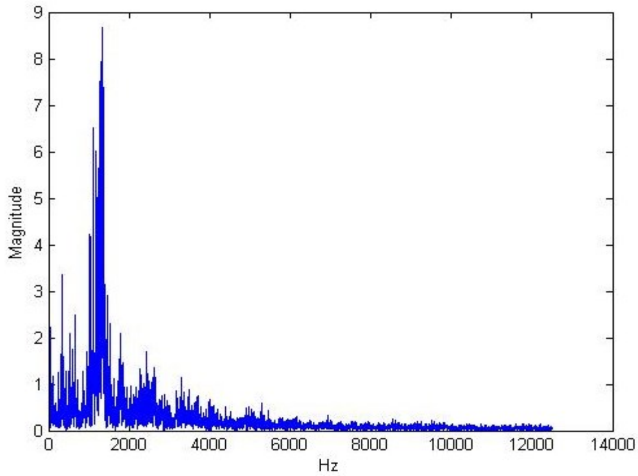

Figure 16. Acceleration in y-direction against frequency

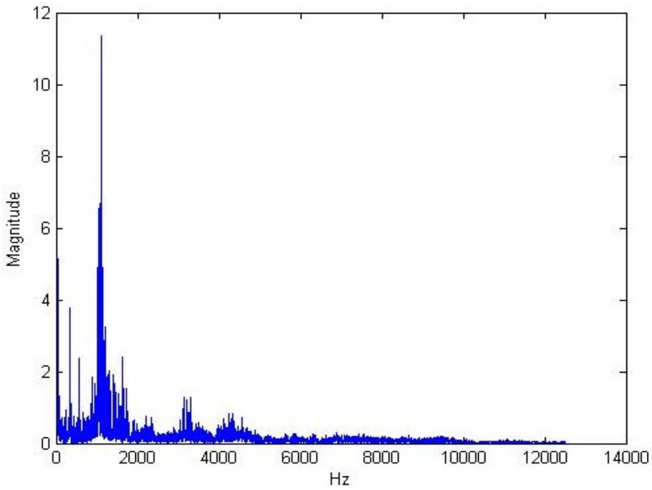

Figure 17. Acceleration in z-direction against frequency

Figure 18. Generator used for this study

The 3D model of the generator is illustrated in Figure (19).

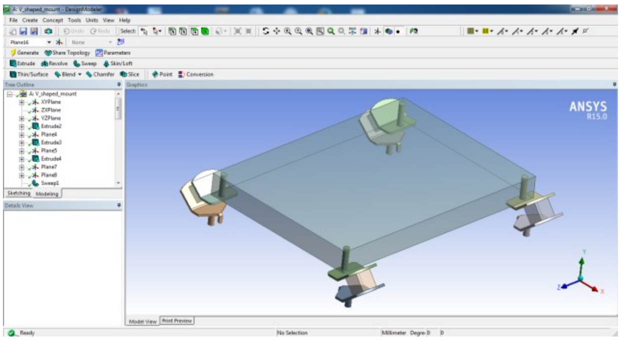

Figure 19. 3D model of the generator modeled using design modeler

8. Viscoelastic behavior of the anti-vibration rubber mounts

The rubber material in generator mounts behaves as a viscoelastic materials, this term is used usually for vibration damping problems. They are nonlinear materials that have both elastic and viscous behavior [3]. The simplest form for modeling viscoelasticity is Kelvin-Voight model [4] which is obtained by connecting a dashpot and spring in parallel and series. In this text the parallel mode was chosen which is illustrated in Figure (20).

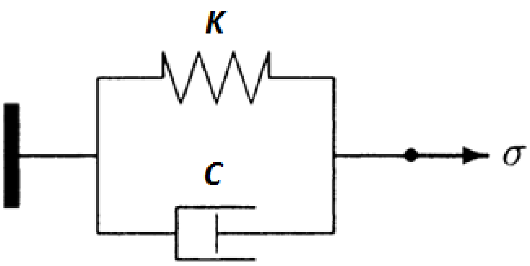

Figure 20. Kelvin-Voight model for viscoelasticity

9. Material selection for anti-vibration rubber mounts

Rubber was treated as viscoelastic material and its properties set in ANSYS were found out experimentally after defining spring element for each rubber mount in the model as shown in Figure (21). The properties were listed in the table below:

Table 1. rubber properties

Poisson ratio	ρ Kg/m3	K N/m	C N.S/m
0.3	1267	600000	715.73

And the standard structural steel material properties was set as shown in table (2).

Table 2. Standard structural steel properties

Young' Modulus Gpa	Poisson ratio	Density Kg/m^3
200	0.3	7800

10. Result and discussion

Transient response solver in ANSYS 15.0 was used to analyze anti-vibration rubber mounts. Mesh sizing set to 0.002 m with nodes count 78295 and elements count 28376. Force load was applied to the whole four mounts as shown in Figure (22) with frequency range of 10-2000 Hz. Areas in red in Figure (22) refer to the region effect of the force applied.

Figure (23) shows the Von-Mises stress induced during operation, while Figure (24) illustrates the maximum principal stress. Figures (25) and (26) show the Von-Mises strain and maximum principal strain respectively. It is found that the stress induced in the anti-vibration rubber mounts is less than the ultimate stresses for the mounts materials.

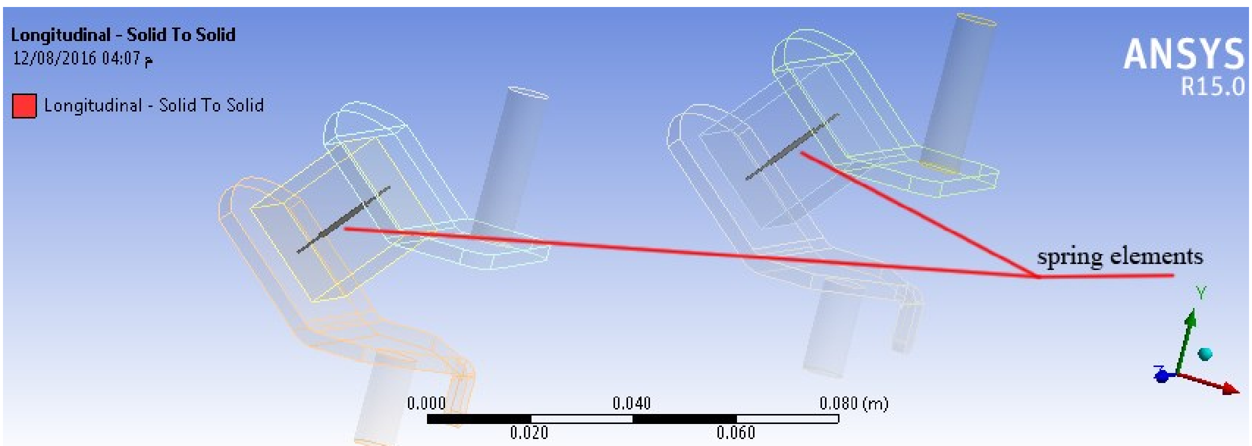

Figure 21. Spring elements

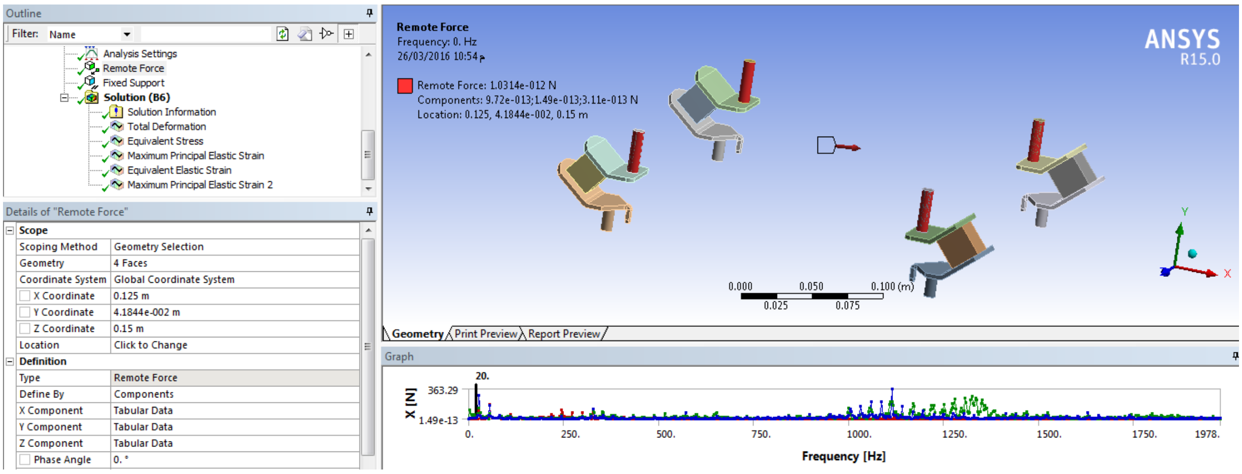

Figure 22. Regions where the force is applied

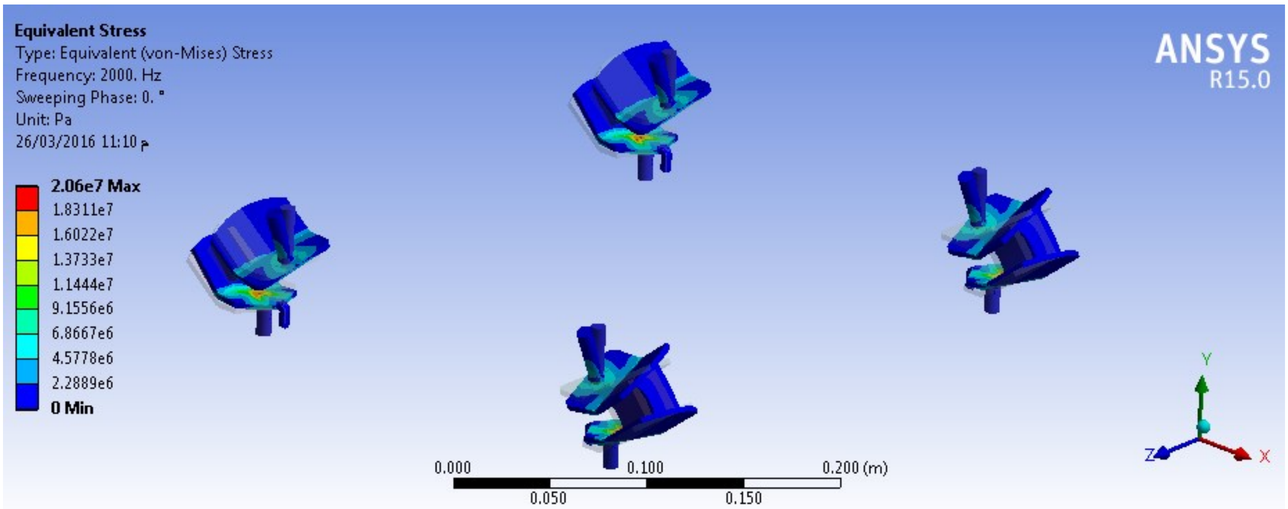

Figure 23. Von- Mises stress due to harmonic load (acceleration)

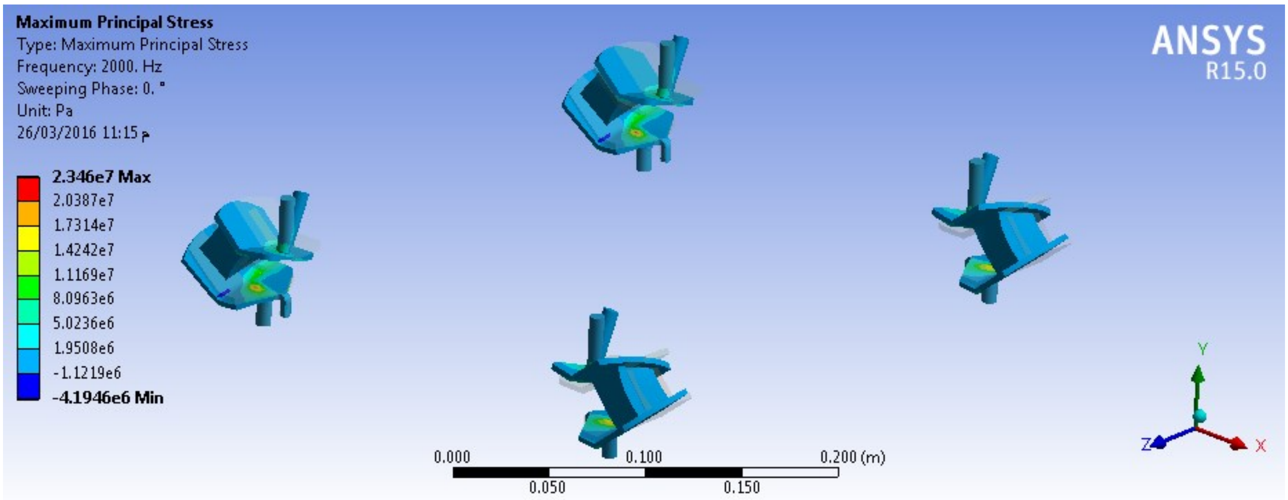

Figure 24. Maximum principal stress

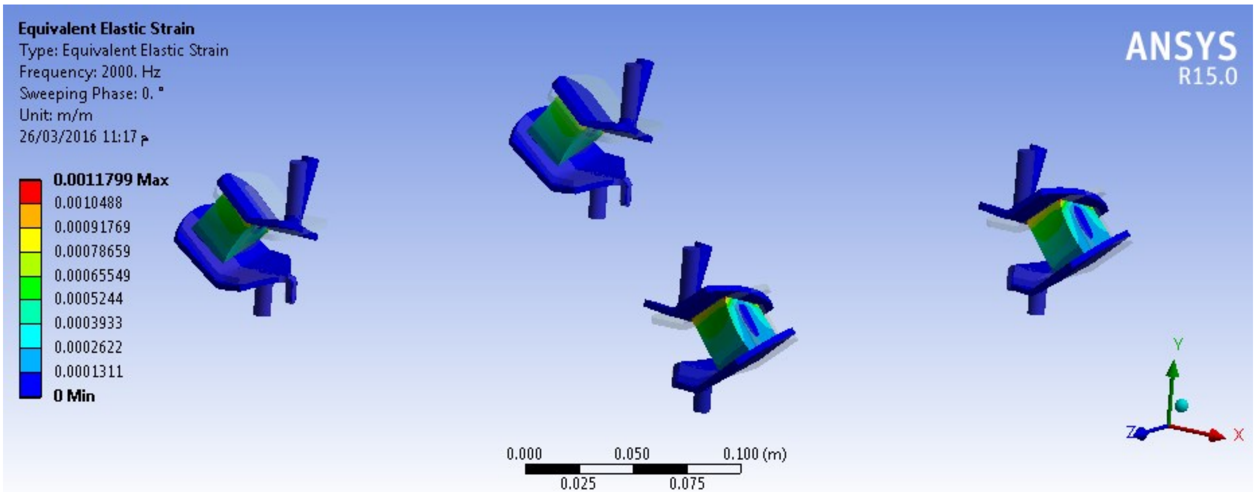

Figure 25. Von-Mises strain due to harmonic load

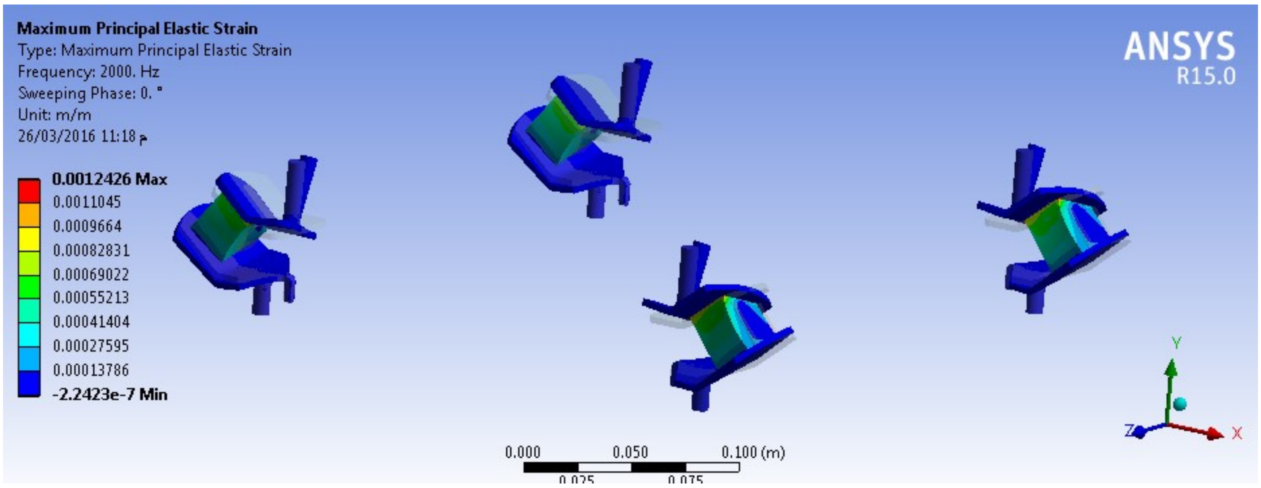

Figure 26. Maximum principal strain

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