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STRESS ANALYSIS WITH FINITE ELEMENT METHOD OF LARGE BEARINGS WITH HOLLOW ROLLERS USED IN WIND TURBINE SYSTEM

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Abstract: *In this paper is presented a method for increase the large bearings durability using hollow rollers, demonstrated through computer modeling.. Implementation of large bearings with hollow rollers in wind turbine system lead to significant gains in constant power plant operation simplifying management of inertial forces. The mathematical model has been implemented in an algorithm, by which it is possible to study stress distribution for various models of hollow rollers. The analysis with finite element method (FEM) is developed with MD Nastran software.*

Key words: *large bearings, Von Mises stress, hollow rollers, FEM.*

1. INTRODUCTION

With increasing the power of wind turbines also increases their size and inertial mass so, leading to increase the speed of the starting. The control of high speed is more difficult and the reduction of the inertial forces becomes mandatory.

The objective of the paper is to show a solution for increase the large bearings durability using weight reduction in large bearings construction, accomplished by the use of hollow rollers. The bearing life is strongly influenced by the stress and its reduction is conditioned by the specifics of the rollers geometry.

Reduction of inertial forces in wind power system is beneficial, leading to starting and braking more accurate and reducing the wear of parts in motion.

Because of the large moment of inertia of the rotor, the main design challenges include the starting, the speed control during the power producing operation, and stopping the turbine when required.

The main issue in the tower design is the vibration appearance. The tower vibration decrease with reduction of inertial mass.

2. DYNAMICS OF A WIND TURBINE SYSTEM

The energy of a wind turbine is the sum of the kinetic energy of the rotor, the gearbox and the generator (Fig.1). The inertia of the turbine blades will be much higher than that of the electrical generator. The latter will have a much higher rotational speed however, which will also result in a large amount of kinetic energy. The inertia *J* of a body is:

$$J = \sum m_i \cdot r_i^2 \qquad (1)$$

where r_i is the radial distance from the inertia axis to the particle of mass m_i and the summation is taken over all particles of the body.

The total moment of inertia [1] for a three-bladed turbine is given by:

$$J = 3 m_b r^2 \qquad (2)$$

unde m_b este masa unei pale, □i r este raza centrului de greutate al palelor. Pentru o centrală eoliană tipică de 2MW masa totală a rotorului este în jur de 40t. [12]

Can write the equations of equilibrium [2]:

$$T_R - T_{CV_1} = J_R \frac{d\omega_R}{dt} + B_R \omega_R \qquad (3)$$

$$T_{CV_2} - T_G = J_G \frac{d\omega_G}{dt} + B_G \omega_G \qquad (4)$$

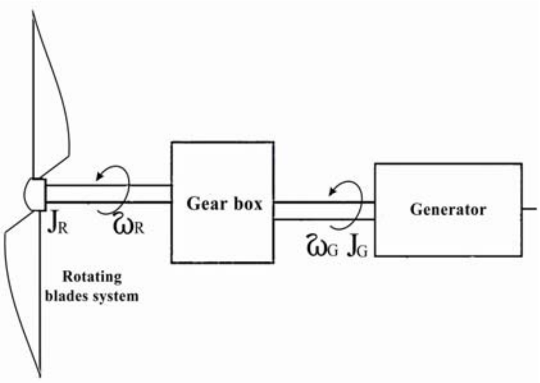


Fig. 1. Dynamic scheme of a wind system

where:

B_R and B_G are the constant friction in the rotor and generator system;

T_R is the rotor torque;

T_{CV1} is the gear box torque to the start;

T_{CV2} is the gear box torque to the end;

T_G is the generator torque to the end;

J_R is the moment of inertia of the rotor;

J_G is the moment of inertia of the generator;

ω_R is the rotational speed of the main shaft;

ω_G is the rotational speed to the start of generator.

From (3) result:

$$\frac{d\omega_R}{dt} = \frac{T_R - T_{CV_1} - B_R \omega_R}{J_R} \qquad (5)$$

Relation (5) leads to the obvious conclusion: importance of decreasing moment of inertia of rotor hence its inertial mass, will lead directly to increased rotational speed of main shaft so will enable faster start at low speed wind of power plant.

One of the ways we can reduce moment of inertia is given by reducing weight of bearings positioned in blades rotational system. For this, we propose the use of bearings with hollow rollers.

3. FINITE ELEMENT MODELING OF PROPOSED HOLLOW ROLLER

Practicability of hollow rollers in construction of large bearings was examined for a material type aiming deep carburization effect. The used material is allied steel: 15NiCr13. Roller dimensions are: L=220mm, R=60mm, [3]. Solid and hollow rollers have been modeled.

The finite element software MD Nastran is used to determine the values of Von Mises stress. Model of this simulation validate form of the roller and give to the researchers the possibility to choose the appropriate design.

The roller profile has a significant influence on the distribution of the contact stress hence, on the bearing load-carrying capacity and life. The best profile of the contact geometry is logarithmic [6]. The advantages of the logarithmic profile have been largely confirmed in research applications.

A recent study also suggested that an crowned profile would be better profile, which can be used to eliminate stress concentration [7]. In both cases manufacturing precision is very high and affect the cost of the bearing. If the manufacture of the roller surface is not accurate enough, the uniform stress distribution may not be achieved, because the edge stresses cannot be eliminated, as expected.

Research has been conducted to an alternative roller structural form, hollow roller with different hollowness which is more flexible when responding to variations of bearing loads, and it should have less strict requirements on manufacturing precision.

To achieve the logarithmic profile and exponential profile, high manufacturing precision is required. For the hollow roller the profile remains cylindrical. There are two ways by which contact-stress distributions can be influenced: the surface geometry of the roller and the stiffness of the roller. The elasticity to the roller ends decrease contact-stress in this sections. The end-stress concentration can be reduced by using hollow rollers. FEM analysis was made to evaluate the design concept of the hollow roller and comparing it with traditional solid profiles of rollers.

In the simulation was considered an uniform load. The rolling element fatigue life [8],[9] is inversely proportional to the maximum stress to the ninth power (Lundberg-Palmgren) (6) or Zaretsky equation (7).

$$L \approx \left(\frac{1}{\sigma_{\max}}\right)^9 \tag{6}$$

$$L = A\left(\frac{1}{\tau}\right)^{\frac{c}{e}}\left(\frac{1}{V}\right)^{\frac{1}{e}} \approx \frac{1}{\sigma_{\max}^n} \tag{7}$$

where:

A is a constant factor of material;

τ is critical shear stress;

c/e is Lundberg-Palmgren parameter (val. 9);

e is Weibull slope (val.1.1);

V is elemental volume;

S_{max} is maximum Hertz stress;

n is Hertz stress life exponent (val. 9…12).

Barnsby starting from the Ioannides-Harris theory [10], introduces the stress factor K_c.

$$K_c = 1 + \left(1 - C_L^{\frac{1}{4}}\right)\frac{\sigma_{VM,\lim}}{\sigma_{VM,\max}} \tag{8}$$

where:

$\sigma_{VM,\lim}$ is fatigue limit of Von Mises stress;

$\sigma_{VM,\max}$ is maximum of Von Mises stress;

C_L is lubricant parameter.

In this research was used for simulate the comportment of rollers, Von Mises stress criterion. The MD Nastran software found all stress components on each point and the results are conform with Von Mises stress equation [11]:

$$\sigma_{VM} = \frac{1}{\sqrt{2}}\sqrt{\left[(\sigma_x-\sigma_y)^2+(\sigma_y-\sigma_z)^2+(\sigma_z-\sigma_x)^2+6\cdot(\tau_{xy}^2+\tau_{yz}^2+\tau_{zx}^2)\right]} \tag{9}$$

The magnitude of uniform load-distribution used for the analysis 275 KN. Distributions of maximum Von Mises stresses are shown for solid cylindrical roller (fig.2), hollow roller with D_i=80mm (fig.3) and hollow roller with D_i=100mm (fig.4).

Crossing to the hollow roller bearings do not require major changes in technology. The present study deals with the problem of bearing resistance in assemblies bearings - wind power plant. New problems that face hollow roller bearings, are larger deformations and larger contact stresses.

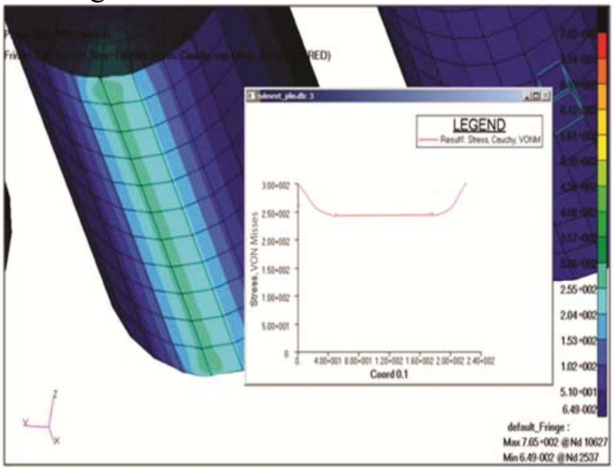
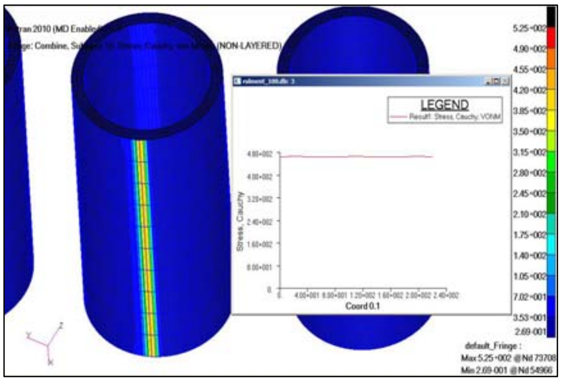


Fig. 2. Von Mises stress chart for solid cylindrical roller [Mpa]



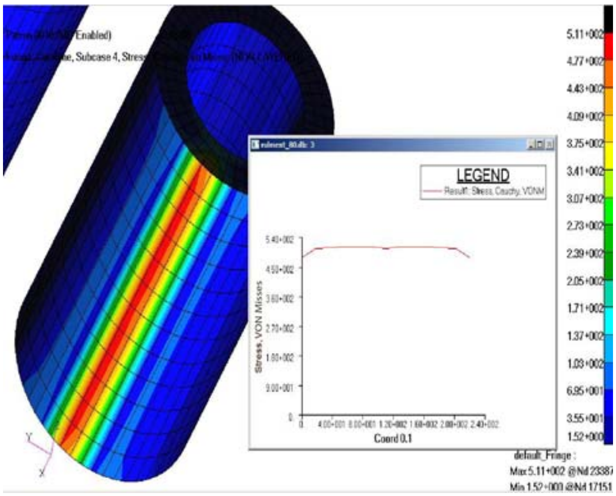


Fig. 3. Von Mises stress chart for hollowcylindrical roller with D_i=80mm [MPa]

Fig. 4. Von Mises stress chart for hollowcylindrical roller with D_i=100mm [MPa]

The research was conducted taking into account the real radial force of a wind power plant but the calculation was made on a single row bearing with cylindrical roller. In reality, these bearings are used rarely, generally using the bearings on two or four rows.

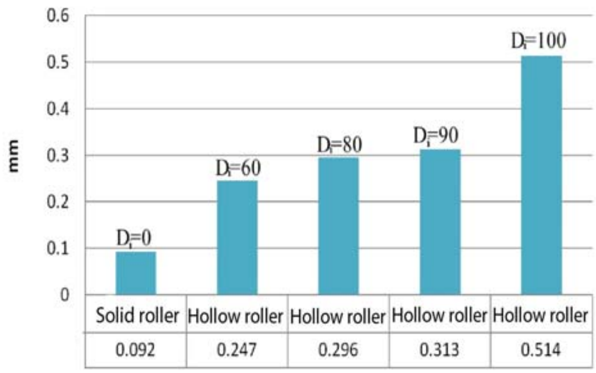


Fig. 5. Graph of deformations of bearing depending of hollowness-results obtained by finite element analysis

The research clearly shows that different hollow rollers, tested in simulations, not only reduces the inertial mass, but behave as good in deformations and much better in the contact stress.

In fig.5 and fig.6 can see the increasing of strains on the basis of the cavity. It is noted that this increase is lower for a bearing with outer diameter D = 1900mm.

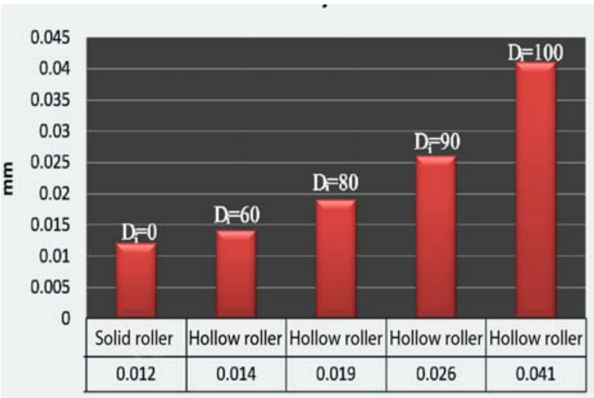


Fig. 6. Graph of deformations of roller depending of hollowness-results obtained by finite element analysis

4. CONCLUSIONS

The results of MD Nastran FEM for the Von Mises stress distributions for the hollow cylindrical roller under identical loads are interesting. Due to increased elasticity to the ends of hollow roller, the Von Mises stress decrease, the graphic being constant (Figure5 and Figure 6). Under the same conditions, the stress caused by the contact between the race and the hollow roller is less than the stress caused by the contact between the race and the solid roller.

Further, FEM simulation shows that a stress concentration occurs at the ends of solid cylindrical roller but for the hollow roller the stress concentration is the same on the entire length.

The results may indicate that for certain range of the roller geometry the use of hollow rollers will not weaken the strength of the bearing, on the contrary, increased its sustainability. In addition, about 50% of roller weight of a bearing could be reduced due to the introduction of the hollowness. FEM analyses demonstrate that the roller would render superior performance which could enable the elimination of the stresses at the ends of the roller. As a result, the roller could function similarly to a logarithmic-profile roller, but the manufacturing procedures of the roller are simplified.

The bearings with hollow rollers have advantages such as reduction of the inertial loads, weight reduction, material saving increasing of durability.

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