Development of laboratory model of wind turbine’s tower-nacelle system with magnetorheological tuned vibration absorber

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Abstract. The paper addresses the consecutive development stages of laboratory model of wind turbine’s tower-nacelle system with horizontally aligned tuned vibration absorber at its top. To cope with system uncertainties and possibly multiple modes of vibration, tuned vibration absorber is equipped with MR damper instead of passive viscous one. Several laboratory model constraints have to be fulfilled. Discrete frequency-based and Comsol-Simulink analyses were conducted to determine and verify model parameters. Finally, sketch of laboratory test rig design was presented.

Introduction

Wind turbines sector is one of the most emerging ones among renewable energy extraction sources. The structures build nowadays are getting higher and higher, while generator power values are of the order of megawatts [6]. The wind (and possibly sea waves) load that is varying in time as well as rotation of turbine elements, are the major sources of structural vibration of tower and blades. Present paper deals with tower vibration only, while all turbine components (nacelle, blades, hub, shaft, generator and possibly gearbox) are represented by nacelle mass and mass moments of inertia.

Damping ratio of the first two tower bending modes is significantly low, usually less than or equal to 0.5%, excluding aerodynamic damping [2][5][12], while aeroelastic damping is usually greater than 5% for the 1st tower longitudinal mode, and greater than 0.5% for the 1st tower lateral mode (9% and 1%, respectively, for 10 MW wind turbine at wind speed of 25 m/s) [1][5]. That is why lateral tower vibration reduction is often of the major concern, while its excitation is due to Karman vortices, generator operation, sea waves variable load and rotation elements unbalance rather than due to direct wind load variation. It is also worth to note that the first mode has dominant modal mass participation – for 0.5 MW wind turbine analysed in [2] participations of the first five modal masses are as follows: 0.609, 0.124, 0.078, 0.024, 0.030 (respectively). The tower deflection amplitude of typical 2 MW wind turbine is of the order of 0.4 m.

Two main solutions utilised to reduce wind turbines’ towers vibration are: collective pitch control of the blades (cancellation of 3p excitation arising due to: differences in inflow conditions for each of the blades [13] and blade passing efect), and tuned vibration absorbers (TVAs) [4]. TVAs are widely spread structural vibration reduction solutions for slender structures, including towers, buildings, chimneys, etc. In the standard approach, TVA is being installed at the top of the slender structure (where the 1st mode deflection is maximal), and is equipped with an additional moving mass, spring and viscous damper, which parameters are tuned to the selected (most often first) mode of structure vibration [4][11]. However, frequency response of such low-damped structures as wind turbines’ towers exhibits significant fluctuations [2], thus more advanced TVA approaches consider adaptive stiffness and damping solutions to change/tune TVA operating frequency. Among these solutions, magnetorheological (MR) TVAs are placed. MR TVAs are TVAs equipped with MR dampers instead of passive viscous dampers.

MR dampers are semiactive solutions characterised with simplicity of construction and minor energy requirements as compared with active systems. When such a damper deteriorates, it usually still behaves as a passive device, while failure of active system requires urgent repair service. MR dampers utilise specific properties of MR fluid, which is a suspension of high-concentration soft magnetic particles in nonmagnetic carrier, thus it changes its apparent viscosity in the presence of magnetic field. MR dampers that are filled with such a fluid and equipped with electrical windings to generate magnetic field, provide a wide range of resistance force, millisecond response times, low sensitivity to temperature change and fluid contamination. They give the possibility to shape operation characteristics with the use of real-time control systems [7].

The aim of current research project is development of wind turbine's tower-nacelle model and vibration reduction system for it. Vibration reduction system comprises MR TVA with appropriate MR damper real-time control algorithm. The disturbance load concentrated at the nacelle is provided by the dedicated shaker. The schematic diagram of the system idea is presented in Fig. 1.

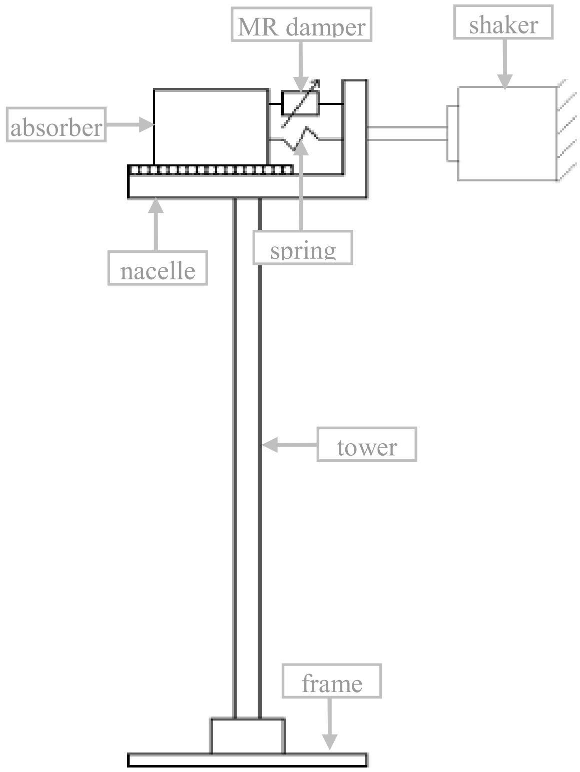


Fig. 1. Schematic of the laboratory model

Laboratory model constraints

The consecutive stages of the development process of wind turbine's tower-nacelle laboratory model equipped with MR TVA are presented here. The design constraints of the laboratory setup arise based on the rig requirements. These are as follows:

(a) possibility to apply disturbance load of changeable maximum value, and changeable maximum travel stroke, according to the shaker specifications and testing conditions' requirements,

(b) possibility to unplug the disturbance load after vibration excitation, to acquire free vibration response of first bending mode (and possibly also second bending mode),

(c) value of bending mode(s) frequency(ies) should be reasonably low, to cope with time delays of feedback loop (MR damper and sensors/conditioners time constants, sampling period, etc.) while being reasonably high to cope with shaker and sensors operating ranges; thus the first bending frequency should be located at 3÷5 Hz range,

(d) adequate damping ratio of system's first bending mode(s) to limit tower deflection amplitude while MR TVA system is not active or MR damper was locked / applied current to high (limit due to MR damper stroke and tower bending strength) and applied disturbance load is adequately high to observe reasonably big tower deflection amplitude for active MR TVA system (minimum reasonable MR damper operation displacement amplitude constraint),

(e) reasonably high yield strength (Re) of tower material and reasonably high tower section modulus in bending (limit due to high bending torque at tower bottom section),
(f) mass of the gondola has to be adequately high to lower values of bending frequencies of the tower-nacelle system, while maintaining reasonably high factor of safety for tower buckling,
(g) mass of the absorber has to be adequately high to enable reduction of tower deflection amplitude, while respecting limited MR damper stroke,
(h) MR damper resistance force constraints: minimal and maximal,
(i) MR damper maximum piston velocity,
(j) maximum shaker force, and maximum shaker stroke limitations,
(k) height of the laboratory test rig is limited by the laboratory ceiling height of 3.29 m, considering future possibility to lay rig down on the horizontally moving platform, modelling excitation of seismic-type, or sea-waves-type (for buoy floating wind turbine model),
(l) mass of the laboratory test rig is limited by the foundation specification – cyclic dynamic loads should be minimised to eliminate the risk of foundation motion,
(m) the laboratory setup, including its mechanical hardware, shaker, and MR damper used should be of commercially available type to fit within budget limitations, and possess adequate reliability and repeatability of operation characteristics,
(n) at least partial dynamic similarity (similarity of motions of one pair of points – tower tips) between full-scale wind turbine's tower-nacelle system, and its laboratory model, has to be fulfilled, which adds additional constraints to the above requirements – this is a subject of separate publication.

MR damper, shaker and tower material selections

As dynamic similarity analysis resulted in conclusion that number of parameters to be determined is greater than number of equations to fulfil, some experimental setup parameters (concerning e.g. scale of the model, cross-section shape) may be assumed arbitrary. Regarding assumption (m), the most crucial element of the system is MR damper to be build in the TVA system, as number of commercially available types, including force and stroke ranges, is very limited. According to the preliminary simulation analyses of various laboratory model configurations that can fulfil assumptions both (k) and (l), because of the relatively low force required for TVA operation, RD-1097-01 damper type by Lord Co. was selected. The data sheet of RD-1097-01 damper is presented in Tab. 1 [10].

Tab. 1. Data sheet of RD-1097-01 damper (selection) [10]

Compressed Length	7.68 inches (195 mm)
Extended Length	9.96 inches (253 mm)
Body Diameter	1.26 inches (32 mm)
Weight	1.1 pound (0.48 kg)
For Installation on Pin	0.31 inches (8.0 mm)
Input Current (continuous)	0.5 amps maximum
Input Current (intermittent)	1.0 amps maximum
Resistance (25° C)	20 ohms
Damper Forces: (Peak to Peak) 2 in/sec at 1 amp 8 in/sec at 0 amp	 > 22 pounds (100 N) < 2 pounds (9 N)
Maximum Operating Temperature	160°F (70°C)
Durability	2 million cycles @ ± 0.5 inches (± 13 mm), 2 hertz with input current varying between 0 and 0.5 amps.
Response Time (amplifier and power supply dependent)	< 25 msec – time to reach 90% of max level during a 0 to 1 amp step input @ 2 in/sec (51 mm/sec).

Data from Tab. 1 enables to define values for constraints (d)&(g), (h), and (i) to be, respectively: damper stroke 58 mm pk-pk, damper resistance force: minimal of 2÷9 N (depending on the piston velocity) and maximal continuous of the order of 100 N, and damper maximal piston velocity of the order of 200 mm/s (order of maximum velocity value previously tested, and also mentioned in [10]).

Based on assumption (c) and TVA damper force and stroke ranges, shaker selection was done. To cope with low frequency operation (also below the first tower bending frequency), and potential possibility of both vertical and horizontal alignment, modal type of shaker was selected, namely The Modal Shop's lightweight electrodynamic exciter of 2060E series [14]. Its specifications are grouped in Tab. 2.

Tab. 2. Data sheet of 2060E exciter (selection) [14]

Output Force, sine pk, ambient air cooling	30 lbs (133 N)
Output Force, sine pk, forced air cooling	60 lbs (267 N)
Stroke Length, pk - pk	1.4 in (36 mm)
Frequency Range, nominal	DC - 6,000 Hz
First Resonance Frequency, nominal	> 4,000 Hz
Maximum Acceleration, bare table	100 g (1000 m/s2) pk
Maximum Velocity	120 ips (3 m/s) pk
Effective Armature Mass	0.6 lb (0.272 kg)
Dimensions (H x W x D), nominal	10.8 x 12.6 x 6.5 in (273 x 319 x 165 mm)
Weight	37 lbs (17 kg)
Operating Range	40 - 100°F (4 - 38°C), < 85% RH

The general view of RD-1097-01 damper and 2060E shaker is presented in figure 2 (a) and (b).

(a)

(b)

Fig. 2. Test rig elements: (a) Lord RD-1097-01, (b) TMS 2060E [10][14]

The absorber mass is selected to be 10% of the modal mass of the first bending mode of tower-nacelle model (with the possibility to decrease it to 5% or 2% of the modal mass of the first bending mode). This percentage is known to give close to maximum vibration reduction efficiency of TVA system – further increment of it has minor effect. To investigate absorber/modal mass percentage of 2% and 5%, absorber is to be build as a set of steel plates with possible removal of some of them with simultaneous TVA's spring replacement and MR damper control alteration. After acceptance of that assumption and previous selection of damper and shaker types, already defined requirements (g) – (n) enforce fixed demands on the laboratory test rig, i.e. on tower material properties and dimensions/geometry as well as on mass of the gondola, to fulfil assumptions (c) – (f) which are unspecified till now. Assumptions (a) and (b) will be fulfilled by means of the appropriate mechanical configuration of the excitation transmission system (described in the *Test rig design* section).

Selection of tower material is an important question. The main requirements are relatively high yield strength Re and adequate internal friction Q^{-1}. Generally, alloy steels with high Re value exhibit very low internal friction, and vice-versa. Additionally, requirement (h) of MR damper minimal force value leads to the demand of adequately high stiffness of the tower structure (i.e. adequately high value of Young modulus E and section area moment of inertia I for the assumed tower height L), which also guarantees holding MR damper stroke and piston maximum velocity limitations under adequately high disturbation loads. Tower stiffness augmentation by shortening its height leads to the increase of bending modes frequencies (see (c)), which can be decreased back to the certain point by nacelle mass augmentation.

The selection of alloy steel seems an inevitable compromise at this point. Considering actual commercial availability (see (m)) and literature presence of internal friction tests results, compromise duplex steel of 1.4462 (X2CrNiMoN22-5-3) type was selected. Its properties are as follows: Re > 450 MPa (typically 550 MPa), E = 200 GPa, $Q^{-1} \approx 4.4 \cdot 10^{-4}$ [-] at 1.5 Hz excitation [9]. As a more expensive alternative, titanium grade 5 (Ti Gr. 5) alloy (Ti-6Al-4V) was proposed. Its properties are: Re > 825 MPa (typical 910 MPa), E = 110 GPa, Q^{-1} may be assessed to be ≈ $4.4 \cdot 10^{-4}$ [-] (at ca. 1.5 Hz excitation) based on [3]. The yield strength of Ti Gr. 5 is superior to that of 1.4462 steel, while internal friction is supposed to be pretty much similar ([3][8] [9]), and Young modulus is higher for 1.4462.

Taking into account 1.4462 steel and Ti Gr. 5 tower material variants, two configurations were considered: Conf. 1 and Conf. 2, respectively. As a next stage, calculation analysis for these configurations was conducted with discrete frequency method (1)-(8) applied for the 1st bending mode only, full continuous-discrete analytical approach (infinite number of modes) based on [11] (and presented in separate publication covering dynamic similarity), and method using Comsol Multiphysics FEM tower-nacelle model embedded within MATLAB/Simulink environment with TVA (MR TVA) model build there (analysis of the first two bending modes).

Mass and geometry parameters selection

For the first mode of vibration analysis only, discrete equations of dynamics (1)(2) were written assuming m_1, c_1, and k_1 are modal mass, damping, and stiffness associated with this mode of vibration. By m_2, c_2, and k_2, mass, damping and stiffness parameters of TVA system were designated. The parameters of TVA were tuned for the first bending mode [4][11].

$$m_1\ddot{x}_1(t) + c_1\dot{x}_1(t) + k_1x_1(t) + c_2\left(\dot{x}_1(t) - \dot{x}_2(t)\right) + k_2\left(x_1(t) - x_2(t)\right) = P(t) \tag{1}$$

$$m_2\ddot{x}_2(t) - c_2\left(\dot{x}_1(t) - \dot{x}_2(t)\right) - k_2\left(x_1(t) - x_2(t)\right) = 0 \tag{2}$$

where:

$$x_1(t) = A\cos(\omega t + \varphi) \tag{3}$$

$$x_2(t) = B\cos(\omega t + \psi) \tag{4}$$

are horizontal displacements of nacelle (tower tip) and absorber with amplitudes A and B, respectively, assuming small angles of tower bending, while:

$$P(t) = P_x\cos(\omega t) \tag{5}$$

is horizontal disturbance force of amplitude P_x (Fig. 1). Transforming (3) and (4) one may obtain:

$$x_1(t) = A_1\cos(\omega t) + A_2\sin(\omega t) \tag{6}$$

$$x_2(t) = B_1\cos(\omega t) + B_2\sin(\omega t) \tag{7}$$

where: $A_1 = A\cos\varphi$, $A_2 = -A\sin\varphi$, $B_1 = B\cos\psi$, $B_2 = -B\sin\psi$

Substituting (5)(6)(7) to side-by-side difference of equations (1)(2), and to equation (2), and comparing cosine terms and sine terms separately, one may obtain a matrix equation:

$$\begin{bmatrix} k_1 - m_1\omega^2 & c_1\omega & -m_2\omega^2 & 0 \\ -c_1\omega & k_1 - m_1\omega^2 & 0 & -m_2\omega^2 \\ -k_2 & -c_2\omega & k_2 - m_2\omega^2 & c_2\omega \\ c_2\omega & -k_2 & -c_2\omega & k_2 - m_2\omega^2 \end{bmatrix} \begin{bmatrix} A_1 \\ A_2 \\ B_1 \\ B_2 \end{bmatrix} = \begin{bmatrix} P_1 \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{8}$$

to get amplitudes: $A = \sqrt{A_1^2 + A_2^2}$ and $B = \sqrt{B_1^2 + B_2^2}$ as functions of angular frequency ω.

The selections of system elements and properties (MR damper, shaker, tower material variants) described above, together with assumptions (c) – (f) and calculations based on [11] and on discrete frequency method (1)-(8), all made it possible to determine tower geometry (thus also mass) and mass of the nacelle. Although the ratio of area moment of inertia to cross-section area is higher for circular tube than for circular rod, greater deflection amplitudes at the same maximal bending stress and the same tower's height are obtained for circular rod and this is the selected tower's shape. Additionally, commercial price for Ti Gr. 5 tube of comparable stiffness and length is higher than for rod.

Apart from Conf. 1 and Conf. 2, as a result of dynamic similarity analysis (published as a separate paper), Conf. 3 based on Ti Gr. 5 tower material was also introduced. Parameters of the three model configurations which were determined on the basis of above mentioned calculations and requirements are all presented in Tab. 3. The amplitude frequency characteristics obtained by discrete method for the three laboratory model configurations are presented in Fig. 3 and Fig. 4.

Tab. 3. Parameters of the three model configurations

Parameter	Conf. 1	Conf. 2	Conf. 3
Tower material	1.4462 steel	Ti Gr. 5	Ti Gr. 5
Tower height [m]	2.50	1.50	1.50
Tower diameter [m]	0.90	0.70	0.65
Tower mass [kg]	124.05	25.57	22.05
Nacelle mass w/o TVA [kg]	149.69	150.59	149.89
TVA absorber mass [kg]	17.78	15.64	15.49
TVA stiffness [N/m]	10 220.8	9 523.9	7 080.7
TVA damping [Ns/m]	143.12	129.56	111.17

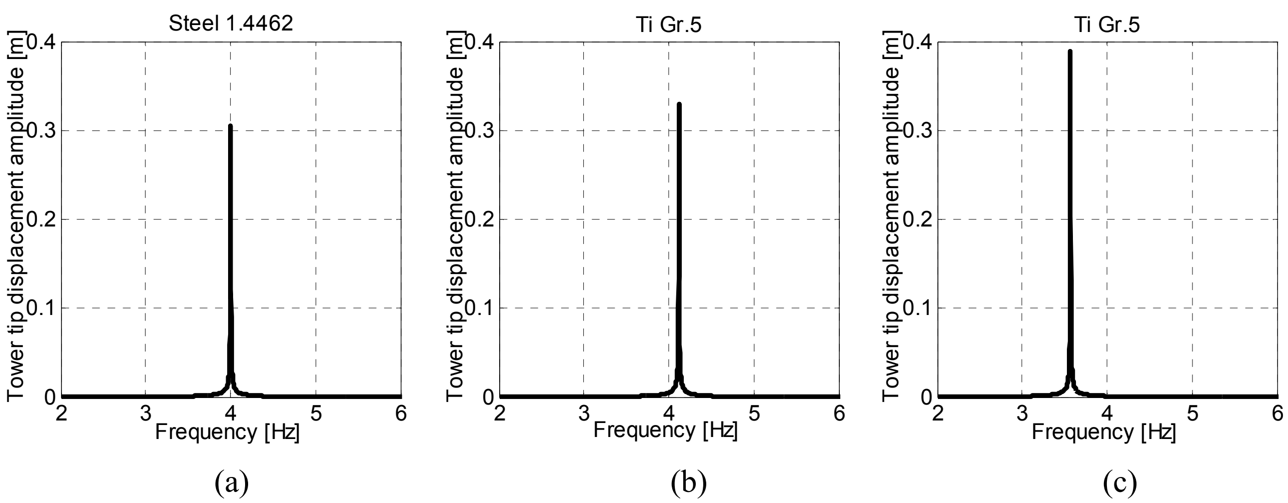
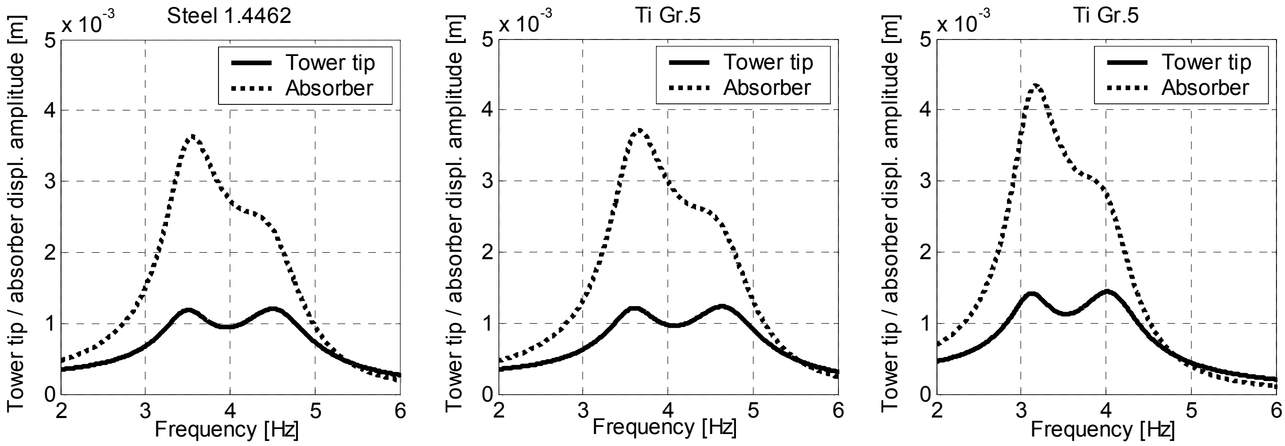


Fig. 3. Tower tip displacement amplitude – frequency characteristics (TVA locked): (a) Conf. 1, (b) Conf. 2, (c) Conf. 3



(a) (b) (c)

Fig. 4. Tower tip / absorber displacement amplitude – frequency characteristics (TVA active): (a) Conf. 1, (b) Conf. 2, (c) Conf. 3

Each of the configurations provides at least 1 mm tower tip displacement amplitude for MR TVA system being active (Fig. 4) when applied sine disturbance load of the 1st bending mode frequency and maximum amplitude of force (P_{x1}=32.0 N for Conf. 1, P_{x2}=30.5 N for Conf. 2, and P_{x3}=26.5 N for Conf. 3) that is safe in the case of MR TVA system being locked (e.g. when incidental overcurrent is fed to MR damper's coil) (see (d)). The maximum amplitudes of tower tip oscillations, when MR TVA system is active, are in detail: 1.21 mm for Conf. 1, 1.23 mm for Conf. 2, and 1.44 mm for Conf. 3. The maximum force values and travel ranges listed above are obtained when shaker / nacelle travel ratio (see *Test rig design* section) is $1:4^{1}/_{7}$ (Conf. 1), $1:4^{1}/_{3}$ (Conf. 2) or 1:5 (Conf. 3) – these ratio values assure respecting shaker travel limits while TVA system is incidentally locked, for all configurations.

For MR TVA system being not active (or locked), maximum amplitudes of tower tip oscillations (Fig. 3) are: 0.31 m for Conf. 1, 0.33 m for Conf. 2, and 0.39 m for Conf. 3. However, for laboratory test rig design purpose, maximum amplitudes of tower tip displacement were assumed smaller so as: (i) excitation frequency was shifted from 1st bending mode frequency by few milihertz. The assumption (i) was justified by several conditions. Firstly, nonlinearity of the material and thus nonlinearity of the resonance curve makes theoretical amplitude peak value impossible to be observed at constant excitation frequency (e.g. for Ti Gr. 5 it is known that fast strain hardening phenomenon occurs, thus the theoretical 1st bending mode frequency moves to the right while deflection amplitude increases, and peak amplitude frequency will be shifted from the theoretical one). Secondly, theoretical calculations consider damping due to the internal friction of the material only, while in a real case external friction due to interfacial slip at all joints adds more damping to the structure. Thirdly, excitation frequency resolution of the available sine pattern generator device is limited to 10 mHz. Recapitulating, the assumption (i) seams quite conservative and it is hardly expected to observe peak amplitudes of displacement and bending stress presented below to be exceeded, although end stop collision bumpers are considered to eliminate any damage risk. The assumed maximum amplitudes of tower tip oscillations (calculated based on [11] and (i)) are: 0.073 m for Conf. 1, 0.077 m for Conf. 2, and 0.081 m for Conf. 3. The maximum bending moments and bending stresses at tower basis are respectively: 23.06 kNm and 323.9 MPa (Conf. 1), 13.34 kNm and 400.0 MPa (Conf. 2), 11.59 Nm and 387.4 MPa (Conf. 3) [11]. The critical buckling forces, calculated according to the Euler condition, are: 254.3 kN (Conf. 1), 142.2 kN (Conf. 2), and 105.7 kN (Conf. 3), what means they are two orders of magnitude higher than actual axial loads.

Regarding above listed numbers (especially tower height and mass, and bending moment at tower basis) and assumptions (k)(l), Conf. 2 and Conf. 3 were selected for further analysis only, both with titanium tower material. Smaller scale of the two titanium-based configurations (with deflections, strokes, frequencies, shaker and damper forces similar to that of Conf. 1) and smaller basement moments, all that made overall flexibility and price of Conf. 2 and Conf. 3 more favourable.

Comsol-Simulink verifying analysis

Based on the assumptions and results stated above, Comsol-Simulink co-simulation analyses were conducted to verify laboratory model parameters. Using Comsol Multiphysics environment, the model of tower-nacelle system was build as a beam fixed at the bottom and free at the top, with an additional mass and mass moments of inertia defined at its top. Two external concentrated loads along horizontal x-axis were defined, of amplitude Px at the top (maximum deflection for the 1st mode) and of amplitude Fx at the half of tower's height (ca. maximum deflection for the 2nd mode; with an additional shaker to be introduced in further stages). Such a model was than exported to Simulink as a MATLAB structure with tower tip displacement $x1$ and velocity $v1$ along x-axis as two output signals. The Comsol model was embedded in Simulink using *COMSOL Multiphysics Subsystem* block with *Sine Wave* generators Px and Fx as input signals, and two *To Workspace* blocks as outputs. When TVA system was active, signals $x1$ and $v1$ were fed additionally to the dynamics of passive TVA system (if MR damper is used, then it emulates passive damper characteristics determined according to [4][11]), which output forces act at the top along x-axis, thus are added with appropriate signs to the force of amplitude Px. Fig. 5 presents Simulink model of such a described system with TVA.

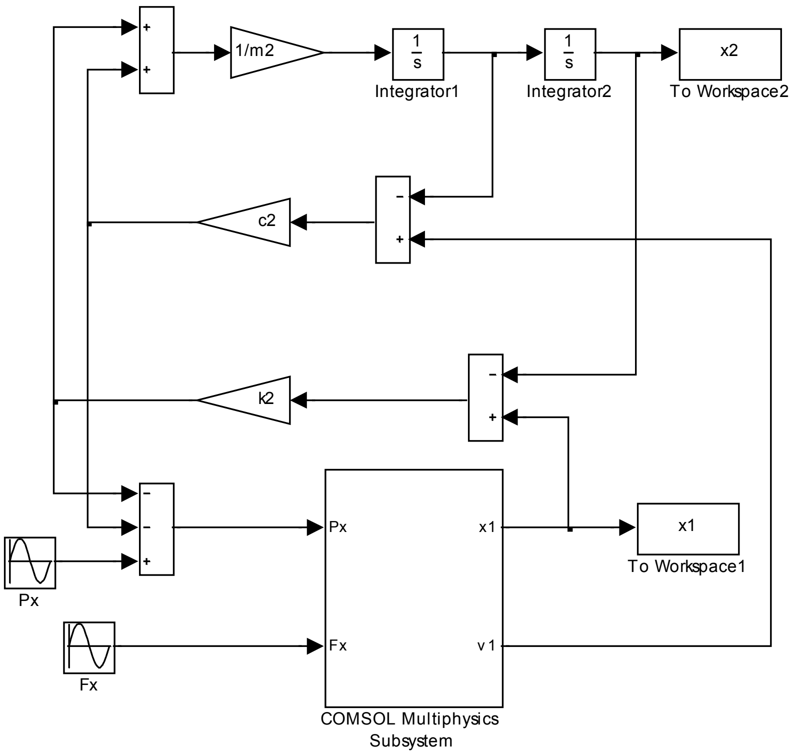
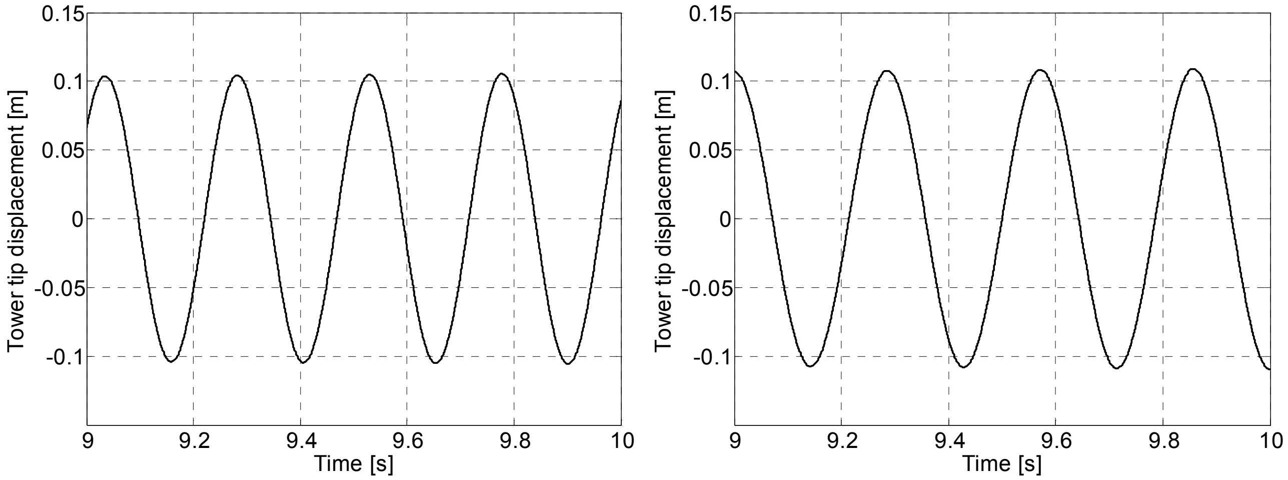


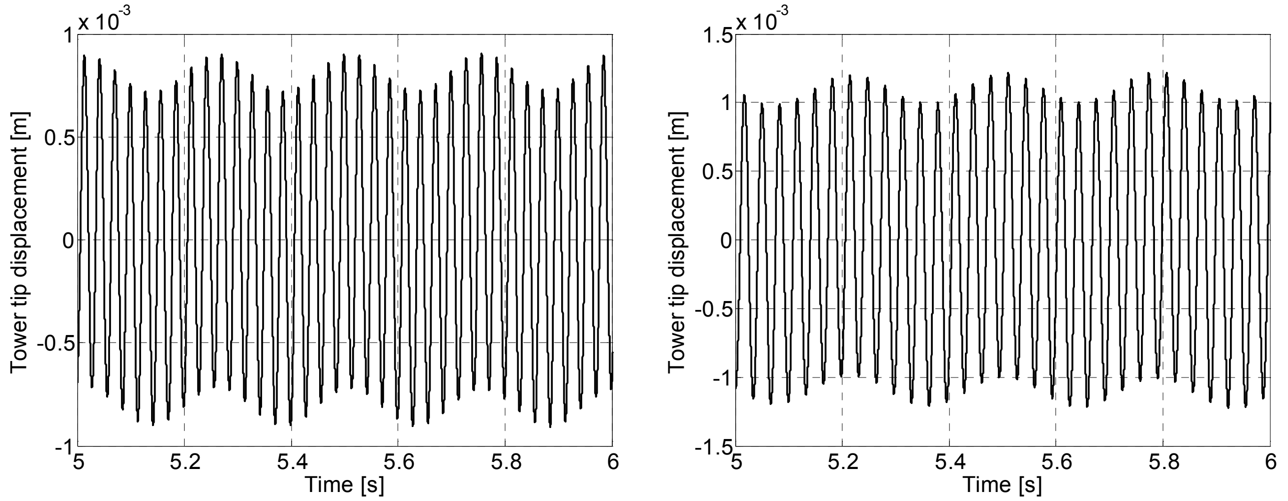
Fig. 5. Simulink diagram of the FEM Comsol tower-nacelle model with TVA

The simulations were conducted at sine excitation frequencies of the 1st (with shift according to (i)) and 2nd bending mode of vibration of Conf. 2 and Conf. 3 (frequencies determined for TVA locked) separately. These frequencies, given by Comsol Multiphysics Postprocessor, are respectively: 4.04 Hz and 35.01 Hz for Conf. 2, while: 3.50 Hz and 30.40 Hz for Conf. 3. To excite the 1st mode, it was assumed $Px=P_{x2}$, $Fx=0$ (Conf. 2) or $Px=P_{x3}$, $Fx=0$ (Conf. 3), while to excite 2nd mode: $Px=0$, $Fx=10\cdot P_{x2}$ (Conf. 2) or $Px=0$, $Fx=10\cdot P_{x3}$ (Conf. 3). The results as time patterns are presented in Fig. 6 and 7 (TVA locked), and Fig. 8 and 9 (TVA active). The 1st bending mode frequency values calculated by Comsol Multiphysics FEM software differ insignificantly from values: 4.09 Hz (Conf. 2) and 3.47 Hz (Conf. 3) calculated with full continuous-discrete model based on [11], and values: 4.12 Hz (Conf. 2), 3.57 Hz (Conf. 3) calculated on the basis of discrete approach (1)-(8) (Fig. 3). Despite some acceptable deviations from the values of frequencies and deflection amplitudes calculated with full continuous-discrete model, Comsol/Simulink co-simulation analyses give the possibility of testing sophisticated MR damper control algorithms for all (including 1st and 2nd) bending modes, in opposition to the simplified methods of discrete analysis (as frequency method presented above) and full analytical models.



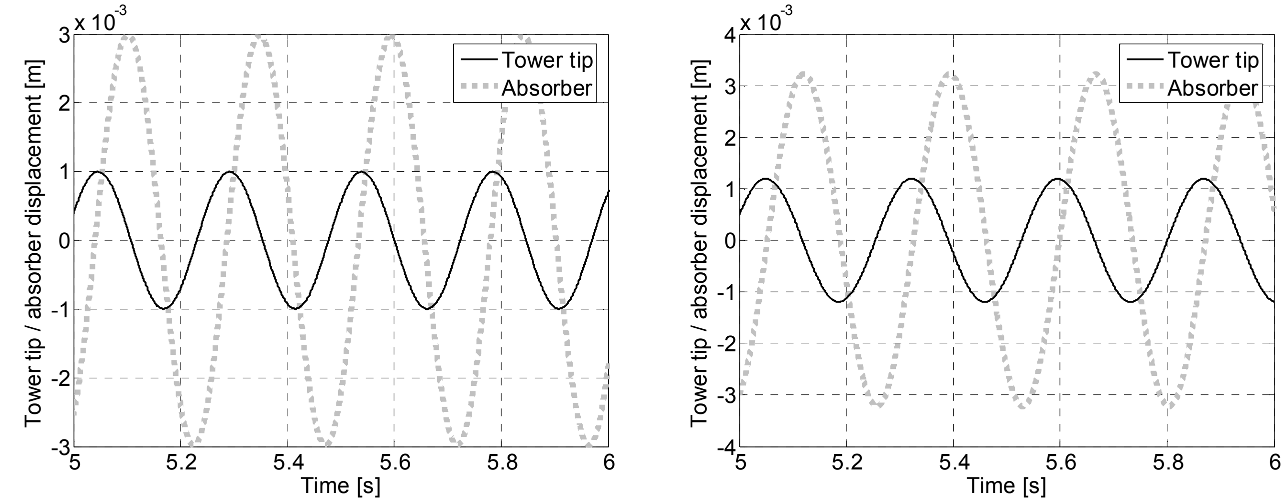
(a) (b)

Fig. 6. Time patterns for TVA locked: (a) Conf. 2 at 4.04 Hz, (b) Conf. 3 at 3.50 Hz



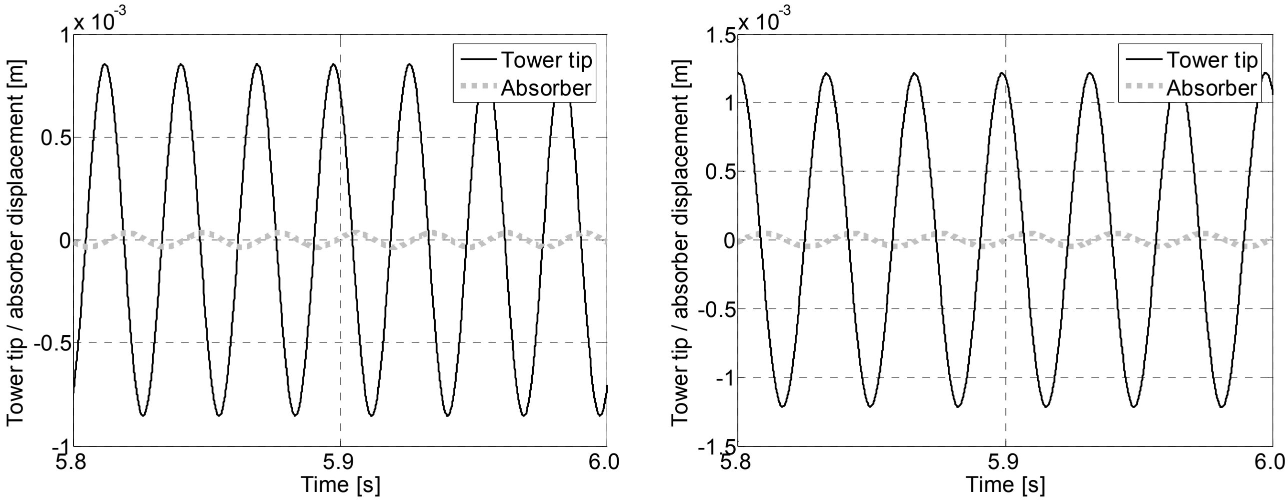
(a) (b)

Fig. 7. Time patterns for TVA locked: (a) Conf. 2 at 35.01 Hz, (b) Conf. 3 at 30.40 Hz



(a) (b)

Fig. 8. Time patterns for TVA active: (a) Conf. 2 at 4.04 Hz, (b) Conf. 3 at 3.50 Hz



(a) (b)

Fig. 9. Time patterns for TVA active: (a) Conf. 2 at 35.01 Hz, (b) Conf. 3 at 30.40 Hz

The presented preliminary results prove the potential of TVA in tower-nacelle system bending vibration reduction at frequency of the 1st bending mode, without significant deterioration at frequency of the 2nd bending mode. Moreover, for the 2nd bending mode frequency, TVA action cancels the 1st mode component of 4.04 Hz (Conf. 2, Fig. 9 (a)) and 3.50 Hz (Conf. 3, Fig. 9 (b)) visible in Fig. 7 (a)(b). Substitution of passive TVA with MR TVA will open the field for further vibration reduction by application of dedicated control solutions.

Test rig design

To cope with necessity of maximum disturbance force variation and tower deflection range (pk-pk) being greater than 36 mm (assumption (a)), shaker force is applied (horizontally) to the nacelle via right-angled triangular lightweight element (Fig. 10) enabling to increase maximum stroke within the range of 1:1 to 1:6 by moving the shaker along one of the leg of the triangle. Thus the shaker itself is aligned vertically and fixed to the auxiliary support structure serving also as a human foot-pace for nacelle access. The triangular element is supported by the auxiliary structure as well with a pin joint. Shaker excitation is being transferred via auxiliary rods, articulated joints and triangular element to the nacelle, with the help of additional system of mandrel capable of sliding inside the bushing when free vibrations are necessary to observe, according to assumption (b). When forced vibrations are required, relative position of mandrel and bushing is being fixed with the help of electromagnetic catch.

The nacelle is build as a set of steel plates fixed to the frame. Nacelle mass is distributed so as its centre of gravity is right above tower's tip. The absorber (being also a set of steel plates) is moving with the help of slide bearings and linear guides fixed to nacelle bottom frame, which is in turn fixed to tower's tip. Nacelle and absorber direction of motion is the same as direction of applied excitation (assuming small bending angles). The tower itself is aligned vertically and fixed to the ground with the help of additional plate and horizontal foundation frame made of section steel, to minimise loads transferred to the ground. Test rig side view is presented in Fig. 10 (Conf. 2), while its detailed configuration will be presented during International Carpathian Control Conference 2013 and in its proceedings.

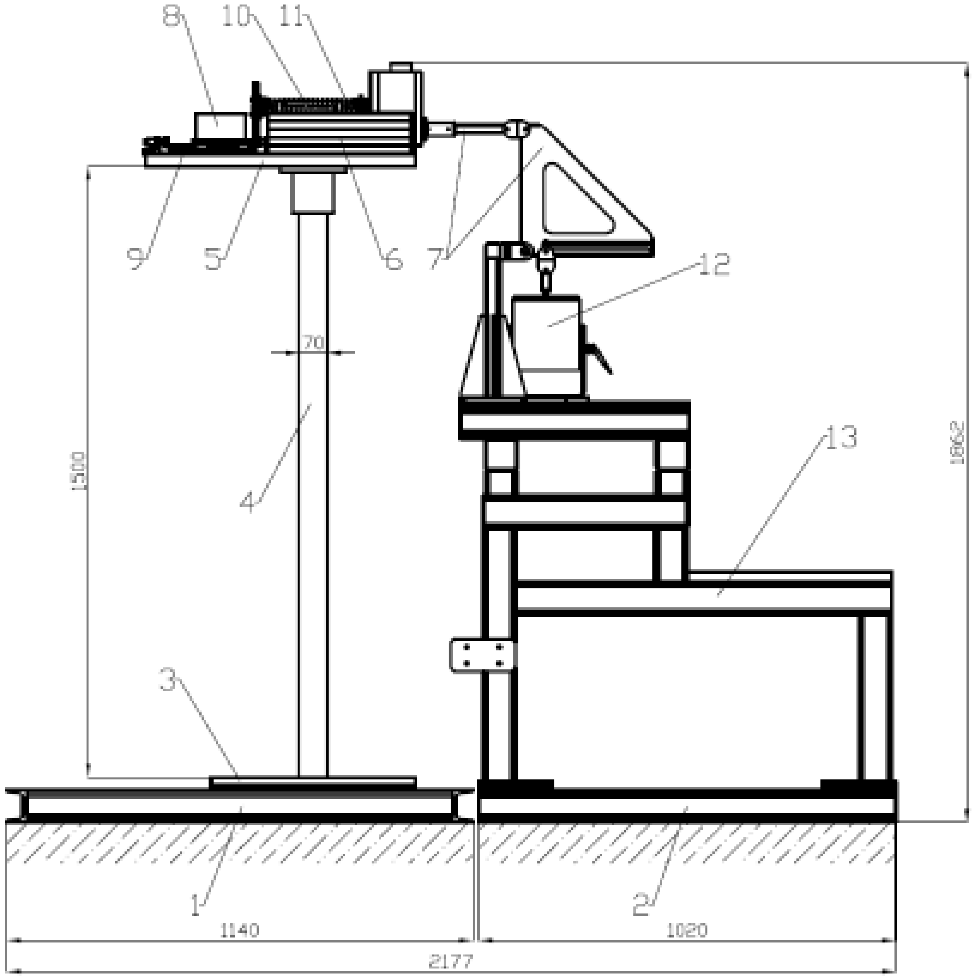


Fig. 10. Laboratory test rig, side view (Conf. 2): 1-tower foundation frame, 2-auxiliary foundation frame, 3-tower fixing plate, 4-tower, 5-nacelle frame, 6-nacelle steel plates, 7-excitation transfer system (with electromagnetic catch), 8-absorber, 9-slide bearings / linear guides, 10-MR damper, 11-spring, 12-shaker, 13-auxiliary support structure

Summary

The designed laboratory model fulfils all the requirements and constraints stated at the beginning of the development stage, according to the analytical and numerical calculations. The rig will enable analysis of the 1st and possibly the 2nd bending mode of tower-nacelle system vibration as well as implementation of MR TVA. MR TVA may operate according to the well known Den Hartog principle [4] where MR damper emulates passive damper (and possibly stiffness, to change/tune TVA resonant frequency), or MR damper may realise one of the dedicated control strategies (ground-hook, LQR, SMC, etc.) to reduce vibration to even further extend than standard TVA does. Although no data for the system with MR damper model is presented here, MR damper was considered within the whole scope of the development process.

As a separate publication, simulation data obtained thanks to embedding MR damper model within the Comsol-Simulink application, will be presented. After completing the laboratory test rig and setting it in motion, thorough tests are expected to deliver more data for identification and control analyses.

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