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FEM NONLINEAR CONTACT VIBRATION ANALYSIS OF A DISC ADAPTER

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Abstract: *In order to simulate a launch, all space instruments undergo a vibration tests. Due to the fixing points in vibrating machines being different, adapters are used for each space instrument. In random environments the load is defined as stochastic data. Only linear analysis can be used. This article describes the method for FE modeling of nonlinear contact transient analysis. The load is defined as random stochastic data depicted as a graph versus time and applied to space instruments. For base excitation, the large-mass method was used. The contact is formed by GAP elements. From the results, the PSD (Power Spectral Density) spectrum and g_{RMS} values of vibration are computed and evaluated.*

Keywords: ***Vibration, space instrument, contact, FEM.***

1. Introduction

To simulate a launch in space, all space instruments are tested by vibration test equipment. Because the fixing points on the vibration machine are in a different position than that of the space instrument, adapters are used. In the micro-accelerometer (MAC) of the SWARM mission a disc adapter was used. The required adapter is required to have minimum variation of transmissibility between test item and mounting points. A random spectrum is defined for each instrument itself. In the MAC, the random spectrum range is from 20 Hz to 2000 Hz. The overall random g_{RMS} variation is maximally ± 10%. This imposes a high demand on the vibration machine. The machine facilities under heavy acceleration are limited by the maximum weight of measuring instrument and adapter. In our case the adapter's maximum weight was set to 10 kg. The transmissibility variation shall not exceed a change of +3 dB between 5 to 500 Hz and +6 dB between 500 to 2000 Hz. During test, the disc adapter and the vibration machine equipment are not constantly in contact, this being the reason why the nonlinear contact analysis must be used.

2. Design

Three designs were modeled. The first a disc adapter with a 25 mm thickness and a 450 mm diameter connected to the vibration machine by 8 screws in a circular formation, radius of which is 203.2 mm. The second a disc adapter with a 35 mm thickness and a diameter of 350 mm. Due to the position of the MAC on the adapter prohibiting the use of 8 screws, 6 screws were used instead in a circular pattern with radius 152.4 mm. The last design used countersunk screw heads, 12 screws can be used, 8 with a radius of 152.4 mm and 4 with a radius of 101.6 mm.

The FE model is computed by NASTRAN software. The disc adapter is modeled as a 3D solid, the vibration machine as a 2D surface on the base. For base excitation, the large-mass method (Leftheris, 2006) was used. The 2D surface on the base was connected with point mass (CONM2) by a rigid RBE2 element. The disc adapter is fixed to the vibration machine by screws in a circular pattern. The screws are modeled as a rigid RBE2 element. The contact is formed by GAP elements with nonlinear behavior (CGAP) not forming classical gap contact as defined in Laursen, 2003. The element is in continuous contact time as node-to-node contact element (Wriggers, 2002). The advantage of node-to-node contact in comparison to slave-master contact is it being a fast solution without iteration. The different properties in compression and tension are defined as nonlinear stiffness material property.

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GAP tension - open stiffness is 100MPa, compression - close stiffness is 45000MPa. The close stiffness corresponds to material stiffness of Electron (vibration test machine material (magnesium alloy)). Material properties of used material are shown in Tab. 1.

Tab. 1: Material properties of disc plate (duralumin) and vibration test machine plate (Electron).

no.	Description	Material	E_{11} [MPa]	G_{44} [MPa]	ν [1]	ρ [kg/m^3]	Damping G
1	Duralumin	7075 T651 (Al alloy)	71700.	26900	0.33	2810	0.005
2	Electron	Electron A8 (EN 1753 MC21110)	45000	17000	0.34	1800	

3. Methods

Firstly, a modal analysis (SOL 103) was made. The frequency range of interest is from 20 Hz to 2000 Hz. The first mode was a calculated circle mode (0,0) on frequency 1119 Hz and a second bending mode (1,0) on frequency 2063 Hz. Next the nonlinear transient analysis (SOL129) was used. The overall acceleration of a random test is 15.23 G_{RMS} (root mean square value multiple by gravitational acceleration) having stochastic character. The maximum PSD of vibrations between 100 and 300 Hz are 0.405 G^2/Hz. Between 20 Hz and 100 Hz, the +3 dB/Oct rate is defined. Between 300Hz and 2000Hz, the -5dB/Oct rate is defined. The defined spectrum is shown in Fig. 3. Instead of acceleration, a force load is applied at the large-mass point. The sensitivity analysis of the large mass value was made. It has exponential effect on the g_{RMS} value. Over the value $10^6 * m_0$, the g_{RMS} value doesn't change. Due to limitations for large-mass being only in numeric overflow, the value of $10^7 * m_0$ was used (recommended value by NASTRAN software is $10^6 * m_0$).

The transfer of stochastic spectra to deterministic values of acceleration versus time by the time averaging was used in (Ariaratnam, 1988). In our case the random load spectra is transferred to acceleration versus time data using MATLAB software. White noise was filtered according to the required spectrum properties using a band pass filter. The final time history load has a greater g_{RMS} (value of 20 G). A greater value is necessary for a conserved solution. The workflow for computing the random behaviour in nonlinear transient analysis is depicted in Fig. 1.

Stochastic environment → Window time load → Solution → Window time result → Stochastic result

Fig. 1: Workflow for transient analysis with stochastic load.

A 0.05s overall time was used in $5e^{-5}$s intervals. According to Umashankar, material damping ξ_{Al} measured 0.0025 using $f_{resonance}$ of 383Hz (GE = 0.005, W4 = 2406). The material damping constant of G_E and W4 are used by NASTRAN software in a damping matrix [B]. Random vibration with symmetric geometric nonlinearities is described in (Roberts, 1990). In our case, the GAP element forms asymmetric geometric nonlinearity. The equation of motion is given as:

$$[M]\ddot{u} + \frac{G_{Al}}{W4}[K]\dot{u} + [K]u = m\{a\} \quad (1)$$

Using a free system, the general displacement u does not correspond to the vibration between the disc adapter and a rigid platform. The vibration displacement u' is defined as:

$$u'(t) = u_{adapter}(t) - u_{ground}(t) \quad (2)$$

4. Results

From transient nonlinear analysis, acceleration and displacement at each point was computed. The results for the displacement of the vibration u' for the above-described designs is shown in Fig. 2. The actual value of u' varies greatly in time. The lowest vibration is shown with the model having a 35 mm thickness and 12 screws.

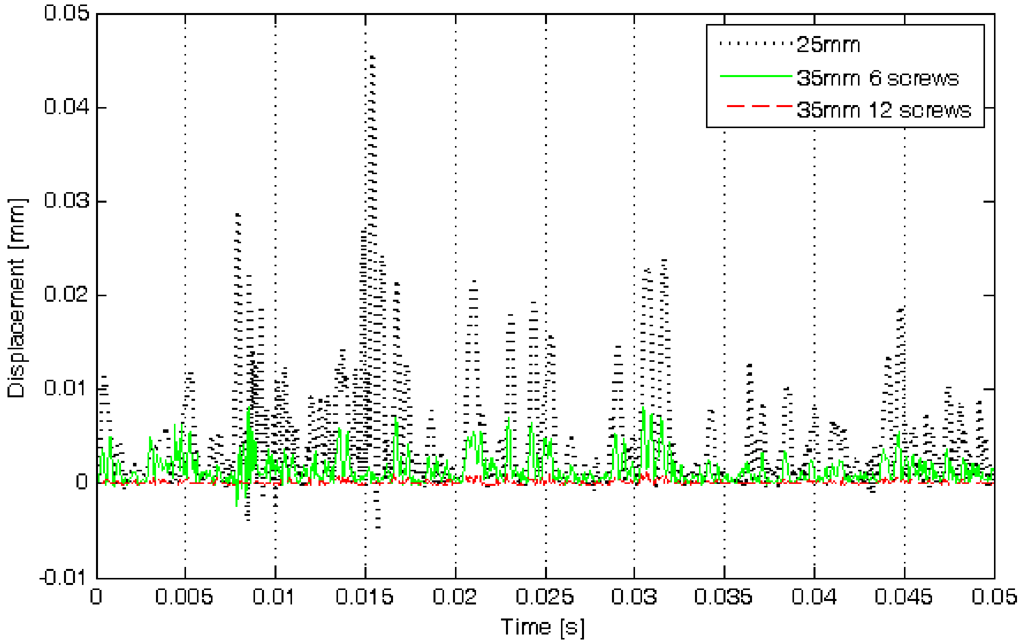

Fig. 2: Computed displacement u'(t) between adapter and ground with nonlinear transient response (legend describes the thickness of the disc adapter).

We change the stochastic data to a graph Due to this, we cannot evaluate the deterministic values; and must use a stochastic estimation. One such estimation is the root mean square (RMS) value (Lalanne, 2002). The RMS values of acceleration with the most intense vibration point g_{RMS} are shown in Tab. 1. From a spectral analysis of transient record, we can determine the resonance frequency (Reddy, 2004). The nonlinear parameters of the GAP element in contact with a rigid ground forms local surface deformation comparable with a non-rigid platform. In low frequency dynamics, the local surface deformation can significantly affect the results (Laursen, 2003). We can compute the mean value 2.345 as the ratio between nonlinear contact resonance and model resonance.

Tab. 1: Calculated modes, nonlinear resonances and effective values of acceleration g_{RMS}.

	Mass [kg]	Mode no.1 [Hz]	Mode no.2 [Hz]	Nonlinear resonance no.1 [Hz]	Nonlinear resonance no.2 [Hz]	g_{RMS} [g]
Random load						15.2
Transient load						20
25 mm$_{Ø450}$	10.326	1119	2061	2600	4720	86.4
35 mm$_{Ø350}$ 6screws	8.689	1914	2267	4400	5620	38.3
35 mm$_{Ø350}$ 12screws	8.689	5161	5857	over 10000	over 10000	21.4

The disc adapter with a 35 mm thickness and a diameter of 350 mm was manufactured and tested. For the random vibration test, a vibration machine LDS V850-22kN was used. The difference between 6 and 12 fixing screws were investigated. The measured spectrum of 6 screws crosses the tolerance of +6 dB at frequency 1400 Hz. The 12 screw design withstood to the defined requirements. The PSD of vibration are shown in Fig. 3. Because of our 0.05 s interval record, we have no results below 100 Hz (with respect to 5 points being a minimum to evaluate the sinusoidal behavior).

Three designs of disc adapters widely change the transferred spectrums when compared with load. The 25 mm thickness adapter changes the spectral density of +3 dB from 480 Hz onwards. The overall g_{RMS} value was 86.4 g because the first resonance of the disc around 2600 Hz. The 35 mm thickness adapter with 6 screws changes the spectral density of +3 dB at 1560 Hz with the experiment showing the limit at frequency 1170 Hz. The 35 mm thick adapter with 12 screws does not change the spectral density +3 dB from 2420 Hz, the experiment shows a frequency of 1500 Hz. Even the experiment shows +3 dB at 1500 Hz, +6 dB don't come until 2000 Hz. The overall g_{RMS} value is 21.4 g. The variation value (+7%) is in accordance with the requirements. The design with 12 screws is acceptable for use in a random vibration test.

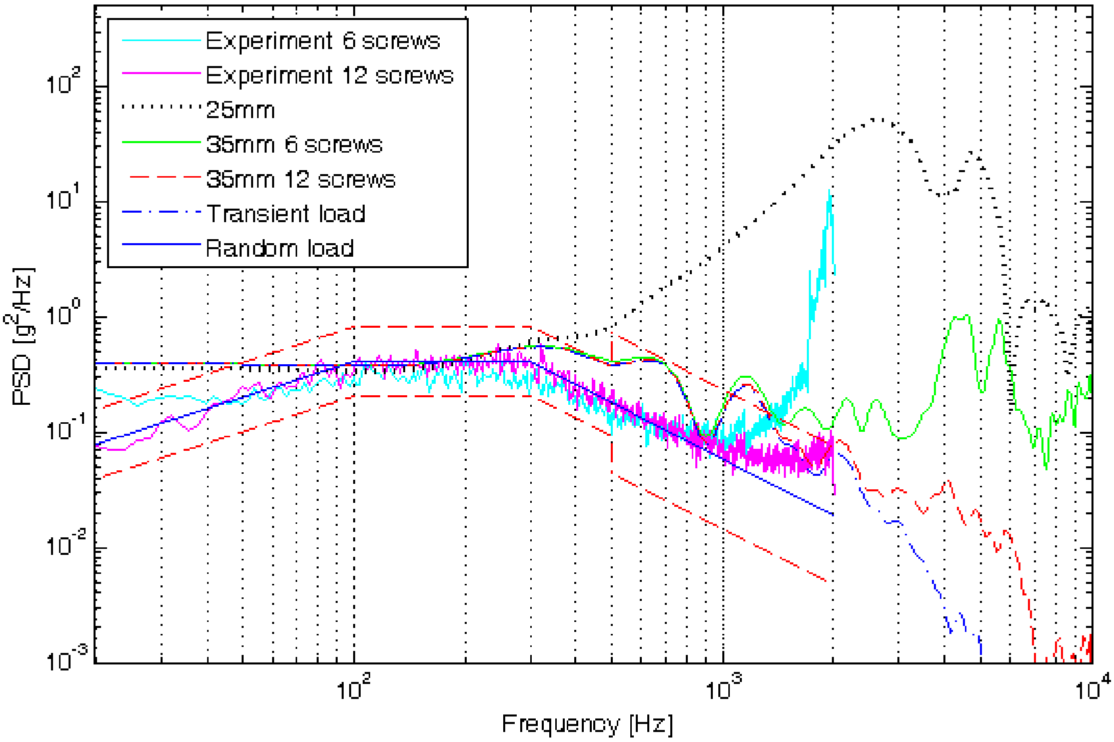

Fig. 3: Measured spectrums (Experiment) and computed spectrums of the 3 designs of a disc adapter.

5. Conclusions

The work describes the FE nonlinear contact random analysis. The process was demonstrated on the three adapter designs for vibration testing. For random vibration, a wide range of frequencies, between 20 Hz and 2000 Hz were used. The only acceptable adapter in this range is the one with 350 mm diameter and 35 mm thickness with 12 mounting screws. To analyze the spectrum, nonlinear transient contact analysis must be used with a sufficient time length and sampling rate. To evaluate the result defined by stochastic behavior, only the stochastic estimated results shall be used. The g_{RMS} and PSD values were used. Nevertheless, to calculate the results we need to view the load as a graph versus time data. The MATLAB software was used for that by filtering white noise. The results were verified by vibration experiments.

Acknowledgement

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