STRUCTURAL VIBRATION ANALYSIS OF INTERMODALLY TRANSPORTABLE SPACE SIMULATION CHAMBER

Nikolajs Glizde, Ilmars Blumbergs, Pauls Irbins
Riga Technical University, Latvia
nikolajs.glizde@edu.rtu.lv, ilmars.blumbergs@rtu.lv, irbins@gmail.com

Abstract. The research was conducted in the frame of the project "Metamorphosis". The project objective is to develop a prototype of an intermodally transportable space environment testing facility. The research is focused on development of the Vacuum Chamber for the Space environment simulation for smallsize satellite system CubeSats testing purposes. The Vacuum Chamber should be designed to be mobile or intermodaly transportable. A basic transport unit for the Vacuum Chamber was taken a vehicle that will be designed as a mobile laboratory. This research phase focuses on the Vacuum Chamber rigidty testing on transportation loads caused by vibration. The testing is conducted using the analytical approach into virtual environment with ANSYS software. The analyses procedure in the virtual environment is completed in several steps that are necessary to obtain the desired results. The main work is in preparation of the model, defining materials, connection types. The obtained results will be avaluated and used in the further design process. If it is necessary, revisions will be made in the Vacuum Chamber CAD design model to make the system design rigid enough to withstand transportation vibration loads like truck transportation, railroad transportation and others as well. The obtained results showed that some system construction redesign is necessary.

Keywords: transportation, vibration loads, random vibration, intermodal transportation.

Introduction

The work for possibility to make survailance of the earth from space environment continues uninterruptedly.

Already from 1999, mutually by the California Polytechnic State University and the Stanford University, the initial technical specification of a small size satellite system called the CubeSat was developed to give a possibility for universities, high schools and private compenies design small size satellite systems for launching them in the Low Earth Orbit for research purposes. One module of this small satellite has a volume of precisely one liter and the mass up to 1.33 kg. The full size of such CubeSat can have up to 12 (twelve) modules. The first such small size satellite was launched in June 2003 [1]. The problem in launching some developed CubeSat is that it should coincide with the time frames given by nations which are capable to develop and keep up launching sites. And also, the launch of the CubeSat is rather costly if there is some smaller nation university which desires to launch it. So, the work also continues to develop systems which would be less costly and give more possibility to launch satelites in Low Earth Orbit as aircraft platforms for satellite launch vehicles [2; 3].

The current research is part of the project *Metamorphosis*. The project is focused on developing of a mobile laboratory based on a transportation vehicle. This is the first testing facility developed for intermodal transportation. There are more and more organizations that are wishing to enter the space bussiness or research environment. The small satellite technology develops very rapidly. Not all the players in this area can afford to develop facilities for full proceses that are necessary to complete for launching a satellite. In this case we could refer to a satellite testing facility. The satellite system and subsystem testing are necessary to evaluate its survivability in harsh space environment. This is where such mobile test laboratory could help coming to your site and completing the necessary tests for lower expenses [4; 5].

The laboratory equipment is very sophisticated and requires its preservation in environment with defined temperature, humidity, vibration, and other factor ranges that are defined by the manufacturer or governmental standards. The perspective mobile laboratory vehicle will be equipped with devices that preserve the necessary environmental conditions. The thing that should be considered during the design is vibration load during transportation that can not be diverted with the devices that preserve environmental conditions, but should be checked during the design and mounted on the vehicle with the *Vacuum Chamber*.

The intermodal transportation means moving large-sized goods through two or more modes of transport. Intermodal transfer may involve truck, rail, ship, and then truck again, and also can include air transport as well. Usually, the goods are loaded in a container and then this container is reloaded on a different type of transport until it reaches the destination point. In the current research case, the vehicle mounted laboratory will be as a container with goods that can reach the necessary inland destinations by itself. The air transport will not be considered in this case as it rises the expenses more in comparison with the other transportation means. On the vehicle laboratory mounted equipment most severily will suffer from vibration loads during its own relocation, that is from working rotary devices like the engine, gearbox, transmission, and road bumps as well. If the vehicle laboratory is transported with a fraight train or ship, its own structure will serve as a damping mechanism. Considering the above mentioned the *Vacuum Chamber* construction should be checked on vibration loads that it would suffer from their own transportation unit. Environmental test procedures include vibration, acoustic and thermal tests that can be completed separately to determine different type environmental phenomena influence on the test object but not all of them can be addressed to current research step [6; 7].

It continues with the design of the vacuum chamber. The project is for construction of the vacuum chamber for smallsize satellite systems, the so called CubeSats testing on close to space environments to ensure that all systems function correctly in that extreme environment. According to the posed requirements the vacuum chamber system should be constructed rigid enough to stand multiple transportation type loads. During transportation mechanical systems are subjected to vibration loads caused and coming from different sources, like the engine, gearboxes, axels and other more unpredictable from road bumps and uneven parts of them. The most common transportation system to use for the vacuum chamber transportation was taken to be road transportation by truck or by railway.

Initially, research on the mentioned subject was conducted to get information about vibration levels which appear during transportation on the truck and train platforms. The research papers were examined which describe examination of vibration levels on vehicle and rail platforms during transportation. The vibration recording sensors were mounted on the test vehicle platforms in such a way that alowed to make records for the highest levels of vibration. For trucks the highest levels of vibrations usually happen at the rear of the platform. The recorded measurements showed that vertical vibration levels are stricter. Though the lateral vibration levels were weaker, they should be used together with the vertical vibration levels to evaluate the vibration influence on the cargo packages that are not ideally placed and have free spaces between each other. The research showed that the truck and the rail vibrations create considerable lateral and longitudinal movement and are different in separate regions of the world and also are different for the truck and that for the train.

In spite of the mentioned problem, the data of the vibration levels were taken to conduct random vibration simulation. According to the mentioned research truck and railroad transportation frequencies compose the range from 1 to about 120 Hz [8-10].

In this research the analyses system natural frequency range is determined, analysed and compared with frequencies caused by transportation, its impact on the system determined and solutions searched to reduce the negative influence on the system for safe transportation.

The system natural frequencies are the frequency levels at which undamped vibrations naturally occur. The system natural frequencies are determined with the assumption that the system free response is periodic with a specified frequency. When the system is exposed to harmonic vibration frequency, which is near its natural frequencies, that results to a large amplitude steady state vibrations that can result in the system destruction. The solution to this problem could be to change the system properties so that the system natural frequencies differ from the vibration frequencies to which the system is subjected. For example, adding rigidity to the system will raise its natural frequency level. Also, another solution to this problem could be adding a damping mechanism as a vibration absorber that causes difference between the system natural frequencies and the frequencies the system is subjected to. Using only springs for the system damping is not recommended as springs can augment the excitation level.

According to the requirements of the design system the *CAD* model will be tested in *ANSYS* software environment. To check the system on vibrational loads, it is necessery to complete simulation on loads in the following *ANSYS* modules: *Static Structural Analyses* –which makes preparations for prestressed modal analysis, *Modal Analyses* – during which the system natural frequencies will be determined, *Random Vibration Analyses* – during which the structure response under random loading will be determined.

Static Structural Analyses – in the static structural analyses the construction inertial reaction forces are determined under standard earth gravity load that prepares the construction model in prestresed condition for the following modal analyses.

Modal Analyses – in the modal analyses the system construction natural frequencies are determined. In modal analyses the system rezonanse frequencies are determined. The rezonanse frequencies depend on the object geometric shape and constraints. When these dengerous rezonanse frequencies are determined, it is necessary to clarify if this geometric model is initiated in these frequencies and what deformations it causes. During the modal analyses the *Pre-Stress* condition from the *Static Structural* analyses will be used.

Random Vibration Analyses – in the random vibration analyses the system construction is checked on loads caused by transportation vibrations, that in turn could come from rotational equipment, engines, road bumps and are iregular in nature.

The vacuum chamber weight is 349.35 kg, the material is stainless steel, Young's Modulus of Elasticity – 1.93e + 11 Pa, Poisson's Ratio – 0.31, density – 7750 kg·m^{-3}, Tensile Yield Strength – 2.07e + 08 Pa, Tensile Ultimate Strength – 5.86e + 08 Pa.

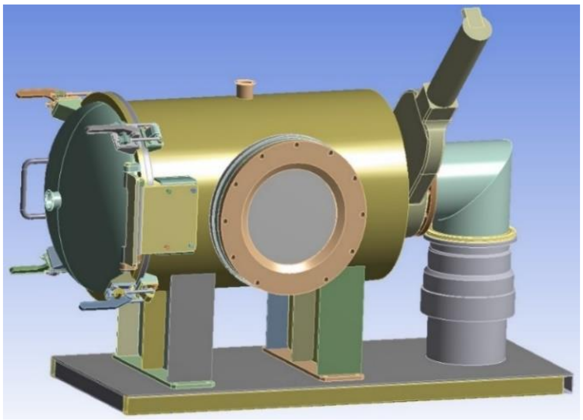

Fig. 1. **CAD drawing model of the vacuum chamber**

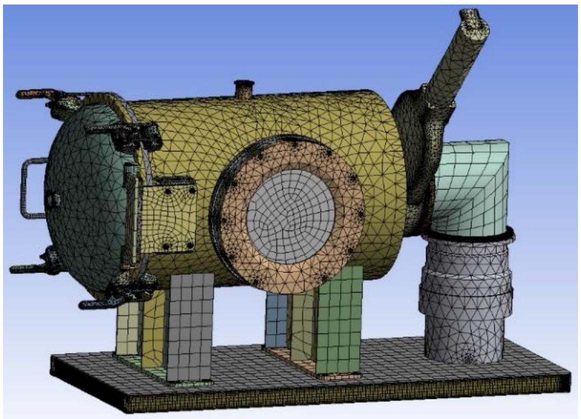

Fig. 2. **Finite element model of the vacuum chamber**

First, it was necessary to prepare the model in ANSYS and complete the *Static Structural Analysis* which makes preparations for prestressed model. To get good simulation results it is necessary to prepare a good finite element model of the construction, or in other words, that is a good construction component mashing. The construction mashing affects the simulation results. The construction and its mashing or the finite element model are shown in Figures 1 and 2 [11; 12].

The factor of safety

The safety factor indicates how much stronger the system is comparing to the intended load during exploitation. The safety factors usually are determined by calculation. Detailed testing is impractical usually in such cases as the bridge or construction, but the load carrying capacity should be determined to acceptable accuracy at least because of economic reasons for using the correct amount of material. Another reason why many constructions are built much stronger than it is necessary is for emergency situations like earthquackes and also if during the construction design the useful construction life is long the material degradation should be taken into account.

The safety factor can be defined in two ways:

- realized factor of safety: that is the ratio of the construction absolute strength to the actual applied load;
- design factor of safety: the value required by related law, standard, specification, etc.

Comparing both above-described factors of safety, it can be indicated that for successful design the realized safety factor should be at least equal or exceed the design safety factor.

The design factors of safety for different applications are indicated by standard, law or contract regulations. The values of the safety factor differ by application: for buildings 2.0 for each structural member, for boilers and pressure vessels 3.5 to 4.0, for cars – 3.0, for aircrafts 1.2 to 3.0 depending on the material.

If the construction has dynamic loading, that is periodic, cyclic or repetitive, then it is necessary to consider material fatigue when choosing the safety factor.

The vibration loads are cyclic and repetitive, as well as vessels for use under pressure (in this case negative pressure), so for the current design the vacuum chamber structure is considered to be subjected to metal fatigue. Also, taking into account the above mentioned factor of safety for pressure vessels, the factor of the safety value of 3.5 is taken for the current design.

Currently also, different developments are made in the field of Non-destructive testing, where sensors are used for real time passive monitoring of systems with sensors which are in waiting state for the signal, which could come from the structure material under control [13].

The static structural analyses

In the linear static structural analysis, the global displacement vector $\{x\}$ is solved with the following matrix equation:

$$[K]\{x\} = \{F\}, \tag{1}$$

where $[K]$ – global stiffness matrix, constant;
$\{F\}$ – global load vector, statically applied.

Completing the linear structural analysis the following assumptions are made:

- material bahavior is assumed as linear elastic;
- small deflection theory is used;
- it is considered that forces are no time-varying;
- damping is not used.

The model deformation can be plotted which is a scalar quantity and calculated by the following equation:

$$U_{total} = \sqrt{U_x^2 + U_y^2 + U_z^2}\ , \tag{2}$$

where U_x – deformation in X axis direction;
U_y – deformation in Y axis direction;
U_z – deformation in Z axis direction.

In this research step the system inertial response forces were identified followed by the system analyses on natural frequency range. The system inertial response forces were identified in the Static Structural analyses applying the system Standard Earth Gravity. The Static Structural analyses results show the biggest directional deformation from the structure inertial load at the tank upper part, but the values are not critical. Also strain energy concentrates in the tube part with right angle, the values also are not critical. Total deformation shows its concentration in the part of the structure where the vacuum pump and shatter are mounted. Though the values are not critical, they can point to problems on these parts caused by vibration loads [14].

Table 1

Static Structural Analyses result summary

Results	Minimum	Maximum	Units
Normal stress	-5.89E + 06	2.05E + 07	Pa
Directional deformation	-4.04E-05	1.05E-04	m
Total deformation	0	1.48E-03	m
Equivalent stress	0	2.67E + 07	Pa
Safety factor	7.7428	7.7428	-

In the construction model for the *Static Structural Analyses* the probe input was made that showed the following reaction forces of -2.2042e-004 N, 3414.6 N, 3.0342e-004 N and 3414.6 N correspondingly for X, Y and Z axis and total.

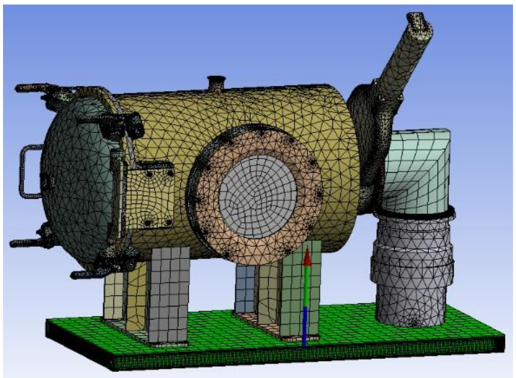

Fig. 3. **Vacuum chamber with the reaction force depicted**

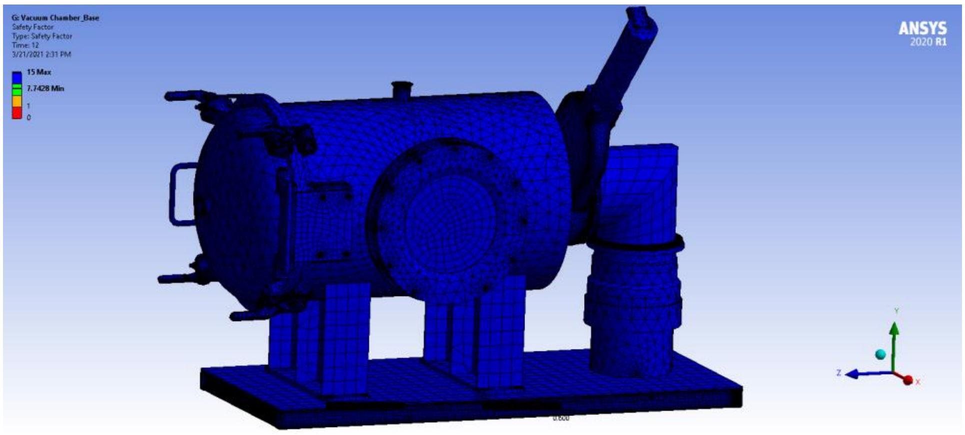

Fig. 4. **Vacuum chamber analysed in the *Static Structural* module**

In Table 1, we can see the simulation results where the maximum stress is 20.5 MPa and maximum deformation is 0.105 mm in the Z-axis direction and total deformation of 1.48 mm. Also, in Table 2 we can see the reaction force, it is 3414.6 N.

The Modal Analyses

The modal analyses are the simulation process during which dynamic response of a system by its modes of vibration or natural frequencies is determined. Any periodic function can be represented as a series of sinusoidal functions, each individual sinusoid is defined by its amplitude, frequency and phase.

$$\sum_{n=1}^{N} A_n \sin\left(a\pi f_n t + \phi_n\right), \quad (3)$$

where A_n – amplitude;
f_n – frequency;
ϕ_n – phase angle;
t – time.

The structure vibrations can be expressed as arrays of vibration modes. The vibration modes are characterized by natural frequency, damping and shape. The *Modal Analysis* analytical method is based on generation of the equations of motion of a construction system through initially prepared finite element or mashed construction model (see Figures 2 and 3).

The *Frequency Response Function* (FRF) is characterized by the output response of the construction and the applied force ratio. Both *FRF* variables are measured simultaneously. The time domain data is transformed to frequency domain by the *Fast Furier Transform* (FFT) function.

The construction system oscillates at the frequency of the applied force that in this case comes from different sources during transportation as discussed above. If the applied force frequency is close to the construction system natural frequency, it dramatically effects the amplitude of oscillation, which becomes very large and can lead to destruction of the construction system.

In the current study there are determined twenty natural frequencies (Table 4) corresponding to the twenty normal frequencies. From the theoretical point, to determine the system durability, it is necessary to take a wider range of frequancies that system will be subjected to in real life with 1.5 probabillity that is about 180 Hz. Thus, from Table 4 we can assume that the construction will show large amplitude oscillations when the exciting frequency is close to either of fourteen natural frequencies.

Undamped free (natural) vibrations are carctarized by equation:

$$m\frac{d^2x}{dt^2} = -kx\,, \tag{4}$$

where m – object mass;
x – displacement;
t – time;
k – stiffness factor.

That can be solved with the following equations:

$$x(t) = x_0 \cos \omega t + \left(\frac{v_0}{\omega}\right)\sin \omega t\,, \tag{5}$$

$$x(t) = A\cos(\omega t - \phi), \tag{6}$$

$$A = \sqrt{x_0^2 + \left(\frac{v_0}{\omega x_0}\right)^2}\,, \tag{7}$$

$$\phi = \tan^{-1}\left(\frac{v_0}{\omega x_0}\right), \tag{8}$$

where x_0 – initial position;
ω – angular frequency;
t – time;
v_0 – initial velocity;
ϕ_n – phase angle;
A – amplitude.

The angular natural frequency (rad·s^{-1}):

$$\omega = \sqrt{\frac{k}{m}}\,, \tag{9}$$

where k – stifness factor;
m – object mass.

And the linear natural frequency is (Hz):

$$f = \frac{\omega}{2\pi}\,, \tag{10}$$

where ω – angular frequency.

The system natural frequencies were determined by modal analyses. The modal analyses method uses the system global coordinates to separate the differential equations [15].

During *Modal Analyses* of the system it was determined that natural frequencies for it appear in the range from 6 Hz to 286 Hz separated in twenty modes, but as was mentinned above, to transportation vibration loads applay fourteen of the mentioned twenty modes in that range.

The results show that there is strong response in the X and Z axis directions. If it were on the support that is under the chamber structure it could be considered as acceptable because of the material strength, but in this case the load direction can influence proper functioning of the vacuum pump and the gate valve. The system itself is strong enough to withstand those loads, but as it refers to technically sophisticated assemblies that can influence their proper functioning, careful considerations should be taken, and it is very desirable to redesign that part of the system.

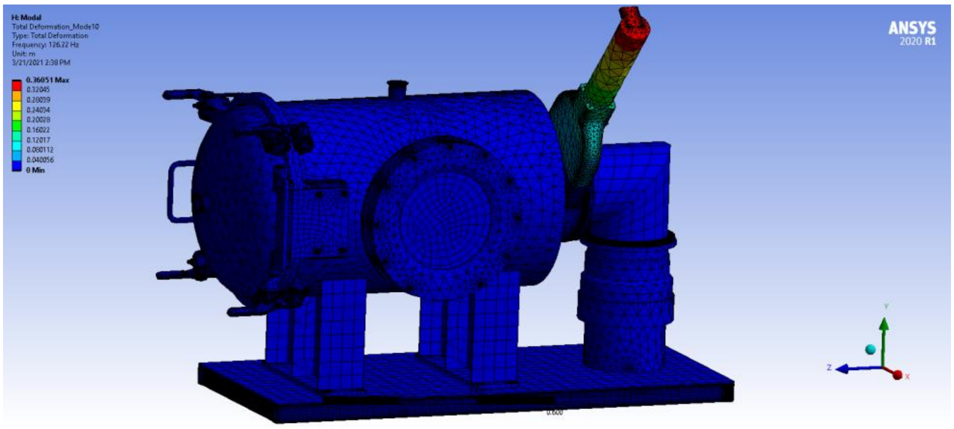

Fig. 5. **Vacuum chamber analysed in the *Modal* module with total deformation depicted during 10th mode**

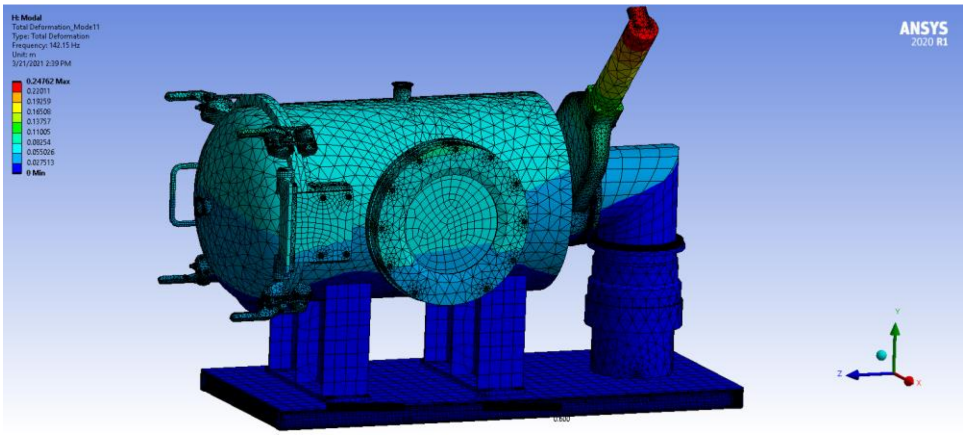

Fig. 6. **Vacuum chamber analysed inthe *Modal* module with total deformation depicted during 11th mode**

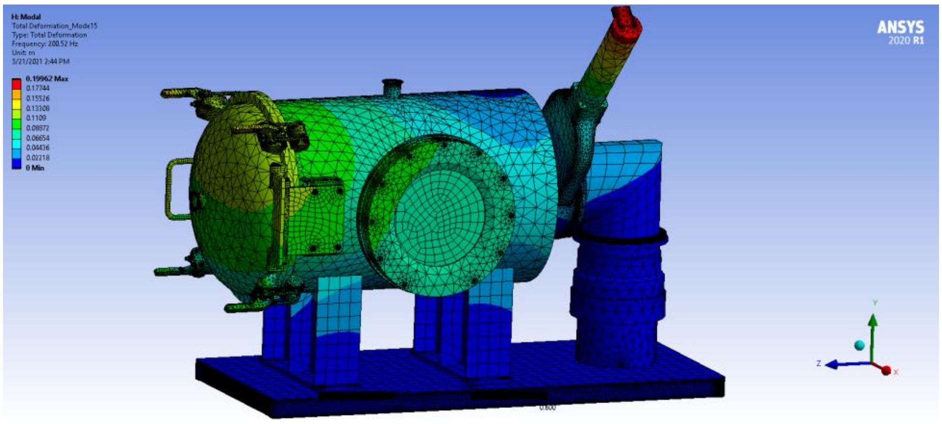

Fig. 7. **Vacuum chamber analysed in the *Modal* module with total deformation depicted during 15th mode**

The calculation results as are shown in Table 2. From Table 3 of the modal analysis results we can see that the *vacuum chamber* has modal natural frequencies in the range from ~6 to 286 Hz. In images 5 to 8 frequency modes are depicted that influence the vacuum chamber structure in the most severe way and are 126.22, 142.15, 200.52 and 247.48 Hz correspondingly.

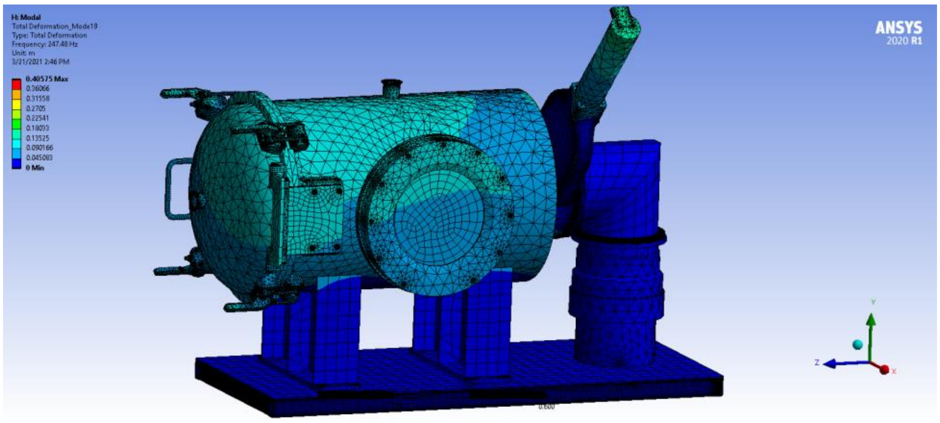

Fig. 8. **Vacuum chamber analysed in the *Modal* module with total deformation depicted during 19th mode**

Table 2

Modal Analyses result summary

Results	Minimum	Maximum	Units	Reported frequency, Hz
Maximum shear elastic strain	0	0.18285	$m \cdot m^{-1}$	6.6923
Normal elastic strain	-5.63E-02	7.56E-02	$m \cdot m^{-1}$	6.6923
Shear elastic strain	-5.80E-02	8.54E-02	$m \cdot m^{-1}$	6.6923
Maximum shear stress	0	1.35E + 10	Pa	6.6923
Normal stress	-1.85E + 10	1.31E + 10	Pa	6.6923
Shear stress 2	-4.28E + 09	6.29E + 09	Pa	6.6923
Directional Deformation_Mode1	-5.30E-02	6.40E-03	m	6.6923
Directional Deformation_Mode3	-0.31106	4.59E-02	m	16.602
Directional Deformation_Mode10	-0.10989	7.74E-02	m	126.22
Directional Deformation_Mode11	-0.14774	9.94E-02	m	142.15
Directional Deformation_Mode14	-2.38E-02	2.60E-02	m	186.6
Directional Deformation_Mode17	-1.00E-02	1.07E-02	m	217.88

Table 3

Natural frequency ranges from Modal Analyses

Mode	Frequency, Hz	Mode	Frequency, Hz
1.	6.6923	11.	142.15
2.	13.418	12.	169.59
3.	16.602	13.	185.24
4.	33.225	14.	186.6
5.	38.435	15.	200.52
6.	61.888	16.	210.37
7.	66.526	17.	217.88
8.	112.69	18.	233.17
9.	120.6	19.	247.48
10.	126.22	20.	286.41

The Random Vibration Analyses

The vibration load influence from transportation is determined by the Random Vibrations analyses.

In the *Random Vibration Analysis* amplitude can be described as a single complex number which is called the *frequency response function* (FRF ($H(\omega)$)) with the following equation:

$$H(\omega) = A(\omega) - iB(\omega), \tag{11}$$

where A – amplitude;
B – amplitude;
ω – angular frequency.

The magnitude of FRF is equal to the amplitude ratio, and the ratio of the FRF imaginary part to its real part is equal to the tangent of the phase angle that is described by the following equations:

$$|H(\omega)| = \sqrt{A^2 + B^2} = \frac{a_{out}}{a_{in}}, \tag{12}$$

$$\frac{\mathrm{Im}[H(\omega)]}{\mathrm{Re}[H(\omega)]} = \frac{B}{A} = \tan\phi, \tag{13}$$

where a_{out} – calculated sinusoidal output;
a_{in} – sinusoidal input;
ϕ – phase angle.

Referring to the theory of random vibration, the response of the system to a single input PSD is described by the following equation:

$$S_{out}(\omega) = |H(\omega)|^2 S_{in}(\omega), \tag{14}$$

or

$$S_{out}(\omega) = \left(\frac{a_{out}}{a_{in}}\right)^2 * S_{in}(\omega), \tag{15}$$

where S_{out} – spectral density response;
S_{in} – spectral density input (value from PSD curve);
a_{out} – calculated sinusoidal output;
a_{in} – sinusoidal input.

The response PSD (RPSD) is calculated by multiplying the input PSD with the response function that is described by equation (7) or graphicaly it can be depicted in the following way (Fig. 9).

$$S_{out}(\omega) \quad = \quad \left(\frac{a_{out}}{a_{in}}\right)^2 \quad * \quad S_{out}(\omega)$$

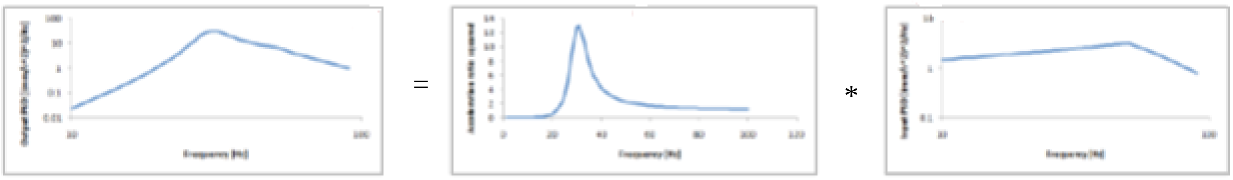

Fig. 9. **Graphical representation of the response function calculation**

Or also:

$$RSPD = \left(\frac{a_{out}}{a_{in}}\right)^2 (input\ PSD). \tag{16}$$

Usually, it is necessary to know the average response of the system that is described by the “root mean square” (RMS). The RMS is the average, or one standard deviation (1-sigma) of response.

$$RMS = \sqrt{\int_0^{\infty} S(\omega) d\omega}, \tag{17}$$

where S – spectral density;
ω – angular frequency.

It is not known what the response will be, but it is known that it will be average *RMS* response on given input.

Assuming that the input is *Gaussian*, and the system is linear, then the output also will be *Gaussian*. The deviations of the response [15-18]:

- $1\sigma \times$ *RMS* (1-sigma) accounts for ~ 68.27% of the total response;
- $2\sigma \times$ *RMS* (2-sigma) accounts for ~ 95.45% of the total response;
- $3\sigma \times$ *RMS* (3-sigma) accounts for ~ 99.73% of the total response.

The *Sigma* stress levels mean that these stress levels will occur 68.27%, 95.45% and 99.73% of the time correspondingly. Usually 1-sigma, 2-sigma and 3-sigma levels are used, but they can be more if it is necessary. For fatigue evaluation the 3-sigma stress level is essential. The stress levels from the *Random Vibration* are real, vary with time and have meaning. Similarly, the *Von Mises* stress is an indication of the stress level severity at a point and its closeness to failure levels.

In the *Random Vibration Analysis* system the possible damages caused by vibration loads of transportation were checked, specially in the range of frequencies that coincide with the structure natural frequency. The analyses results also show strong structure response in the Z Axis and YZ Component directions when vibration frequency is close to the structure natural frequency. Also, the result values do not exceed the material (stainless steel) yield strength the factor of safety should be taken into account.

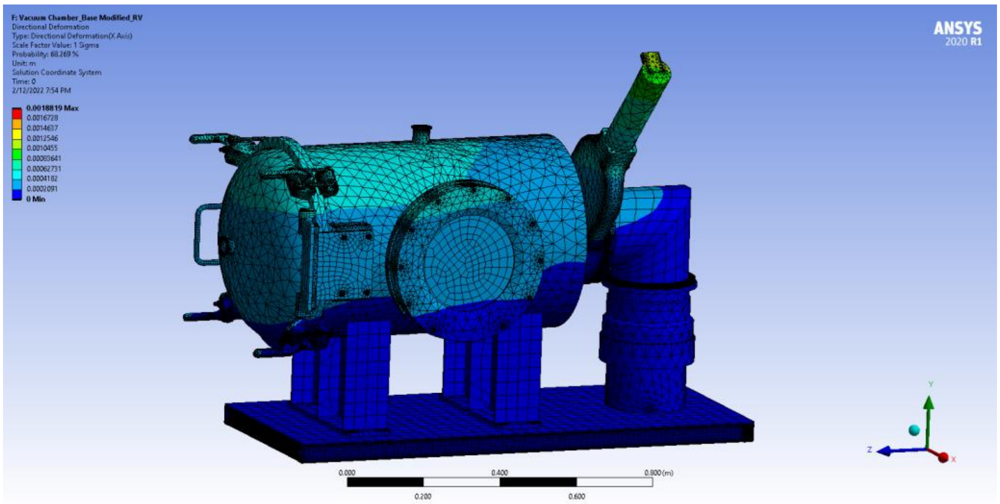

Fig. 10. **Vacuum chamber analysed in *Random Vibration*. Directional deformation depicted**

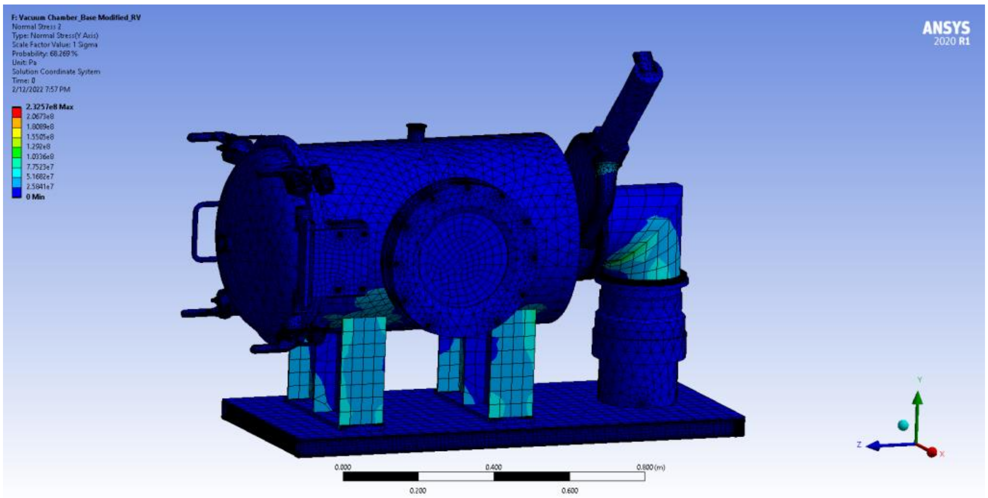

Fig. 11. **Vacuum chamber analysed in *Random Vibration*. Maximum stress points depicted**

Table 4

Random Vibration Analyses result summary

Results	Minimum	Maximum	Units
Directional deformation	0	6.85E-03	m
Normal elastic strain	0	1.30E-03	m/m
Shear elastic strain	0	8.89E-04	m/m
Normal stress	0	3.44E + 08	Pa
Shear stress	0	6.55E + 07	Pa
Equivalent stress	0	2.026E + 08	Pa
Directional velocity	6.19E-02	0.94465	$m \cdot s^{-1}$
Directional acceleration	39.392	711.16	$m \cdot s^{-2}$

In Table 4, we can see the simulation results where the maximum stress is 344 MPa and maximum deformation is 6.85 mm in the Z-axis direction.

Results and discussion

Completing the *Vacuum Chamber* analyses in *ANSYS*, it was possible to identify deficiencies in it. From the *Static Structural* analyses it was identified that the *Safty Factor* is too high, it is equal to 7.74, that is more than twice of the necessary value defined at the beginning – 3.5. That also can be identified by the *Equivalent von-Mises Stress* value (see Table 1) 2.67e + 07 Pa. The system is too bulky. The changes could be made in the support construction, that will save same material and make the construction lighter.

Also, from the *Random Vibration* results we can conclude that the *vacuum chamber system* is safe for transportation, however, as originally the system is designed as a transportable unit. If this is taken into account, then the safty factor of 3.5, should be also addressed in the analyses results from the *Random Vibration* that show *Equivalent von-Mises Stress* value 2.026e + 08 Pa (see Table 5). This points to too week safety factor in case the system is subjected to transportation vibration. Taking into account the mentioned above, a possibility to redesign the *vacuum chamber* base construction including some damping mechanism should be considered.

From the *Modal Analyses* and the *Random Vibration Analyses* it can be seen that there is rather big vibration influence on the gate valve (see Figures 6 and 10). As it is rather a sophisticate device and this could influence its proper functioning, the consideration should be taken to change it for another type of valve more appropriate to the design situation (e.g., electro-pneumatic angle valve).

Conclusions

The research on vibration load analyses in software environment has provided more knowledge about its influence on structural elements, their behaviour and strength. The current findings prove that the structure safety factor is critical for its safe use during the lifecycle and defined by stress values. Another thing that influences the safety factor is the use of the material in its construction that turns to the economical factor of the project in general.

Also, the research helped evaluate the usefulness of the software test environment for the system design. Using software to evaluate the construction during its design can help avoid unnecessary expenses in the further design steps when the real construction is made. The construction can be evaluated in software environment that helps avoid destructive mistakes during its service life usage.

However, to evaluate truthfulness of the software analyses results, the real system structure should have been built and tested in real local transportation environment (vehicle road transportation), and the results compared with the obtained from analyses in software environment.

Acknowledgements

This research was financed by the Institute of Aeronautics of the Riga Technical University project "Prototype development of transportable in intermodal traffic mobile space testing facility "Metamorphosis" (Metamorphosis, project No. 1.1.1.1/18/A/133) budget.

Author contributions

Supervised the research and co-wrote the paper, I.B.; designed and performed virtual experiments, analysed data, and co-wrote the paper, N.G.; assisted in interpretation of analyses data and co-wrote the paper P.I. All authors have read and agreed to the published version of the manuscript.

References

[1] Kaminskiy M., CubeSat Data Analysis, National Aeronautics and Space Administration, Goddard Space Flight Center Greenbelt, Maryland 20771, Prepared by: GSFC/Code 371, November 2015. 21 p.

[2] Kravchenko S., Kuleshov N., Shestakov V., Panova N., Blumbergs I., Comparative Analysis of Possible Aircraft Payload Transportation Method, Suitable for the LatLaunch Reusable Launch Vehicle Operation, Transport Means - Proceedings of the International ConferenceVolume 2021-October, pp. 818-825, 2021 25th International Scientific Conference Transport Means 2021 Kaunas, Virtual 6 October 2021 through 8 October 2021; ISSN 1822296X. 8 p.

[3] Kravchenko S., Kuļešovs N., Šestakovs V., Panova N., Blumbergs I., Advantage of Using Aircraft-Type Stages for Creating Latlaunch Launch Vehicle. In: 20th International Scientific Conference “Engineering for Rural Development”: Proceedings. Vol.20, Latvia, Jelgava, 26-28 May, 2021. Jelgava: Latvia University of Life Sciences and Technologies, 2021, pp.891-898. ISSN 1691-5976. Available from: doi:10.22616/ERDev.2021.20.TF202. 8 p.

[4] Cryogenic and Vacuum Systems, SIA. Presentation-Metamorphosis-the-mobile-space-testing-facility, 23 p. [online][11.03.2021] Available at: http://www.cvsys.eu/wp-content/uploads/2017/10/Presentation-Metamorphosis-the-mobile-space-testing-facility)

[5] Kravchenko S., Panova N., Urbahs A., Urbaha M. The Importance of the Mobile Space Testing Facility “METAMORPHOSIS” Prototype Development. Proceedings of 25th International Scientific Conference PART III. Transport Means 2021 Sustainability: Research and Solutions. October 06-08, 2021 Online Conference - Kaunas, Lithuania. Pages 1098 to 1102. 5 p.

[6] Stevenson, R., Chisabas, S., Loureiro, G. “Space Thermal and Vacuum Environment Simulation”, In (Ed.), IntechOpen, 2017. 22 p. [online][11.03.2021] Available at: https://www.intechopen.com/chapters/59192

[7] Stevenson, R.., “Space Simulation Chambers State-Of-The-Art,” 67th International Astronautical Congress (IAC), Guadalajara, Mexico, 2016. 16 p. [online][11.03.2021] Available at: https http://mtc-m21b.sid.inpe.br/col·s-1id.inpe.br/mtc-m21b/2016/10.04.19.34/doc/IAC-2016%20Space%20Simulation%20Chambers.%20IAC-16-C2-IP-10-35703.pdf

[8] Garcia-Romeu-Martinez, M., Singh, P., Cloquell-Ballester, V. “Measurement and Analysis Levels for Truck Transport in Spain as a Function of Payload, Suspension and Speed,” In Print. Published online in Wiley InterScience, 2007. 10 p. DOI: 10.1002/pts.798

[9] Singh, S. P., Singh, J.,Gaur, P., Saha, K., Measurement and Analysis of Vibration Levels on Warehouse and Retail Store Material Handling Equipment, Journal of Applied Packaging Research, Vol. 2, No. 2, December 2007. 18 p.

[10] Singh, S. P,, E. Joneson, J. Singh and G. Grewal, “Dynamic Analysis of Less than Truck Load Shipments and Test Method to Simulate this Environment”, Packaging Technology and Science, 2007. 14 p.

[11] Modeling Connections, ANSYS, Inc., 2015. 44 p.

[12] Best Practices for Contact Modeling using ANSYS, ANSYS, Inc., 2016. 44 p.

[13] Hauka M., Pavelko V., Kubuliņš R., Blumbergs I., Automated Sensory Monitoring System Prototype for Continuous Monitoring of Material and Structure State. In: Environment. Technology. Resources: Proceedings of the 13th International Scientific and Practical Conference. Vol.2, Latvia, Rezekne, 17-18 July, 2021. Rezekne: Rezekne Higher Education Institution, ISSN 1691-5402. DOI:10.17770/etr2021vol2.6660, 2021, pp.58-61.

[14] Static Structural Analysis, ANSYS, Inc., 2015. 48 p.

[15] Matalevich J., Vibration and Modal Analysis Basics, Jefferson Lab, Thomas Jefferson National Accelerator Fecility, 2015. 12 p.

[16] Module 08: Random Vibration Analysis, ANSYS, Inc., 2019. 45 p.

[17] Grishin A., Estimating Structural Response to Random Vibration: Reaction Forces, Phonex Analyses & Design Technologies, www.padtinc.com. November 27, 2017. 31 p.
[18] Results and Post Processing, ANSYS, Inc., 2015. 44 p.