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RESEARCH ARTICLE

Simplified modeling for needle roller bearings to analyze engineering structures by FEM

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Abstract

The rolling bearings are frequently applied elements of the mechanical structures. During finite element modeling of rolling bearing supported structures to define the adequate accuracy of bearing rigidity behavior enlarge immensely the FE model size and the related calculation time. So it is an understandable effort to use simplified substituting model for rolling bearing which carries the bearing rigidity character and does not make an unnecessary complicated structural model. In this paper the authors present a substituting technique for needle roller bearing.

Keywords

needle roller bearing · bearing rigidity · substituting FE model · finite element modeling

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1 Introduction

During the finite element modeling of complex structures it is difficult to model the rolling bearings in general and the needle roller bearings in particular. Due to the Hertz contact it is needed to apply very dense mesh on one hand and contact examination on the other hand. According to our examinations for a single needle roller bearing at least 43 hours running time was required (on an average configured PC) for a "more or less" accurate rigidity examination of needle roller bearing. Quite obvious that in this way it is not possible to build the model for a complex structure. Such kind of substituting models should be found – for the complex structure – which describe the needle rolling bearing rigidity behavior in the required accuracy. At the same time there are also required: not creating the structural model by unnecessarily large number of elements (and node numbers) and the possibility to leave out the contact examination.

To substitute needle rolling bearings for finite element modeling of complex structures we elaborated two substituting models, such as: one solution with spring and an other one with bush.

2 Analytic solutions of the elastic deformation and the stiffness of needle rolling bearing

The basis of the substituting model elaboration is the calculation of the Hertz-kind elastic deformation calculation. We applied in our study the results of three authors – Palmgren, A; Kovalszkij, B. Sz.; Eschamann, P. – found in the references [2–4].

In case of two cylinders having parallel axes the elastic deformation, i.e. the approximation of two (for located) distant stress-less points of the bodies is (according to Palmgren) [1,2]:

$$\delta = 1.360 \frac{1}{L^{0.8}} \frac{Q^{0.9}}{E_r^{0.9}} \quad [\text{mm}], \tag{1}$$

where Q [N] is the compressive force, L [mm] is the effective length of the two cylinders, E_r is the equivalent modulus of elasticity having the value for steel-steel cylinders:

$$E_r = 109\,890 \text{ MPa}. \tag{2}$$

As it is seen in formula (1) the Hertz-kind elastic deformation does not depend on the radii of the cylinders in case of two contacting cylinders according to Palmgren [2]. (This was proved by us in numerical examinations.)

According to Kovalszkij [3] the elastic deformation can be calculated by the next formula:

$$\delta = p_{\max} b \frac{1-\nu^2}{E}\left(\ln\frac{d_1}{b}+0.407+\ln\frac{d_2}{b}+0.407\right), \tag{3}$$

where p_{max} [MPa] is the maximum surface pressure formed, such as the Hertz-stress, b [mm] is the half width of the contact surface, d_1 and d_2 [mm] are the diameters of the two cylinders, E [MPa] is the modulus of elasticity , and υ is the *Poisson's* ratio.

Eschmann [4] gives the elastic deformation of the inner and outer rings together:

$$\delta_b+\delta_k=\left(\frac{1}{26300}\cdot\frac{1}{L^{0.92}}\right)^{1/1.08} Q^{1/1.08}\,[\text{mm}]. \tag{4}$$

The spring stiffness of the needle roller (considering both the inner and the outer contact together) is:

$$s_g=\frac{\partial Q}{\partial\delta}\quad, \tag{5}$$

where $\delta=\delta_b+\delta_k$.

We carried out calculations for one RNA 35 × 50 × 27.7 type needle roller bearing having the following geometric parameters: inner race diameter is D_b = 35 mm; outer race diameter is D_k = 45 mm; rolling element diameter is d_g = 5 mm, the length of the needle roller is L = 16 mm. According to these three authors the change of the spring stiffness in terms of the roller load is shown in Fig. 1.

At F_r = 1250 N radial load of the bearing the highest spring stiffness values of the examined bearings – according to the authors – are as follows:

Palmgren: s_g = 229 600 N/mm;

Kovalszkij: s_g = 219 400 N/mm;

Eschmann: s_g = 217 000 N/mm.

According to the recommendation of the three authors the spring stiffness values of the roller at F_r = 76 000 N radial load are:

Palmgren: s_g = 347 000 N/mm;

Kovalszkij: s_g = 326 000 N/mm;

Echmann: s_g = 295 000 N/mm.

The relationship between the radial bearing load and the elastic deformation [1]:

$$F_r=K_n\delta^n, \tag{6}$$

where $\delta=\delta_b+\delta_k$, and the coefficient K_n is:

$$K_n=\left[\frac{1}{\left(\frac{1}{K_b}\right)^{1/n}+\left(\frac{1}{K_k}\right)^{1/n}}\right]^n, \tag{7}$$

where n = 10/9 (in case of ball bearing n = 3/2). For the inside and the outside rings K_b and K_k are the contacting constants and

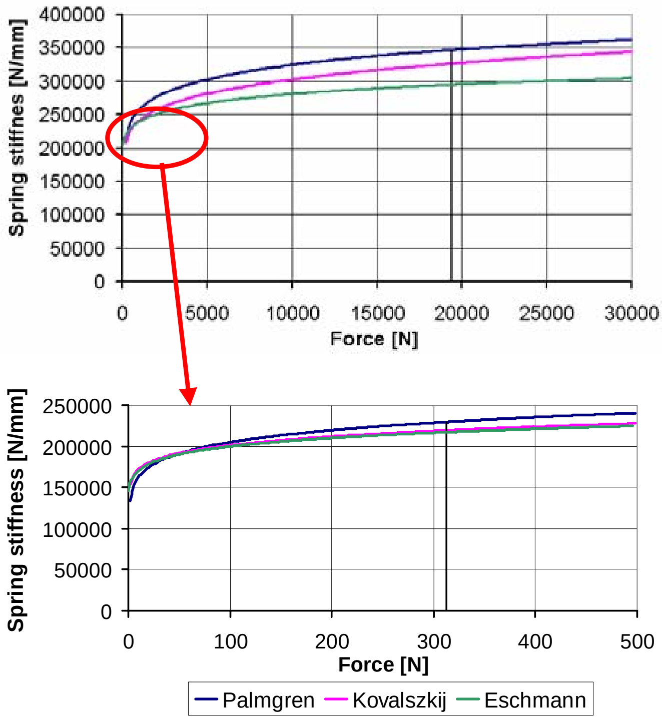

Fig. 1. The spring stiffness of the needle roller bearing in the function of the roller load

their values can be determined – according to Palmgren – by the following formula:

$$K_b=K_k=\left(\frac{1}{1.360}\cdot L^{0.8}\cdot E_r^{0.9}\right)^n. \tag{8}$$

If $K_b=K_k$ then the relationship (7) is changed to the following:

$$K_n=\frac{K_b}{2^n}\quad. \tag{9}$$

The relationship between the F_r radial load and the maximum load Q_{max} on the rolling element is:

$$F_r=ZQ_{\max}J_r\quad, \tag{10}$$

where Z is the number of the rolling elements and J_r is the so called Sjövall integral [5], which provides the value for line contact clearance-free bearing as follows:

$$J_r=0.2453. \tag{11}$$

The radial elastic displacement of the inner ring center (shaft) is:

$$u=\left(\frac{Q_{\max}}{K_n}\right)^{1/n}. \tag{12}$$

The spring stiffness of the bearing can be calculated from the following relationship:

$$s_{cs}=\frac{\partial F_r}{\partial u}. \tag{13}$$

For example the spring stiffness of RNA 35 × 50 × 27.7 type needle roller bearing is shown in the function of radial load in Fig. 2.

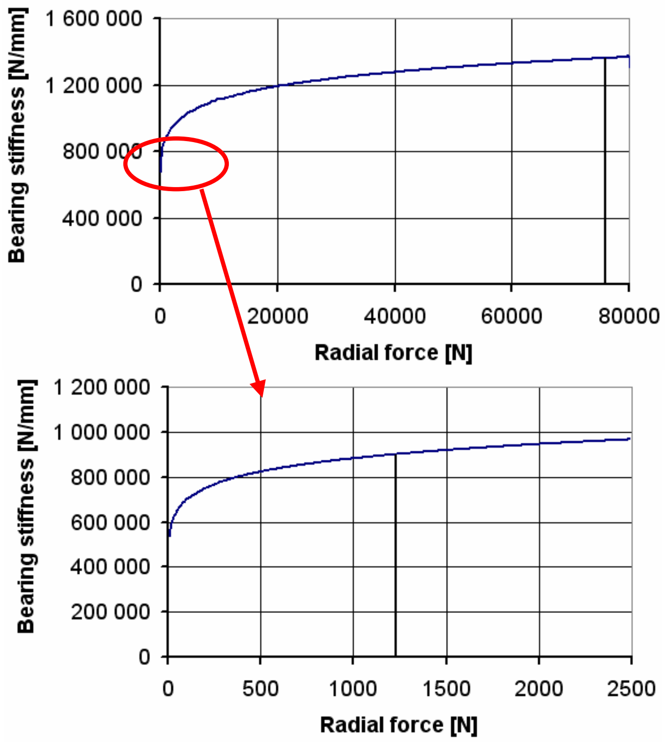

Fig. 2. Radial spring stiffness of the bearing in the function of radial load

For F_r = 1250 N radial load the spring stiffness of the bearing is:

s_{cs} = 900 700 N/mm ,

and in case of radial load F_r = 76 000 N:

s_{cs} = 1 360 000 N/mm .

3 FE calculation for cylinder-cylinder contact

In case of cylinder-cylinder contact the Hertz contact parameters can be determined (first of all for elastic displacements) by numerical models as well. The aims of the examinations were to verify the calculation methods found in the references and also to give orientation to selecting models for the numerical modeling of contact problems.

Analytical and numerical models used to calculate the Hertz mechanical characteristics and the results are compared in Table 1. The compressive force of the two cylinders is: 315 N. For the elastic deformation the analytic model was based on Palmgren model.

Tab. 1. Comparing the results of analytic and FE calculation

Contacting cylinders	Ø35 - Ø5 convex-convex		
Hertz characteristics	b [mm]	p_{max} [MPa]	u [mm]
Analytic model	0.044502	559	0.0007564
FE model	0.048433	549	0.0007345

For the FE calculation the MSC.Marc program was used. To obtain the tabulated finite element results 9 level adaptive mesh densities should be used [7] which need several hours running time.

4 Finite element modeling of needle roller bearing

We carried out the numeric analysis of a real needle rolling bearing "built in a structure". Truly, due to the contact surfaces the examination needs reasonable running time. However, the results obtained this way give basis to evaluate the accuracy of the substituting models. The building in of the examined needle rolling bearing is shown in Fig. 3.

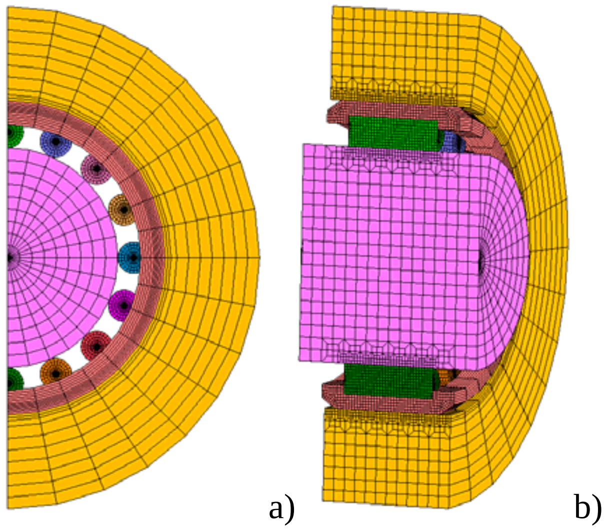

Fig. 3. The examined built in needle roller bearing (RNA 35x50x27.7). a) side view; b) isometric view

The behavior of needle roller bearing for radial load (this may be compared to the analytic calculation) and for bending load were examined too.

The bearing load is 1250 N which is applied through the shaft. Three different adaptive mesh density level examinations were carried out for the given model and conditions. The main characteristics of the examinations are summarized by Table 2. As the result of the calculations the vertical displacement of the shaft axis is shown in Fig. 4 along the shaft length.

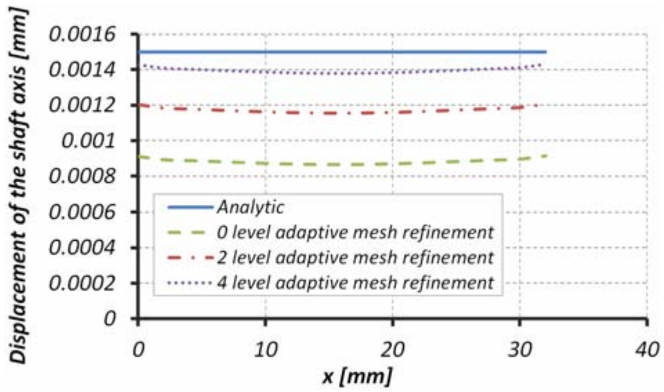

Fig. 4. Vertical displacement of the shaft axis along the shaft

Fig. 4 shows well that to describe the required accuracy of bearing rigidity behavior requires at least 4 level adaptive mesh density. Here the CPU time is about 43 hours. (It is to be noted that the mentioned CPU time relates to "half bearing" utilizing the condition of symmetry in bearing geometry. For a complete

Tab. 2. The main characteristics of three different mesh density level models

	Case 1	Case 2	Case 3
Mesh density level	0	2	4
Number of elements	44 352	96 782	321 608
Number of node	48 865	133 224	502 240
CPU time	**0.35 hour**	**1.78 hour**	**42.7 hour**

bearing examination the expected CPU time is multiple of the given one.)

The needle rolling bearing behavior under bending moment can not be decided analytically except with reasonable neglecting. Thus the only acceptable way is the numerical modeling. Here the bending moment on the shaft was applied on the two endplates of the shaft by force couples having the value of 19.6 Nm on the hole shaft.

The shaft's angular rotation results obtained by the examination on the three different level models (see Fig. 5).

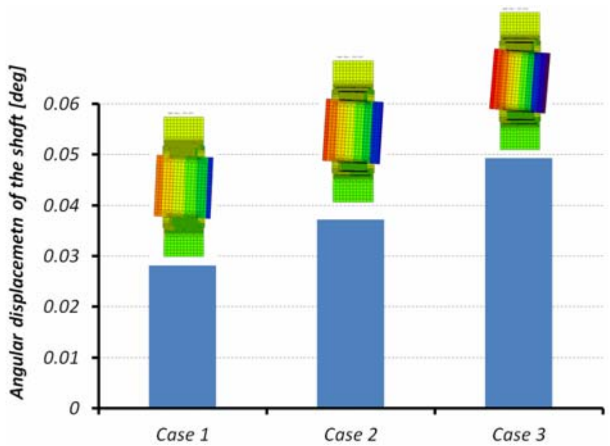

Fig. 5. Angular rotation values obtained for different level adaptive net density.

The numeric and analytic results of bending moment loaded needle roller bearings can not be compared (due to lack of data). However it can be seen from Fig. 5 that at least 4 level adaptive net densities are needed to obtain the required accuracy. Here the CPU times are larger by about 10% compared to the values given in Table 2.

5 Substituting models of bearings

For the finite element models of complex structures two substituting models of needle roller bearings were elaborated by us. One was a solution with spring and the second by bush. The models should deal with:

- the load distribution within the bearing;
- the deformations of the bearing rings besides the Hertz-kind deformations;
- the bearing clearances;
- the bending moment as well as the radial load.

5.1 Spring substituting model

The essence of the spring model is that the rolling elements are substituted by a series of linear springs. These spring elements carry the rigidity characteristics of the contacting rollers and bearing rings (Fig. 6).

The spring stiffness should be determined in the following way: the resultant of the parallel connected spring element series answers to the "contact" rigidity of both contact surroundings of the rolling elements, i.e. inside and outside. The rolling element stiffness can be calculated by formula (2). In case of F_r = 1250 N radial load one spring element has the stiffness of 13 750 N/mm (Fig. 7) calculated by 220 000 N/mm rolling element stiffness and 17 substituting spring elements in one row.

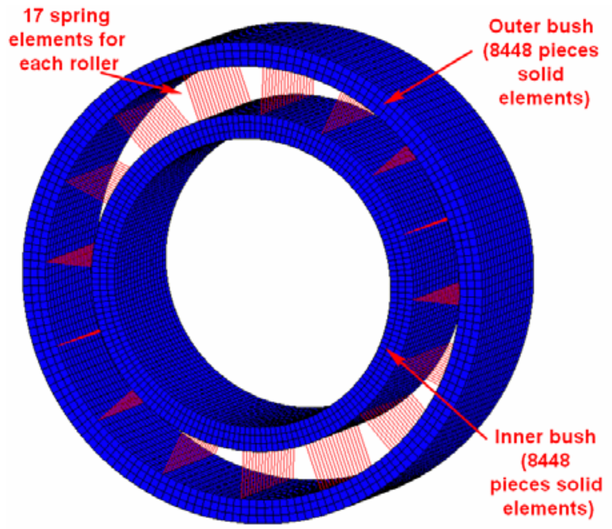

Fig. 6. Substituting spring model of the needle roller bearing

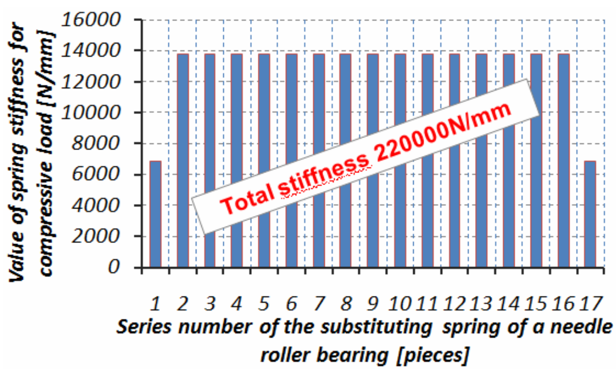

Fig. 7. Spring stiffness distribution for springs in a row (along the rolling element length)

The spring element behaves differently for tension and compression loading. Outside of the loaded zone the elements show tension but these elements do not take part in the load transmission. Therefore the spring stiffness of these springs has 0 value (Fig. 8).

There is a chance to consider the bearing clearance though it is not needed during the modeling. The bearing clearance can be considered via the characteristic of the spring element in such a way that within the bearing clearance the spring stiffness is

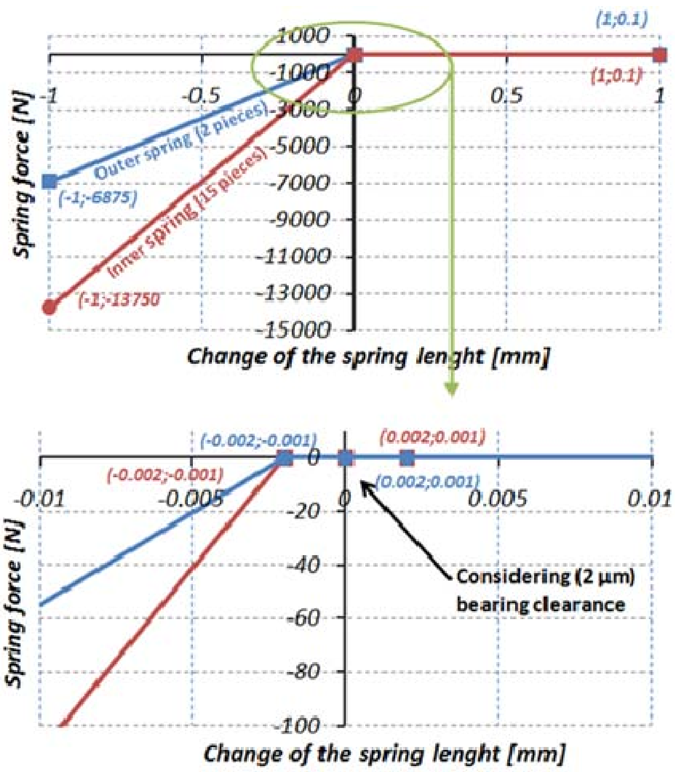

Fig. 8. Load-displacement characteristics of spring element. The lower enlarged figure shows the possibility to consider the bearing clearance.

negligible small. The spring stiffness characteristic of bearings with clearance is shown in Fig. 8.

The load direction displacements are calculated different ways for the inner ring and/or the shaft are shown for two kinds of radial loads in Table 3, while the calculated values of the bending moment loaded substituting model are shown in Table 4.

Tab. 3. The elastic displacement of the inner ring in case of loading.

Bearing radial load	1250 N	76 kN
Calculation method	The radial displacement of the inner ring [μm]	
Analytic, according to Palmgren	1.51	62
Real bearing according to FE	1.41	67
Substituting spring model	1.71	67

Tab. 4. The elastic angular rotation of inner ring in case of bending moment applied.

Calculation method	Angular rotation of the inner ring [minute]
Realé bearing according to FE	3.0’
Substituting spring model	4.6’

For radial loading in case of smaller loading the deviation is only about 20%. However it should be noted that for the bigger radial load the CPU time increased from 42.7 hours to 97 hours having the same model parameters. It was caused by the increased contact area for the bigger load which increased the contact examination time requirement.

In case of bending moment loading the difference between the two calculation values is 50%. The reason of the relatively big difference is that the real bearing would require increased adaptive net density for the FE model (see Fig. 5) which was not possible as the CPU time could not be increased further.

The application of the substituting spring model is shown by the block diagram (Fig. 9).

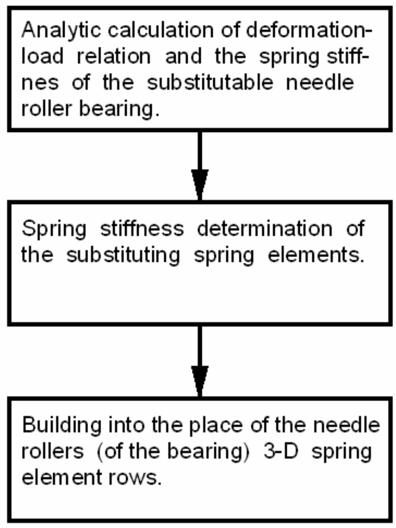

Fig. 9. The activity scheme of the spring model application.

5.2 Bush substituting model

The essence of the bush substituting model is: the row of the rolling elements is substituted by a bush having the required fictitious material characteristic which produces the same deformation for loading than the original row of rollers (Fig. 10).

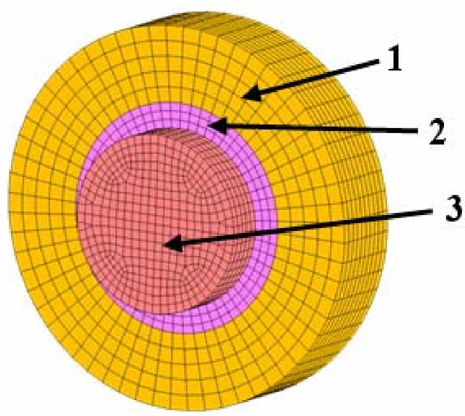

Fig. 10. Substituting bush model of needle roller bearing. **1** – Outer ring together with the housing; **2** – Substituting bush; **3** – Inner ring together with the shaft.

Comparing the two models the advantage of the bush model is that it is simpler to build in to the structure. On the other hand bush model does not make possible the characteristic fine setting (e.g.: changing spring stiffness for the different rolling elements) of the spring model. This is disadvantageous for the bush model. However in the practice fine settings are not required in most cases.

The modulus of elasticity for the bush should be determined so that the deformation of the bushing for a given load should be the same as the deformation of the rolling elements' row. For determining the modulus of elasticity it is enough to apply a 2-D model where the inner and the outer rings are considered completely rigid.

The bush material is a so called hyper elastic material which can be loaded by compression only (tension can not be applied).

Between the outer ring and the substituting bush there is a glued connection while between the shaft and the substituting bush there is a contact connection, in order to reduce the CPU time.

The bush substituting model was examined on rough mesh and fine mesh models (Fig. 11). The characteristics of the two models are given in Table 5.

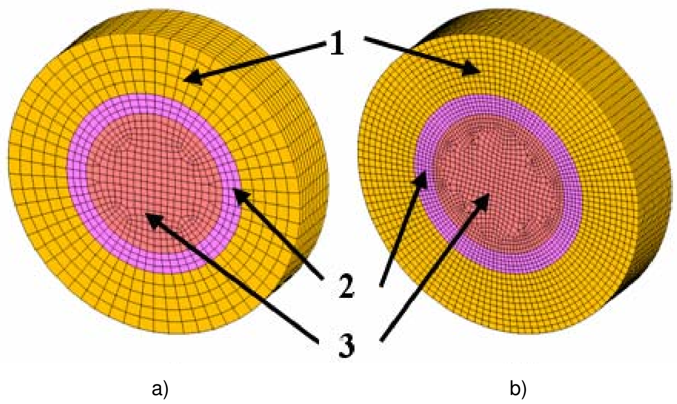

Fig. 11. Substituting bush model of needle roller bearing. **1** – Outer ring together with the housing; **2** – Substituting bush; **3** – Inner ring with shaft; a. rough mesh model; b. Fine mesh model.

Tab. 5. Main characteristics of the two kinds of bush models

	Rough net	Fine net
Element type	solid	solid
Average element size	1.5 mm	0.75 mm
Element number	8 650	53 600
CPU time	**0.4 minute**	**30 minute**

The data of Table 3 are completed by the calculated displacement values and these results are shown in Table 6.

At 1250 N radial load the fictitious modulus of elasticity becomes to be E_f = 8000 MPa and at 76 kN radial load it becomes to be E_f = 12 500 MPa.

It can be seen from the calculated results that there are hardly any difference between the values calculated by rough mesh and calculated by fine mesh model though the fine mesh calculation time requirement was 100 times larger than the time requirement for the rough mesh model. Furthermore it can be seen that the calculated values by bush models fit well into the values calculated by other methods.

Tab. 6. The inner ring displacement for radial loading

Radial bearing load	1250 N	76 kN
Calculation method	Radial displacement of the inner ring [μm]	
Analytic, according to Palmgren	1.51	62
Real bearing according to FE	1.41	67
Substituting spring model	1.71	67
Bush model, rough mesh	1.41	60
Bush model, fine mesh	1.54	64

The calculated values of angular rotation for bending moment are shown in Table 7.

Tab. 7. Elastic angular rotation of the inner ring in case of bending moment.

Calculation method	Angular rotation of the inner ring [minute]
Real bearing according to FE	3.0'
Substituting spring model	4.6'
Bush model, rough mesh	3.7'
Bush model, fine mesh	4.1'

For the application of substituting bush model of needle roller bearing – at the finite element calculation of the complex structures – an activity series should be carried out as it is given in the block diagram of Fig. 12. The first step is to determine the elastic deformation of the bearing by the help of the known loading and the geometric data of the substituting needle rolling bearing. For the above the relationships of (4), (5), (6) and (7) are required.

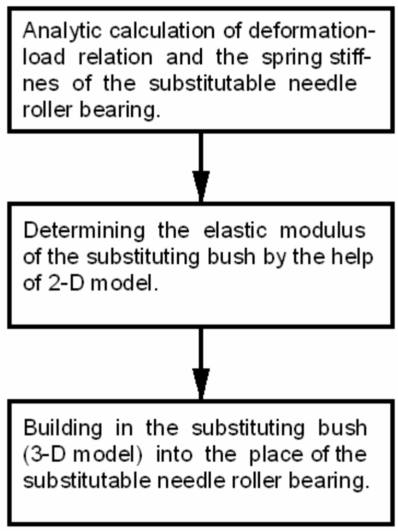

Fig. 12. Main activity steps for the application of substituting bush model

For the second step the fictitious elastic modulus must be determined by applying a 2-D model and the elastic deformation of the bush should be agreed with the analytic calculation of deformation. This calculation process needs an iteration.

The substituting bush should be built in replacing the rolling elements row of the needle rolling bearing of the complex structure FE model. Here the inside and outside bush diameters are coinciding with the inner and outer race diameters of the bearing. The bush and the roller widths are also considered to be equal. For substituting bush hyper elastic material the value of the compressive elasticity modulus is determined in the second step while the tensile elasticity modulus is considered with a small value.

6 Summary

For substituting needle roller bearings in complex structures two kinds of simplified substituting bearing models were elaborated. The aim of the substituting bearing model application is reducing the length of the time consuming calculations by considering the rigidity behavior of the bearing and eliminating the contact examination within the bearing.

The basis of the substituting model elaboration is the Hertz deformation calculations.

For the verification of the elaborated models finite element calculations were carried out on true needle roller bearings where the applied loadings were radial and bending moment loadings. These calculations showed that very long calculations were required for obtaining the sufficient accuracy which can not be permitted for structures having rolling bearings.

Within the frame of the present research and development work for the substitution of needle roller bearings two kinds of models were elaborated and both were found usable for practical application.

Handling the spring model needs more complicated FE data preparation but it is possible to set finer model (e.g.: varying spring stiffness for the rolling elements). The FE data preparation work for bush model is simpler and provides adequate accuracy for practical examinations. Therefore substituting bush model with rough mesh is recommended for finite element modeling of common structures.

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