Analysis of Static Load Bearing Capacity of the Ball Bearing by using ANASYS

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Abstract— The study was conducted in order to generate a systematic and more accurate system that can be used to determine the static load bearing capacity of the ball bearing without destructive test of the costly bearing which can help to reduce the cost time and efforts required. The findings of the research were that the deviation between the static load bearing capacity of a ball bearing obtained by finite element soft wares and the result that were obtained by the practical test that were conducted on Universal testing machine of the Laboratory has a linear relation with size of bearing under test. When the relation between this deviation and the bearing size was established using statistical tool of Linear Interpolation the following relation were obtained.

Key words: Asymmetric Gear; Bending Strength; Accurate Stress; Safety Factor; Deformation

I. INTRODUCTION

The gearbox is continuously being developed for future demands of increased loads, longer lifetime and reductions of noise and weight. To do this in an effective way it is necessary to evaluate designs virtually, e.g. with CAE methodologies such as FE-analysis. A useful FE-analysis requires a good knowledge of the loads that the gearbox will be subjected to during operation. It is also necessary to be able to correctly model how these loads affect the structure of the gearbox.

Bearings are key components in this aspect as they transmit forces between the components in the gearbox, e.g. gear wheels, shafts and housings. Using full FE-models of the bearings with contacts and solid rollers is not feasible, from the view of computational cost, when modeling a significant part of the gearbox with several bearings. Thus it is necessary to have simplified bearing models. The demand on a bearing model varies with the position of the bearing relative the area of interest in the analysis. If the position of the bearing is not close to the area of interest it is sufficient that the bearing model acts as a pivot point and allow for the correct total force to be transmitted. This requires that the model provides the correct overall stiffness and correct forces.

When the position of the bearing is close to the area of interest, then the detailed deformation and force distribution in the bearing will affect the critical area. This yields different and significantly more complex requirements on the bearing model than for the previous case.

Having bearing models of suitable complexity that is numerically stable and thoroughly tested and verified both decreases the computational time and ensure that the results are adequate. If they can be created automatically from a limited number of parameters then also the time for setting up models of gearboxes can be decreased. The intention with this thesis work is to develop bearing models with different degree of complexity and computational cost. This gives a possibility to choose the proper model with respect to the purpose of the analysis. The effect of hoop stresses on the rolling-element fatigue life of angular-contact and deep-groove ball bearings was determined for common inner-ring interference fits at the ABEC–5 tolerance level [1].

This paper determines the method to find out the shoulder height in an angular contact ball bearing using a 3D contact analysis. The load analysis was performed by calculating the distributions of internal loads and contact angles for each rolling element. The effects of axial load on contact pressure at the inner and outer raceways were evaluated and the critical axial loads in the present shoulder height were calculated. The results showed that the initial shoulder heights of the example bearing were designed to be excessively high, and the critical shoulder heights were presented under the load condition of practical use. The proposed methodology is generally applicable for the purpose of reducing the material cost and improving the efficiency of the bearing design process [2].

The numerical approach is based on the coupling between a semi-analytical code and a finite element (FE) model which computes the deformation of the rings and housings. The experimental setup allows investigation of the load-displacement behavior of a four-point contact ball bearing submitted to a static thrust loading [3].

II. BEARING FAILURE

Bearings are typically rated based on the life, stated in revolutions (or in hours of operation at the design speed), that 90% of a random sample of bearings of that size can be expected to reach or exceed at their design load. In other words, 10% of the batch can be expected to fail at that load before the design life is reached.

There are two basic types of bearing failure-breakage of parts like races or cage and surface destruction. The fracture in the other race of the ball bearing occurs due to overload. When the bearing is misaligned the load acting on some ball sharply increases and may even crush them. The failure of the cage is due to centrifugal force acting on that ball. The complete breakage of the parts of the ball bearing can be avoided by selecting the correct ball bearing, adjusting the alignment between the axis of the shaft and the housing and operating within permissible speeds.

The ball bearings themselves act as a source of vibration, even if there are no defects present and they are perfectly aligned and adjusted. A defect on one of the elements of a ball bearing can cause the vibration level to increase. There are several types of defects that can occur on a ball bearing, such as cracks or pits on rotating surface or rolling elements, distributed defects such as roughness or misaligned races. Those distributed or localized defects form the vibration pattern that can be detected by a transducer and then analyzed and processed with the algorithm, which can

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enable the condition monitoring system to detect even the occurrence of a failure before it damages the machine or interrupt the production. When a rolling element strikes to a defect on one of the races, this strike creates an impulse. Since the rolling element bearing rotates, those impulses will be periodic with a certain frequency [5].

In case the defect occurs on the inner or outer race, how frequent each rolling element strikes to the defect is called “Ball-pass frequency” and determined by the bearing geometry and rotation speed. The failure of antifriction bearing occurs not due to breakage of parts but due to damage of working surfaces of their parts. The principal types of surface wear are as follows:

A. Abrasive Wear

Abrasive wear occurs when the bearing is made to operate in an environment contaminated with dust, foreign particles, rust or spatter. Remedies against this type of wear are provision of oil, seals, increasing surface hardness and use of high viscosity oils allows fine particles to pass without scratching.

B. Corrosive Wear

The corrosive of the surface of bearing parts is caused due to corrosive elements present in the extreme pressure (EP) additives that are added in the lubricating oils these elements attack the surfaces of the bearing resulting in fine wear uniformly distributed over the entire surface. Remedies against this type of wear are, providing complete enclosure for the bearing free from external contamination, selecting proper additives and replacing the lubricating oil at regular intervals.

C. Pitting

Pitting is the main cause of the failure of antifriction bearings. Pitting is a surface fatigue failure which occurs when the load on the bearing part exceeds the surface endurance strength of the material. This type of failure is characterized by pits, which continue to grow resulting in complete destruction of the bearing surfaces. Pitting depends upon the magnitude of Hertz’ contact stress and the no. of stress cycles. The surface endurance strength can be improved by increasing the surface hardness.

D. Scoring

Excessive surface pressure, high surface speed and inadequate supply of lubricant result in breakdown of the lubricant film. This results in excessive frictional heat and overheating at the contacting surface. Scoring is a stick-slip phenomenon, in which alternate welding and shearing takes place rapidly at high spots. Here, the rate of wear is faster. Scoring can be avoided by selecting the parameters, such as surface speed surface pressure and the flow of lubricant in such a way that the resulting temperature at the contacting surfaces is within permissible limits.

III. THEORY APPLIED TO BEARING DIAGNOSTICS

Static load is defined as the load on the bearing when the shaft is stationary. It produces permanent deformation in balls and races, which increases with increasing load. The permissible static load depends upon the permissible magnitude of permanent deformation. From past experience, it has been found that a total permanent deformation of 0.0001 of the ball or roller diameter occurring at the most heavily stressed ball of roller diameter occurring at the most heavily stressed ball and race contact can be tolerated in practice, without any disturbance like noise or vibrations. The static load carrying capacity of a bearing is defined as the static load which corresponds to a total permanent deformation of balls and races, at the most heavily stressed point of contact, equal to 0.0001 of the ball diameter. Steinbeck’s equation gives the static load capacity of bearing. It is based on the following assumptions:

1) The races are rigid and retain their circular shape.
2) The balls are equally spaced.
3) The balls in the upper half do not support any load.

It is assumed that there is a single row of balls. Considering the equilibrium of forces in the vertical direction,

$$C_O = P_1 + 2P_2 \cos\beta + P_3 \cos(2\beta) + \ldots \ldots$$

As the races are rigid, only balls are deformed.

Suppose δ_1 is the deformation, the inner race is deflected with respect to the outer race through δ_1.

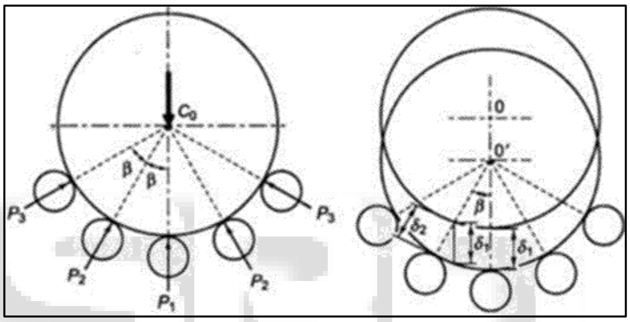

Fig. 1: Load Distribution on Ball Bearing

1) Forces acting on inner race,
2) Deflection of inner race

As the race of the ball rigid, only balls are deformed. Suppose δ_1 is the deformation at the most heavily stressed Ball No.1. Due to this deformation, the inner race is deflected with respect to the other race through δ_1. The center of the inner ring moves from O to O’ through the distance δ_1 without changing its shape. Suppose δ_1, δ_2…… are radial deflections at the respective balls.

$$\text{Also, } \delta_2 = \delta_1 \cos\beta \text{ or } \frac{\delta_2}{\delta_1} = \cos\beta$$

According to Hertz’s equation, the relationship between the load and deflection at each ball in given by,

$$\delta \propto (P)^{2/3}$$

Therefore,

$$\delta_1 = C_1 P_1^{2/3} \quad \text{and} \quad \delta_2 = C_1 P_2^{2/3}$$

$$\frac{\delta_2}{\delta_1} = \left(\frac{P_2}{P_1}\right)^{2/3}$$

From eq.(b) and (c),

$$\left(\frac{P_2}{P_1}\right)^{2/3} = \cos\beta$$

$$P_2 = P_1(\cos\beta)^{3/2}$$

Or In the similar way,

$$P_3 = P_1(\cos 2\beta)^{3/2}$$

Substituting these values in eq. (a)

$$C_0 = P_1 + 2\left[P_1(\cos\beta)^{3/2}\right]\cos\beta + 2\left[P_1(\cos 2\beta)^{3/2}\cos 2\beta\right] + \cdots$$

$$\text{Or } C_0 = P_1 M$$

Where, $M = [1 + 2(\cos\beta)^{5/2} + 2(\cos 2\beta)^{5/2} + \cdots]$

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If z is the no. of balls,

$$\beta = \frac{360}{z}$$

Nearly all bearing manufacturers' catalogs provide the static radial load capacity for any bearing size.

Principal dimensions (mm)			Basic load ratings (N)		Designation
D	D	B	C	Co	
25	52	15	14000	6950	6205
	62	17	22500	11400	6305
	80	21	35800	19600	6405

Table 1:

In this we carried out experiments on the ball bearings of the size that we drafted in Catia and analyzed practical maximum static load carrying capacity of ball bearing. The specification of the ball bearing of 3 different sizes are following are following Material

Bearing no	Bearing Outer dia	Ball dia	Bearing no	Bearing Outer dia
6205	52 mm	8 mm	6205	52 mm
6305	62 mm	10 mm	6305	62 mm
6405	80 mm	20 mm	6405	m

Table 2:

IV. Analysis Results

A. *Bearing no. 6205*

Bearing Outer dia = 52 mm	Ball dia = 8 mm	Req deformation = 0.0001 X 8 = 0.0008mm	
	178000 N	120000 N	60000N
TOTAL DEFORMATION	7.9557e-004 mm	5.3634e-004 mm	2.6817e-004 mm
EQUIVALENT STRESS	9.3106e-003 MPa	6.2768e-003 MPa	3.1384e-003 MPa
MAX SHEAR STRESS	5.1771e-003 MPa	3.4902e-003 MPa	1.7451e-003 MPa

Table 3:

B. *Bearing no. 6305*

Bearing Outer dia = 62 mm	Ball dia = 10 mm	Req deformation = 0.0001 X 10 = 0.001 mm	
	853000 N	570000 N	290000 N
TOTAL DEFORMATION	1.0019e-003 mm	6.6951e-004 mm	3.4063e-004 mm
EQUIVALENT STRESS	1.7272e-002 MPa	1.1542e-002 MPa	5.8721e-003 MPa
MAX SHEAR STRESS	9.3686e-003 MPa	6.2604e-003 MPa	3.1851e-003 MPa

Table 4:

C. *Bearing no. 6405*

Bearing Outer dia = 80 mm	Ball dia = 20 mm	Req deformation = 0.0001 X 20 = 0.002mm	
	2000000 N	1400000 N	700000 N
TOTAL DEFORMATION	2.1682e-003 mm	1.5177e-003 mm	7.5887e-004 mm
EQUIVALENT STRESS	4.5071e-002 MPa	3.155e-002 MPa	1.5775e-002 MPa
MAX SHEAR STRESS	2.5008e-002 MPa	1.7505e-002 MPa	8.7527e-003 MPa

Table 5:

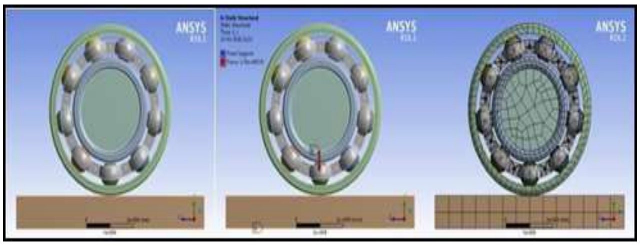

Fig. 2: Model, Boundry Conditions, Mesh Model

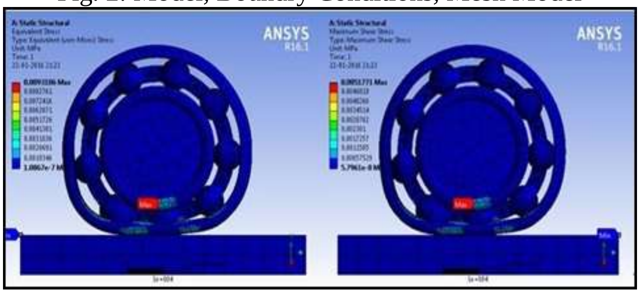

Fig. 3: 6205

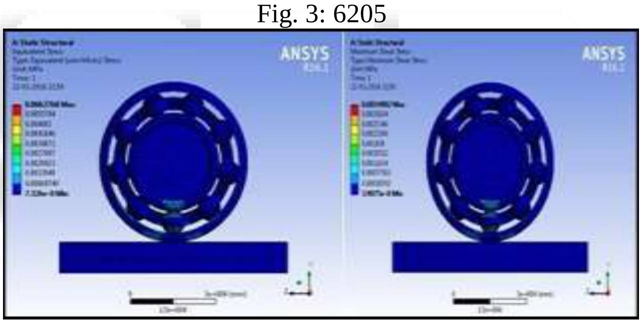

Fig. 4: 6305

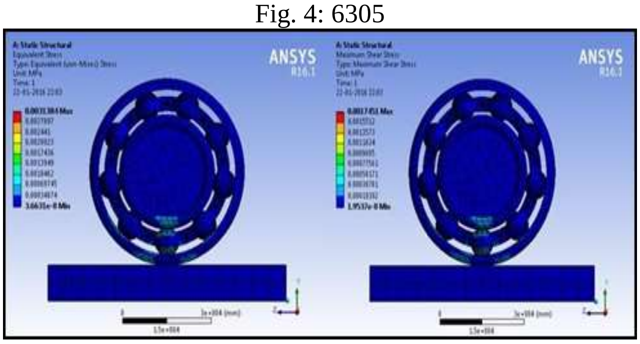

Fig. 5: 6405

V. Conclusions & Future Work

The study was conducted in order to generate a systematic and more accurate system that can be used to determine the static load bearing capacity of the ball bearing without destructive test of the costly bearing which can help to reduce the cost time and efforts required.

The findings of the research were that the deviation between the static load bearing capacity of a ball bearing obtained by finite element software and the result that were obtained by the practical test that were conducted on

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Universal testing machine of the Laboratory has a linear relation with size of bearing under test.

When the relation between this deviation and the bearing size was established using statistical tool of Linear Interpolation the following relation were obtained

A. Summery Output

Regression Statistics	
Multiple R	0.883
R Square	0.780
Adjusted R Square	0.559
Standard Error	4.268
Observations	3

ANOVA

	df	*SS*	*MS*	*F*	*Significance F*
Regression	1	64.450	64.450	3.538	0.311
Residual	1	18.217	18.217		
Total	2	82.667			

	Coefficients	*Standard Error*	*t Stat*	*P-value*	*Lower 95%*	*Upper 95%*	*Lower 95.0%*	*Upper 95.0%*
Intercept	2.5301	5.9258	0.4270	0.7431	-72.764	77.825	-72.764	77.825
X Variable 1	0.0003	0.0001	1.8809	0.3111	-0.0015	0.0020	-0.0015	0.0020

The results shows that the relation of deviation between the result obtained by the two method and the bearing size is linearly related with the

- Coefficient of Correlation 0.0003
- Coefficient of Regression 2.5301

This deviation is of Diverging in nature and the significance is of 0.311. The results can be interpreted as that if the static load carrying capacity of the bearing is to be determined by the relation developed in this research following equation can be used

$$Y= 0.0003X+2.5301$$

Where X= bearing life

Y= Deviation in the static load bearing capacity bearing capacity by the to methods

The effect of the other factors like manufacturing method of bearing and the manufacturing defect is not considered in this bearing.

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