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The natural frequency analysis of a wind turbine with vertical axes

Abstract

The natural frequency analysis of four analytical models of a home wind turbine is presented in this paper. These models are: the rotor of a generator without housing, the rotor of a generator with housing and a stator, the rotor of a wind turbine, and the entire assembly of the wind turbine. By comparison of the received results of these models, evaluation of the variation in natural frequencies was performed. Additionally, interaction of the natural frequencies of the models with excitations was investigated. Moreover, various forms of vibrations, such as bending and torsional mode of the wind turbine rotor, rotor generator and blades, were found. Simplified analytical calculations for evaluation of the bearing stiffness applied to the vibration models were also carried out. The natural frequency analysis was performed in the analytical environment of the Siemens NX software based on NX Advanced module by the use of NASTRAN solver.

Keywords: natural frequency analysis, wind turbine, vibrations.

1. Introduction

Parts of the stator, magnets as well as the rotor of a generator during operation are affected by continuous deflection, which can gradually deteriorate their technical condition. These parts are rarely deflected. However, such deflections can have serious consequences related to the operator's health, integrity of the adjacent equipment and the financial aspect.

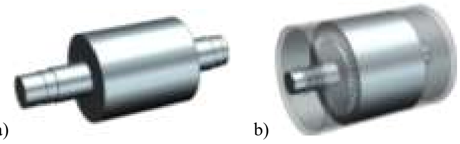
Failures always occur in the area of the disc and rotor in the generator, which becomes considerably deformed, at the same time being displaced, which leads to unbalance, increase in vibrations and torsional breaking or detachment of the generator rotor shaft as a consequence of the contact with the stator.

In order to avoid failure and to estimate dangerous frequencies, the paper presents the dynamic analysis of a 2 kW generator connected to a home wind turbine with a vertical axis. The analysis includes the evaluation of natural frequencies of the generator and the whole turbine unit. It is also very important to understand loads during wind turbine operation to avoid its failure [1, 2].

Moreover, this paper focuses on the estimation of decrease or increase in natural frequencies after the wind turbine has been connected to the generator and the influence of the way the turbine unit has been installed. The analysis was performed with FEM by means of the Siemens NX software applying Nastran solver.

2. Description of analytical models

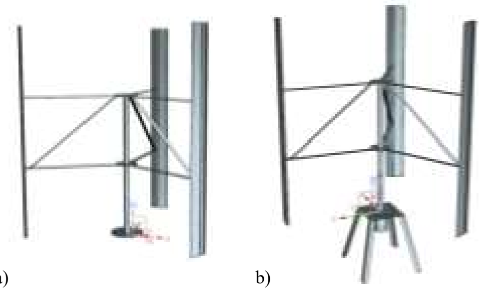
Natural frequencies were analyzed in four types of models. In the first model (Fig. 1a), the rotor was modeled in the form of a shaft with sets of sheet metals mounted on the shaft. In the second model, the generator stator was additionally taken into consideration in the form of a simplified cylinder (Fig. 1b).



a) b)

Fig. 1. a) model of the generator rotor (first model), b) model of the rotor with housing and stator (second model)

The third model was made of a separate wind turbine rotor with the vertical axis (Fig. 2a). Moreover, in order to assess the dynamics of the whole turbine unit, calculations were made in relation to the turbine rotor and the whole supporting structure (Fig. 2b).



a) b)

Fig. 2. a) separated wind turbine rotor (third model), b) whole turbine unit with the supporting structure (fourth model)

Figure 3 shows a CAD model of the connection between the generator and the axis of the turbine. The whole structure is stabilized by four metal brackets, spread evenly in a circle.

Fig. 3. CAD model of the connection of the generator with the wind turbine rotor and supporting structure

3. Bearing stiffness

Bearings are one of the most reliable and widely used machine elements. Their main task is safe transfer of the working loads from a rotating element to the housing of the machine having low resistance motion at the same time. Most of the machines are rotary machines having a rotor supported by bearings. Operating characteristics of these machines as a part of the entire system are determined by the dynamic of the scheme: rotor-supporting bearing-housing. The point is to determine the value of the force transmitted from the shaft to the bearings and housings.

For the purpose of this paper, stiffness of the bearing was evaluated by means of the formulas provided by the reference [3] considering the stiffness of elements in contact with the inner race, outer race and rolling elements. Bearing stiffness is a non-linear function and depends on the force acting on the bearing.

In the present analysis, the stiffness was modelled by stiffness elements with stiffness values calculated for 2000 N force acting on the bearing. Brief evaluation of the stiffness methodology is presented below. In the contact area of the curved surfaces of bodies pressed against each other, high values of the contact stress occur within a small area. The outline of this area depends on the shape of the bodies being in contact. For theoretic consideration of two bodies in contact, the approximate method can be used.

For stiffness evaluation certain assumptions were made:

- bodies are made of homogenous and isotropic material,
- bodies are in contact with smooth surfaces with regular shape,
- small deflections arise in the area of contact,
- the area of contact is significantly smaller than the area of the bodies in contact,
- only normal stress occurs in the area of contact.

The first step to determine the bearing stiffness is to determine the dimensions of semi-axes of the contact ellipse in the area of contact between the rolling element and the inner and outer ring of the bearing. For this purpose it is necessary to determine values A and B for both contacts based on geometrical dimensions of the inner and outer races and rolling elements of the bearing, R_{1ij} and R_{2ij}. Angle φ determines mutual position of the normal planes containing the biggest curvatures of both surfaces.

$$A+B=\frac{1}{2}\left(\frac{1}{R_{1xx}}+\frac{1}{R_{1yy}}+\frac{1}{R_{2xx}}+\frac{1}{R_{2yy}}\right) \tag{1}$$

$$B-A=\frac{1}{2}\left\{\left(\frac{1}{R_{1xx}}-\frac{1}{R_{1yy}}\right)^2+\left(\frac{1}{R_{2xx}}-\frac{1}{R_{2yy}}\right)^2+ +2\left(\frac{1}{R_{1xx}}-\frac{1}{R_{1yy}}\right)\left(\frac{1}{R_{2xx}}-\frac{1}{R_{2yy}}\right)\cos(2\varphi)\right\}^{\frac{1}{2}} \tag{2}$$

Knowing values A and B, it is possible to calculate semi-axes a and b of the contact ellipse. For this purpose it is necessary to determine additional values m and n. Coefficients m and n can be determined from the following polynomial relations:

$$\begin{aligned} m &= 1.24466\cdot\theta^4-6.51974\cdot\theta^3+13.3336\cdot\theta^2+ \\ &\quad -13.48444\cdot\theta+6.97567 \\ n &= 0.1026\cdot\theta^3-0.21861\cdot\theta^2+0.57586\cdot\theta+0.2366 \end{aligned} \tag{3}$$

whereas θ can be calculated from the following formula

$$\theta=\arccos\left(\frac{B-A}{B+A}\right) \tag{4}$$

Thus semi-axes of the contact ellipse

$$a=m\cdot\sqrt[3]{\frac{3\pi}{4}\frac{F(k_1+k_2)}{A+B}}\,,\quad b=n\cdot\sqrt[3]{\frac{3\pi}{4}\frac{F(k_1+k_2)}{A+B}} \tag{5}$$

where

$$k_1=\frac{1-\nu_1^2}{\pi E_1}\,,\quad k_2=\frac{1-\nu_2^2}{\pi E_2} \tag{6}$$

ν_1 and ν_2, are the Poisson coefficients and E_1, E_2, are the Young's modulus. Finally, the maximum contact stress

$$p_0=\frac{3}{2}\frac{F}{\pi ab} \tag{7}$$

Based on the stress value, it is possible to determine the value of displacements of the bearing races and, finally, the bearing stiffness for the assumed value of the force acting on the bearing.

4. Boundary conditions and discrete mesh

In the first model, the shaft was supported by two bearings which were modeled by spring elements in the X and Y directions in the ball bearing in front and in the X, Y and Z in the other ball bearing (at the back). In this model, it was assumed that the outer bearing rings were rigidly mounted (details of the model and support are presented in Fig. 4). The bearings were modeled by spring elements with stiffness values equal to 5.87E8 N/m, determined by the analytical method described above. Moreover, displacements in a tangential direction were fixed at the end of the shaft (fixing tangential displacements allows calculating torsional vibration modes). As a result of fixing these displacements, the calculated torsional vibration frequencies were overestimated because the shaft was connected to the remaining part of the wind turbine structure. However, the shaft support in bearings allowed identifying the frequency of bending modes of the shaft [4, 5].

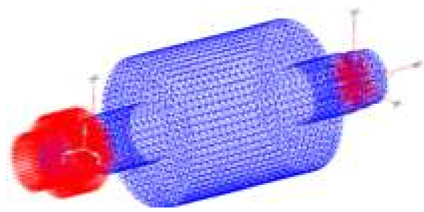


Fig. 4. Discretization grid with the boundary conditions set in the places of bearings mounting and connection of the shaft with the rotor turbine (x marks)

5. Determination of free vibrations by means of the MES method

The finite element method describes non-damped free vibrations with the following equation $\{[K]\text{-}\omega^2[M]\}\{u\}=\{0\}$ [4, 5], where $[K]$ and $[M]$ stand for stiffness and mass matrix, respectively. Solutions of the equation are: $f=\omega/2\pi$ and $\{u\}[m]$, i.e. frequencies of the natural vibration and the generalized vector of the vibration mode, where ω is a circular frequency in rad/s.

Application of FEM makes it possible to calculate natural frequencies for complex geometrical systems, and also to take into consideration pre-stress effect of the construction load and generator fixing.

Apart from the natural frequencies of the rotor, the second model includes additional vibration modes of the generator housing and stator. Sample results of the radial vibration modes to the order of r=2 and r=3 of the stator are presented in Fig 5.

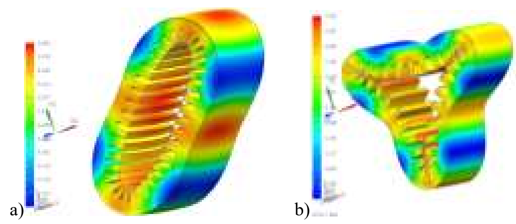


Fig. 5. a) radial mode of the stator to the order of r=2 (1243 Hz), b) radial mode of the stator to the order of r=3 (2636 Hz)

The analytical results of the natural frequencies calculation for the first and second model with their description are presented in Table 1.

In order to evaluate the dynamics of the whole turbine unit, the analysis of the turbine rotor with the generator was performed. For the purpose of the comparison the analysis of the separate turbine rotor was performed, too. Examples of the results for the third and fourth models are presented in Fig. 6.

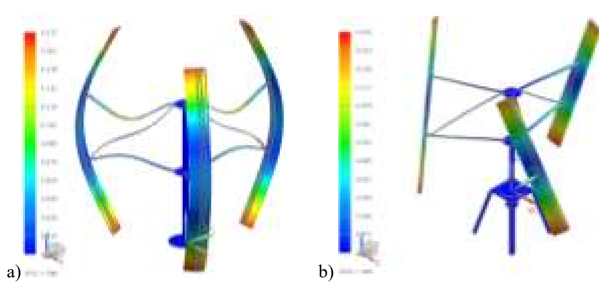


Fig. 6. a) bending form of the blade vibrations 1st mode, all blades in the same phase, the third model at 24 Hz, b) bending form of the whole turbine unit with respect to the base, the fourth model at 2.348 Hz

The natural frequencies for the third model are (in Hz): 3,16; 3,40; 5,99; 7,00; 15,56; 20,00; 24,00; whereas these frequencies for the fourth model are: 2,35; 2,52; 5,62; 6,02; 6,05; 6,84; 14,26.

Tab. 1. Specification of the natural frequencies with their description for the first and second model

Rotor without housing - model I		Rotor with housing – model II	
Frequency Hz		Frequency Hz	
1102	Torsional vibrations of the rotor with respect to shaft fixing	1102	Torsional vibration of the rotor with respect to shaft fixing
		1243	Vibrations of the stator and housing with second order radial deflection $r = 2$
		1308	Vibrations of the stator and housing with bending 1 mode in Y direction
1656	Bending mode of the shaft first mode	1736	Bending mode of the shaft first mode
		1765	Torsional vibrations of the stator with housing
		1822	Vibrations of the stator and housing with bending 1 mode in X direction
		2311	Longitudinal vibration of the stator with housing
		2636	Vibrations of the stator and housing with third order radial deflection $r = 3$
3474	Bending mode of the shaft second mode	3574	Bending mode of the shaft II mode
3978	Longitudinal vibration of the shaft	4045	Longitudinal vibration of the shaft
10760	Torsional vibration of the shaft second mode		

6. Conclusions

Natural frequencies of a mechanical system depend on distribution of mass and stiffness. In general, increase in mass without changing stiffness leads to decrease in natural frequencies, while increase in stiffness causes increase in frequencies. To put it simply, a natural frequency depends on the square root of the k/m, where k stands for the stiffness and m for mass.

Bending frequency of the rotor shaft with the housing (the second model) increased by 5% for the first mode in comparison to the first model, whereas for the second mode it increased by 2.8%. The torsional frequency remained almost the same.

The natural bending frequencies of the whole turbine unit are about 35% lower than the separate rotor turbine. The third model was fixed at the base of the mast, and that is why it can be stated that it is caused by the influence of the turbine base. Bending modes of the blades for the fourth model, in turn, are about 6-16% lower than for the third model.

In the presented results of the analysis for the fourth model, no natural frequencies of the generator are given because their values are about 100 times higher than those of the wind turbine rotor and require calculation of a large amount of natural frequencies.

In complex mechanical systems, it is difficult to estimate beyond any doubt whether in the case of redesign of the mechanical system or change in mass distribution, decrease or increase in natural frequencies will occur. It would be necessary again to perform many analyses to obtain satisfactory results in this case.

Based on the performed analyses it can be stated that there is no interaction of the generator natural frequencies with the excitation of the generator frequencies. Bending stiffness of the shaft is sufficient because the resonance speed of the bending form is more than ten times higher than the nominal speed of the generator rotor.

The FEM allowed determining different forms of natural frequencies. They are a consequence of 3D aspect of the model and the way the housing is fixed. The vibrations take the form of bending, torsional and mixed shapes. We have proved that the finite element method can be used with success to determine natural frequencies of the generators working together with e.g. a wind turbine [6].

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