ISSN 2348 – 7968

Deformations Analysis of Hollow Rollers for Large Bearings using Finite Elements Method

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Abstract

The objective of the paper is to show some of the results of investigation into hollow roller implementation in large bearings construction as a solution for weight reduction. The finite elements analysis made with Nastran software highlighted the contact stress and deformations for a hollow roller with different levels of hollowness. The method allows choosing the best roller for bearing. The weight of large bearing decrease which leads to lower inertial forces and increase sustainability.

Keywords: *Hollow Rollers, Deformations, Large Bearings, Finite Elements Analysis*

1. Introduction

Using hollow rollers instead of solid rollers bring real benefits in large bearing industry. Determination of their cavity is the most important issue, according to the resulting sustainability of the entire system. Design optimization of hollow rollers and determination of dimensional typology for each functional situation leads to increase operating time, with significant economic benefits. Determination of resistance to stress and the comparison of deformations of hollow and solid rollers allow choosing the optimal variant of roller. It accepts the validity of Hooke's law for elastic deformations, joint frames were ideal geometric surfaces, d eformations are completely elastic and contact area is small compared to the size of surfaces in contact. The optimal setting of parameters with influence in power system was made by optimization techniques and probabilistic approach. Practicability of hollow rollers in construction of large bearings was examined for 2 material types [5]. The used material is allied steels: SAE 3310.. The use of hallow rollers has mainly been to achieve high speed, accuracy and low maintenance, also is unconventional and new.

Both the dynamic analysis performed with the help of finite element method as well as the results of the analytical model calculations lead to the conclusion that hollow cylindrical rollers can replace rollers with logarithmic profile, more expensive and heavy, bringing in the same time an increase of life of the bearing through the reduction of uneven wear of rolling elements.

2. Theory

The calculation uses the basic assumptions of the theory of linear elasticity, admitting additionally lack of adhesion forces, resulting so that the pressure on the contact area are positive or null.

$$u(x,y) = u_I(x,y) + u_{II}(x,y) \qquad (1)$$

The analytical model used is defined by the next group of three equations[1],[2]:

Geometrical equation of the elastic contact:

$$g(x,y) = h(x,y) - u(x,y) - h_o \qquad (2)$$

The equation (2) expresses the size of the distance between surfaces after deformation, as an amount between the initial separation of h(x, y), carried out in the absence of load, the composed deformation of u(x, y) and minimum close h_0 between surfaces, size also unknown.

Integral equation of the elastic contact (equation Boussinesq - Flamant)[6]:

$$u(x,y) = \frac{1}{\pi}\left(\frac{1-\upsilon_I^2}{E_I} + \frac{1-\upsilon_{II}^2}{E_{II}}\right)\iint_{A_r} \frac{p(\xi,\eta)}{\sqrt{(x-\xi)^2 + (y-\eta)^2}}\, d\xi\, d\eta \qquad (3)$$

where: E_I, E_{II} ,are Young modulus for each bodies in contact;

v_I and v_{II} are Poisson's ratio for each bodies in contact;

p(x, y) is the pressure of the common area..

Equilibrium equation:

$$\int_{A_r} p(x,y)\,dxdy = Q \qquad (4)$$

where Q is the normal force [N].

Bearings durability is based on equation of the theory developed by Lundberg and Palmgren and on which can determine basic bearing dynamic loads [8]:

$$\ln\frac{1}{s} = \frac{N^e \sigma_{\max}^c V}{z_0^h} \qquad (5)$$

where: N is the number of load cycles;

S is contact reliability;
σ_{max} is maximum orthogonal stress;
V is significant volume of material requested;
z_0 is the depth oa wich is realizing σ_{max}.

$$L_{10} = \left(L_\mu^{-e} + L_\gamma^{-e}\right)^{\frac{1}{e}} \quad (6)$$

L10 [hours]=L [million revolution]*10^6/(60*N)
where durability of rotating driveway L_μ and $_a$ fixed driveway Lγ are provided so:

$$L_\mu = \left[\frac{Q_{c\mu}}{Q_{e\mu}}\right]^p ; L_\gamma = \left[\frac{Q_{c\gamma}}{Q_{e\gamma}}\right]^p \quad (7)$$

Contact loads roller/driveway, Qj (j=1.. Z), are equivalent loads of roller level, and driveway lavel and Qeμ, Qeγ, can be calculated with the relations:

$$Q_{e\mu} = \left(\frac{1}{Z}\sum_{j=1}^{j=Z} Q_j^p\right)^{\frac{1}{p}} ; Q_{e\gamma} = \left(\frac{1}{Z}\sum_{j=1}^{j=Z} Q_j^{pe}\right)^{\frac{1}{pe}} \quad (8)$$

For large bearing durability in millions rotation is [3],[4]:

$$L_{na} = a_1 \times a_2 \times a_3 \times a_v \times a_r \times a_c \times a_a \times \left(\frac{C}{P}\right)^p \quad (9)$$

where: L_{na} is nominal durability in millions rotations[9];
a_1 is factor of reliability;
a_2 is factor for material;
a_3 is factor for running conditions;
a_v is speed factor;
a_r is a roughness factor;
a_c is a factor for lubricant;
a_a is factor of coaxiality ;
C - basic dynamic load [N];
P - equivalent dynamic load [N];
p - exponent for roller bearings: p=10/3;
The ratio C/P is direct dependent on contact stress.

3. Application of FEM for Optimization the Hollowness of Rollers

Stress analysis by finite element consists in replacing the studied ensemble through a structural system composed of subregions, called finite elements wich, in fact, are a part of that whole.
Were chosen rollers with D_{ext} = 120 mm and L = 180mm from SAE 3310, carburized at 65HRC. The diameter of bearing is D_{rul} = 1900 mm. For the inside diameter (hole diameter) were chosen six cases according to the following values: solid roller, hollow roller with Di_1 = 60 mm, Di_2 = 80 mm, Di_3 = 90 mm**.** Also it was applied FEM on a hollow roller with Di_2 = 80 mm, equipped with covers (2 mm and 4 mm, thickness)

For bearing with solid rollers and for all bearings with hollow rollers was made the finite element analysis with Nastran software.
When applied finite element analysis method to rollers, was used iteration cinematic method, that assumes that the nodal forces are known by evenly distributing, remaining to calculate the contact forces. The value of these forces is determined by successive iterations resulting when the speed of the node is null.
The results obtained by applying the finite element method were calculated in the position of maximum stress when the loading force is perpendicular to roller. Redistributed components of normal loading forc e is considered lower, in the angular positions, without influencing the results. [7]
It was considered the case of variable loads with maximum value Q=275 kN

Table 1: The values of contact elements when applying a load Q=50kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2391,5	0,3236	1031,8	0,0145
2	2396,2	0,3257	1036	0,0163
3	2400,8	0,3442	1039,6	0,0175
4	2404,7	0,3592	1042,2	0,0197
5	2401,6	0,3337	1038,2	0,0165
6	2399,6	0,3489	1037,7	0,0166

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

Table 2: The values of contact elements when applying a load Q=100kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2393,8	0,3836	1033	0,0153
2	2398,2	0,3857	1036,8	0,017
3	2403,1	0,4242	1041	0,0183
4	2406,7	0,4392	1043,4	0,0209
5	2403,9	0,3937	1039,4	0,0178
6	2401,6	0,4489	1039,4	0,0173

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

Table 3: The values of contact elements when applying a load Q=150kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2395,8	0,4436	1034,2	0,0157
2	2399,9	0,4457	1037,4	0,0176
3	2405,1	0,5042	1042,3	0,0187

4	2408,3	0,5192	1044,3	0,022
5	2405,9	0,4537	1040,6	0,0188
6	2403,3	0,5489	1040,9	0,0179

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

Table 4: The values of contact elements when applying a load Q=200kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2397,6	0,5136	1035,6	0,0163
2	2401,6	0,5157	1038	0,0184
3	2406,9	0,5942	1043,4	0,0193
4	2409,8	0,6092	1045,1	0,0229
5	2407,7	0,5237	1042	0,0198
6	2405	0,6189	1042,3	0,0187

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

Table 5: The values of contact elements when applying a load Q=250kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2399,2	0,5536	1037,2	0,0175
2	2403,4	0,5557	1038,6	0,0196
3	2408,5	0,6942	1044,4	0,0205
4	2411,2	0,7092	1045,9	0,024
5	2409,3	0,5637	1043,6	0,0212
6	2406,8	0,7189	1043,9	0,0199

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

Table 6: The values of contact elements when applying a load Q=275kN

Type	Contact pressure (N/mm^2)	Contact lenght (mm)	Von Mises stress (N/mm^2)	Deformation (mm)
1	2400	0,5736	1038	0,0184
2	2404,3	0,5757	1038,9	0,0204
3	2409,3	0,7442	1044,9	0,0214
4	2411,9	0,7592	1046,3	0,0247
5	2410,1	0,5837	1044,4	0,0221
6	2407,7	0,7689	1044,7	0,0207

1-solid roller; 2-hollow roller (D_i=60mm); 3-hollow roller (D_i=80mm); 4- hollow roller (D_i=90mm); 5- hollow roller with covers (D_i=80mm, B=2mm); 6- hollow roller with covers (D_i=80mm, B=4mm).

In the tables 1-6 are presented values resulted after applying FEM analysis with Nastran software

4. Conclusions

After FEM analysis, the optimal roller is the hollow roller with D_i= 60 mm. This roller allows weight reduction of large bearing with 23-27%, showing a very good behaviour in operation. Von Mises stress, to maximum load is approximate with stress of solid roller (Fig. 1) and deformations shows insignificant increases (Fig. 2).

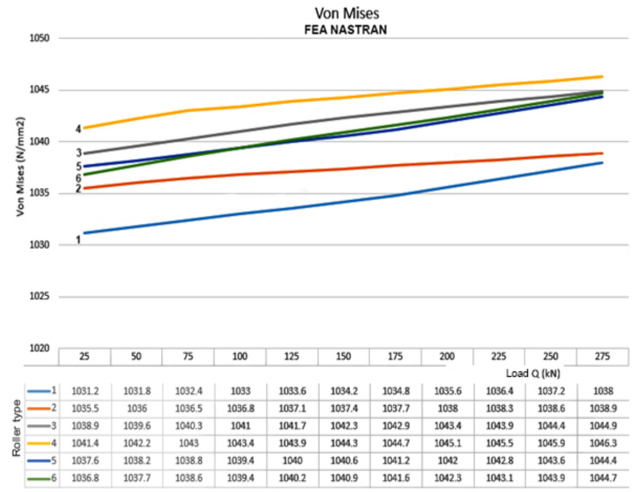

Fig. 1 Von Mises stress for 6 types of rollers depending on the loading force

From the graph shown in Fig. 1 are observed positions 1, 2, 3, 4 with an upward trend, with increasing cavitation. Mounting of covers makes stresses to decline due to stiffening of system.

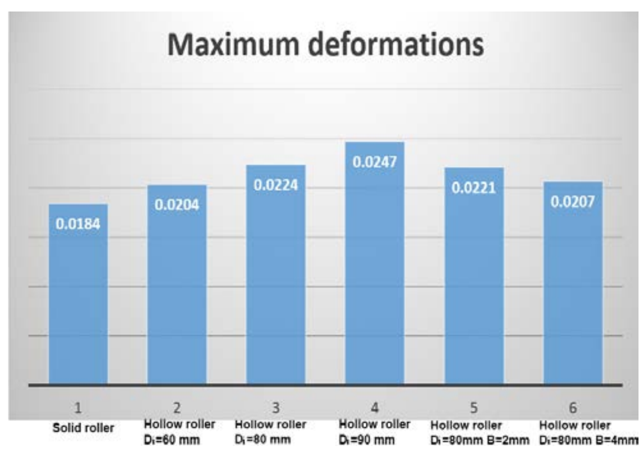

Fig. 2 Graph of deformations for Q=275kN

An analysis made of the results clearly shows slight increase of stresses in hollow rollers. The results are interesting from another point of view, namely that advantages of using of hollow rollers, completely reduces the disadvantage of increasing Von Mises stress and deformations.

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ISSN 2348 – 7968

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