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ADAPTIVE TECHNIQUES FOR SUPPRESION OF FORCED VIBRATIONS

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Abstract. Adaptive structures, equipped with so-called structural fuses (based on fast responding piezo-devices), able to connect/disconnect instantly selected structural interface, allows effective protection against resonance induction via externally forced vibrations. The presented case study demonstrates haw forced vibrations with modifiable frequencies can be smoothly received, if structural fuses are properly activated/deactivated when the external excitation approaches the structural eigen frequencies.

1 INTRODUCTION

During last two decades a rapid growth of various solutions aimed at absorbing impact energy or protecting structures against external sources of vibrations is observed. Those systems usually constitute an entire spectrum between the extremes of passive and active systems.

In passive solutions, optimum design for crashworthiness is equivalent to maximization of absorption capacity in predetermined impact scenarios only [1], while for other load conditions (different mass, velocity or direction, etc.) structural response can be suboptimal [2]. The most often scenario considered in optimization corresponds usually to the most dangerous load, which is infrequent in practice. As a result, a passively optimized structure responds suboptimally for most of its operational lifetime. This is especially clear in the case of landing gears, which are designed for the most demanding touchdown, but whose everyday usage involves much smaller vertical velocities and loads [3].

The opposite extreme are active absorption systems, which comprise actuators to generate external forces. Active strategies are theoretically very effective and (at least for linear systems) well-researched [4]. But those solutions have important disadvantages, especially in impact absorption, which include application of large external control forces, the correspondingly high energy consumption, as well as their susceptibility to instabilities in case of power loss, control delays or actuator failure.

Adaptive systems, located in-between the extremes of passive and active systems, are based on semi-active control strategies [5].The same as active systems, adaptive systems areoften designed to cope with various types of loading but they act in different way. The absorbers, or the actuators, are not used to generate external forces but are generally used to change properties of the structure in predetermined relation to identified loading scenario. In this way adaptive systems only need energy for recognizing type of external loadand supplying adapters with controlling signals. The ability to quickly identify and determine type of external load as well as the absorber design and properties are very important for the efficiency of adaptive systems.There are many papers which describe recent investigations of impact identification methods and various types of absorbers, which are usually based on the use of so-called smart materials. For example, the group of pneumatic/hydraulic flow control shock-absorbers with actively controlled piezo-valves determines the first class of AIA (Adaptive Impact Absorption)requiring real time feed-back control of piezo-actuators[6], [7]. The other class of absorbers [8] requires precise in time switching between two active interfaces of adaptive inerter (so-called SPINMAN), causing switching of the spin of rotating inertial cylinders for conversion of linear impact energy into rotation of the inerter.

Other problem related to performance of adaptive structures and stabilization of their dynamic response is identification (usually in real time) of dynamic excitation. It is especially a challenging issue for the impact loads, when in few milliseconds not only the kinetic energy of the impact, but also the impact velocity has to be determined [9].

As it is mentioned above, the strategy of controlling absorbers is also crucial one and have direct impact on efficacy of adaptive systems. In this paper we discuss the problems related to algorithm, methods and techniques which can be used for supressing forced vibrations of the adaptive structure shown in Fig. 1. For example, concentrating on satellite aerospace engineering, the following requirements are crucial: a) necessity of artificial vibration damping system due to flexibility of the structure and its low natural damping, b) necessity of low weight of the proposed damping system, c) necessity of high reliability (simplicity) of the proposed system, d) preference of fluid-less techniques because of leakage danger. Therefore, semi-active (or even smart-passive) AIA damping systems will be preferable for aero-applications, where phyro-activated deployment on the orbit (typically applicable technique) causes problems for sensitive instrumentation.

Our previous investigations [10], [11] have shown that many problems with suppression of forced vibrations can be resolved by introducing the concept of pre-designed pre-stressing techniques. Feasibility study for control of eigen vibrations in flexible truss structures of parabolic mirrors on the orbit, by means of inducing self-stresses has been already performed [12]. It has been demonstrated that ca 15% reduction of local vibration amplitude is available via proper pre-stressing of the structure. This techniques, so-called PAR (Prestress Accumulation-Release), used in adaptive system for damping of impact born vibrations is extremely effective.

Finally, in this paper we discuss and analyze the case study of the flexible structure shown in Fig. 1, exposed to excited vibrations and equipped with PAR absorbers in the form of switchable joints [11]. We introduce the new concept of suppressing local vibrations via proper in-time control of these absorbers. Also the results of numerical simulations, as well as the results of experimental investigations, are presented and discussed.

2 BASIC FORMULAS

The general form of equations of motion can be expressed as follows:

$$M\ddot{x} + C\dot{x} + Kx = F \tag{1}$$

where M, $C$and$K$are symmetric $n \times n$mass, damping and stiffness matrices, respectively, $\ddot{x}$, $\dot{x}$, $x$denote n-dimensional column vectors of space variables (accelerations, velocities and displacements, respectively), while the vector Fdenotes external forces acting on the system. Neglecting damping effect and external forces, the above equation takes the following form:

$$M\ddot{x} + Kx = 0 \tag{2}$$

which describes free vibration of an undamped system. The general solution of equation (2) can be expressed as follows:

$$x = \phi e^{j\omega t} \tag{3}$$

where$\phi$is an n-dimensional vector called a mode shape,ω denotes the frequency of vibration and t denotes time. Substitution of the above relation to the equation (2) leads to:

$$(K - \omega^2 M)\phi = 0. \tag{4}$$

We are interested in non-trivial solutions satisfying the following formula:

$$det(K - \omega^2 M) = 0 \tag{5}$$

which is known as the characteristic equation. The roots of equation (5) are known as the eigenvalues while their square roots are referred to as the natural frequencies of the system. Each distinct eigenvalue $\omega_i^2, i = 1,2, \dots, n$ has a corresponding mode shapeϕ_i, which satisfies equation:

$$M^{-1}K\phi_i = \omega_i^2\phi_i. \tag{6}$$

The mode shapes are often used to pair corresponding natural frequencies, obtained from numerical and experimental modal analysis. In this paper the MAC (Modal Assurance Criterion) coefficient has been used as a measure of correlation between two eigenmodes of vibration:

$$MAC = \frac{|\phi_{EM}^T\phi_{NA}|^2}{(\phi_{EM}^T\phi_{NA})(\phi_{NA}^T\phi_{NA})} \tag{7}$$

where ϕ_{NA},ϕ_{EM}-are mode shapes of vibration obtained from numerical and experimental modal analysis, respectively.

3 NUMERICAL MODEL OF FLEXIBLE STRUCTURE

A numerical model (Fig. 1) of flexible structure mockup has been elaborated taking advantage of Abaqus program. The finite element model has been assembled from 350 beam elements, two connector elements (revolute joint) and six point masses, added in nodes corresponding to the structural joint locations.The housing of semi-active joints has been modeled in a simplified manner using beam elements, whereas a connector element has been used to model bending stiffness of the active end of the joints, i.e. the ends connected by a transverse element. Those ends can operate in two states, referred to in this paper as a framemode (transferring the bending moment), anda truss mode (not moment-bearing).The boundary conditions have been defined by fixing two supporting points on the left hand side of the structure.

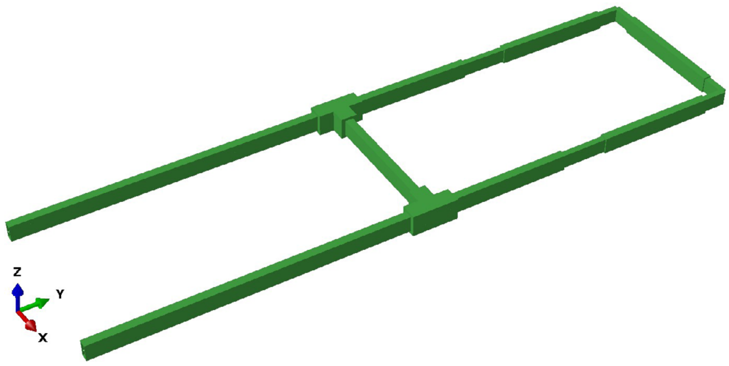

Figure 1: Numerical model of the experimental setup.

In order to identify the properties of the semi-active joints, the model updating procedure has been conducted in two steps. In the first step, the housing stiffness of the joint has been identified, whereas in the second stage, thebendingstiffness of the active endhas been investigated.The updating procedure has been based on the results obtained from the experimental modal analysis, however, only those natural frequencies for which the mode shapes remained in the XY plane of the structure have been taken into account (neither the torsional forms nor the flexure forms in the ZY plane have been included in the process of identification).Moreover, it has been assumed that the semi-active jointsare identicalandhave the same stiffness parameters. For the purpose of the first step, thesemi-active nodesduring the experiment have been switched to truss modeto minimizethe influence of the bendingstiffness of the active end on the natural frequencies. Then, the optimization problem has been formulated in which equivalent stiffness of the housing joint is found by minimizing the differences between the appropriate natural frequencies.Good agreement between the numerical and experimental results has been achieved as shown in Table1. Figure 2 shows the first four natural frequencies and mode shapes determined from the updated model.

Table 1: Comparison of experimental and numerical natural frequencies obtained in the first step.

Mode	Measured frequency [Hz]	Computed frequency [Hz]	MAC value
1	11.3	11.30	0.98
2	39.4	39.44	0.99
3	121.7	119.30	0.90
4	152.8	162.40	0.80

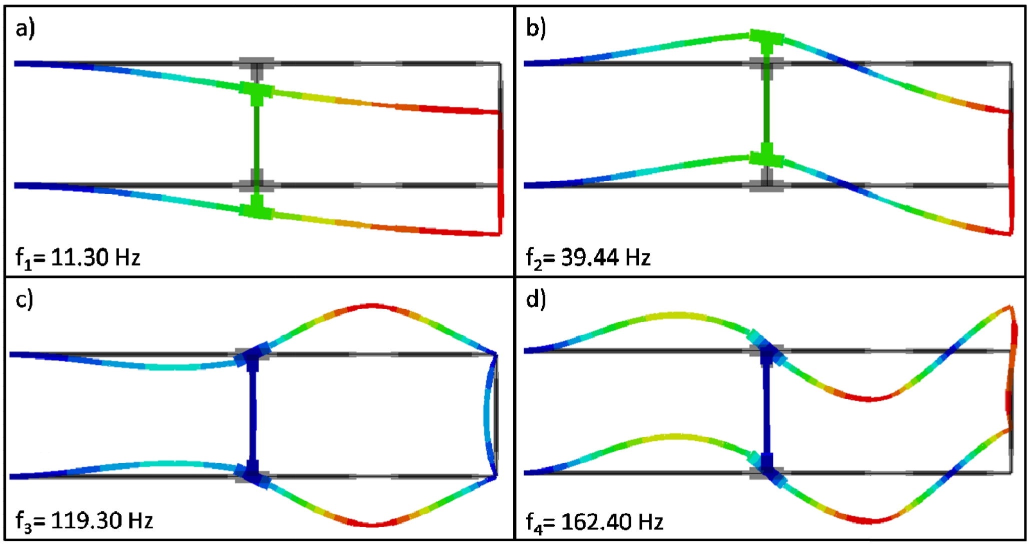

Figure 2: First four mode shapes of numerical model.

In the second step, the semi-active nodes have been switched to frame mode in order to perform identification of bending stiffness of the active end. The change of the operating state of nodes increases the rigidity of the structure, which is reflected in the natural frequencies as shown in Table 2. It has been observed that the all first fournatural frequency have increased by 28%,1.6%, 5%, 2.3% respectively, but the forms of vibrations have remained similar in its nature.

Table 2: Comparison of experimental and numerical natural frequencies obtained in the second step.

Mode	Measured frequency [Hz]	Computed frequency [Hz]	MAC value
1	14.5	14.47	0.99
2	40.1	40.7	0.99
3	125.3	126.67	0.91
4	166.3	177.98	0.89

4 EXPERIMENTAL INVESTIGATION

The experimental verification of the investigated concept was conducted in a laboratory environment. The objective was to propose for tests a slender structure with a low level of natural material damping in order to increase its susceptibility to mechanical excitations. The chosen mechanical structure was a steel frame mounted in a cantilever configuration, as depicted in Fig. 3. The frame was structured as two segments of the dimensions of 300 mm x 600 mm, which gave the total span of the structure equal to 1200 mm. The individual elements were designed as steel profiles 15 mm x 30 mm in a cross-section. The frame connections have been made by passive joints positioned in points P8 and P16 and semi-active joints positioned in points P4 and P12 (see in Fig. 3). The complete structure was mounted on a stiff support in points P1 and P9.

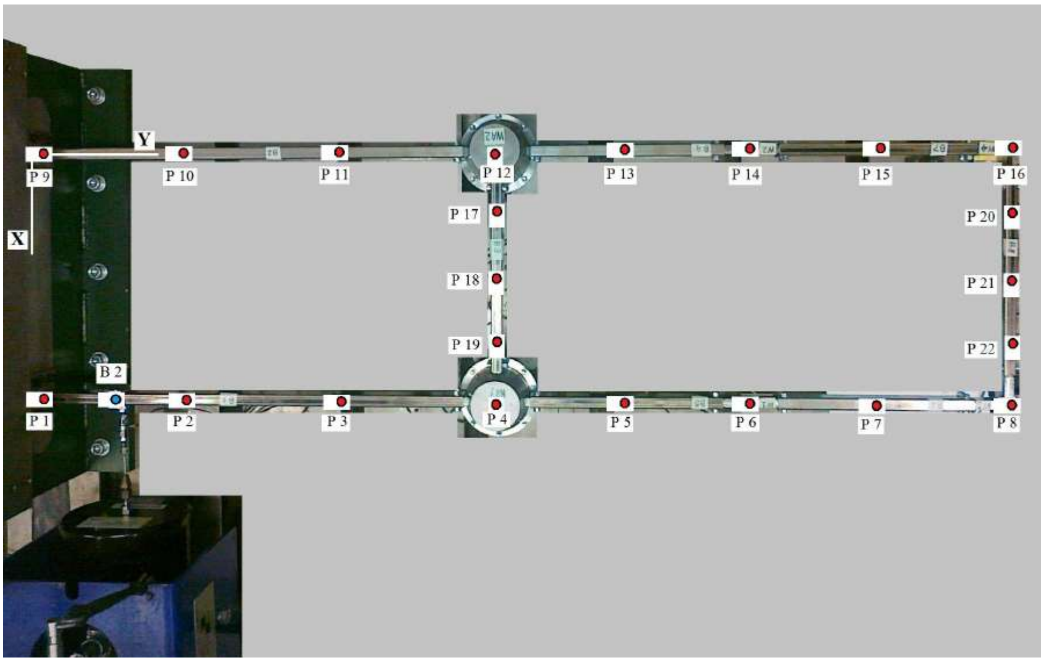

Figure 3: Experimental setup.

The semi-active joints were designed with three connecting ends, two of which were passive, elastic and the third one was semi-active, allowed to obtain an elasto-ideally plastic characteristic in angular movement in OXY plane (Figure 4). The semi-active joints were designed as dry friction devices with operating surfaces controlled with a piezoelectric actuator. The general mechanical response of the frame was modified in the domain of structural stiffness in the OXY plane. The semi-active joints were operated in two modes: frame and truss. In the case of the frame mode, the semi-active ending behaved in accordance with an elastic constitutive law. When the semi-active joints were operated in the truss mode, the joints did transmit 20 times lower torque in comparison to the another mode.

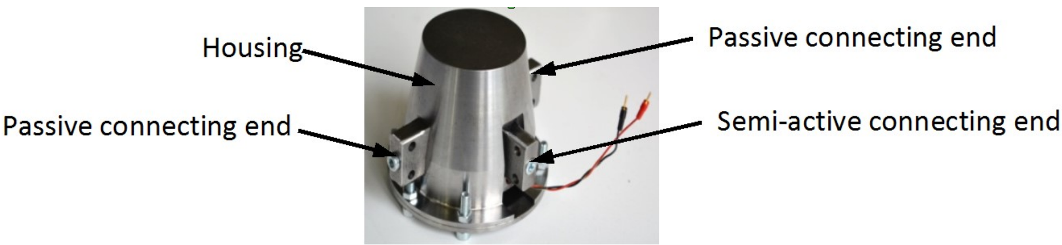

Figure 4: Experimental setup.

The frame was instrumented with a set of vibration type accelerometers positioned in points depicted as P1 – P22 (Fig. 3). The excitation of the frame was carried out by a modal shaker and a modal hammer.

The objectives of the experimental tests conducted on the frame were: a) to characterize the object in the domain of the frequency response and b) to reveal the controllability of the system. The frame was tested in accordance with two routines: operational modal analysis and frequency response function. The operational modal analysis was conducted in order to characterize the dynamical response of the structure and the FRF analysis was conducted in order to define the control space of the object and to characterize the semi-active control strategy.

Assuming danger of externally forced critical harmonic excitation (eg. close to the first eigen-frequency 14.5 Hz), reduction of the vibration amplitude for the monitored frame by deactivation of the semi-active joints and transition between frame and truss modes has been analyzed. The moment of deactivation of the joints were chosen on the basis of online FFT analysis, and the increasing of the amplitude. The control strategy adopted for the demonstration task was based on on-line monitoring of the dominant frequency of the excited vibration. Figure 3 depicts a comparison between the frequencyresponses of the frame in frame and truss modes. The transition in values of the eigen frequencies in dependence on the mode of operation are reflected in the position of the peaks on the plot.

The result of the control program execution for the 1st eigen mode is depicted in Fig. 6 and for the 2nd eigen mode in Fig. 7. The control strategy was to switch the semi-active joints operation from the truss to the frame mode at the moment dependent on the frequency response of the structure.

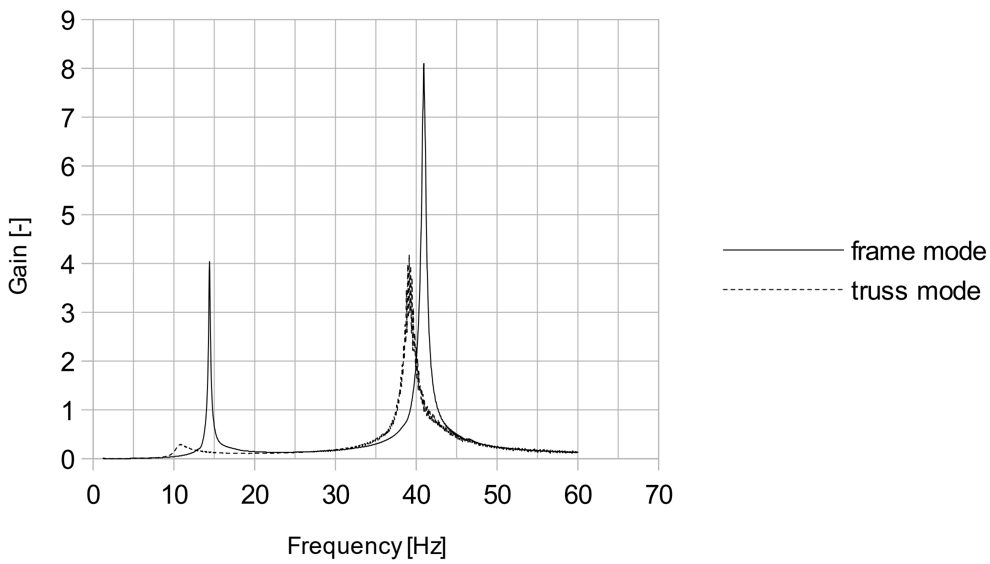

Figure 5: Frequency response of the frame for the 1st and 2ndeigen mode (measurement point P16).

The presented results reveal that the proposed semi-active control approach allows to reduce the amplitudes of the structure vibration of the resonance transition phase effectively. The gain attenuation magnitudes are presented in Table 3.

Table 3: Gain attenuations as a result of the control implementation.

	1^{st}eigen frequency	2^{nd}eigen frequency
Gain attenuation	23.76	4.02

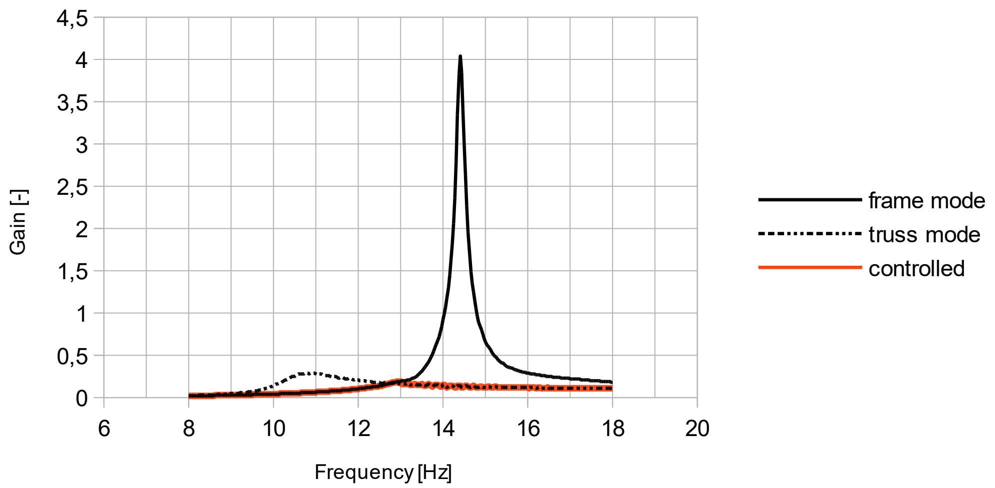

Figure 6: Frequency response of the controlled frame for the 1^{st} eigen mode (measurement point P16).

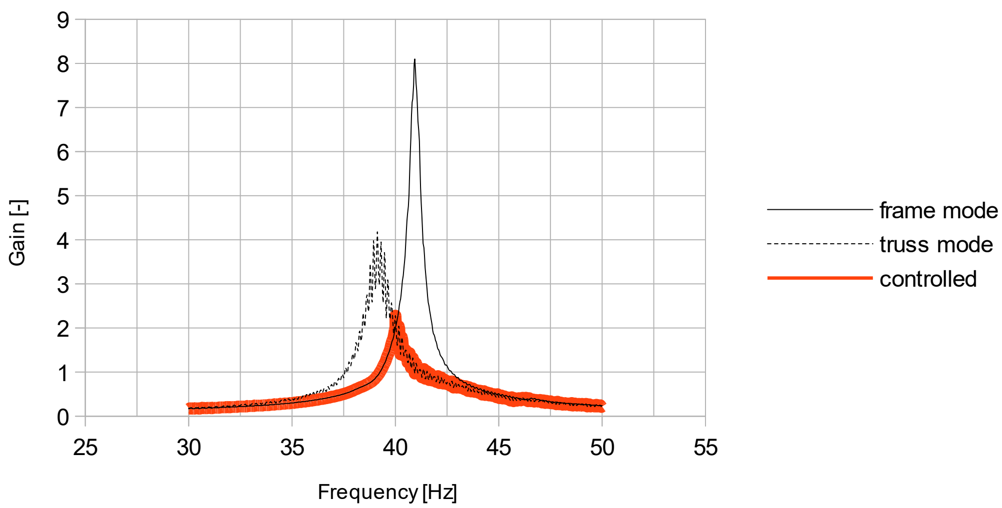

Figure 7: Frequency response of the controlled frame for 2^{nd}eigen mode (measurement point P16).

5 CONCLUSIONS

The case study demonstrated in this paper is motivated by the following, practical situation. Externally forced harmonic excitation (with frequency over 45 Hz) transmitted to the flexible frame structure (eg. satellite on the orbit) can be dangerous if slowing down and passing with decreasing frequencies through the second structural eigen frequency (40,1 Hz) and the first one (14,5 Hz). Our proposed switching off the adaptive joint allows us to pass safely the critical frequencies. The analyzed case study shows that the corresponding small shifting of the structural eigen-frequencies (between the frame-like and the truss-like structures) gives us as the result dramatic improvement in gain attenuations (4 times in the case of second mode and over 23-times in the case of the second mode.

It can be also demonstrated, that properly tuned initial stresses induced in the structure via initial distortions (incompatibilities) forced in one location can effectively reduce vibration amplitudes in other, selected location. However, the side effect of such operation can be increase of vibration amplitudes in other location on the structure. One can say, that prestressing technique can be effective in local isolation of structural vibration amplitudes, caused by some critical, external excitation. Important advantage of the proposed technique, especially for aeronautic and aerospace applications, is a new option for reducing unwanted vibrations, without increasing the structural mass.

The magnitude of modification of the dynamic, structural response depends on overall structural stiffness and sensitivity of defined objective function for admissible structural modifications in selected localizations. The above results motivate authors for further, more methodological studies on control of structural dynamic response, based on modifiable prestressing technique, what will be published in a separate paper. The challenges to be overcome require including material damping effect, what will make the computational task more heavy, but also will lead to more realistic results. Also, the problem of prestress optimization (location for generation of prestress distortions and their magnitude) and technologies for prestress inducing (preferable in a modifiable manner) are open questions.

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