UDC 624.974.014.2.042:621.548.3]:519.67
Preliminary communication
Received: 16.09.2015.

Simple dynamic model of wind turbine tower with experimental verification

Predrag Đukić(1), **Hinko Wolf**(2), **Jani Barle**(3)

(1) *University of Split, University Department of Professional Studies, Kopilica 5, HR-21000 Split, CROATIA*
e-mail: pdukic@oss.unist.hr

(2) *University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, Ivana Lučića 5, HR-10000 Zagreb, CROATIA*
e-mail: hwolf@fsb.hr

(3) *University of Split, Faculty of Electrical Engineering, Mechanical Engineering and Naval Architecture, R. Boškovića 32, HR-21000 Split, CROATIA*
e-mail: barle@fesb.hr

SUMMARY

The main goal in designing the wind turbine is ensuring structural safety and optimal operational performance. Key aspects to be addressed are: long design life, sensitivity of structure to vibration and resonance, non-deterministic wind loads with significant time variations and micro-location dependence. In order to achieve these goals, a simple dynamic lumped-mass model of a conic tubular steel wind turbine tower has been developed in this paper. The tower stiffness matrix is determined by considering the stiffness of a cantilever beam with variable cross-section, and lumped masses for each conical segment are determined in such a way that the centres of gravity of conical segments match their actual position in the structure. There is a very good agreement between the measured field data and the results obtained by the developed simple numerical model.

KEY WORDS: *wind turbine, dynamic model, verification, monitoring.*

1. INTRODUCTION

For an optimal performance of the wind turbine in Figure 1, two main requirements should be met, the first of which comes from the main reason why the plant is built and this is to produce as much energy as possible given the wind available. The other requirement is associated with the fact that the energy can be produced only if the turbine is fit to operate. Therefore, in the control subsystem those two aspects are taken into account simultaneously in a way that managing the turbine operating in the energy-optimal condition, loads on the structural and power-producing subsystems must be kept below the load levels determined by durability criteria. All of the possible parameter values must be classified as load cases and are well defined in [1] and [2] as essentially a combination of external conditions (e.g. wind speed, turbulence intensity etc.) and wind turbine operational modes. Turbine operational modes are generally considered as normal (e.g. power production, start-up, shut-down and parked at extreme wind speeds) or fault (e.g. generator short-circuit, control system fault, rotor emergency stop and rotor overspeed). Due to the structural sensitivity to vibration and resonance build-up, the loads in normal operating conditions are transferred both from nacelle to the supporting structure and back. Certain input values to the control system, which in normal operation is a way to minimize stresses, can have an opposite effect: mechanical overload, vibrations or failure. Because of the complexity of the structure, and the fact that the loads are non-deterministic and location-dependent, models and parameters taken into account during the design stage must be verified [3-5]. A wind turbine originally has its own anemometers with other atmospheric sensors and accelerometer located in the nacelle. Those and the signals from the set of sensors for internal operating parameters are used by the control system and recorded in the event log. The wind speed and accelerometer data are the main signals used by the control system to achieve a balance between maximal production and protection from excessive damage to the mechanical components.

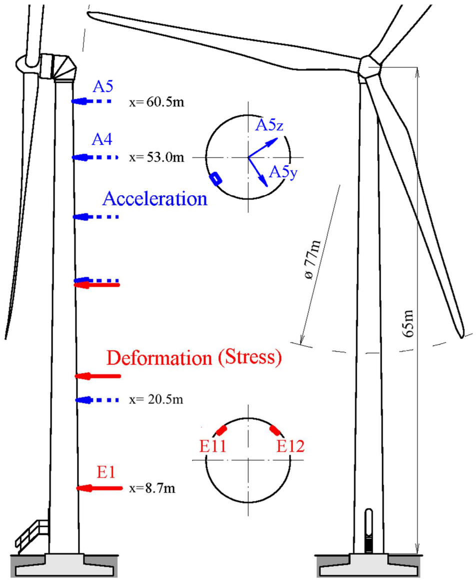

Fig. 1 *Wind turbine with positions of used sensors*

Geometry of a recently erected *1.5 MW* turbine is shown in Figure 1. In order to support research on this subject, an additional data recording system was added to the already existing system. The monitoring setup presented here is based on requirements stipulated in [6]. The turbine is instrumented with 5 dual-axis accelerometers and 5 rectangular strain gage rosettes for measuring accelerations and strains at defined locations.

The accelerometers were positioned at 5 distinct levels in height ranging from *20.5* to *60.5 meters*, denoted with A1 through A5. The accelerometers are marked with blue-dashed arrows. As an example, directions at level A5, A5y and A5z are illustrated on the cross section in the Fig 1. Their directions were fixed and did not change with the position of the nacelle. Accelerometer A5 is nearly collocated to the turbine's own accelerometer used by the control system. A second set of monitoring sensors consists of five delta (*0/60/120 deg.*) strain gage rosettes applied to the structure at three levels, marked with red-solid arrows. At levels E1 and E2, the two rosettes were mounted as shown in the Figure 1 for the level E1. These rosettes are denoted with E11 and E12. These six signals are sufficient to determine internal torsional and bending moments acting on their respective heights. An additional rosette E3 is needed to estimate the shape of internal forces distribution along the tower height. With this positioning of sensors (with accelerometers "up" and strain gages "down"), the whole structure is adequately covered and all sensors have optimum sensitivity for their respective inputs. As well as accelerometer A5, nacelle angle signal is also doubled in monitoring system by an additional sensor. All 26 monitoring signals were acquired with *200 Hz / 16 bit* sample rate over a period of two years. During this time, a number of operational sequences patterns were detected. Those sequences did not yield expected production efficiency and they needed improvement. Also, an excessive number of stopping was recorded which, as the following analysis shows, were not necessary at all. Furthermore, turbine operation was often set at a point where it is protected from mechanical damage, or where mechanical loads are decreased to extend its lifetime.

The upper diagram of Figure 2 shows six strain signals from level E1. Superimposed to the upper diagram are control events marked with squares. That event sequence is chosen to represent a characteristic problem. Yellow circles corresponding to the legend on the right side represent operational states: idle, stop, start-up and run. Wind speed during that period was in the range of *10 m/s,* meaning that the production should have been kept in the optimal range and uninterrupted.

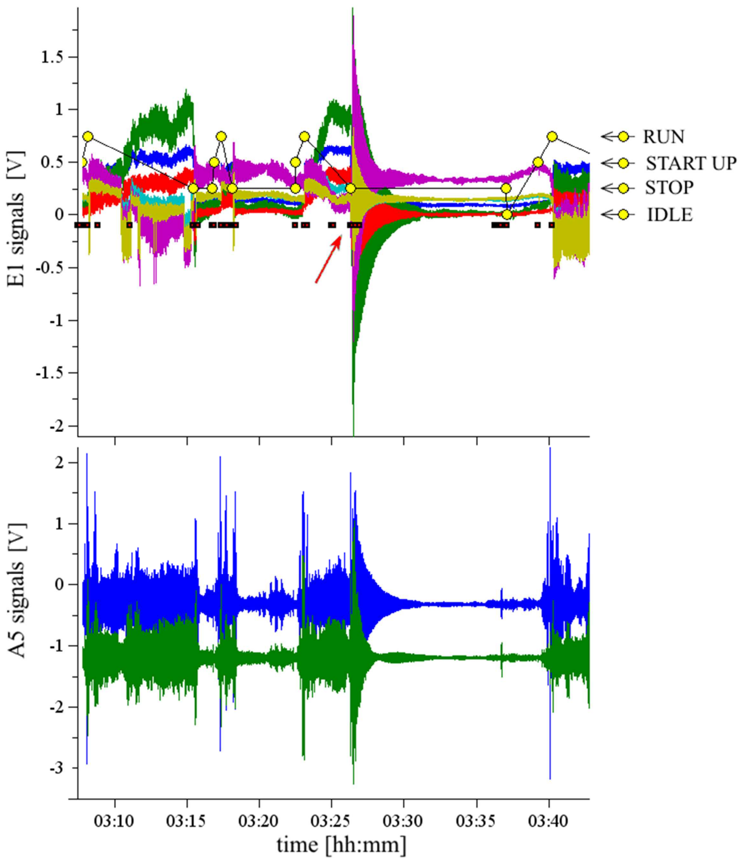

Fig. 2 *Signals from sensors at levels E1 and A5*

Signal values from the A5 level in the lower half of Figure 2 seem unrelated to the actual stress signals and therefore acceleration signals cannot be reliably used as control input for simple methods as treshold level. Within the period of approximately *40 minutes* (3:05 to 3:45AM) the turbine was started 3 times and it produced for a duration of *18 minutes*. During this period, wind speeds were between *8* and *13 m/s*, which means that the turbine should have produced with approximately *60 %* of installed power. Red arrow in the upper diagram marks a moment when the turbine was brought to the STOP state with exactly *9.7 m/s* wind speed, *16.8 rpm* and production of *837 kW*. After that, a control system procedure called "Tower vibration safety chain" kicked in, and brought the rotor to a complete still in about *20 seconds*. This is only a simplified description of a number of automatic decisions that happened prior or during the event. Such abrupt stopping caused extreme loads to the structure, which were, in this case, unnecessary, all due to control system set points and limits.

In this case, it is clear that the control system was the root cause of excessive dynamic loads that were far greater than the loads detected as unacceptable. A great number of possible control sequences similar to the one shown here exist in reality, while their combinations with external loads are almost unlimited. Using a more complex structural model of the turbine to simulate its dynamic behaviour leads to enormous computational efforts resulting in huge amounts of data, and there is still a good possibility of missing real, important points. The solution is some simpler model of supporting structure, but complex enough to give a correct answer to the main questions. The main purpose of this paper is to include all relevant values to achieve a simple, efficient and applicable dynamic model. The model presented here is verified by full-scale, in-field data.

2. EXPERIMENTAL DATA FOR VERIFICATION

For analysis presented in this section the acceleration signals from all five levels were used. The turbine was idle with wind under *3 m/s*. This low level of excitation loads is chosen for comparison purposes with the dynamic model presented here. In Figure 3, the magnitude of spectral density between signals from A5x and A4x clearly reveals the first two eigen frequencies of $f_1 = 0.402$ *Hz* and $f_2 = 3.323$ *Hz*. The peaks in the range between f_1 and f_2 produce weak modal indicators and they are the result of the blade vibrations. Considering the quality of the signals acquired at a low level of excitation, the last two frequencies denoted in this figure $f_3 = 9.852$ *Hz* and $f_4 = 17.43$ *Hz*, are neither clear nor accurate as the first two.

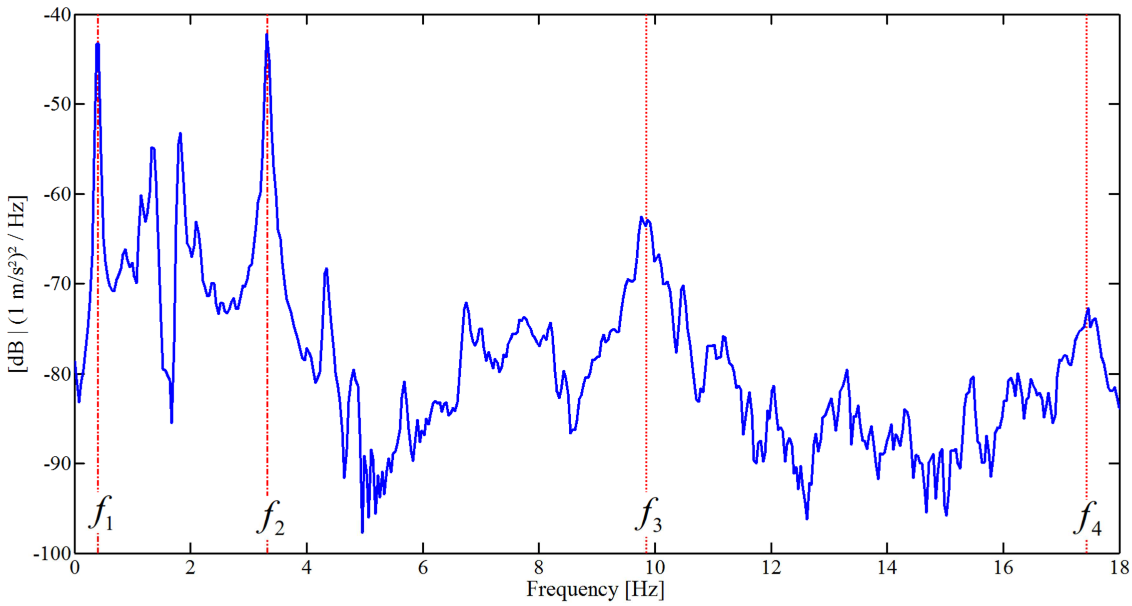

Fig. 3 *Magnitude of spectral density between signals from A5x and A4x*

3. NUMERICAL MODEL OF THE SUPPORTING STRUCTURE

The wind turbine's conic tubular steel tower was modelled as a cantilever beam with lumped masses of *N-1* conical segments (Figure 4). The last N^{th} segment of the length l_N is not a conical tubular steel segment. It represents the distance from the top of the tower to the centre of gravity of the wind turbine's nacelle and rotor. It was considered to be rigid. On the top of the tower there is a nacelle of mass m_N and mass moment of inertia J. Diameters of the conical segments are denoted as D_i and thickness of their walls as t_i ($i=1,\ldots, N-1$). Lumped masses for each conical segment, m_B and m_T were determined in such a way that centres of gravity of conical segments matched their actual position in the structure, shown in Figure 5. After determining m_B and m_T (of each conical segment) the segments were connected in one unit. All structural parametres were taken from the turbine technical documentation with the exception of mass moment of inertia of the nacelle with rotor. Due to a lack of data, this value was approximately estimated. In accordance with the technical documentation, mass of flanges, cables, ladders and coating colour were taken into consideration.

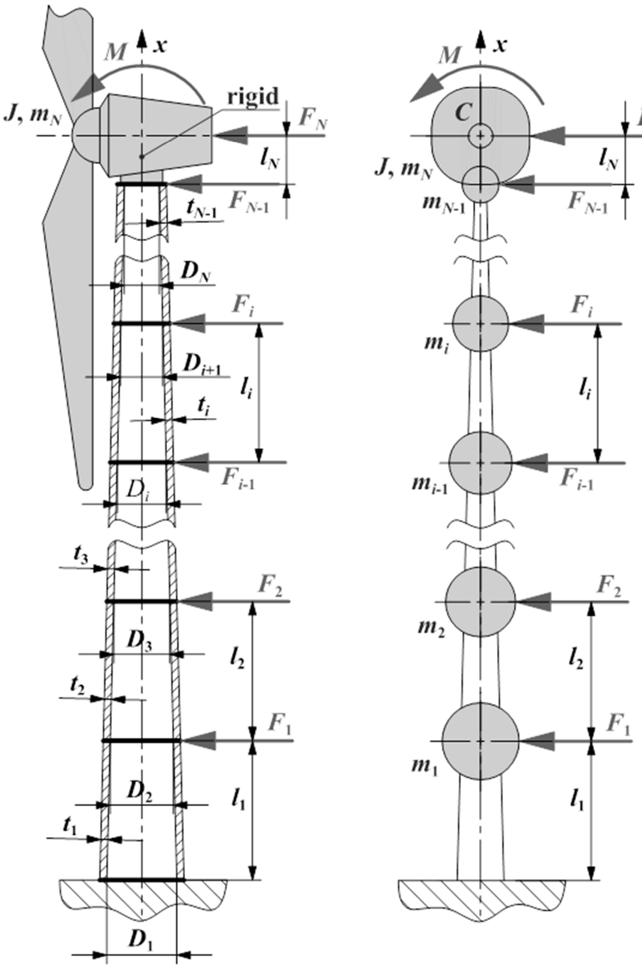

Fig. 4 *Lumped-mass model of a wind turbine's conic tubular steel tower*

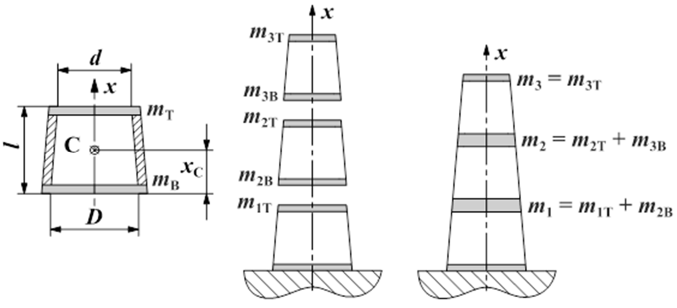

Fig. 5 *Discretization of the wind turbine's conic tubular steel tower*

The forces and bending moment acting on the structure due to wind are denoted as F_i ($i=1,\ldots, N-1$) and M (Figure 4). The stiffness matrix of the wind turbine tower modelled as a cantilever beam with variable cross-section is obtained by inversion of its flexibility matrix. Flexibility matrix elements (flexibility influence coefficients) are obtained by numerical solving of differential equation of the beam's deflection curve. One segment of the wind turbine's conic tubular steel tower is shown in Figure 6.

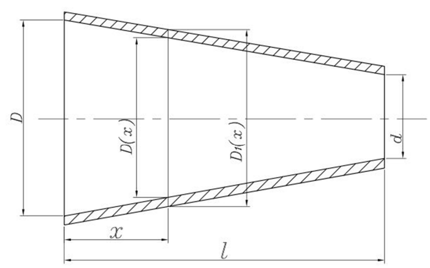

Fig. 6 *Conical tubular steel segment modeled as a beam with a variable cross-section*

Differential equation of the elastic line of the considered conic segment is:

$$\frac{d^2w}{dx^2} = -\frac{M(x)}{EI_y(x)} \tag{1}$$

where:

$$I_y(x) = \frac{\pi}{64}\left(D_1^4(x) - D^4(x)\right) = \frac{\pi}{64}\left(C_0 + C_1x + C_2x^2 + C_3x^3\right), \tag{2}$$

$$C_0 = A^4 - D^4, \quad C_1 = -4B\left(A^3 - D^3\right), \quad C_2 = 6B^2(A^2 - D^2), \quad C_3 = -4B^3(A - D), \tag{3}$$

$$A = D + 2t, \quad B = \frac{D-d}{l}. \tag{4}$$

Since the integrals:

$$\int \frac{dx}{C_0 + C_1x + C_2x^2 + C_3x^3}$$

and:

$$\int \frac{x\,dx}{C_0 + C_1x + C_2x^2 + C_3x^3}$$

obtained by introducing (2) into (1) have no simple analytical solution, they were solved numerically. After determining the inertia and stiffness matrices, natural frequencies of undamped wind turbine tower vibrations and corresponding mode shapes were calculated [7].

Results for the first four natural frequencies (f_1 – f_4) obtained in the case when the mass moment of inertia of the nacelle with rotor is taken into account, and when it is not, are compared and presented in Figure 7. The results are related to wind turbine tower models with *26*, *50*, *98* and *194 DOFs*. One can see that a very good agreement among numerical and measured results is obtained: *1.86%* difference compared to the measured values at f_1, *0.26%* at f_2, *13.75%* at f_3 and *11.44%* at f_4. These results change insignificantly with the increasing of DOFs. Significant differences between numerical and measured results, which are observed at the third and fourth natural frequencies, could be the result of inadequate approximation of the mass moment of inertia of the nacelle and rotor or of the insufficient quality of the signals acquired at a low level of excitation (as mentioned in Section 2). As one can see from the numerically obtained results, the influence of the mass moment of inertia of the nacelle and rotor to the first natural frequency f_1 is practically negligible (*0.29 %*), but its impact on higher natural frequencies can be significant: *7.32%* at f_2, *22.53%* at f_3 and *35.86%* at f_4. Compared to the results shown in Figure 7, natural frequencies obtained without masses of cables, ladders and coating colour differ approximately *0.3%* at f_1 and f_2 and *2%* at f_3 and f_4. Figure 8 presents the first two numerically obtained mode shapes of the wind turbine tower that correspond to frequencies f_1 and f_2 in Figure 3. Circles indicate positions of the lumped masses i.e. the endpoints of conical segments as shown in Figure 4. From the Figures 3 and 7 one can see that a very good agreement between the measured field data and the data obtained by the considered simple numerical model has been obtained. In general, the accuracy of the calculation of natural frequencies and mode shapes can be increased by using larger number of DOFs, i.e. by discretization of a wind turbine conic tubular steel tower with larger number of lumped masses and conical segments. However, Figure 7 shows that the increase of DOFs in this case insignificantly affects the accuracy of the results. In other words, the model with 26 DOFs can be considered adequate. In this example, the wind turbine tower is divided into 24 conical segments that are necessary for detailed modelling of flanges, tower diameter changes and wall thickness changes. Since one uses modal analysis to simulate the wind turbine tower movements, the number of DOFs does not significantly affect the efficiency of the numerical calculations [8].

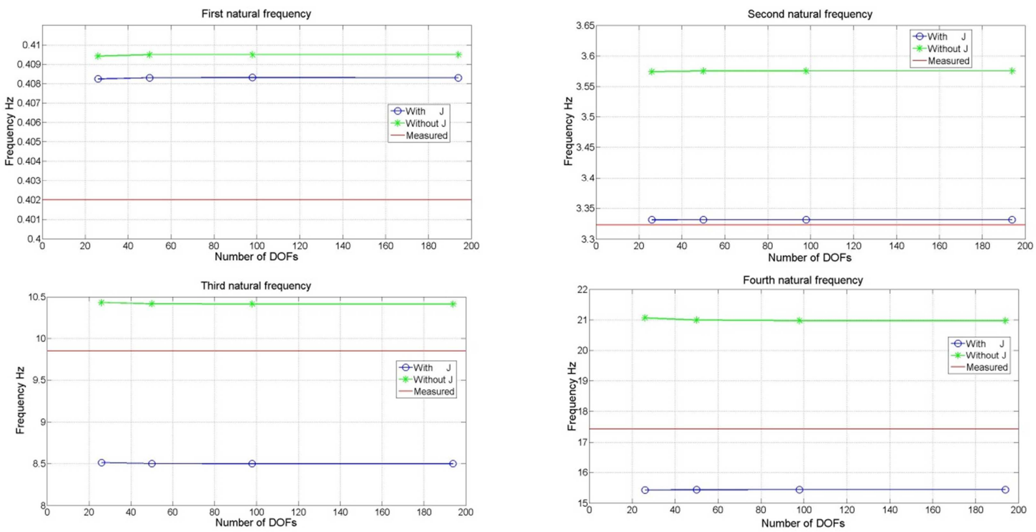

Fig. 7 *Comparison of numerical and measured results*

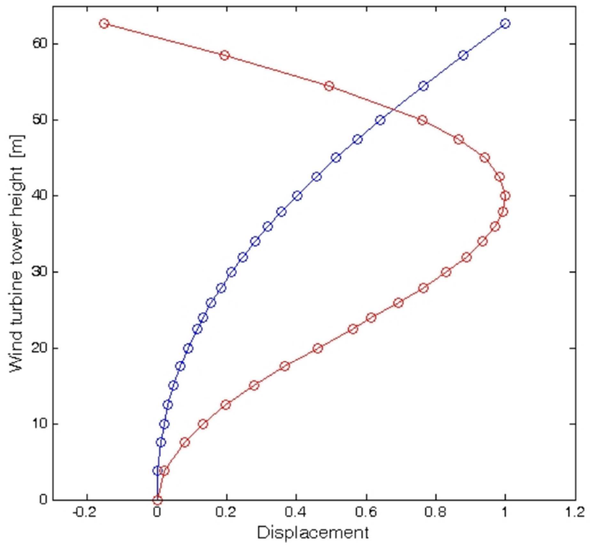

Fig. 8 *First two mode shapes of the wind turbine tower (f_1 = 0.40819 Hz, f_2 = 3,3309 Hz)*

One must emphasize that, in this study, damping (both structural and aerodynamic) as well as elasticity of the tower foundation were not taken into account (the foundation of the tower is modelled as absolutely rigid).

4. DISCUSSION

In this section, a period even shorter then the one shown in Figure 2 is used. The event sequence corresponds to the 3:26 timestamp, which is marked by a red arrow in the upper diagram in Figure 2. Data in Figure 2 is now processed to show bending moment amplitude on the level E1, Figure 9.

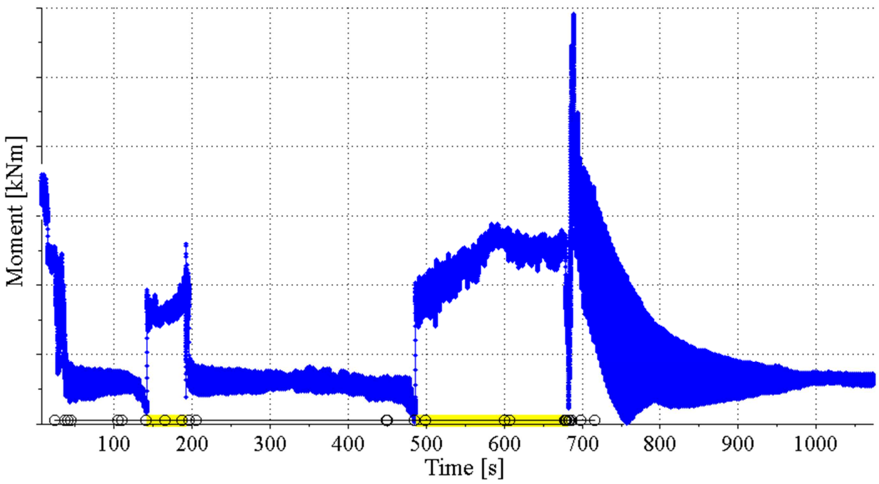

Fig. 9 *Bending moment at E1 during a sequence of runs and emergency shutdown*

On that figure the two production runs are marked with yellow colour on abscissa. The first run is from approx. *150-th* to *200-th* second, while the second is from *500-th* to *700-th* second. During these intervals, the production reached *90%* of the expected output. After that, emergency stop occurred which caused high cyclic loads, from *700-th* second onwards. It is obvious that stopping done in this manner is damaging the structure more than operation in the regime control system that was assumed harmful. In the upper diagram of the Figure 10, the emergency shutdown moment magnitude is examined in more detail. The lower diagram shows longitudinal and transversal components of the bending moment relative to the actual nacelle position. Sequence of control events, depicted by a row of red dots and circles inserted on the upper abscissa, show how the situation unfolded. The first red dot is a control event that stopped production, which is visible in a drop of the bending moment magnitude and shift of the transversal moment to zero. Transversal moment keeps oscillating around the zero value, while bending moment magnitude oscillates around the nacelle overhang value. The fourth control event stopped the rotor by engaging the mechanical brake. What follows is a longer period of free vibrations with amplitudes significantly higher than those normally encountered during periods of normal production.

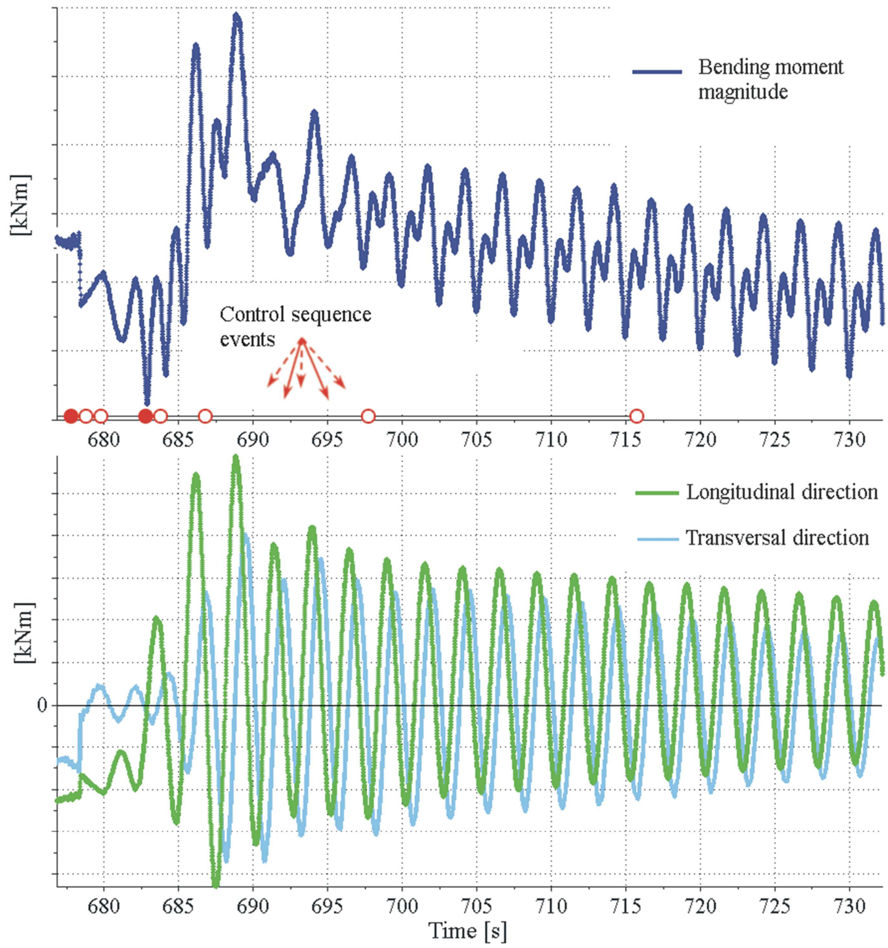

Figure 10. *Bending moments and control events during an emergency shutdown*

Considering the problem addressed in this paper, it is evident that dynamic loads resulting from bending in the first mode have the greatest influence on the operational life.

In these diagrams, according to the NDA agreement with the owner of the wind turbine park, actual values of the bending moments were not revealed, but the ratio of the bending moments in different operation regimes is sufficient to realize that this should be avoided. Finding out the mentioned anomalies and new data acquired, helped to adjust the existing control system and reduce the problem to levels considered to be acceptable.

To improve the situation in the future (to allow for smoother stopping in attempt to avoid excessive free vibrations, or reaching a decision whether the stopping is required at all), the following changes/additions would be of great help: (1) to include the dynamic model presented here into the control algorithm/model and (2) to estimate the influence of the acceleration signals on the fatigue life of the supporting structure during normal operation or during every stopping.

5. CONCLUSION

This paper points out to a need of having reliable experimental data for description of the behaviour of wind turbines.

Data similar to the one presented here can be used to verify design and/or in-field performance of a wind turbine. Besides the need to avoid such an event, it is obvious that if they happen anyway, first-mode loads have the greatest influence. Since the model presented here is verified on first two modes, its use in simulations will lead to the results that are sufficiently close to the reality, without the need for a more complex model. With its simplicity, this model allows examining numerous possible states, checking and fine-tuning of the control system algorithms and parameters together with sensors used in its operation.

Control strategy is responsible for dealing with the problem of dynamic instability, but incorrectly adjusted or missing key data can also be its main cause. The described simple numerical model is validated in field and it will be further upgraded and used in future research to simulate response of the wind turbine tower subjected to external loads, like wind speed or operation regime and to link acceleration data from the top of the tower with parameters of structural damage, in this case represented by the bending moment at level E1.

6. REFERENCES

[1] N 61400-01: Wind turbine generator systems - Part 01: Design requirements, 2010.

[2] Germanischer Lloyd, Hamburg, Germany: Guideline for Certification of Wind Turbines, 2010.

[3] M.G. Mollineaux, K.L. Van Buren, F.M. Hemez and S. Atamturktur: Simulating the dynamics of wind turbine blades: Part I, model development and verification, *Wind Energy*, Vol. 16, No. 5, pp. 694-710, 2013.

[4] K.L. Van Buren, M.G. Mollineaux, F.M. Hemez and S. Atamturktur: Simulating the dynamics of wind turbine blades: Part II, model validation and uncertainty quantification, *Wind Energy*, Vol. 16, No. 5, pp. 741-758, 2013.

[5] I. Fleming and D.J. Luscher, A model for the structural dynamic response of the CX-100 wind turbine blade, *Wind Energy*, Vol. 17, No. 6, pp. 877-900, 2014.

[6] EN 61400-13: Wind turbine generator systems - Part 13: Measurement of mechanical loads, 2001.

[7] S.S. Rao, *Mechanical vibrations*, Prentice Hall, New York, 2011.

[8] M.R. Hatch, *Vibration Simulation Using Matlab and Ansys*, Chapman & Hall/CRC, Boca Raton, London, NewYork, 2001.

JEDNOSTAVNI DINAMIČKI MODEL TORNJA VJETROTURBINE SA EKSPERIMENTALNOM VERIFIKACIJOM

Primarni ciljevi koje treba ostvariti kod projektiranja vjetroturbina su njezina sigurnost i optimalne performanse tijekom eksploatacije. Glavni aspekti koje pri tome treba riješiti su: dug životni vijek konstrukcije, osjetljivost konstrukcije na vibracije i rezonanciju, ne-deterministička priroda opterećenja uzrokovanih vjetrom (koja značajno variraju tijekom eksploatacije), te značajna ovisnost tih opterećenja o mikro-lokaciji na kojoj je smještena vjetroturbina. Kako bi se ostvarili ovi ciljevi, razvijen je jednostavni diskretni dinamički model konusnog čeličnog stupa vjetroturbine. Matrica krutosti određena je razmatranjem krutosti ukliještene grede promjenjivog poprečnog presjeka, a diskretizacija mase stupa napravljena je na način da položaji težišta diskretiziranih segmenata stupa odgovaraju njihovim stvarnim položajima na stupu. Dobiveno je vrlo dobro poklapanje između izmjerenih vrijednosti i rezultata dobivenih s razvijenim jednostavnim numeričkim modelom.

KLJUČNE RIJEČI: *vjetroturbina, dinamički model, verifikacija, monitoring.*