Technical Sciences 17(3), 2014, 249–258

PASSIVE VIBRATION DAMPING IN A TRUSS TELECOMMUNICATION TOWER

***Józef Pelc*[1], *Bronisław Kolator*[2]**

[1] Department of Mechanical Engineering and Fundamentals of Machine Design
[2] Department of Construction, Maintenance of Vehicles and Machines
University of Warmia and Mazury in Olsztyn

Received 16 April 2014, accepted 24 September 2014, available on line 25 September 2014

Key words: truss, tower, vibration, damping, finite element modeling.

Abstract

A dynamic model of a truss telecommunication tower was developed. A viscoelastic absorber was proposed for reducing structural vibrations caused by wind. Small linear system vibrations were assumed. A finite element three-dimensional model was used to determine the damper viscoelastic parameters which guarantee the highest damping effectiveness. Simulation results indicate that a damper can reduce the amplitude of tower vibrations by half and produce a similar reduction in forces acting upon the most loaded members of the structure.

Introduction

Truss structures are characterized by significant stiffness and low weight, and they are often used as supports for power transmission lines (ALBERMANI et al. 2009, SHEA, SMITH 2006) and telecommunication towers (DA SILVA et al. 2005, SULLINS 2006, BARLE et al. 2010). The structure of a telecommunication tower should safely bear the load of the service platform and the antennas, and it should effectively resist wind action. The load bearing requirements for telecommunication towers exposed to static load and wind load are set by the applicable regulations, and the forces acting upon the tower are determined in view of the relevant climate zone and orography of the location. Telecommunication towers are characterized by a considerable slenderness ratio, which is why truss structures have to be additionally checked for sensitivity to dynamic wind loads. The structure has to be protected against fluctuating character of wind-induced forces causing significant dynamic response, which may lead

* Correspondence: Józef Pelc, Katedra Mechaniki i Podstaw Konstrukcji Maszyn, Uniwersytet Warmińsko-Mazurski, ul. M. Oczapowskiego 11, 10-719 Olsztyn, phone: +48 89 523 49 31, e-mail: joseph@uwm.edu.pl

to damaging effects (HOLMES 2007). Excessive displacement of the top part of the tower can also distort signal transmission (JONES et al. 2007). According to applicable design codes, the tower sensitivity to wind loading is determined based on the values of its fundamental vibration period and the logarithmic decrement of damping. A coefficient that increases static wind load is adopted for wind-sensitive towers.

In wind-sensitive truss towers, the adverse effects of dynamic wind action are minimized by increasing the cross-section of rods, and thus the fundamental frequency in order to avoid resonant responses of the structure. Consequently, the operation increases weight of structure. Vibration dampers pose a solution for structure amplitudes decrease and they do not compromise the low weight of telecommunication towers, which is the main attribute of truss structures. In very large self-supporting lattice towers, dampers are an essential part of the structure which significantly reduce material requirements. QU et al. (2001) relied on the bi-model method to perform a dynamic analysis of a television tower with the total height of 339 m, including the mast. The tower was provided with friction dampers.

In this paper, a damped dynamic absorber was used to reduce the vibrations of a telecommunication tower. A dynamic model was developed to analyze the tower dynamic attributes and determine damper parameters. The vibration of a 3D model had to be analyzed to account for the possibility of omni-directional wind. For this reason, the analytical approach proposed by ŁATAS and MARTYNOWICZ (2012) could not be used in the study, and numerical methods implemented in Finite Element Method (FEM) software (RAKOWSKI, KACPRZYK 2005) were applied instead.

The method of reducing the vibration of a truss tower was presented on the example of a real structure comprising a 40 m-tall tower with an equilateral triangle cross section. The width of the tower sides is reduced with height. The tower is divided into seven segments which are determined by leg cross-sections.

Static and dynamic behavior of a computational model

A service platform and antennas with the combined weight of 450 kg are installed at the apex of the truss tower (Fig. 1 *a*, *b*). It has been assumed that identical point masses with total mass equal to the mass of the service platform exist at three apex points D, E and F. The tower is made of steel with Young's modulus of $E = 2.0 \cdot 10^{11}$ Pa and Poisson's ratio of $\nu = 0.3$.

The analyzed structure comprises rods which are equal-sided angle sections joined with screws. For this reason, the computational model should be

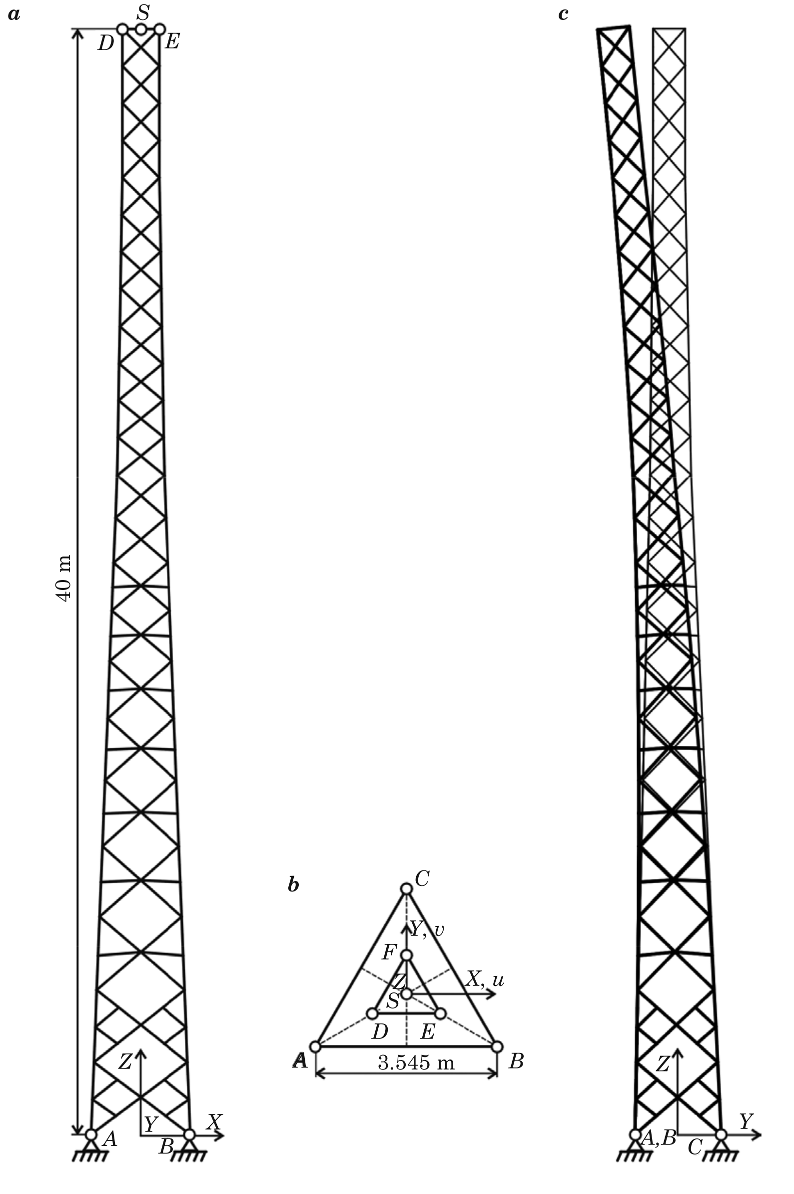

Fig. 1. Transmission tower model: *a* – *xz* view, *b* – *xy* view, *c* - 1st eigenmode

a 3D truss, but such structures cannot be analyzed by FEM in classic way due to the presence of out-of-plane nodes that contribute to the model mechanical instability. In this case, the movement of out-of-place nodes can be automatically blocked (PELC 2012) or a frame model with reduced flexural stiffness of rods can be applied. The tower is to be treated as a truss structure, and it was assumed that rods are circular cross section elements with cross-sectional area of angle sections and the corresponding moments of inertia. Three-dimensional 2-noded linear truss finite elements and elastic straight beam finite elements were applied in truss model and in frame model of the structure, respectively. Proportional internal (material) damping model was used. A structural damping coefficient of $\xi = 0.07$ was adopted for steel structures fastened with screws (ADAMS, ASKENAZI 1999).

The MSC Marc FEM program was used in simulations. An initial analysis of the tower mechanical properties involved determination of its compliance with a horizontal force of variable direction applied to apex point S (Fig. 1). The displacement δ of an apex point of an equilateral triangular truss was identical to that observed at the end point of an axisymmetric cantilever column (Fig. 2).

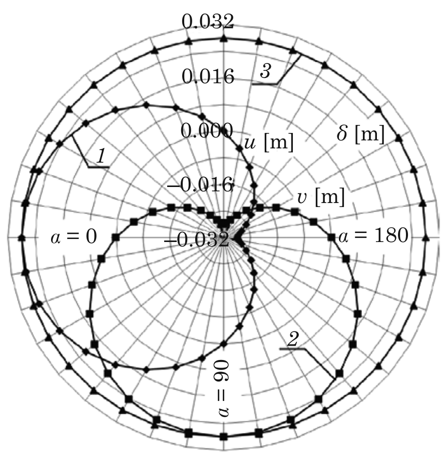

Fig. 2. Displacements of the tower apex subjected to a force of varying direction: *1* – x-axis displacement, *2* – y-axis displacement, *3* – displacement in the direction of the applied force, α – force angle to the x-axis

The eigenfrequencies of various models were analyzed to define the dynamic model of a truss tower. Eight initial eigenvalues of the tower with the service platform, and displacements of the apex point caused by static wind loads in the direction of the y-axis, perpendicular to one of the tower faces, are shown in Table 1. The forces acting upon truss nodes due to wind pressure were determined based on the area of structural elements exposed to wind and wind speed profiles at the tower location. The resultant force of static wind loads in the direction of the y-axis was 27.22 kN. This value is a result of the action of wind pressure on the structure under consideration according to applicable design code. The wind pressure values (Pa) were assumed for the consecutive height intervals (m): 0-10: 491; 10-16: 537; 16-40: 688. Percentage differences between the results calculated for frame models and the truss model were given. The author's method for automatic elimination of mechanical instability was applied in the analysis of truss model. Calculations were performed for the following tower models:

– truss model with blocked movement of out-of-place nodes,

– frame model with identical moments of inertia of rod cross-sections (I_identical),

– frame model with minor moments of inertia of rod cross-sections ($I = I_{\min}$),

– frame model with minor moments of inertia reduced 5-fold ($I = 0.2 \cdot I_{\min}$).

The minor moments of inertia of rod cross-sections are the minor moments of inertia of cross-sectional area of equal-sided angle sections which are the structural elements of the analyzed tower. In cases where identical moments of inertia were determined for all rods, the value of the minor moment of inertia of an angle section with the largest cross-section was adopted.

The shape of 1-4 eigenmodes was characteristic of a cantilever beam. Eigenmodes with nearly identical frequencies correspond to the curvature of the tower axis in two mutually perpendicular planes.

The global stiffness of the frame model with reduced rod flexural stiffness is nearly identical to that of the truss model, but its dynamic behavior differs most considerably from that of a truss structure. For this reason, a frame model with identical moments of inertia of rod cross-sections was used in vibration analysis.

Table 1

Dynamic and static parameters of different tower models

Eigen-mode	Truss	Frame I-identical	Frame $I = I_{\min}$	Frame $I = 0.2 \cdot I_{\min}$	Frame I-identical	Frame $I = I_{\min}$	Frame $I = 0.2 \cdot I_{\min}$
No.	Frequency [Hz]				Difference [%]		
1	1.414	1.372	1.365	1.363	–2.9	–3.4	–3.6
2	1.414	1.372	1.365	1.363	–2.9	–3.4	–3.6
3	6.325	5.965	5.866	3.910	–5.7	–7.3	–38.2
4	6.325	5.965	5.866	3.910	–5.7	–7.3	–38.2
Displacement of the tower apex [m]							
v_{st}	0.1355	0.1339	0.1353	0.1355	–1.23	–0.21	–0.05

Dynamic vibration absorber

A dynamic vibration absorber installed directly under the service platform in the top part of the tower was proposed. The mass element of the absorber m was connected with apex nodes of the tower legs with the use of three viscoelastic elements as shown in Figure 3. The elements were modeled with the use of MSC Marc SPRINGS model definition option in the form of springs/dash-pots. Appropriate values of stiffness and damping coefficients were being entered.

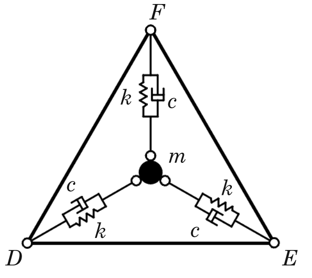

Fig. 3. Diagram of a damped dynamic absorber

The most dangerous vibrations result from fundamental eigenmode associated with the lowest natural frequency. A damped dynamic absorber has to be tuned to reduce vibrations caused by wind gusts with near fundamental frequency. A dynamic absorber for damping vibrations in a system with one degree of freedom was analyzed in detail by DEN HARTOG (1956). The tower analyzed in this study is characterized by multiple degrees of freedom, but the formulas proposed by the above author can be used to determine the parameters initial values for a dynamic damper. A dynamic vibration absorber can be precisely tuned during numerical experiments. Based on the theory for dynamic dampers, it was assumed that absorber mass $m = 76.8$ kg is equal to $\mu = 1/10$ of damped mass M, in this case, the modal mass of the tower first eigenmode. When the above mass was incorporated into the tower and viscous elements were blocked (high values of parameter c), successive system frequencies decreased by 4% and 1%, respectively. When a simplified model with two degrees of freedom (modal mass – damper mass) is assumed, the values of parameters k and c can be estimated, on the requirement that the amplitude magnification factor has the lowest value at extreme points. The above can be achieved if the circular frequency of vibrations of isolated damper mass ω_t relative to the fundamental eigenfrequency of a system without a damper ω_1 fulfills the following condition (DEN HARTOG 1956):

$$\varphi = \frac{\omega_t}{\omega_1} = \frac{1}{1 + \mu} \tag{1}$$

Since

$$\omega_t = \sqrt{\frac{k_y}{m}} = \varphi \omega_1 \tag{2}$$

then

$$k = \frac{2}{3} k_y = \frac{2}{3} \varphi^2 \omega_1^2 m \tag{3}$$

In a dynamic vibration absorber with viscous damping, the optimal value of damping coefficient c_y can be determined with the use of the formula:

$$\frac{c_y}{c_{kr}} = \frac{c_y}{2m\ \omega_1} = \sqrt{\frac{3\mu}{8(1 + \mu)^3}} \tag{4}$$

and coefficient c can be calculated for every viscous element:

$$c = \frac{2}{3} c_y = \frac{4}{3} m\ \omega_1 \sqrt{\frac{3\mu}{8(1 + \mu)^3}} \tag{5}$$

The parameters necessary for the creation of the computational model of a tower with a dynamic vibration absorber were calculated with the use of the above formulas. Obtained values are as follows: $k = 3141$ N/m and $c = 141$ Ns/m.

The initial values of the frequency spectrum of a tower with a dynamic damper were determined at 1.13, 1.13, 1.51, 1.51, 5.96, 5.96 Hz. Frequencies related to the attached damper were noted in a tower frequency spectrum, and the fundamental frequency relating to the modal mass of the first eigenmode was changed. Higher eigenfrequencies, i.e. the third eigenfrequency and upwards, did not change when a damper was incorporated into the analyzed structure.

The model was subjected to wind loads within the frequency range of 1.0-1.7 Hz. Frequency variations of forces acting upon truss nodes were established based on the previously determined spatial distribution pattern and the resultant amplitude of $F_y = 27.22$ kN.

The maximum relative displacement of the tower apex S was $v_S/v_{st} = 3.1$, and it was observed in the first resonant zone of 1.13 Hz. Relative displacement of damper mass reached $v_t/v_{st} = 5.9$ at the same frequency. Attempts were made to reduce the displacement by changing the value of damping coefficient c. The system response to the introduced changes is presented in Figure 4.

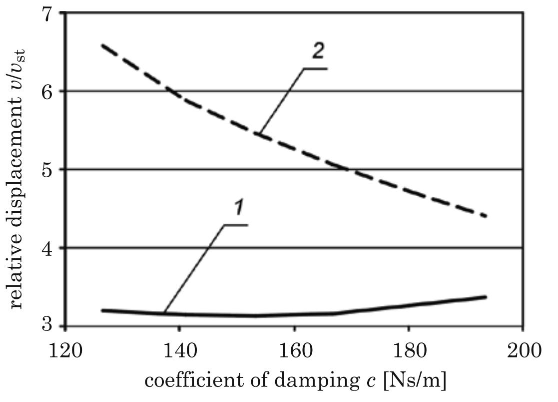

Fig. 4. Relative displacement amplitude of the tower apex and the damper with respect to the tower as a function of the damping coefficient: *1* – tower apex point, *2* – damper

Within the analyzed range of values of damping coefficient c, the relative displacement of damper mass was highly sensitive to the introduced changes, whereas minor changes were noted in the displacement amplitude of the tower apex.

Dampers with tuning ratio φ close to unity have been discussed in literature. For this reason, the effect of tuning on the amplitude of fundamental vibrations of the tower and damper mass were analyzed. Damper mass was modified. Damper parameters c and k were calculated with the use of formulas (1)-(5). Changes in parameter μ led to insignificant changes in the values of the first four eigenvalues, but yet the changes were taken into account in calculations. The relative values of amplitudes in the system first resonant zones are presented in Figure 5.

The data shown in Figure 5 indicates that the tower vibration amplitude is lowest when the damper mass-modal mass ratio equals 0.1. When damping parameters reach the values suggested by DEN HARTOG (1956), the amplitude of damper vibrations is high. Based on the diagrams presented in Figure 4, the value of parameter c should be increased to c = 193 Ns/m when μ = 0.1. In this case, vibration amplitude is 7% higher than the minimum value, but damper vibrations are significantly reduced, which has important sense in its structural implementation.

Based on the performed calculations, the relative amplitude of vibrations caused by wind gusts with the first resonant frequency (1.37 Hz) was determined at 7.3 in a tower without a damper. In a tower with a blocked absorber mass, vibration amplitude was only 0.01 lower at first-order resonance (1.32 Hz).

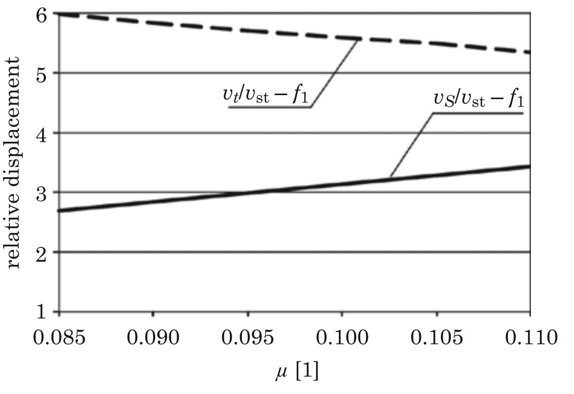

Fig. 5. Relative displacement amplitude of the tower apex (v_S) and the damper with respect to the tower (v_t) as a function of coefficient μ: f_1 – first-order resonance

Conclusions

A dynamic vibration absorber significantly reduces the amplitude of a tower fundamental vibrations. In the analyzed case, extreme displacements were reduced by 54% when damper movements were restricted. Similar reductions are observed in internal axial forces of truss rods. As the result, rods with a smaller cross-section can be applied, and the above solution reduces material requirements. The theory describing the principles of dynamic vibration damping in a system with one degree of freedom can be used in preliminary analyses of vibration damping in systems with multiple degrees of freedom. FEM software can be used to tune dampers for complex 3D mechanical systems.

References

Adams V., Askenazi A. 1999. *Building Better Products with Finite Element Analysis*. OnWord Press, Santa Fe.

Albermani F., Kitipornchai S., Chan R.W.K. 2009. *Failure analysis of transmission towers*. Engineering Failure Analysis, 16: 1922–1928.

Barle J., Grubisic V., Radica D. 2010. *Service strength validation of wind-sensitive structures, including fatigue life evaluation*. Engineering Structures, 32: 2767–2775.

Da Silva J.G.S., Vellasco P.C.G.D.S., De Andrade S.A.L., De Oliveira M.I.R. 2005. *Structural assessment of current steel design models for transmission and telecommunication towers*. Journal of Constructional Steel Research, 61: 1108–34.

Den Hartog J.P. 1956. *Mechanical vibrations*. McGraw-Hill, New York.

Holmes J.D. 2007. *Wind Loading of Structures.* Tylor & Francis, London.

Jones G.A., Layer D.H., Osenkowsky T.G. 2007. *National Association of Broadcasters Engineering Handbook* (10th Edition). Focal Press, USA.

ŁATAS W., MARTYNOWICZ P. 2012. *Modelling vibration of wind turbine tower-nacelle assembly with a tuned mass damper*. Modelowanie Inżynierskie, 44: 187–198.

PELC J. 2012. *A strategy for automatic elimination of mechanical instability in structural analysis of spatial truss tower model*. Technical Sciences, 15: 319–330

QU W.L., CHEN Z.H., XU Y.L. 2001. *Dynamic analysis of wind-excited truss tower with friction dampers*. Computers and Structures, 79: 2817–2831.

RAKOWSKI G., KACPRZYK Z. 2005. *Finite element method in structural mechanics*. Oficyna Wydawnicza Politechniki Warszawskiej, Warsaw.

SHEA K., SMITH I.F.C. 2006. *Improving full-scale transmission tower design through topology and shape optimisation*. Journal of Structural Engineering, 132: 781–790.

SULLINS E.J. 2006. *Analysis of radio communication towers subjected to wind, ice and seismic loadings*. MSc Thesis, University of Missouri-Columbia.