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Angular Contact Ball Bearing Modeling with Different Types of Coatings

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Abstract: ***The purpose of this study is to determine the stress distribution of uncoated and coated ball bearings by using finite element analysis. The coatings used in this study are titanium nitride (TiN), titanium carbide (TiC) and chromium nitride (CrN) with a thickness of 5 microns. A contact analysis has been performed on ball bearings to compare the performance between coated and uncoated ball bearings. Boundary loads of 5000 N is used for contact analysis. This study tries to establish a simple, two-dimensional expression for the elastic deformation with the inner ring and ball bearing as a angular or curvature model in terms of the geometry of the coating contact surfaces. The coating of ball and raceway surfaces is a requisite but difficult factor to be determined during design, so it is desirable for engineering to understand the effect of surface coating on the motion of ball and subsurface stresses in bearing. For contact analysis, the maximum contact pressure and maximum stress on the coating, inner ring and ball bearing have been used for comparison. The results of this study show that, among other coatings, TiC provides the best protection for the ball of the ball bearing. This is because the low Poisson's ratio of TiC in other coated ball bearings helps reduce the stress on the ball bearing, even though TiC has the lowest Young's modulus in the coating. When a lower boundary load is applied, high COF will also cause an abnormal increase in the maximum stress on the contact surface between the coated or uncoated ball and the inner ring of the ball bearing.***

Keywords: ***Ball Bearings, Coating, Contact Surface, FEM***

I. INTRODUCTION

Th Rolling bearings are extensively applied in rotary machine of aerospace, railway, construction, mining and other industries because of their convenient installation and reliable operation [1]. Roller bearings are usually used for applications requiring exceptionally large load-supporting capability. Although ball and roller bearings appear to be simple mechanisms, their internal geometries are quite complex. When ball bearing works, it is usually that more than one rolling ball bears the load. When the load is 0, the contact area is a point, i.e., point-contact. When the load increases in running, the bearing inner ring, outer ring and rolling elements bring forth plastic deformation in the contact area, so the point-contact becomes face-contact [2]. Furthermore, contact area gradually becomes ellipse, and generates residual stress [3]. The contact parameters, such as the place, size, shape of contact area, as well as the contact pressure and friction force distribution, will be variable with loads change.

The contact finite element analysis can show bearings' information under contact, such as contact stress, strain, penetration and sliding distance, and so on, which play a significant role in optimum design of complicated rolling bearings. Contact is a complex nonlinear phenomenon, which involves not only change in state, but also accompanies with heat or electricity. Contact problem mainly includes two considerable difficulties at present. Firstly, before solving problems, the specific contact area isn't usually been known. With the change of load, material, boundary condition or the other factors, touch or separation will take place between surfaces. That is hard to predict, even is a abrupt change. Secondly, most frictional effects on contact problems are needed to be considered. They may be disordered as well as nonlinear. Finite element method (FEM) was proposed in for contact analysis of a roller bearings. FEM have proven to be successful in analysis of engineering structures. The study of contact analysis of a ball bearing using FE simulation was presented by Toumi et al. [4]. Analysis was conducted on deep groove ball bearings in a system having multiple bodies.

The dynamic analysis of roller bearings in multi-body systems were presented in, [5 ,6]. A lumped mass-spring model was used by [7] to study the vibration response of a ball bearing. Finite element explicit dynamics analysis of a defective roller bearing was also performed by [8] to analyze the vibration response. Similarly, the ball bearing was analyzed by [9] under high loads to determine the internal dynamics of the ball bearing coatings. Analysis of ball bearings and cylindrical rollers have been conducted by researchers using specially developed software. Similar method was adopted by [10] on an angular ball bearing to calculate the stiffness and loading relationship. However, some important factors that are mainly affected by the contact conditions, did not include the previous research. The crucial factors such as the coating of ball bearing, and the interfacial bond strength of coating structures might affect the performance of ball bearing. To determine the effects of the above factors, the stress distribution of the contact between coated ball bearing and inner ring were investigated.

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Various types of coating materials have been used in the industry nowadays. Most coatings will be used on steel ball bearings as these balls have low hardness compared to ceramic ball bearings [11]. However, the thickness of the coating and the type of coating on the ball surface or ball bearing ring are also significant factors so that the ball coating can provide better protection to the surface. Previous research showed that a ball bearing model assumes the plate (without curvature) as the inner ring [12]. However, the contact surface between the sphere with the sphere will have a different stress distribution compared to the contact between the sphere with the plate[13]. In fact, no previous studies investigated the impact of ball bearing via coated curves surfaces model in a systematic way. Hence, the purpose of this study is to create an FE model, with the inner ring as a angular model, with uncoated and coated ball bearings to evaluate the stress distribution. This paper provides a contact analysis of a roller bearing using Finite Element simulation in ABAQUS.

II. METHODOLOGY

A. Modeling of Uncoated and Coated Ball Bearing

This study will focus on modeling in 2-dimensional (2D) by using the parameters in Table I and the mechanical properties of ball bearing is shown in Table II. The modeling of ball and inner ring of ball bearings using Abaqus software has been shown in Fig. 1 and Fig. 2 for coated and uncoated ball bearing.

Table- I: Parameters of ball bearing modeling

Sections	Parameter (mm)
Diameter of Ball copy	17.462
Outer Diameter of Inner Ring	55.038
Inner Diameter of Inner Ring	45

Table- II: Parameters of ball bearing modeling

Material	Young's Modulus (GPa)	Poisson Ratio	Density (kg/m^3)
100Cr6	210	0.25	8000

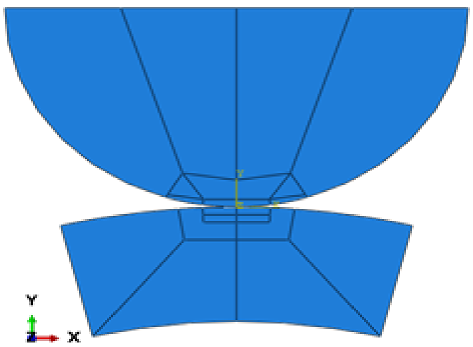

Fig. 1. Model for uncoated ball bearing.

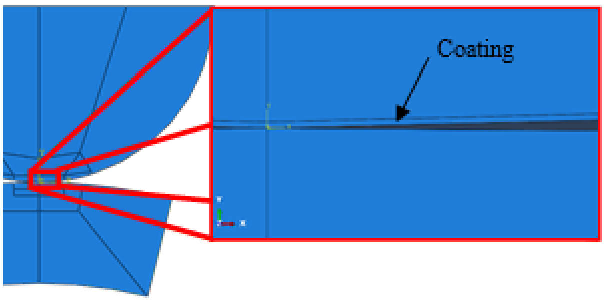

Fig. 2. Model for coated ball bearing.

B. Analysis of Stress Distribution for Contact on Uncoated Ball Bearing

When using the Abaqus software, various settings need to be set so that the results obtained will be more accurate. Different types of analysis such as contact need to use different settings. For contact analysis, the step settings have been set as explicit when conducting contact analysis. The time-period has been set to 0.13 seconds so that the load applied to the ball bearing can start from 0 to the maximum load. The step also needs to be set to a smooth step so that the accuracy of the results will be more accurate compared to the instantaneous step. Amplitude is the speed of the load applied to the ball and the set value is 55. This is because the step and amplitude will ensure that the load applied to the ball is gradual so that this situation can mimic the real cases. The following table is the data set on the Abaqus software.

Table- III: Step data that being set in Abaqus software

Step Setting	Period (s)	Amplitude
Initial	0	0
Step	0.13	55

Interaction is the setting of the contact or interaction between two surfaces, which is the ball surface and the inner ring of the ball bearing. The setting is set to general contact interaction without friction. The boundary condition used for this analysis is shown in Fig. 3. The side and bottom surfaces of the inner ring of the ball bearing have been constrained while the center of the ball has been constrained on the x-axis and the rotation of z-axis. The point of contact between the ball and the inner ring on the ball is also constrained on the x-axis. This is because the point of contact between the ball and the inner ring will not slip when a load is placed on the ball in the direction of the y-axis since the contact interaction is frictionless. Coupling is used to Spair the surface shown in Fig. 3 with the center point of the ball so that the surface will be restrained with the center point of the ball.

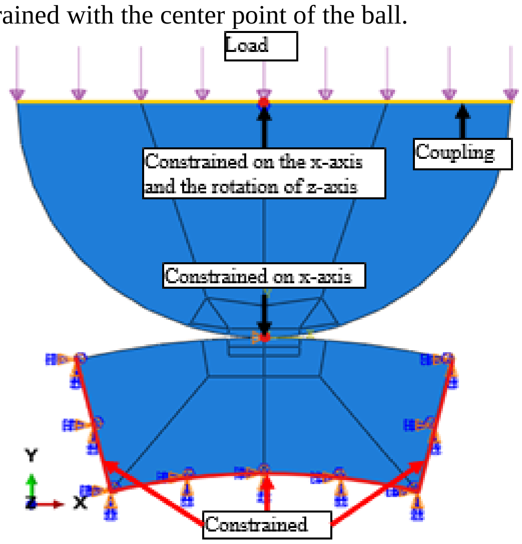

Fig. 3. Boundary condition for contact analysis.

After the boundary conditions are set, a partition has been constructed to ensure that different parts can be assigned different number of elements. Higher number of elements can produce results that have higher accuracy but the time for analysis process will take longer time to complete.

Therefore, partition can help by assigning some areas that are more critical which require a higher element number and some areas that require a lower element number as shown in Fig. 4.

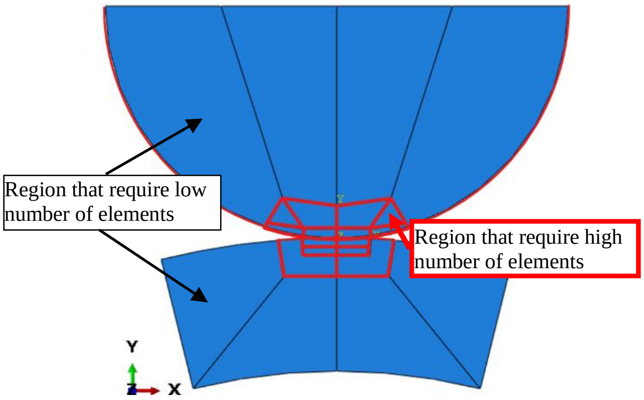

Fig. 4. Boundary condition for contact analysis.

C. Analysis for Different Types of Coated Ball

This analysis will focus on the types of coating as a variable to make comparisons between different types of coating materials. Table IV shows the mechanical properties for different coatings. The mechanical properties of the coating are found based on previous studies. The thickness of the coating will be set as 5 µm as shown in Fig. 5 because the thickness is based on the study of Yau et al. [14,15]. The settings for this analysis are the same as the contact for uncoated ball bearing except for the interaction between the ball and the coating surface is set to surface-to-surface and rigid constraint is needed to assign between the ball substratum surface with the coating surface. This is because the surface between the substratum and the coating has a strong adhesion.

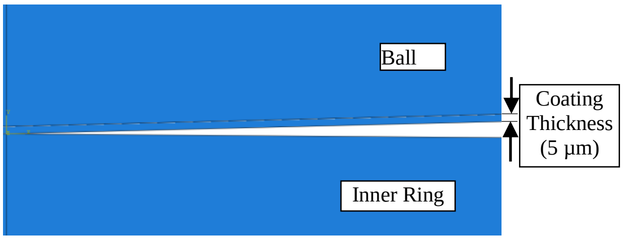

Fig. 5. Model of coating thickness.

The boundary condition of the ball shown in Fig. 6 requires a high number of elements because the thickness of the 5-micron thin layer is rigid constrained on the surface of the ball bearing radius of 17.5 mm. Due to these size differences in geometry, the number of elements needs to be increased so that the mesh distortion of the model can be avoided. The number of elements on the ball bearing should be increased at the contact area between the ball and the inner ring of the ball bearing. Fig. 7 shows the meshing of the coating of ball bearing.

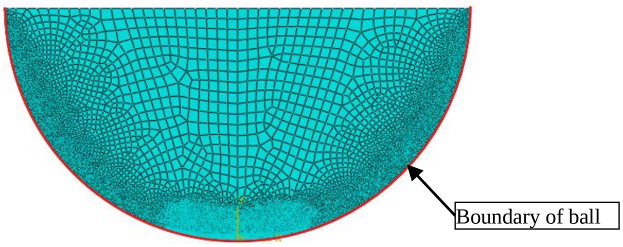

Fig. 6. Boundary of ball which requires high number of elements.

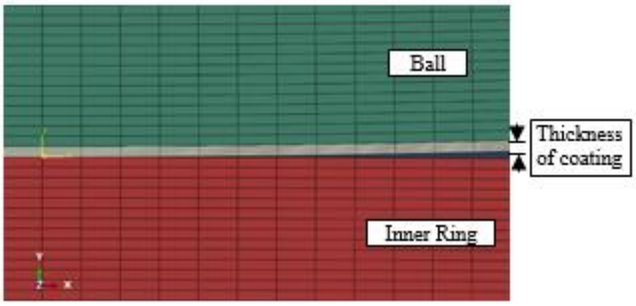

Fig. 7. Mesh of coating.

Table- IV: Mechanical properties of coating materials

Type of materials	Young's Modulus (GPa)	Poisson Ratio	Density (kg/m^3)	References
TiN	376.5	0.25	5430	Tekaya et al. (2014) Wang et al. (2015)
CrN	300	0.28	5900	Mohammadpour et al. (2017)
TiC	272.1	0.17	4920	Su (2014) Shanaghi et al. (2012)

III. RESULT AND DISCUSSION

A. Model Verification

Do not use abbreviations in the title or heads unless they are unavoidable. Mesh convergence is an important step in finite element analysis. The number of elements while the mesh converged plays an important role in obtaining high-precision results. Before increasing the number of elements, the partition labeled with yellow color should be set as a sweep technique in mesh control as shown in Fig. 8. This is because this setting can ensure an increase in the number of elements that have a consistent pattern. Network control on pink maps is negligible because the default option is an "Free" technique.

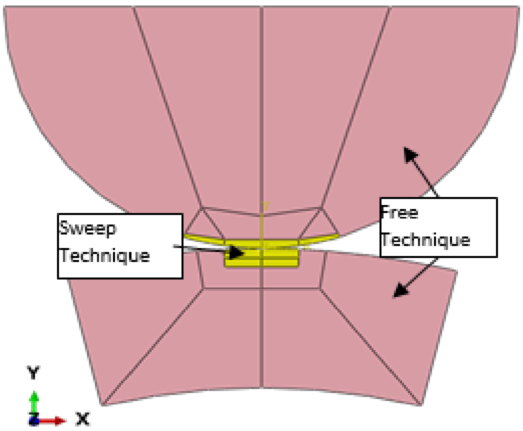

Fig. 8. Mesh control on ball bearing model.

The number of elements starts with 3159 by applying a boundary load of 5000 N on a TiN ball bearing because the boundary load is the maximum load and the coating will produce the widest and maximum stress distribution in the ball bearing model. Stress on the coating, inner ring and ball is used to ensure that all three values can achieve mesh convergence.

Based on Fig. 9, the mesh convergence for maximum stress on the coating was achieved at 7.743 GPa while the 41800 elements were used.

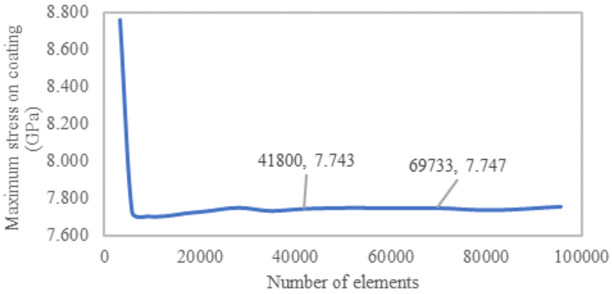

Fig. 9. Graph of maximum stress on coating versus number of elements.

The model of this study will now be verified with the results of the journal Massi et al. (2014) who have a similar model to this study. The journal model is a type of uncoated ball bearing. Therefore, the uncoated ball bearing model will be used to make comparisons with the results of the past journal for verification. The comparison results in Figure 10 show that the finite element analysis results found in this study have a maximum difference percentage of only 3.77 %. This shows that the results obtained in this study can be confirmed accurately because the maximum difference percentage does not exceed 5%. Therefore, this model can be used to conduct analysis to achieve the objectives of this study. Fig. 10 shows a graph of the percentage difference with a load value of 1000 N up to 5000 N.

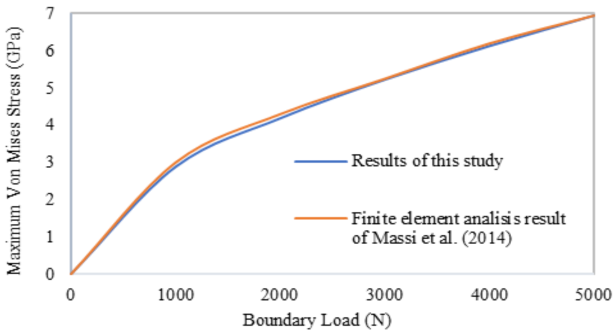

Fig. 10. Result comparison of maximum stress on ball bearing between this study and Massi et al. (2014).

B. Contact Analysis of Coated and Uncoated Ball Bearing

Contact analysis was conducted to identify the maximum stress on the TiN, TiC and CrN-coated ball bearing as well as uncoated ball bearing with substratum of 100Cr6 for comparisons. Comparison of contact pressure, maximum stress for inner ring and ball bearings is also carried out between uncoated ball bearing with TiN, TiC and CrN-coated ball bearings. This comparison is conducted to identify the effect of the coating on the ball bearing and which coating can help reduce the maximum stress and contact pressure of the ball bearing. Fig. 11 shows the Von Mises stress distribution in (a) uncoated and (b) TiN, (c) TiC and (d) CrN-coated ball bearings for step time of 0.0065 seconds and Fig. 12 shows for 0.013 seconds Fig. 13 shows for 0.13 s with a boundary load of 5000 N. At a step time of 0.0065 seconds, uncoated ball bearing has a wider stress distribution than coated ball bearings. However, the maximum stress on TiN, TiC and CrN-coated ball bearings has exceeded the maximum stress on coated ball bearings starting at 0.013 seconds. The maximum stress location on TiN and TiC-coated ball bearings is located at the contact point of the ball bearing while CrN-coated ball bearings are the same as the uncoated ball bearing which are located on the inner ring of the ball bearing starting from the step time of 0.013 seconds. The evolution of maximum stress value of Von Mises for the ball bearings can be clearly seen in Fig. 13. The TiN-coated ball bearing have the highest maximum stress among all other ball bearings due to highest Young's modulus of mechanical properties. This value may indicate a significant prediction in design of coatings.

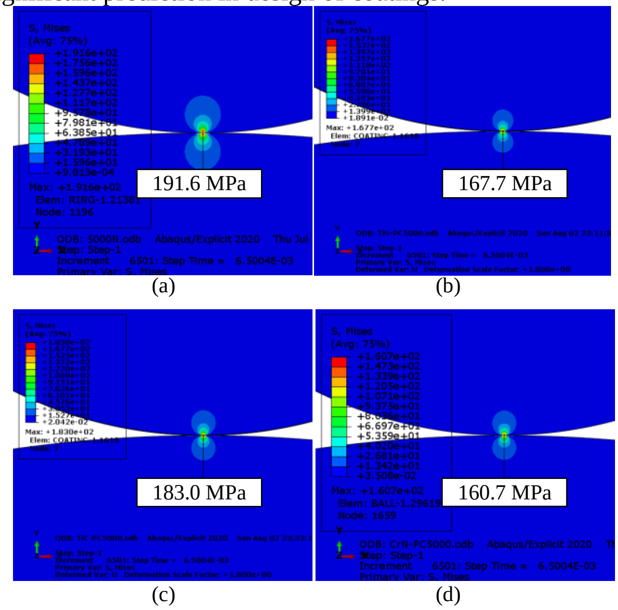

Fig. 11. Von Mises stress distribution in (a) uncoated and (b) TiN, (c) TiC and (d) CrN-coated ball bearings for step time of 0.0065 s.

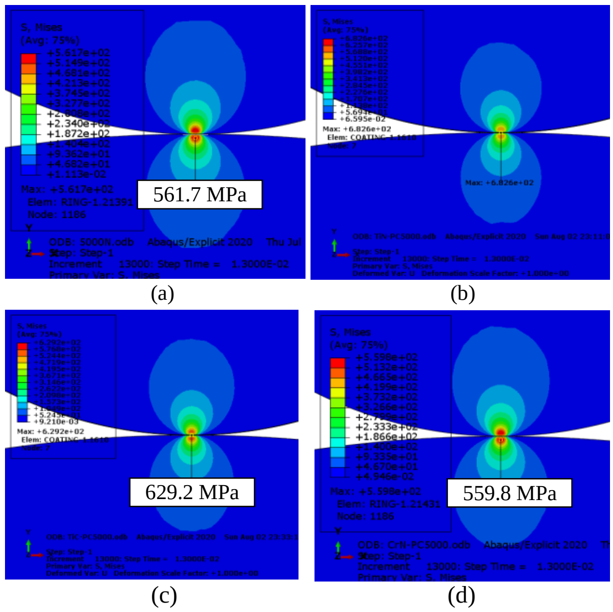

Fig. 12. Stress distribution in (a) uncoated and (b) TiN, (c) TiC and (d) CrN-coated ball bearings during step time of 0.013 s.

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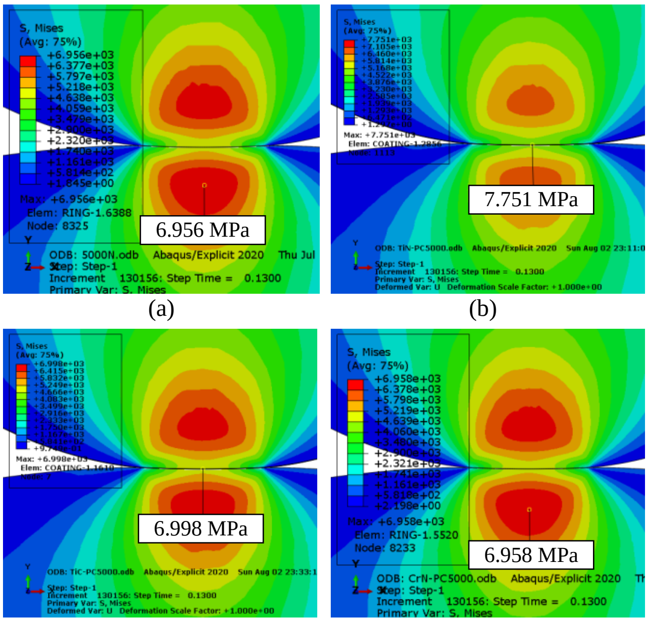

(a) (b) (c) (d)

Fig. 13. Stress distribution in (a) uncoated and (b) TiN, (c) TiC and (d) CrN-coated ball bearings during step time of 0.13 s.

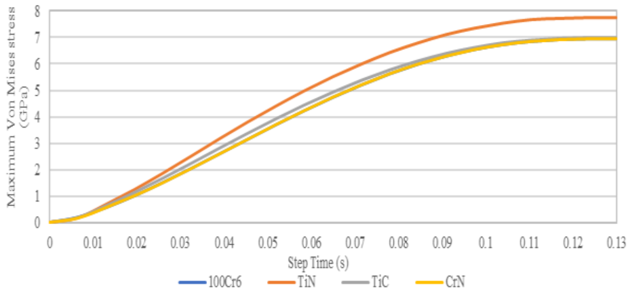

Fig. 14. Graph of maximum Von Mises versus step time.

IV. CONCLUSION

The objective of the first study was to identify the stress distribution for the contact of the uncoated ball bearing (100Cr6) by using a curved modeling. For the contact analysis, the Von Mises stress distribution for TiN, TiC and CrN-coated ball bearings with a boundary load of 5000 N shows that the coated ball bearings have a wider stress distribution than the coated ball bearings at a step of 0.0065 seconds. However, the maximum stress on TiN, TiC and CrN-coated ball bearings has exceeded the maximum stress on uncoated ball bearing after step time of 0.013 seconds. The maximum stress location on the TiN and TiC-coated ball bearings is located at the contact point of the ball bearings while the CrN-coated ball bearings are the same as the uncoated ball bearing which are located on the inner ring of the ball bearing starting at step time of 0.013 seconds. The maximum stress value on the coated ball bearings is lower than the maximum stress on the uncoated ball bearing since the start till 0.026 seconds of step time. The maximum stress on TiN-coated ball bearings has exceeded the maximum stress on uncoated ball bearing after step time of 0.026 seconds while for CrN-coated ball bearings, the maximum stress on ball bearings began to exceed the maximum stress on uncoated ball bearing after step time of 0.0455 seconds. However, TiC-coated ball bearings always have a lower maximum stress on ball bearing compared to uncoated ball bearing because TiC coating can help reduce the maximum stress on ball bearings compared to TiN and CrN coatings. These maximum stress value actually offers significant interpretation when these value have been used to calculate the safety factor. The safety factor is the factor that a material's maximum strength value is divided by it to find the design strength.

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