FIFTH INTERNATIONAL CONGRESS ON SOUND AND VIBRATION
DECEMBER 15-18, 1997
ADELAIDE, SOUTH AUSTRALIA

LOW CYCLE CARRYING CAPACITY OF BEARINGS WITH HARDENED ROLLING LAYER

Ivan Prebil[1], Robert Kunc[1], Matjaž Torkar[2]

[1]University of Ljubljana, Faculty of Mechanical Engineering, Slovenia
[2]Institute of Metals and Tecnology, Slovenia

ABSTRACT

The paper deals with the problem of the actual carrying capacity of a rolling contact in large axial bearings with surface hardened raceway. The carrying capacity of such bearings is usually given with the maximal permissible force on the rolling element with the highest load. The established criteria of maximal permissible plastic deformation of the raceway, and the maximal allowed subsurface stress on the hardened layer boundary give widely varying values for the carrying capacity of the rolling contact. In order to determine the actual carrying capacity of the rolling contact in axial bearings with low speed of rotation, considering base material properties, hardness and thickness of the hardened layer, and the geometry of the contact, we have measured the models of bearing raceways made of 42CrMo4 and C45 materials. The loads were static and low cycle dynamic. We have determined the cyclic curves (force – deformation), limit of fast increase in (elastic and plastic) deformation gradient, and the size of the contact surface. Simultaneously with the experiment, we have checked the sub surface stresses using a FEM model, considering the material nonlinearities. On the basis of the measurement results and the FEM model we would like to set up a simplified mathematical relation that gives the permissible contact force considering the base and hardened material properties, and the geometric model of the rolling contact.

PROBLEM PRESENTATION

The basic theory of two bodies in contact has been set up already in 1882 [1]. The Hertz theory, that is limited to ideally elastic bodies, and no friction in the contact, has been the basis for several successful models developed in 20th century [2], [3], [4], to comply with the needs of the fast industrial and transport development.

The term "rotational connection" describes a machine structure that allows relative rotating or oscillating motion of two substructures. It is loaded with a combination of axial and radial

forces, and a turnover moment, and consists (Figure 1) of a rolling bearing, screw connection, fixing the bearing rings to supporting and upper structures, and frequently also a gearing, integrated with the inner or outer bearing ring and used to drive the rotating substructure. The external loads are transferred from the upper (rotating) substructure over the bearing rings and rolling elements to the supporting (fixed) structure. The loads are distributed over a large number of rolling elements, so for the purpose of rolling element force calculation, the bearing is considered a statically indeterminate system.
The mathematical models for the calculation of load distribution over rolling elements [5], [6], [7], supposing stiff upper and lower structures, show that the force distribution over the elements approximates the form of a cosine function (Figure 2). This supposition proved to be too inaccurate [8], [9]. The actual load distribution can differ significantly from the computed one (cosine function) in shape and magnitude of contact forces (Figure 2).

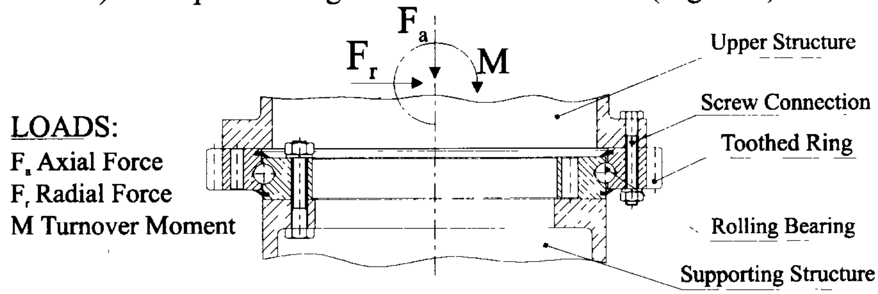


Figure 1: Elements of rotational connection

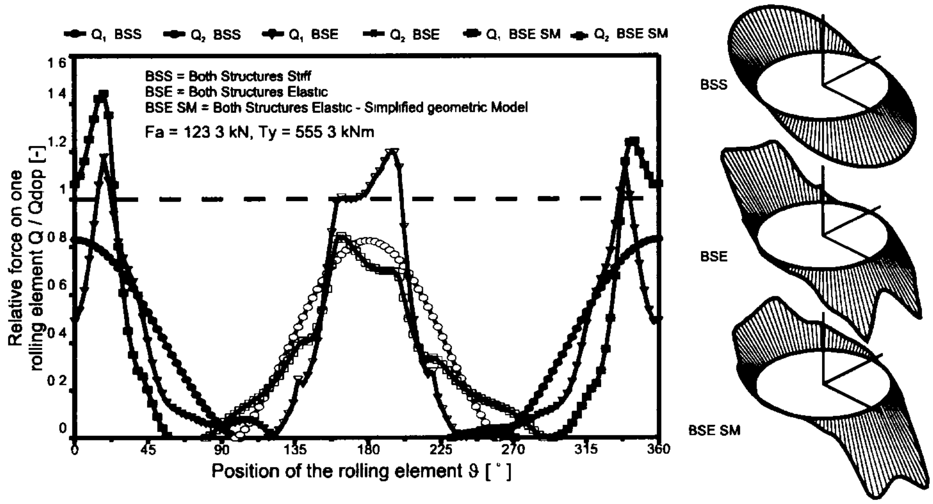


Figure 2: Example of load distribution over rolling elements

THE TWO CRITERIA FOR THE DETERMINATION OF STATIC CARRYING CAPACITY OF A ROLLING CONTACT

Damage appearing on large, slowly running bearings is caused mainly by large static loads and material fatigue [5], [6]. Too high static loads on bearings during standstill periods cause plastic deformations in the contact of rolling element and raceway, and consequently the movement of the rotational connection becomes uneven.
To assure normal operation of a slowly running rotational connection during its predetermined lifetime, two criteria have been established to evaluate the static carrying capacity of a large axial bearing:

- Criterion of allowed permanent deformation (ISO)
- Criterion of allowed shear stress on the edge of the hardened layer.

CRITERION OF ALLOWED PERMANENT DEFORMATION

Permanent plastic deformation in the contact of two bodies (rolling element and raceway) appears when the contact is overloaded. In practice plastic deformations on the raceway surface appear already with small loads, because of the relatively rough surface. This effect causes plastic smoothing of rough surfaces. The influence of local overloads and pertaining plastifications can be neglected in the calculations of the carrying capacity of rotational connections [10].

The basic permissible static load on the non moving bearing, i.e. on the rolling element with the highest load equals the load causing permanent deformation 0.01% of the rolling element diameter in the contact [6].

This size of plastic deformation guarantees smooth vibrationless running of a bearing, and has been adopted in the currently valid ISO standard [11]. The computational procedure [12] for the determination of the highest load on the rolling element of a steel bearing with both rings hardened through the whole section and raceway hardness between 63.5HRc and 65HRc has been developed on the basis of experimental data.

CRITERION OF ALLOWED SHEAR STRESS ON THE EDGE OF THE HARDENED LAYER

The criterion requires a suitable thickness and hardness of the surface. The damage will not appear if the subsurface stress, computed according to the yield criterion of maximal shear stresses, does not exceed the plastic yield limit of the core [13]. The limit of the hardened layer is in the depth, where the hardness does not fall under 50HRc (Figure 3). The plastic yield limit of the core is measured in the depth of three times the thickness of the hardened layer.

The distribution of the subsurface stress, computed according to the yield criterion of maximal shear stresses must be such that the maximum falls in the hardened surface layer. The stress on the limit of the hardened layer must be equal or lower than the plastic yield limit of the core (Figure 3).

$$\sigma_s(depth\ at\ 50HRc) \leq \sigma_{02}(core).$$

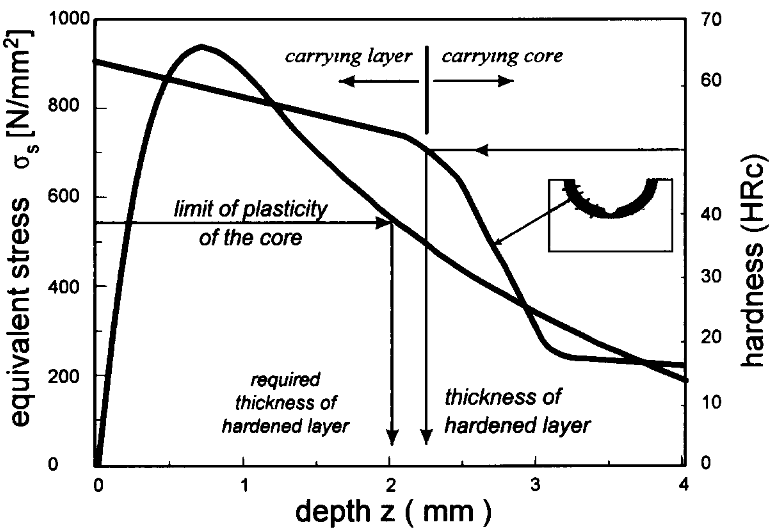


Figure 3: Equivalent subsurface stress in Hertz contact and hardness decrease in hardened layer

DETERMINATION OF THE CONTACT CARRYING CAPACITY

It is our purpose to determine the actual carrying capacity of the contact between the raceway and the rolling element, with the bearing mounted. The bearing should have surface hardened raceways, actual geometric properties (size of rolling element, diameter ratio of rolling element and raceway), material properties (thickness and hardness of hardened layer, type of material), and is loaded with actual loads (load magnitude, amplitude, and number of load cycles).
The actual carrying capacity of a large dimension axial bearing is specified with the largest allowed contact force on the rolling element with the highest load (Figure 2).

SUITABILITY OF THE LISTED CRITERIA

The material for rotational bearings must be sufficiently soft to facilitate the manufacturing process, especially because of the integrated gearing and screw connection. Such materials do not have the surface carrying capacity and strength to prevent the impressions (low carbon steels C45, 42CrMo4 – ISO 683/1). The surface carrying capacity is increased with suitable heat treatment, i.e. hardening of the raceway. The cracks appearing in the base material under the hardened layer and growing towards the surface are typical for bearings with soft core and insufficient hardened layer thickness, and are caused by excessive subsurface stresses.
The stresses that cause larger permanent deformations influence the carrying capacity and, under repeated loading also the lifetime of the bearing. The stresses under the surface (at the approximate depth of 70% of the shorter axis of the contact ellipse) can surpass the material plasticity limit and cause a local plastic deformation. When the load decreases, the residual stresses remain in the material, and on next loading counter the stresses caused by external loads. If the residual stresses are higher than the "external" ones then the plasticized region decreases, otherwise a visible damage, leading to eventual breakdown, appears. The process of cyclic loading, and the pertaining subsurface stresses, appearing in normal rotation of a rotational connection are called plastic cyclic fatigue. The limit of repeated plastification is the shake down limit [14], [15]. The determination of this limit depends on material, geometry of bodies in contact, and distribution of stresses on and under the surfaces, and is therefore a complex task.
The conditions for the use of the criterion of allowed permanent deformation and the criterion of allowed shear stress on the edge of the hardened layer are in the case of large dimension rolling bearings not completely fulfilled, mainly due to the technology. The criterion of allowed permanent deformation supposes that both rings are hardened through the whole section, with very high surface hardness – over 63 HRc. The main deficiency of the criterion of allowed shear stress on the edge of the hardened layer is the computation of the subsurface stresses according to Hertz theory. This computation considers only the elastic region of the material deformation curve, and therefore disregards the plastic material hardening.

GUIDELINES FOR THE DETERMINATION OF THE CONTACT CARRYING CAPACITY

Because the listed criteria are not completely satisfactory, we would like to determine an experience-derived formula for the critical contact force, based on the following steps:

- Experimental determination of the static carrying capacity of the contact
- Experimental determination of the low cycle load carrying capacity
- Computation of subsurface stress with the use of the FEM (Finite Element Method) model

Test specimens are made of C45 and 42CrMo4 (ISO) materials.

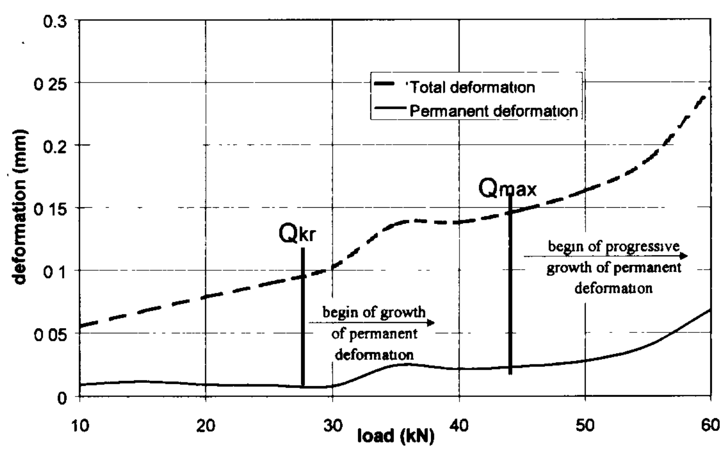
Example of a specimen data:
Material: C45:
- basic ($\sigma_e = 353$ Nmm^{-2}, $\sigma_u = 621$ Nmm^{-2}, $E_{el} = 2.1*10^5$, $\nu = 0.3$,for computation: $E_{pl} = 0$ Nmm^{-2} material ideally plastic)
- hardened layer ($\sigma_e = 2200$ Nmm^{-2} (=σ_u), Thickness of hardened layer = 1mm, $E = 2.58 *10^5$, $\nu = 0.3$)

Geometry of bodies in contact:
- diameter of raceway profile: $D_t = 16.000$ mm
- ball diameter: $D_k = 15.081$ mm
- $S = D_k / D_t = 0.943$

Measurement system:
- INSTRON (Load Cell : Lablow 500 kN, linear displacement transducer Instron 12.5 mm)

STATIC CARRYING CAPACITY

We have measured the change of deformation and contact surface size as a function of the static load. The plastic deformation has been calculated from the relative indications of initial and final positions of the deformation transducer. The size of the contact surface has been determined after unloading. We applied the method of indicating medium, containing micrometric hard particles [16], (Figure 4).
Figure 4 shows the limit load, where the deformation gradient increases steadily. At smaller loads the deformation regains the previous state after unloading (Fig. 8). As soon as the plastic yield limit is surpassed, the plastic deformation increases – critical force Q_{kr}. The plastic deformation is more or less constant in the region of the hardening of base material, after that it increases progressively – maximal force Q_{max}.



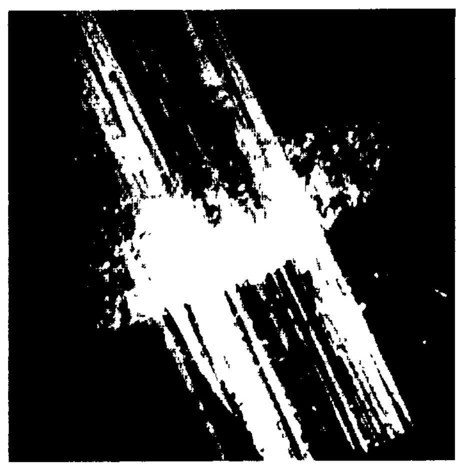


Figure 4: Contact deformation as a function of load and measurement of contact surface size

CARRYING CAPACITY UNDER LOW CYCLE LOADS

The determination of the carrying capacity under low cycle loads is simply an extension of the previous (static) process, with the added fatigue influence. We are loading and unloading the bodies in contact and recording the increase in deformation and contact surface size as a function of the load magnitude and the number of load cycles.

Load conditions:

- Low cycle (number of cycles up to $5*10^4$),
- Load Control: maximum (10kN, 20kN, 30kN, 40kN, 50kN) and minimum (1kN) of load amplitude is constant we are recording the change of deformation
- Load frequency 5Hz
- Load form is sinusoidal.

When recording the deformation as a function of number of load cycles, an increase in plastic deformation (hysteresis shift) and a diminishing rate of plastic deformation increase (due to increase in contact surface and plastic hardening) can be detected (Figure 5). Critical, i.e. maximal load of the rolling contact is derived from the increase of permanent deformation as a function of load magnitude and number of load cycles (Figure 6). Especially important is the load, where the increase of the permanent deformation becomes so small that it can be neglected.

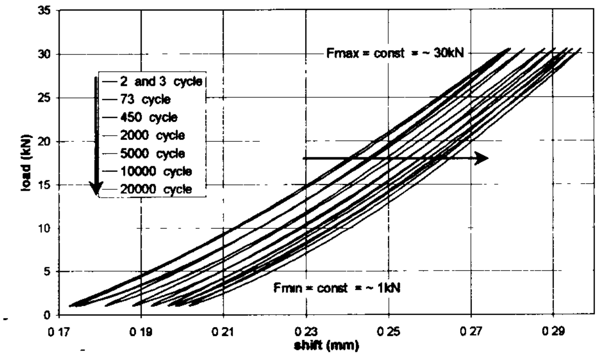


Figure 5: Load graph (increase of deformation)

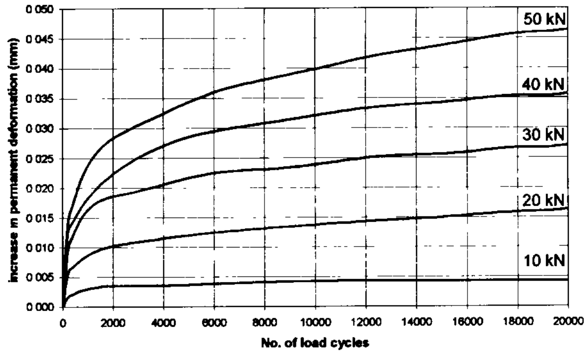


Figure 6: Increase of permanent deformation in the contact as a function of number of load cycles

CHECKING OF THE SHEAR STRESSES

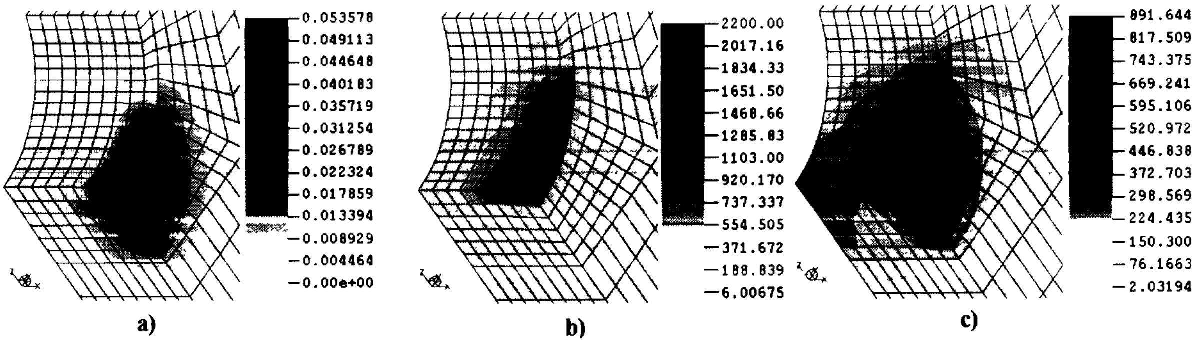
It is very difficult to measure the stress on the hardened layer boundary, and to determine the damage. We can overcome this difficulty with the use of a numeric model (FEM). It is used to determine the magnitude of deformations of the bodies in contact, and the stresses in the hardened layer of the raceway. This computation is done simultaneously with the experiment, which is used to check the mathematical model. For the determination of the carrying capacity of the contact we use the increase in the permanent deformation as well as the magnitude of the shear stress on the boundary of the hardened layer.

The FEM model considers all the geometric and load properties of the contact. The only supposition is a stiff ball. The results shown were obtained from the first load cycle (F_{max} = 50 kN, F_{min} = 1kN, f = 5 Hz).

Figure 7a and 7b shows an example where equivalent stress exceeds the plasticity limit -$\sigma_{0.2}$ in a relatively large area. We can expect that the damage (crack) will emerge on the boundary of the hardened layer and will grow towards the surface. The damage appears because the hardened layer is too thin. Figure 7c shows equivalent stresses after unloading (1kN). On the edge of the plastified region, i.e. on the edge of the contact surface is the greatest stress concentration, due to the permanent deformation of the raceway.

The comparison of computed deformations with the measured ones:

	$\delta_{computed}$ (mm)	$\delta_{measured}$ (mm)
Loaded (50 kN)	0.175	0.165
Unloaded (1kN)	0.052	0.034



a) plastic strain b) Total stress (N/mm^2) c) Total stress after unloading (N/mm^2)

Figure 7: Equivalent (Mises) shear stress and plastic strain

CONCLUSION

We would like to determine an experience based formula, considering all the material (E – elastic modulus, ν - Poisson ratio, $\sigma_{0.2}$ – elastic limit, HV – hardness according to Vickers, h – thickness of the hardened layer), and geometric data (d – diameter of the rolling element, S – ratio of sizes of the rolling element and the raceway, k – ratio of the contact ellipse semiaxes) for the computation of the maximal contact force and carrying capacity curve of a large dimension bearing (Figure 9).

$$Q_{kr,max} = f(E, d, s, k, HV, h, \sigma_{0.2}, \nu)$$

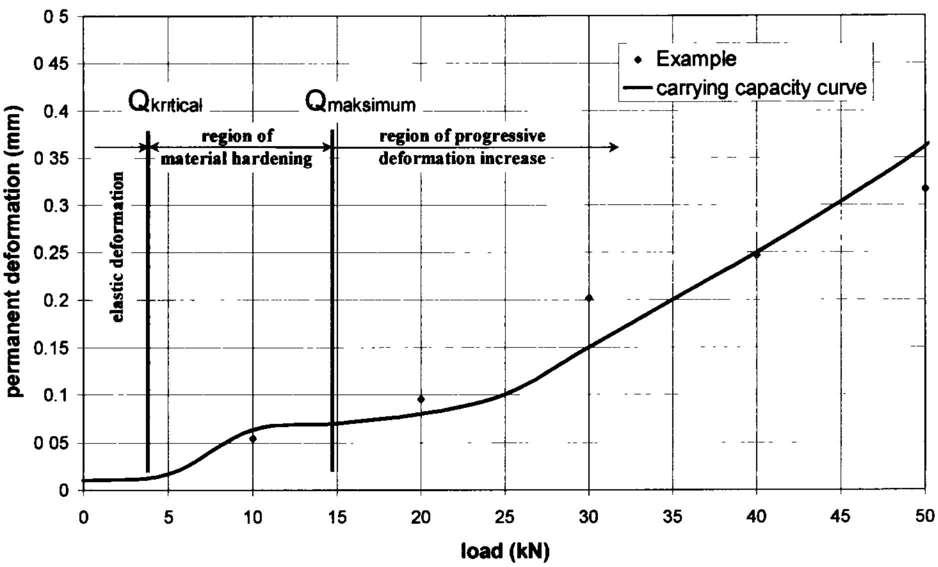


Figure 8: Carrying capacity curve (example)

The given methodology and results are the start of a larger measurement programme. It also confirms the suitability of the determination of the contact carrying capacity with the use of experimentally derived equations, augmented with the analytical computation of the subsurface stresses (FEM).

The determination of the contact carrying capacity is only one step towards the determination of the carrying capacity and lifetime of slowly running large dimension rolling bearings.

LITERATURE

[1] Hertz, H.: Über die Berührung fester elastischer Körper, J. für die reine und angewandte Mathematik, Leipzig, 1881, pp156-171.

[2] Thomas, H. R.; Hoersch, V. A.:Stresses due to the Pressure of one elastic Solid upon Another, University of Illinois, Bull.212, 1930.

[3] Lundberg, G.; Sjovall, H.:Stresses and Deformation in Elastic Contacts, Chalmers University of Technology, Gothenburg, 1958.

[4] Gupta, P. K.; Walowit, J. A.; Finkin, E. F.: Strees Distributions in Plain Strain Layered Elastic Solids Subjected to Arbitrary Boundary Loading, Trans. of ASME, J. of Lubrication Technology, pp427-433, October 1973.

[5] Eschman,P.; Hasbargen, L.; Weigand, K.: Ball and Roller Bearings, John Wiley & Sons Inc., New York, 1985.

[6] Harris, T. A.: Rolling Bearing Analysis - 3rd edition, John Wiley & Sons Inc., New York, 1991.

[7] Mathias, K.: Berechnung der Wälzkörperkrafte in Großwälzlagern, Fordern und Heben, 29, 1979.

[8] Brändlein, J.: Lastübertragung durch Großwälzlager bei elastischen Ringträgern als Unter und Oberkonstruktion, Fordern und Heben, 30, 1980

[9] Prebil,i.;Zupan,S.;Lučič,P.: Lastverteilung auf Wälzkörper von Drehverbindungen, Konstrukcion, Bund 47, Heft 11, 1995.

[10] Palmgren, A.: Ball and Rolling Bearing Engineering, 3rd Edition, Burbank, Philadelphia, 1959.

[11] ISO 76, Rolling bearings - Static load ratings, 1987.

[12] Yhland, E.: Static load carrying capacity, Ball Bearing Journal, Vol.211, 1982.

[13] Pallini, R. A.; Sague, J. E.: Computing Core-Yeild Limits for Case-Haredened Rolling Bearings, ASLE Trans., No.1, pp91-96, Vol.28, 1985.

[14] Buxbaum, O.: Betriebsfestigkeit, Verlag Stahleisen mbH, Düsseldorf, 1986.

[15] Ioannides, E.; Jacobson, B.; Tripp,, J. H.: Prediction of Rolling Bearing Life under Practical Operating Conditions, Tribological Design of Machine Elements, Tribology Series 14, Elsevier Science Publisher B.V., 1989.

[16] G rabec, I.; Prebil, I.: Postopek ugotavljanja naleganja stičnih površin, Patent št. 9200323, Ljubljana, 1994.