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Research Article

Design and Analysis of Anti-vibration mount for G+3 Elevator

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Abstract

Motor and gearbox assembled together form a traction machine assembly and it mainly responsible for vibration and noise in elevator. In this project we aim at isolating the traction machine, cabin and structure with an anti –vibration mount. This mount design comprises between the conflicting requirements of acceptable damping and good isolation. ANSYS software helps to get approximate calculation of dynamic and static parameter of the mount and analytically justify the structure as stable.

Keywords: *The Model Analysis, Transient Analysis, Ansys R14.5, Equivalent Stress, Mode Shapes*

1. Introduction

In today's buildings, elevators are a necessary building component providing vertical circulation and access for residents in buildings with numerous floors. When used in residential facilities, the noise and vibration of the elevator operation can be a potential intrusion for residences that are adjacent to the equipment. The most significant effect can result in lower sound quality, disturbed sleeping conditions, and reduced enjoyment of the residences. To overcome the problem of vibration and noise various methods are used such as vibration damping, isolation, and vibration absorption. Dampers dissipate system energy, and vibration isolations prevent vibration transmission, while vibration absorbers transmit the vibration energy to a secondary system.

1.1 Vibration Isolators

A vibration isolator is defined by ISO standard 2041 (ISO, 1990) as ''a support, usually resilient, designed to attenuate the transmission of vibration in a frequency range''. A vibration isolator typically comprises a resilient element attached to a mounting plate at each end. Resilient elements include rubbers, elastomers, polymers, metal springs, corks, felts and air bags. The dynamic properties (stiffness and damping) of the isolator determine the level of the transmission of vibration through an isolator. The stiffness of a material represents its ability to resist deformation. Stiffness is commonly characterized by slope of the linear region of a stress–strain curve. Stiffness can be represented as

$$k = \frac{\mathrm{dF}}{dX} \tag{1}$$

Where k is the stiffness constant in Nmm, F is the change in force in N, and X is the deflection in mm. Stiffness can be used to estimate both the natural frequency and isolation effectiveness of a lightly damped isolation system made of neoprene, natural rubber or similar materials.

1.2 Vibration isolation Concept

Vibration is isolated by placing properly chosen isolation materials between the vibrating body and the supporting structure. The effectiveness of isolation is measured in terms of the force or motion transmitted at the point of exposure from the source. The first type is known as force isolation and the second type as motion isolation.

The less force or motion transmitted the greater is the isolation. The isolator should support the vibrating system in a static state, prevent its bounce from shock excitation and isolate vibration disturbances in the complete frequency range.

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1.3 Rubber Properties

Rubber is a unique material that is both elastic and viscous. Rubber parts can therefore function as shock and vibration isolators and/or as dampers. Although the term rubber is used rather loosely, it usually refers to the the compounded and vulcanized material. In the raw state it is referred to as an elastomer. Vulcanization forms chemical bonds between adjacent elastomer chains and subsequently imparts dimensional stability, strength, and resilience.

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Rubber has a low modulus of elasticity and is capable of sustaining a deformation of as much as 1000 percent. After such deformation, it quickly and forcibly retracts to its original dimensions. It is resilient and yet exhibits internal damping. Rubber can be processed into a variety of shapes and can be adhered to metal inserts or mounting plates. Rubber will not corrode and normally requires no lubrication.

One of the dynamic properties of rubber includes viscoelasticity:

Rubber has elastic properties similar to those of a metallic spring and has energy absorbing properties like those of a viscous liquid. These viscoelastic properties allow rubber to maintain a constant shape after deformation, while simultaneously absorbing mechanical energy.

2. Selection of materials

The material used throughout the structure was mild steel.

2.1 Mild steel

Steel is made up of carbon and iron, with much more iron than carbon. In fact, at the most, steel can have about 2.1 percent carbon. Mild steel is one of the most commonly used construction materials. It is very strong and can be made from readily available natural materials. It is known as mild steel because of its relatively low carbon content.
Mild steel has a high resistance to breakage. It has high tensile and impact strength. Higher carbon steels usually shatter or crack under stress, while mild steel bends or deforms. Considering the physical properties and requirements of our concern the rubber which we have selected is Butadiene rubber.

2.2 Butadiene rubber (chemical composition-cis. Polybutadiene)

It has excellent abration resistance, good tear and oxidation resistance. It also high Resilience. Rubber has a low modulus of elasticity and is capable of sustaining a deformation of as much as 100 percent. After such deformation, it quickly and forcibly retracts to its original dimensions.
Other important properties were:
Durameter range- 30-90.
Tensile max,psi- 3000.
Elongation max,% -650.

3. Structural Model

In order to conduct the analysis, both the geometry of the structure and the actions and support conditions are idealized by means of an adequate mathematical model, which must also roughly reflect the stiffness conditions of the cross-sections, members, joints and interaction with the ground. The structural models must allow considering the effects of movements and deformations in those structures or part thereof, where second-order effects increase the effects of the actions significantly.

Four rubber mountings made of butadiene rubber (BR) were selected .The properties of the isolator depends upon the application of load ,hence weight of engine is considered for selecting them. We have taken the four base plates of size 160×160 mm having four holes at the corner to it to fix it on the structure of the elevator ,above it resides four square channels of height 100 mm and width 45 mm. Two C-channels of length 1020 mm are welded on each two square channels .On the four corners of C-channel four columns with inclination at top are welded. four rubber mounts are mounted with that inclined faces, two rectangular plate with collar resting on rubber mount are welded. Two rectangular channels are used to support the Rectangular plates. At the top two C- channels are welded. The CATIA v5 was used to make the basic Model of the structure with respect to corresponding dimensions:

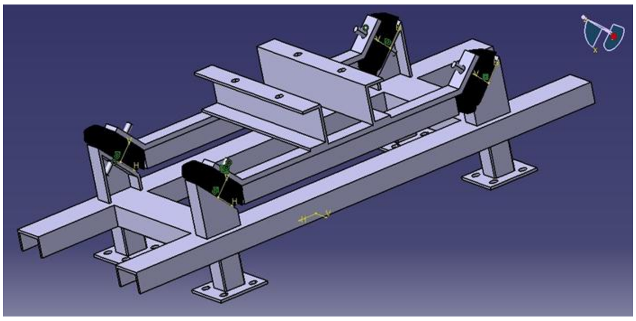

Fig.1 Structural model

4. ANSYS Analysis

ANSYS was used to pre determine the basic two important parameter analysis:
MODAL ANALYSIS, TRANSIENT ANALYSIS

Initially the mount structure is inserted in ansys R14.5 and is futher underwent Meshing as initial stage

4.1 Meshing

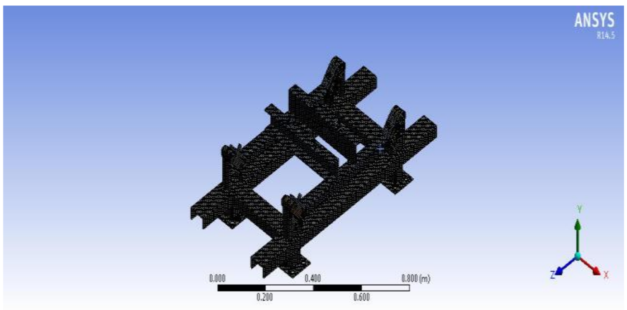

Fig.2 Meshing

It is also called as Discretization.Domain is discredited into a finite set of control volumes or cells. The discredited domain is called the grid or the mesh. General conservation (transport) equations for mass, momentum, energy, etc., are discredited into algebraic equations.The mesh used for our consideration was Quadra-mesh to minutely analyze the mount structure.

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Table 1. Mesh Properties

Object Name	**Mesh**
Sizing	
Relevance Center	Coarse
Element Size	5.e-003 m
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	2.0576e-003 m
Inflation	
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Statistics	
Nodes	316168
Elements	66062

4.2 Stress Analysis

By stress analysis, minimum and maximum normal and shear stress values occurring at the fracture surface during operation is investigated. At first, forces and torques acting on the mount are determined. By analyzing minimum stress value, only the weight of the empty cabin (420 kg) and balance weight (310 kg) is considered. Shear forces, caused due to the loads of empty cabin, ropes and balance weights, forms a shear stress of 7.69 MPa. By analyzing maximum stress value, balance weight, cabin weight with four persons inside (each person is 80 kg and the total weight of the cabin is 740 kg), torsion moment and impact ratio is considered.

4.3 Modal Analysis

We can use modal analysis to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. It also can be a starting point for another, more detailed, dynamic analysis, such as a transient dynamic analysis, a harmonic response analysis.We can do modal analysis on a pre-stressed structure, such as an anti vibration mounts. Another useful feature is modal cyclic symmetry, which allows you to review the mode shapes of a cyclically symmetric structure by modeling just a sector of it. It allows the design to avoid resonant vibrations or to vibrate at a specified frequency. Gives engineer an idea of how the design will respond to different types of dynamic loads.

We applied the boundary conditions and imposed the structure to vibrate at various frequency, it was considered that the frequency at which there would be major deformation ,it would be its natural frequency.

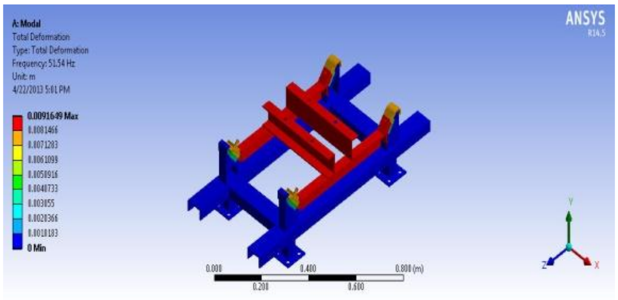


Fig 3 At 51.54 Hz

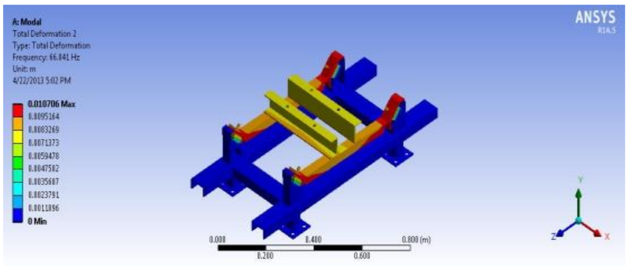


Fig 4 At 66.84 Hz

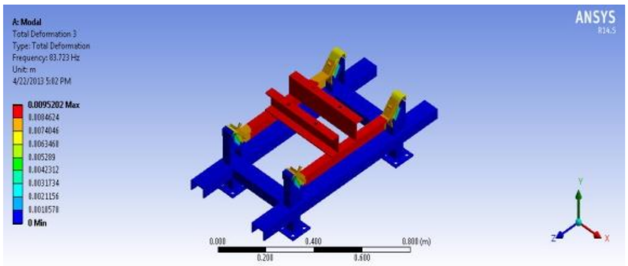


Fig 5 At 83.723 Hz

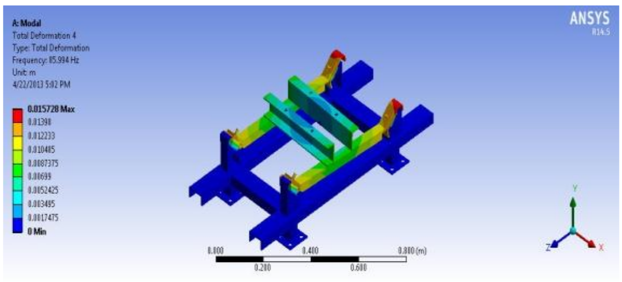


Fig 6 At 85.944 Hz

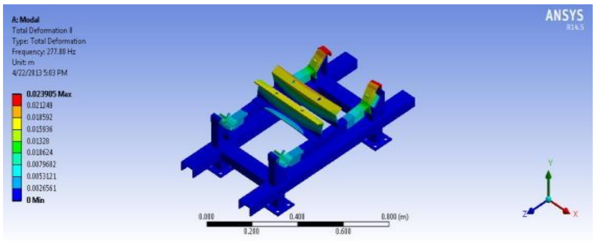


Fig 7 At 277.88 Hz

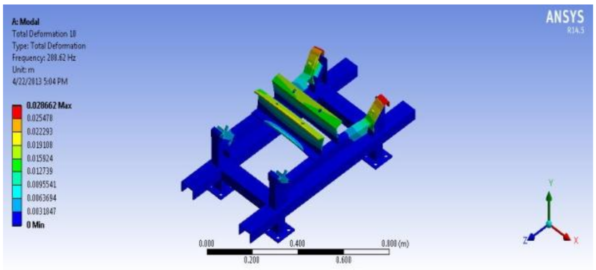


Fig 8 At 288.62 Hz

Table 2 Frequency at each calculated Mode

Mode	Frequency [Hz]
1	51.54
2	66.841
3	83.723
4	85.994
5	93.558
6	167.04
7	178.68
8	277.88
9	285.43
10	288.62

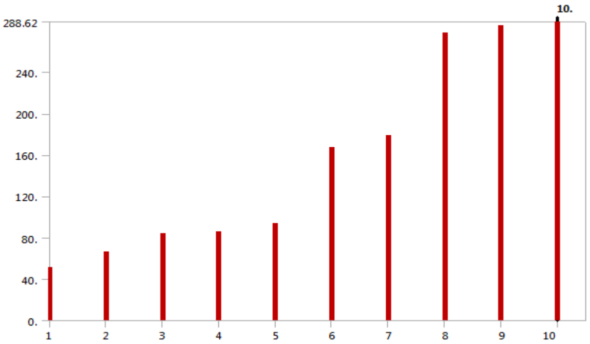


Fig 9 Frequency at each calculated Mode

We did model analysis to find natural frequency of anti vibration mount structure. Following figures shows total deformation in mount structure for each frequency. The frequency at which deflection of structure is maximum gives the natural frequency of the structure of anti vibration mount. In this case maximum deflection of structure shows in fig. 8 at frequency of 288.62 Hz.Thus to avoid resonance ,te mating of natural frequency should be avoided.

4.4 Transient Analysis

Transient dynamic analysis is a technique used to determine the dynamic response of a structure under a time-varying load. The time frame for this type of analysis is such that inertia or damping effects of the structure are considered to be important.Cases where such effects play a major role are under step or impulse loading conditions, for example, where there is a sharp load change in a fraction of time. If inertia effects are negligible for the loading conditions being considered, a static analysis may be used instead. To perform transient analysis of anti vibration mount we have applied force of 3075.8N on two patches one C channel.

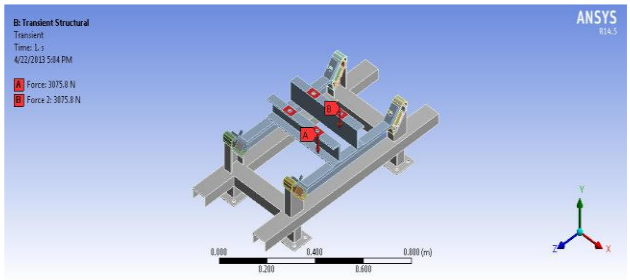


Fig 10 Transient Structural analysis showing applied force

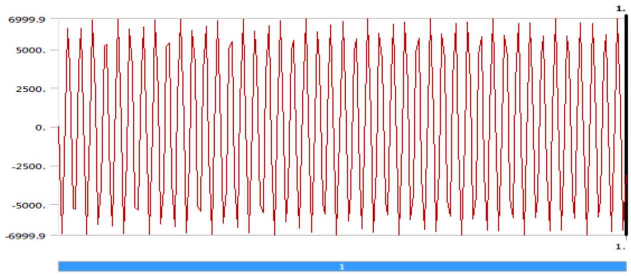


Fig 11 External exciting force on anti vibration mount

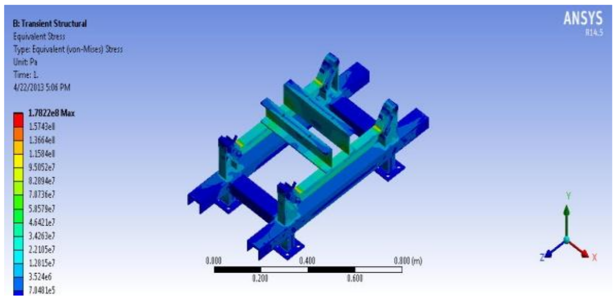
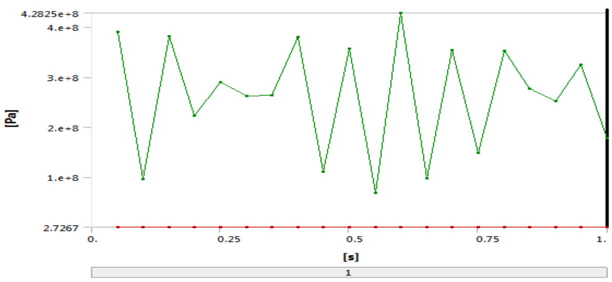


Fig 12 Transient Structural analysis for equivalent stress



Load Vs Time(for Equivalent stress)

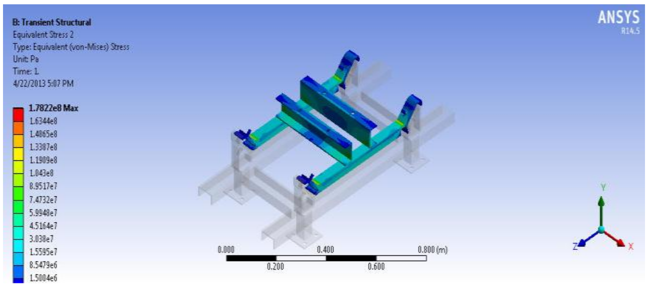
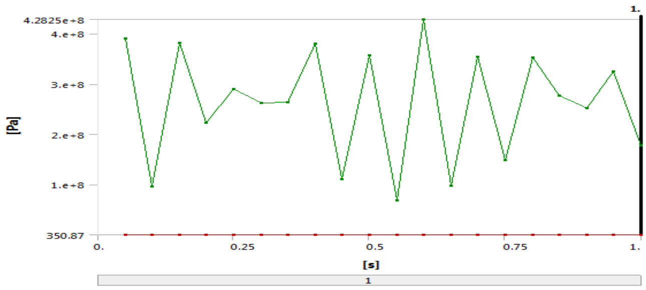


Fig 13 Equivalent stress analysis of jointed upper structure



Load Vs Time (for jointed upper structure)

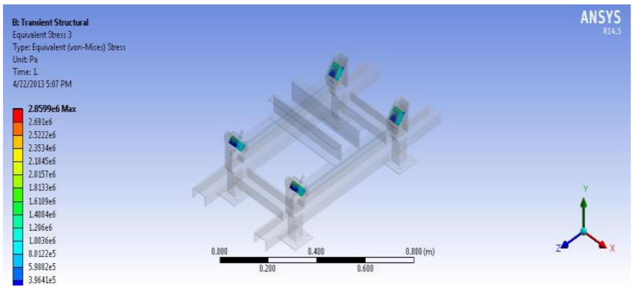
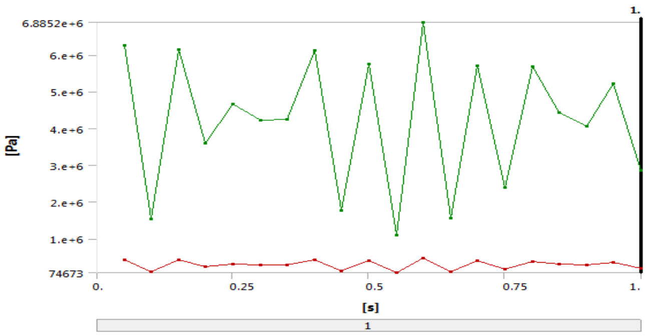


Fig 14 Equivalnt stress analysis for rubber pad



Load Vs Time (for rubber pad)

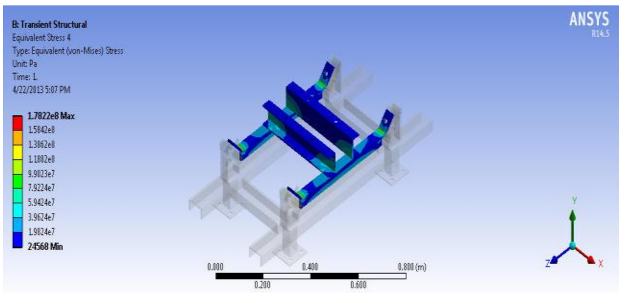
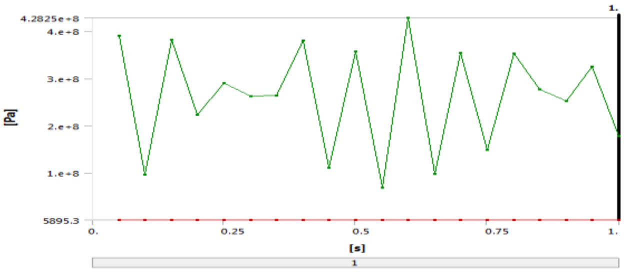


Fig 15 Equivalent stress analysis of upper structure



Load Vs Time(for upper structure)

Object Name	**Equivalent Stress**	**Equivalent Stress 2**	**Equivalent Stress 3**	**Equivalent Stress 4**
Definition				
Type	Equivalent (von-Mises) Stress			
By	Time			
Results				
Minimum	6.4944 Pa	948.6 Pa	1.94e+005 Pa	24568 Pa
Maximum	1.7822e+008 Pa		2.8599e+006 Pa	1.7822e+008 Pa
Minimum Occurs On	10		RUBBER_PAD	
Maximum Occurs On	1		RUBBER_PAD	
Minimum Value Over Time				
Minimum	2.7267 Pa	350.87 Pa	74673 Pa	5895.3 Pa
Maximum	17.066 Pa	2301.9 Pa	4.6918e+005 Pa	58192 Pa
Maximum Value Over Time				
Minimum	6.7996e+007 Pa		1.0937e+006 Pa	6.7996e+007 Pa
Maximum	4.2825e+008 Pa		6.8852e+006 Pa	4.2825e+008 Pa
Information				
Time	1. s			
Load Step	1			
Sub steps	20			
Iteration Number	20			

Thus from following analysis data it can be observed that the anti-vibration mount is able to bear the given cyclic load without deformation.And hence the structure is most stable to bear the load.

4. Conclusion

A study of vibration isolation is carried out analytically and numerically using FEM .For FEM analysis ANSYS 14.5 tool is used.It can be concluded from the analysis that, Natural frequency of anti-vibration mount is 288.62 Hz which is certainly the safe frequency in which no resonance can occur for the particular structure,also the analyzed structure is able to bear the applied cyclic load which states the structure is safe analytically.

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