STRUCTURAL CALCULATION, LUBRICANT CONSUMPTION, AND FEM MODELING FOR 7211 BE AND NUP 211 EC BEARINGS

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ABSTRACT

In the machine-building industry, the reliability and performance of mechanical systems, including bearings, directly depend on the quality and efficiency of lubrication. This paper presents an evaluation of lubricant behavior under specific operating conditions, addressing the principles for lubricant consumption calculation and highlighting relevant methods for calculation, simulation, and testing applied to bearings. Additionally, lubricant selection criteria, lubrication systems, relubrication intervals, and the scheduling of lubricant use in industrial environments are discussed. Through FEM (Finite Element Method) modeling and analysis of bearings embedded in real mechanical assemblies, the effects of various types of lubricants on stress distribution, deformation, and long-term durability are investigated. The objective is to provide an applied perspective on how lubrication contributes to increasing component service life, reducing maintenance costs, and improving the energy performance of bearings.

KEYWORDS: lubrification, bearings, consumption, maintenance, simulation, FEM Analysis

1. INTRODUCTION

Bearings are machine components designed to support shafts and to guide rotational or oscillatory motion, while reducing friction between other components.

Rolling-element bearings are crucial components in modern mechatronic systems, providing support for rotary components under various loads [1].

Although the use of bearings in various functional constructions has a long history, there are still specific aspects to be analysed related to the bearings' strength and performance, such as contact strength, radial stiffness, and, implicitly, bearing life cycle [2-4].

An important factor that has significant influence in the bearing's lifecycle is the processes and tehnologies used to manufacture and control the components and final products.

A series of scientific studies have been conducted to investigate bearing manufacturing technologies [5,6], possibilities for optimizing bearing processing methods [7], production cost management [8], and FEM analysis of bearing behavior under various operating conditions [9,10].

In the machine-building industry, the use of appropriate lubricating fluids is essential to ensure the long-lasting and optimal operation of mechanical components, as well as to reduce associated costs by minimizing friction, heat generation, and wear between moving parts [11].

The proper use of synthetic lubricants [12-14] not only maximizes energy efficiency but also helps extend the operational lifespan of equipment, contributing to a reduction in maintenance interventions [15].

The classification and grading of lubricants are carried out according to numerous criteria (origin, chemical composition, physical properties, application, state of aggregation at ambient temperature, etc.) [16,17].

Examples of common lubricants include mineral oils, obtained from crude oil refining [18]; motor oils, often used in automobiles; synthetic lubricants, chemically manufactured to achieve superior properties, including long-lasting durability and performance under extreme temperature conditions [19]; semi-synthetic lubricants, defined as blends of mineral and synthetic oils [20]; and biodegradable lubricants, developed to reduce environmental impact - such as hydraulic oil [21] used in equipment operating in environmentally sensitive areas.

Engine lubricants [22] possess cleaning properties and reduce friction. Transmission lubricants [23] are essential for providing protection against damage, maintaining performance under high pressures, and ensuring stability at high temperatures. Industrial lubricants [24] are used for equipment such as compressors, pumps, and machine tools, and must guarantee outstanding protection against wear and corrosion.

According to international standards, *SAE* (Society of Automotive Engineers), the lubricants are classified basedon viscosity [25]. *API* (American Petroleum Institute) establishes a classification system for automotive lubricants according to performance, specifying whether they are designed for gasoline or diesel engines (e.g., *API SN, API CF*) [26].

ACEA (Association des Constructeurs Européens d'Automobiles) [27] defines European standards that classify oils according to the requirements of modern engines (*A/B* for gasoline/diesel engines, *C* for engines with particulate filters, and *E* for heavy engines) [28].

A classification of mineral oils (ISO 6743 or ASTM) is also established in Romania. Mineral oils are designated by one or more capital letters, depending on the field of application (*M, AVI, T, LDE, TIN, L, G, Te, Tb, K, F, H, Tr,* etc.), followed by numbers indicating the viscosity grade according to SAE standards (American) for engine and automotive transmission oils. For other types of oils, the numbers represent the kinematic viscosity (*CST*) of the oil at 40°C or 100°C.

Greases are designated according to their main fields of use: general-purpose greases, *U*; bearing greases, *RUL*; open bearing greases, *LD*; extreme-pressure greases, *EP*; and multipurpose greases with mixed barium–aluminum soap dispersed in oil.

The lubricating properties of oils and greases can be defined by two simultaneous effects: a bulk effect and a surface effect. The bulk (physical) effect characterizes the load capacity of the lubricant film and is based on its viscosity index. The surface effect characterizes the adhesion of lubricant molecules to the friction surfaces (lubricant smoothness).

The friction in a lubricated kinematic pair is based on the interaction of the metal–lubricant–metal system (friction triad). However, the effectiveness of lubrication depends on a number of physical, chemical, geometric, and kinematic factors, determined by the geometry of the contacting bodies, the operating environment (chemical action, temperature), and the kinematic (speed) and dynamic (load) parameters of the mechanism.

According to ISO 15243:2004, any pair of materials in contact constitutes a tribosystem and is composed of a basic triboelement, a moving triboelement (sliding or rolling), an interposing material, and the operating environment. The parameters of the tribosystem change over time (dimensions, tolerances, fits, amount of heat generated, vibration levels).

If the relative motion between the elements of the rolling tribosystem is taken as a basis, then bearings fall into the category of free-rolling elements.

In the case of ball bearings, Hertzian contact theory is commonly used to calculate contact pressure and radial stiffness [29]. In the case of radial-roller bearings, due to the presence of edge loads between the two contact surfaces, the contact problem is classified as a non-Hertzian contact, and Hertzian theory cannot be applied to radial-roller bearing contact analysis. To address the contact problem in radial-roller bearings, an approximate contact model considering a roller in contact with an infinitely long surface has been developed [30].

The integration and use of FEM methods in the study of products such as bearings constitute an important tool in the design phase. They help helpi predict bearing behavior based on the characteristics of the materials from which they are manufactured, and assist in estimating the operating conditions in practical applications, including the type of lubricants to be used.

An important advantage of using FEM tools by manufacturers and end-users is the impact of FEM analysis results on the management of assembly and maintenance activities, as well as on the budgeting of resources required for maintenance processes.

The present paper proposes an integrated approach to the analysis of two types of bearings - a 7211 type radial ball bearing and a NUP 211 type radial-roller bearing.

The proposed approach involves integrating the following characteristics: the operating conditions in the application (temperatures, lubrication, stationary and rotating elements), the theoretical calculation of the durability characteristics of the analyzed bearings, and the FEM analysis of the product behavior in two operating scenarios - one in which the inner ring is fixed and one in which the outer ring is fixed.

An integrated approach prior to the decision to purchase or manufacture a bearing can help ensure the safe use of roller and ball bearings over a defined period.

2. CONSTRUCTIVE-FUNCTIONAL ANALYSIS OF 7211 BE AND NUP 211EC BEARINGS

In general, a bearing is composed of an inner ring and an outer ring, with raceways; rolling elements: balls, cylindrical rollers, needles, tapered rollers, spherical rollers, toroidal rollers and one or two cages or semi-cages, which ensure the positioning of rolling elements; sealing elements: made of elastomer, with or without contact; protective elements, without contact, Figure 1.

Fig. 1. *Radial-ball bearings - components* [31]

Depending on the type of movement, bearings can be with rolling or sliding elements, and depending on the direction of the load, they can be radial (loads perpendicular to the axis), axial (loads parallel to the axis), or combined (radial-axial, axial-radial), Figure 2.
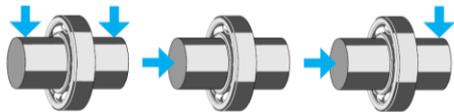

Fig. 2. *Load direction* [31]

The following functional terms are used in the design and manufacture of bearings: contact angle, radial/axial clearance, preload, dynamic and static load capacity, nominal life, speed limit, friction torque, and rigidity. The choice of bearing type depends on the load, speed, required precision, lubrication conditions, and working environment.

According to the type of rolling elements, bearings are classified into: ball bearings and roller bearings (cylindrical, tapered, barrel, needle), Figure 3. According to the number of rows, bearings with one row or more rows of rolling elements.
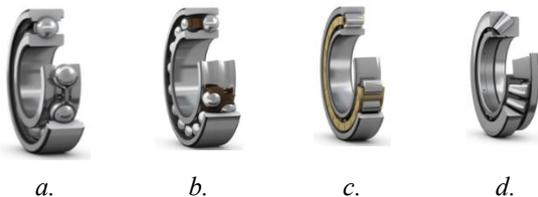

a.   b.   c.   d.

Fig. 3. *Types of bearings: radial-ball bearing (a); radial-axial bearing with balls (b); radial-rooler bearing with cyllindrical rollers (c); spherical bearings (d)* [31]

2.1. Bearings symbolisation

Radial-ball bearings are used for high speeds and moderate loads, while radial-roller bearings are used for heavy loads. Thrust bearings are used to support thrust loads.

The symbolization of radial ball bearings, Figure 4, involves, in summary, specifying the material; bearing type; dimensional elements - outer diameter, width, bore diameter; internal and external design solutions; cage type; clearance, precision, stabilization, and lubricant.
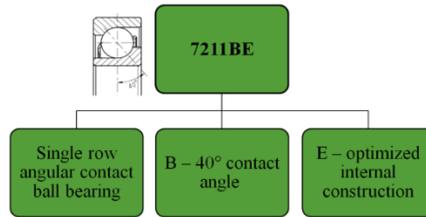

Fig. 4. *Explanation of 7211 BE symbol*

The symbolization of radial-roller bearings, Figure 5, involves specifying the basic elements: design variant (*NU*; *NJ*; *NUP*; *N*; *NU+HJ*; *NJ+HJ*); size; internal construction (*A* – modified internal construction; *EC* – optimized internal construction); cage type (*P* - polyamide; *J* - steel; *M* - brass); internal clearance (*C2*; *C3*; *C4*; *C5*).
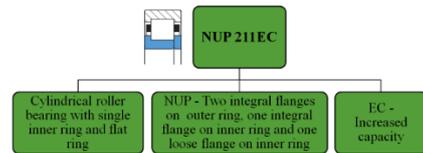

Fig. 5. *Explanation of NUP 211EC symbol* [31]

Each bearing is characterized by a series of technical and geometric parameters obtained through mechanical processing, which ensure the assembly of components and the achievement of clearance in accordance with technical specifications, and subsequently the normal functioning of finished products within the equipment or technical systems in which they are installed.

The technical and geometric characteristics of the products analyzed are presented in Table 1.

Table 1. *Technical and geometrical characteristics for 7211 BE and NUP 211EC bearings*

Characteristics name	Product name	
	7211BE	**NUP 211EC**
Type off rolling elements	Balls	Cyllindricall rollers
Inner diameter [mm]	55	55
Outer diameter [mm]	100	100
Bearing width [mm]	21	21
Number of rows	1	1
Raceway roughness - outer ring [μm]	Max 0.2	Max 0.2
Raceway roughness - inner ring [μm]	Max 0.2	Max 0.2

2.2. Bearings calculation

2.2.1. Calculation of bearing 7211 BE

The organological calculation of the bearings was performed using the following input data: radial load $F_r=5000$ N, axial load $F_a=2000$ N, speed $n=1500$ rot/min, dynamic load capacity $C=52.5$ kN, and static load capacity $C_0=30.5$ kN.

Typical values of coefficients (for $40°$ contact):

$$\frac{F_a}{F_r} = \frac{2000}{5000} = 0.4; \qquad (1)$$

$$X = 0.57; \ Y = 0.93.$$

Equivalent static load:

$$P = X \cdot F_r + Y \cdot F_a \ [\text{N}]; \qquad (2)$$

$$P = 0.57 \cdot 5000 + 0.93 \cdot 2000 = 4710 \ \text{N}.$$

Bearing rating life:

$$L_{10} = \left(\frac{C}{P}\right)^3 \ [\text{mil. rot.}]; \qquad (3)$$

$$L_{10} = \left(\frac{52500}{4710}\right)^3 = 1381.6 \ \text{mil. rot.}$$

Bearing rating life in functioning hours:

$$L_{10h} = \frac{L_{10} \cdot 10^6}{60 \cdot n} \ [\text{hours}]; \qquad (4)$$

$$L_{10h} = \frac{1381.6 \cdot 10^6}{60 \cdot 1500} = 15351 \ \text{hours}.$$

The data was centralized in Table 2.

Table 2. *Calculation results for bearing 7211 BE*

Parameter	Value
Equivalent static load, P [N]	4710
Bearing rating life, L_{10} [mil. rot.]	1381.6
Bearing rating life in functioning hours, L_{10h} [hours]	15351

If correction factors for actual lubrication conditions, cleanliness, and loads are taken into account, the modified rating life, L_{nm}, can be calculated according to ISO 281.

The modified service life is calculated using the following formula:

$$L_{nm} = a_1 \cdot a \cdot \left(\frac{C}{P}\right)^p \ [\text{hours}], \qquad (5)$$

where:
- a_1 is the reliability factor, Table 3;
- a - correction factor for actual operating conditions;
- C - dynamic load capacity;
- P - equivalent dynamic load;
- $p = 3$ for ball bearings.

Table 3. *Fiability factor, a_1*

Fiability [%]	a_1
90	1.00
95	0.62
99	0.21

Factor a – actual conditions, takes into account correct lubrication, degree of lubricant purity, material, and quality.

If it is assumed that lubrication is good, the environment is clean, and the stresses are below the fatigue limit, then $a≈2.5$ (this value can vary between 1 and 50 depending on working conditions).

The calculation of the modified rating life is:

$$L_{nm} = 1 \cdot 2.5 \cdot \left(\frac{52500}{4710}\right)^3 = 3454 \ \text{hours};$$

$$L_{nmh} = \frac{3454 \cdot 10^6}{60 \cdot 1500} = 38378 \ \text{hours}.$$

The results were centralized in Table 4.

Table 4. *Final results for bearing 7211 BE*

Parameter	Value
Equivalent static load, P [N]	4710
Nominal rating life, L_{10h} [hours]	15351
Modified rating life, L_{nmh} [hours]	38378
Correction factor, a	2.5

2.2.2. Calculation of bearing NUP 211 EC

In the case of bearing NUP 211EC in the manufacturers catalogues, the following input data was identified: dynamic load capacity, $C=75.5$ kN, static load capacity, $C_0=75$ kN, and the load under which fatigue does not occur, P_u, which is approximately 8.5 kN.

The equivalent dynamic load for radial cylindrical roller bearings (which do not support significant axial loads) is:

$$P \cong F_r = 5000 \ \text{N}. \qquad (6)$$

F_a is ignored if it is small and does not significantly affect the behavior of the bearing.

The *bearing rating life*, L_{10}, is calculated using Equation (3):

$$L_{10} = \left(\frac{75500}{5000}\right)^{\frac{10}{3}} = 4306 \ \text{mil. rot.}$$

The *bearing rating life in functioning hours* is calculated using Equation (4):

$$L_{10h} = \frac{4306 \cdot 10^6}{60 \cdot 1500} = 47844 \ \text{hours};$$

If correction factors for actual lubrication conditions, cleanliness, and loads are taken into account, the modified rating life is calculated according to ISO 281 with the relation (5):

$$L_{nm} = 1 \cdot 3 \cdot (15.1)^3 = 12912 \text{ mil. rot.}$$

The calculation of the modified rating life is:

$$L_{nh} = \frac{12918 \cdot 10^6}{60 \cdot 1500} = 143530 \text{ hours.}$$

The results were centralized in Table 5.

Table 5. *Calculation results for bearing NUP 211 EC*

Parameter	Value
Equivalent static load, P [N]	5000
Nominal rating life, L_{10h} [hours]	47844
Modified rating life, L_{nmh} [hours]	143530
Correction factor, a	3

The comparative analysis shows that roller bearings have a significantly longer rating life due to the fact that the rollers provide a larger contact surface than the balls (linear contact compared to point contact), reducing contact pressure.

The comparative results are centralized in Table 6.

Table 6. *Comparative results 7211 BE/ NUP 211 EC*

Bearing	Nominal rating life [hours]	Modified rating life [hours]
7211 BE	15351	38378
NUP 211 EC	47844	143530

2.3. Bearings modeling

Solid modeling of the bearings was performed using the bearing calculation and generation module in *Autodesk Inventor* software, which was also used to perform their organological calculation.

The two bearings were generated. The following figures and tables show the input data and the results of the organological analysis for the two bearings.

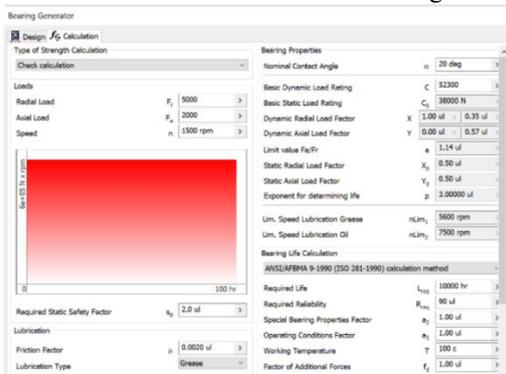Figure 6 and Table 7 show the characteristics and results obtained for the 7211 BE ball bearing.

Fig. 6. *Input data for bearing 7211 BE*

Table 7. *Results for bearing 7211 BE*

Parameter	Value
Basic rating life, L_{10} [hours]	12716
Adjusted rating life, L_{na} [hours]	12716
Calculated static safety factor, s_{0c} [ul]	7.600
Power lost by friction, P_z [W]	43.197
Necessary minimum load, F_{min} [N]	380
Static equivalent load, P_0 [N]	5000
Dinamic equivalent load, P [N]	5000
Over-revolving factor, k_n [ul]	3.733
Life adjustment factor for reliability, a_1 [ul]	1.00
Temperature factor, f_t [ul]	1.00
Equivalent speed, n_e [rpm]	1500
Minimum speed, n_{min} [rpm]	1500
Maximum speed, n_{max} [rpm]	1500
Strength check	Positive

Figure 7 and Table 8 show the input data necessary for generating the analyzed roller bearing and the results obtained from the organological calculation for the NUP 211EC bearing.
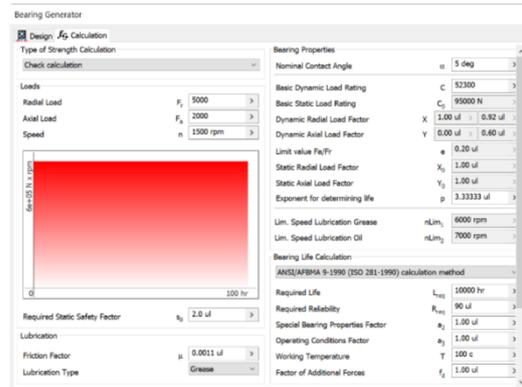

Fig. 7. *Input data for bearing NUP 211EC*

Table 8. *Results for bearing NUP 211EC*

Parameter	Value
Basic rating life, L_{10} [hours]	16956
Adjusted rating life, L_{na} [hours]	16956
Calculated static safety factor, s_{0c} [ul]	13.571
Power lost by friction, P_z [W]	27.560
Necessary minimum load, F_{min} [N]	1900
Static equivalent load, P_0 [N]	7000
Dinamic equivalent load, P [N]	5800
Over-revolving factor, k_n [ul]	4.000
Life adjustment factor for reliability, a_1 [ul]	1.00
Temperature factor, f_t [ul]	1.00
Equivalent speed, n_e [rpm]	1500
Minimum speed, n_{min} [rpm]	1500
Maximum speed, n_{max} [rpm]	1500
Strength check	Positive

2.4. Principles for calculating lubricant consumption in rolling tribosystems

One of the main factors to be considered for bearing usage is wnsuring enough lubricant. The lubrication of bearings can be done manually, by dripping, with a lubricator, or in an oil bath, and the lubricant consumption is determined according to the lubrication system and the size of the bearing, and is given as a guide in Tables 9 and 10.

Table 9. *Oil consumption for bearings - manual / drip feed (D - spindle diameter)*

D [mm]	Oil consumption in 8 hours, considering the bearing series [g]				
	200	300	1500	1600	2400
20	1.70	1.80	2.20	2.30	2.50
25	2.30	2.60	2.70	3.10	3.60
30	2.90	3.40	3.70	4.10	5.90
35	3.50	4.50	5	5.50	6.50
40	4.50	5.50	6	6.50	8
45	5	6	6.50	8	10
50	6	7	7.50	9	12
55	7	8	8.50	10	14
60	8	9	10	13	16
65	9	11	12	15	19
70	10	12	13	18	21
75	11	13	15	21	25
80	12	14	16	23	28
85	14	16	18	25	31
90	16	20	22	27	35
95	18	22	25	28	38
100	20	26	28	36	44
110	25	30	35	40	53

Table 10. *Approximate capacities of lubrication baths and filling requirements for 8 hours (D - spindle diameter; l - bath capacity; g - lubricant consumption)*

D [mm]	Bearing serie					
	200 - 1200		300 - 1300		400	
	l	g	l	g	l	g
20	0.01	0.80	0.02	0.90	0.03	1.20
25	0.01	0.80	0.02	0.90	0.04	1.50
30	0.02	0.90	0.03	1.20	0.05	2
35	0.03	1.20	0.04	1.50	0.06	2.70
40	0.03	1.20	0.05	2	0.08	3.20
45	0.04	1.50	0.07	3	0.10	4
50	0.05	2	0.08	3.20	0.12	4.50
55	0.06	2.70	0.09	3.60	0.13	5
60	0.09	3.50	0.13	5.50	0.19	6.50
65	0.10	4	0.15	6	0.21	7.50
70	0.11	4.20	0.19	6.50	0.3	10
75	0.13	5	0.22	7.50	0.33	10.50
80	0.15	6	0.25	8.50	0.37	11.50
85	0.20	7	0.33	10.50	0.48	13.50
90	0.23	8	0.36	11	0.55	15
95	0.26	9	0.40	13	0.63	16
100	0.29	10	0.47	14	0.68	17
110	0.39	12	0.64	16	0.93	21
120	0.46	13	0.74	20	1.14	26
130	0.49	14	0.86	21	1.38	30
140	0.60	16	0.99	26	1.54	34

In the bearings' lubrication process, liquid lubricants (mineral or synthetic oils), consistent lubricants (greases), or solid lubricants (graphite, molybdenum disulfide, teflon) are used. The lubricant is selected based on an analysis of both the operating conditions and the properties of the lubricant, as there are no universal lubrication systems. When choosing a lubricant and determining re-lubrication intervals, the following factors must be taken into account: bearing size; bearing load; bearing temperature, and actual operating conditions (humidity, saline environment, impurities). When lubricating with thick greases, the free space in the bearing is completely filled with grease, and the bearing housing is filled to approximately 1/2...3/4 of the free space. Guidelines on grease consumption for bearing lubrication are given in Table 11.

Table 11. *Grease consumption (D - spindle diameter [mm]; g - lubricant consumption)*

Barings characteristics					
D [mm]	with 1 and 2 rows	Thrust bearings	D [mm]	with 1 and 2 rows	Thrust bearings
10	40	20	60	192	132
12	48	24	65	208	140
15	60	28	70	224	148
17	68	32	75	240	156
20	80	40	80	256	166
25	100	48	85	272	176
30	108	62	90	288	189
35	128	70	95	304	205
40	144	86	100	320	226
45	160	98	105	336	256
50	176	111	110	352	285
55	184	123	120	364	315

In small bearings, especially those with a single row of rolling elements and a deep raceway, the grease is consumed quite slowly, so that no additional lubricant is required. In the case of medium and large bearings, relubrication is necessary at certain intervals. When relubricating, the bearing is disassembled, washed, dried, and only then relubricated (to avoid mixing of greases).

When the grease used for the previous lubrication is known, the consumption is calculated using the following equation:

$$Q = 0.005 \cdot D \cdot B, \qquad (7)$$

where D is the outer diameter of the bearing [mm] and B - width of the inner ring of the bearing [mm].

Table 12 provides examples of lubricant selection and re-lubrication intervals, taking into account the influencing factors mentioned above.

Table 12. *Examples of lubricant selection and re-lubrication intervals*

Place of usage	Functioning status						Lubrication interval
	D	n	F	t	grease	oil	
Vehicles							
Wheel	2	1	1	1	2a	-	30000 - 50000 km
Crankshaft	2	1	1	2	-	1b.2	15000 - 20000
Differentials	2	1	1	3	-	2.3	15000 - 20000 km
Gearboxes	2	1	2	2	-	2.3	15000 - 20000 km
Clutch	3	2	1	3	2b, 2c, 3	-	No re-lubrication
Electric motors							
Domestic motors	1	1	1	1	2a		No re-lubrication
Medium size	2-3	1	1	1	2a	-	1000-2000 hours
Large size	4	1	1	2	3		500 - 1000 hours
Engine traction	3	2	1	3	2c,3		200000 - 500000 km
Agricultural machinery							
Soil machinery	2-3	1	2	1	1a,2b,3	-	1000 hours
Food choppers	2	1	1	1	1a,2b,3		5000 hours

The main factors considered in the selection of lubricants and the symbols assigned in Table 12 are:

a. bearing size (outer diameter) - size D, in mm;
- for bearings with $D \leq 22$ mm;
- for bearings with $22 < D \leq 62$ mm;
- for bearings with $62 < D \leq 240$ mm;
- for bearings with $D > 240$ mm.

b. bearing rotation speed - n, in rpm:
- for $n \leq 0.8 \, n_{lim}$;
- for $n > 0.8 \, n_{lim}$, where n_{lim} represents the bearing rotation speed limit given in the producers' catalogues.

The numbers indicate the ranges for factors that meet maintenance requirements and influence the choice of lubricant. Conventional symbols have also been established for lubricants: *oils*, depending on their viscosity and operating temperatures, Table 13; *greases*, depending on their operating temperature range and thickening agent (soap), Table 14.

Table 13. *Conventional symbols used for oil*

Symbol	Freezing temperature [°C]	Recommended viscosity [m²/s]
1a	$t < 50°$	$(16-37)\cdot 10^{-6}$
1b	$t < 50°$	$(11.8-60)\cdot 10^{-6}$
2	$50° \leq t < 80°$	$(37-75.8)\cdot 10^{-6}$
3	$80° \leq t < 120°$	$> 75.8\cdot 10^{-6}$
4	$120° \leq t < 150°$	$= 227.4\cdot 10^{-6}$

Table 14. *Conventional symbols used for grease*

Symbol	Operating temperature range	Grease lubricant	Water resistance
1a	-35……+50°	Calcium	Water repellent
1b	-35……+50°	Calcium	Water repellent
2a	-35……+80°	Sodium	Unstable
2b	-35…..+120°	Sodium	Unstable
2c	-35…..+120°	Sodium	Unstable
3	-35…..+110°	Lithium	Stable up to 90°
4a	<60°	Calcium	For sealing
4b	<110°	Sodium	For sealing

3. FEM ANALYSIS OF BEARINGS 7211 BE AND NUP 211EC

The FEM analysis of bearings 7211 BE and NUP211EC was performed in ANSYS in the following two scenarios: *scenario 1* – the inner ring is rotating and the outer ring is fixed; *scenario 2* – the inner ring is fixed and the outer ring is rotating.

The following steps were performed for the FEM analysis:
a) identification of the type and dimensions of the products to be analyzed (7211 BE – radial ball bearing and NUP 211EC – radial-roller bearing);
b) calculation of bearings;
c) creation of material for component elements;
d) imposition of boundary conditions;
e) discretization of component elements;
f) analysis of results regarding the distribution and values of Von Mises stresses.

3.1. Material creation in ANSYS

In bearing manufacturing specific materials are used. Steel type 100Cr6 / AISI 52100 is used to manufacture the analysed bearing. The material was created in ANSYS. 100Cr6 steel is a through-hardening bearing steel intended for rolling contact and other high-fatigue applications [32].

Mechanical properties were identified on one supplier, Table 15 [32]. Chemical composition for a sample was performed using a spectrometer at a bearing manufacturer in Romania, Table 16 [33].

Table 15. *Mechanical properties of 100Cr6 / AISI 52100*

Parameters	Values
Elasticity modulus, E [GPa]	210 - 215
Poisson ratio	0.3
Density [kg/m³]	7810 - 7850
Tensile strength [MPa]	1900 - 2200 (quencing and tempering)
Hardness	61 - 64 HRC

Table 16. *Chemical composition of 100Cr6 / AISI 52100*

Chemical element	Request percentage	Measured percentage
C	0.93 - 1.05	0.95
Si	0.15 - 0.35	0.20
Cr	1.35 - 1.60	1.49
Mn	0.25 - 0.45	0.26
Ni	0.00 - 0.25	0.03

3.2. FEM analysis of 7211 BE bearing

3.2.1. Identification of boundary conditions and discretization of 7211 BE bearing

The FEM analysis of the 7211 BE ball bearing considered the two mentioned scenarios, Figure 8: the first in which the inner ring is rotating and the outer ring is fixed, and the second in which the inner ring is fixed and the outer ring is rotating.

Therefore, the conditions were set taking into account these two scenarios.

a.

b.
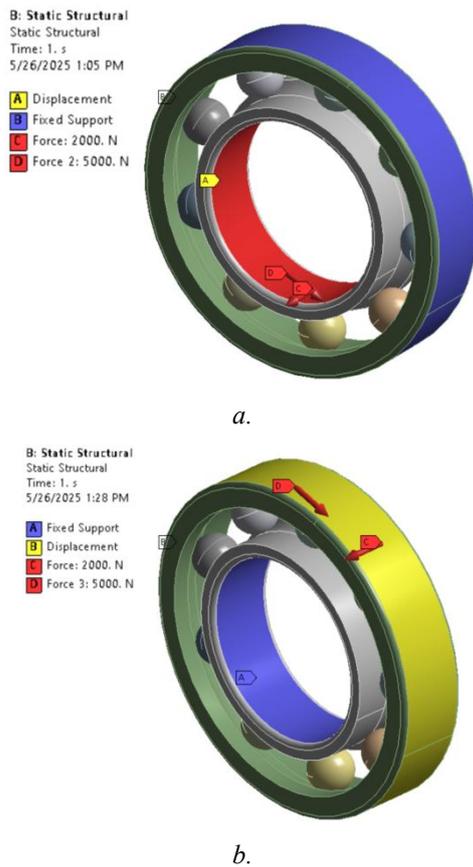
Fig. 8. *Boundary conditions: a. inner ring rotating; b. outer ring rotating*

The discretization of the assembly was performed by establishing different values for the bearing rings and balls, which was discretized into 167.684 nodes and 56.556 elements, Figure 9.
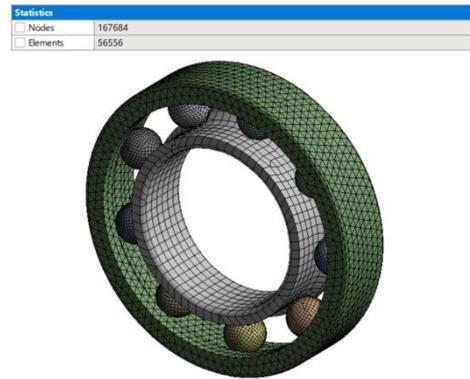
Fig. 9. *Discretization of 7211 BE bearing*

3.2.2. FEM results for 7211 BE bearing

Following the FEM analysis of the ball bearing, it can be observed that the stress distribution depends significantly on the rotating part.

When the outer ring is rotating, the Von Mises stresses are approximately 50% higher than in the case where the inner rings rotates. This is due to the more complex contact conditions between the balls and the raceways generated by the rotating outer ring.

The maximum Von Mises stress value of 5.877 MPa, Figure 10, is below the elastic limit of the selected steel, which shows that the bearing operates in a safe elastic regime in both cases.
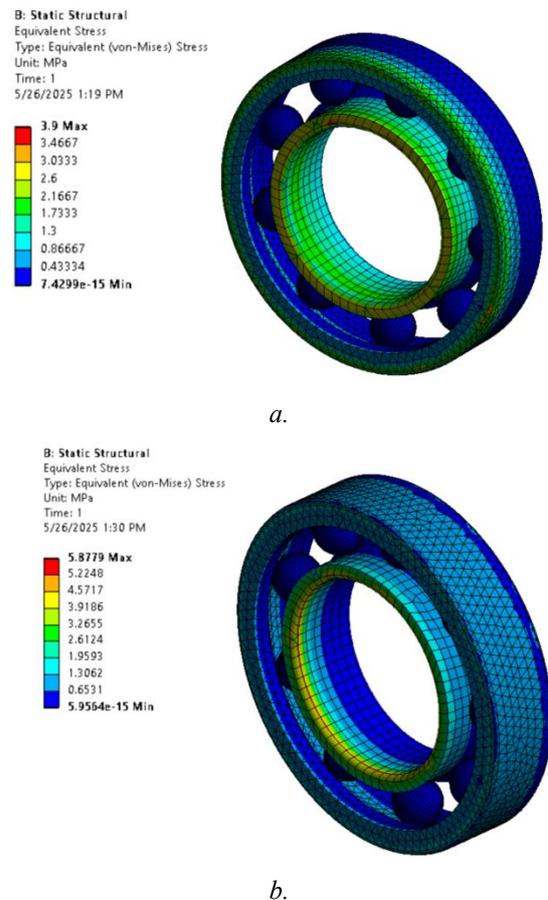

a.

b.

Fig. 10. *Von Mises stress in case of 7211 BE bearing: a. inner ring rotating; b. outer ring rotating*

In terms of strains, Figure 11, it can be observed that these are smaller when the outer ring is rotating, which suggests greater rigidity of the assembly in this case, which is important for precision applications.
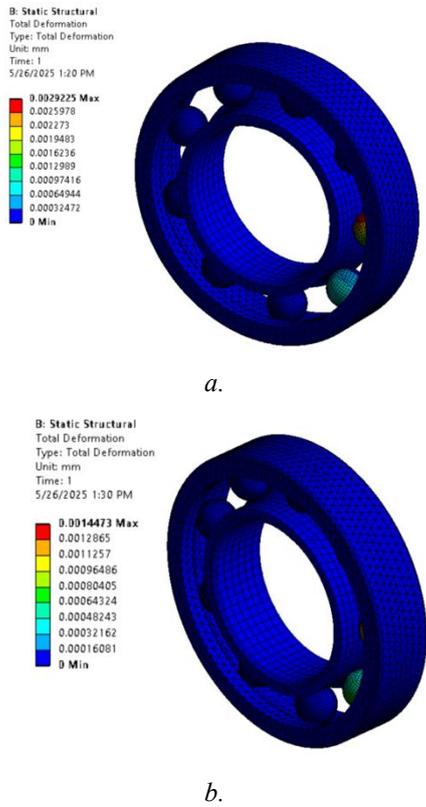

a.

b.

Fig. 11. *Strain distribution in case of 7211 BE bearing: a. inner ring rotating; b. outer ring rotating*

The differences observed in the two scenarios highlight the importance of correct assembly and of the way the bearing is integrated into the system, including the identification of the the fixed component and rotating components. These factors influence both the internal stresses and the deformation behaviour, which in turn affect the durability and dynamic performance of the assembly.

3.3. FEM analysis of NUP 211EC bearing

The FEM analysis of the roller bearing was also performed using *Ansys Workbench*, following the same steps as for the ball bearing. The same material was used for both the bearing rings and the rollers.

3.3.1. Identification of boundary conditions and discretization of NUP 211EC bearing

The FEM analysis of the NUP 211EC bearing was performed using the same two scenarios: the first in which the inner ring is rotating, and the outer ring is fixed, and the second in which the inner ring is fixed, and the outer ring is rotating, the conditions imposed being those shown in Figure 12.
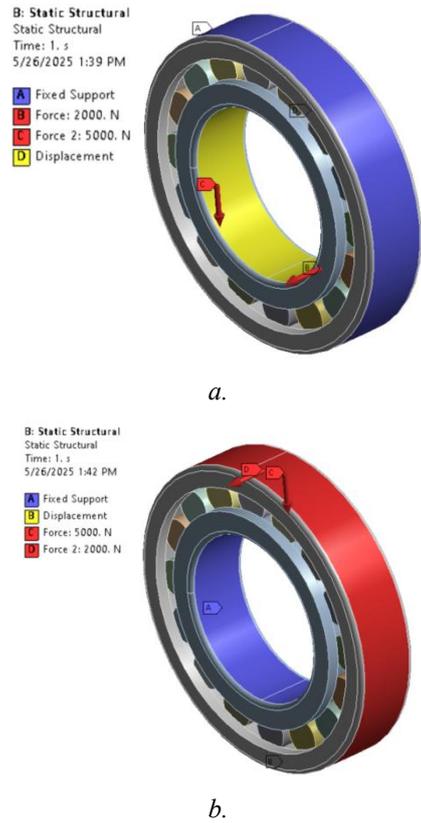

a.

b.

Fig. 12. *Boundary conditions for NUP 211EC bearing: a. inner ring rotating; b. outer ring rotating*

The discretization of the assembly was performed by imposing different discretization values for the bearing rings and rollers in order to obtain the most accurate results possible. The ring-roller assembly was discretized into 20.285 nodes and 5.683 elements, Figure 13.
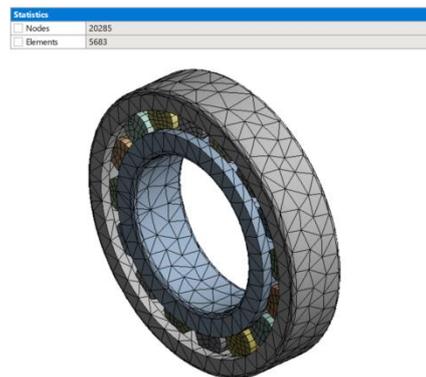

Fig. 13. *Discretization of NUP 211EC bearing*

3.3.2. FEM results for bearing NUP 211EC

The Von Mises stresses are almost identical in both scenarios, with a difference of only 0.35% between the two cases, indicating a similar distribution of forces in both configurations, which may be due to the internal symmetry of the NUP bearing and the linear contact specific to cylindrical rollers.

The stresses are below the limit of the chosen material, with the maximum stress of 13.8 MPa being well below the elastic limit of the steel chosen for the roller bearing, indicating a completely elastic and safe regime, which guarantees reliable operation in static conditions.
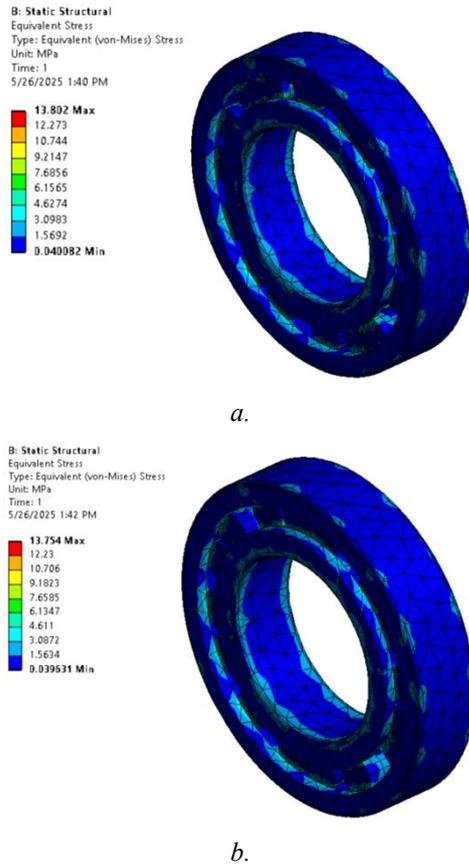

a.

b.

Fig. 14. *Von Mises stresses for NUP 211EC bearing: a. inner ring rotating; b. outer ring rotating*

The deformations obtained, Figure 15, can be considered equal, with a difference of only 0.0001 mm, leading to the conclusion that the assembly behaves almost identically regardless of which part rotates, the inner or outer ring, with the bearing exhibiting very good rigidity.
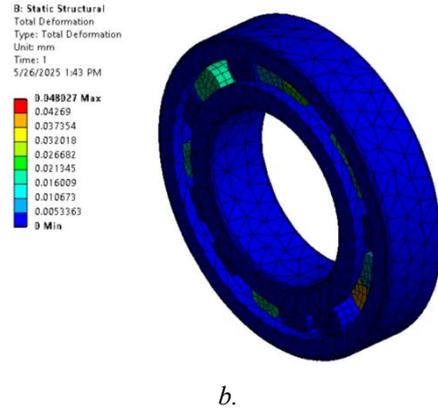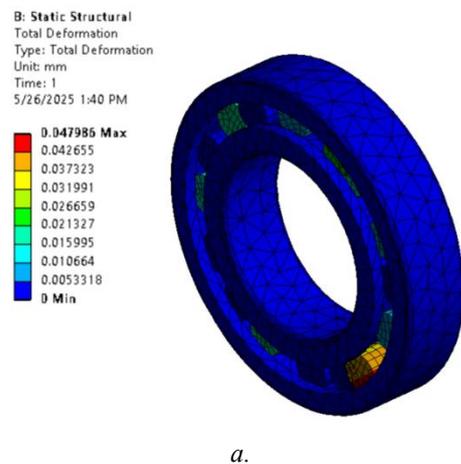

a.

b.

Fig. 15. *Strains distribution in case of NUP 211EC bearing: a. inner ring rotating; b. outer ring rotating*

4. CONCLUSIONS

The present paper developed an integrated approach for the calculation process for bearings, taking into account the identification of working conditions (temperature and lubricants), theoretical product calculation, and FEM analysis. The comparative FEM analysis of the behavior of two bearings in two working scenarios led to the identification of safe elastic behavior in both cases, with Von Mises stresses remaining well below the elastic limit of the material for both bearings, indicating safe operation in static conditions.

The roller bearing is more rigid and less sensitive to the mounting configuration. The differences between the two scenarios, inner rotating ring and outer rotating ring, are negligible, making it suitable for applications where structural stability and predictability are essential. The ball bearing recorded lower Von Mises stresses, but is more sensitive to configuration; when the inner ring is rotating, the deformation is much greater compared to when the outer ring is rotating, making it more suitable for applications where axial precision is important and where axial loads are significant.

The choice between the 7211 BE ball bearing and the NUP 211 ECP roller bearing should be made based on the nature of the load and the rigidity requirements. If axial precision is required, the ball bearing is more suitable, and if high rigidity and insensitivity to mounting are required, the roller bearing is superior. FEM analysis is also an important tool in terms of the sustainable approach to a company's processes, as predicting how a product will behave in a system or piece of equipment helps to determine the material, financial, and human resources needed to keep that system or piece of equipment in working order. The deformations obtained can be considered equal, with a difference of only 0.0001 mm, leading to the conclusion that the assembly behaves almost identically regardless of which part rotates, the inner or outer ring, with the bearing exhibiting very good rigidity.

The Von Mises stresses are almost identical in both scenarios, with a difference of only 0.35% between the two cases, indicating a similar distribution of forces in both configurations, which may be due to the internal symmetry of the NUP bearing and the linear contact specific to cylindrical rollers.

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