Mechanical Analysis of Rolling Mill Support System Based on ANSYS-Workbench

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Abstract. In the rolling of four-roll mills, there are often damages in bearing such as the burning and adhesion failure. Domestic and foreign scholars have done a lot of calculations and experiments on this, and believe that when the support system of the roll neck (roll neck, bearing and bearing seat) suffers the roll bending force, the four-row cylindrical roller bearing suffering the radial bending force may be eccentric due to the hyperstatic structure of the mechanism in which the bearing block cannot be synchronously bent with the roller neck, thereby causing bearing damage. In this paper, the integrated model of the roll neck support system of the roll is established by SolidWorks, which is introduced into ANSYS-Workbench for contact surface setting, loading and constraint, and finally the roll neck bending deflection and bearing roller contact stress are analyzed and verified, which confirmed the validity of integrated deformation and mechanical analysis of roll neck support system by finite element method.

1. Introduction

With the increasing requirements for the precision of the strip material in the rolling industry, the widening material and the process needs for controlling demand a bigger bearing capacity, and a higher mechanical design requirements, especially the strict working environment of bearings on the neck. Compared with general bearings, roll bearings have the following characteristics: large working load, large difference in running speed, and poor working environment. Since the diameter of the roller should ensure the strength, the outer dimensions of the bearing block should not be larger than the minimum diameter of the roll barrel, and the length of the roll neck is shorter, the bearing on the roll neck would bear a heavy rolling load under the condition of limited radial size. Therefore the rolling bearings on the rolling mill mostly are multi-row roller bearings. This kind of bearing possesses a small radial dimension and good impact resistance. A common form of support is a four-row cylindrical roller bearing that is subjected to radial forces and an additional thrust bearing which affords the axial loads. Four-row cylindrical roller bearings are widely used in high-speed heavy-duty rolling mills for their small radial size, large bearing capacity and high permissible speed [1].

During the rolling process, the bearing failure of the operating side often occurs. After analysis by experts at home and abroad, it is found that the mechanism of the rolling mill support system is...
hyperstatic when subjected to hydraulic bending force, that is, the bearing block is axially restrained by
the axial retaining baffle, which prevent its synchronously deflection with the roller neck. And that leads
to the result that the inner and outer ring of the four-row cylindrical roller bearing of the radial bearing
supporting the bearing block generate uneven gap, so that the four rows of rollers are not subjected to
uniform load affecting the service performance of the bearing seriously [2]. At present, the calculation
model of the rolling process analysis usually assumes the force at the roll neck as a concentrated load or
a uniform load, which is inconsistent with the actual bearing force, and accordingly the model does not
simulate the true stress distribution and deformation of the support system [3].

In this paper, based on the actual size of the support system of the four-roll cold rolling mill work
roll of rolling factory in Suzhou, an integrated model of the roll neck, four-row cylindrical roller bearing
and the bearing block is established. The bending force is much smaller than the rolling force, so the
end of the edge of the contact area between the work roll and the support roll can be considered as the
fixed end of the cantilever beam in the roll neck support system. The bending force and constraint are
applied to the assembled support system by the finite element simulation software ANSYS-Workbench
to simulate the bending state on the roll neck which supports the bearing block and the four-row
cylindrical roller bearing. This work provides a certain theory support for analyzing the eccentric load
of the four-row cylindrical roller bearing and the structural static analysis of the supporting system by
finite element method (FEM) [4].

2. Calculation of bending deflection for roll neck cantilever model

When the work roll neck is subjected to the bending force, the separation point between the work roll
and the backup roll can be seen as the cantilever of the roll neck model for the rolling force is much
larger than the bending force. The calculation model is shown in Fig. 1.

![Figure 1. Calculation model of the bending force of the work roll cantilever beam model.](image)

Considering the theoretical calculation of the bending deflection of the roll neck bending force,
according to machinery’s handbook, the Mohr integration in energy method can be used for only a few
specific points on the axis need to be accurately calculated [5]. The formula of Mohr integration is as
follows:

\[
\Delta = \int_l \frac{M(x)}{EI} dx
\]

Where \( \Delta \) represents the amount of deformation in the given point(deflection or deflection angle), \( M \)
represents the bending moment of the shaft, \( \overline{M} \) represents the bending moment caused by the additional
unit force or the unit moment at the given point, \( E \) represents the elastic modulus, and \( I \) represents the
moment of inertia of the section. The four stepped shaft dimensions from the fixed end are: \( \phi 260 \times 45 \text{mm} \),
\( \phi 185 \times 100 \text{mm} \), \( \phi 170 \times 206 \text{mm} \), and \( \phi 150 \times 187 \text{mm} \). A uniform bending force of 550 KN was applied to
the third shaft, and the Mohr integration was used to obtain a deflection of the edge of the third section
from the right hand. And the final deflection of the roll is 0.283 mm and the deflection angle is 2.2'.
According to the machinery’s handbook [5], the four-row cylindrical roller shafts in the rolling mill only
bear radial force, but not axial force. The maximum allowable roll neck deflection angle is 2'. The results
obtained in this paper exceed the allowable limit, which would cause the decrease of bearing fatigue life
and affect the normal operation and roll changing period of the rolling mill.
3. Building an integrated model by SolidWorks

The three-dimensional modeling software SolidWorks was used to build the cantilever beam model based on the roll neck data on the working roll side of a four-roll aluminum foil cold rolling mill. The process is as follows:

1. Build the stepped shaft model of the roll neck. Ignore chamfers and other small sizes.
2. Build the four-row cylindrical roller bearing model. For merely limiting the circumferential movement of the roller under the radial load, the bearing cage is ignored during modeling. Since the outer ring of the bearing is connected closely to the bearing block, it was seen as a whole with the bearing block for simplifying the model.
3. Build the bearing block model. The bearing block and the four-row cylindrical roller outer ring are built as an integral model, and the bending cylinder lugs was built to bear the bending force. Ignore chamfers on the block and other small dimensions.

The above three models are assembled into a whole part in SolidWorks, where the four-row cylindrical roller bearing has a size of $\phi 170 \times 230 \times 180$mm, the inner ring raceway diameter is 185mm, and the outer ring raceway diameter is 215mm. The roller-diameter is 15mm with a length of 30mm, and the number of rollers is 30 in one row.

The following explanation is given about the gap problem of the contact surface during modeling. Since the working clearance of the bearing is the actual clearance, which is generated by the theoretical clearance minus the amount of fit compression, the clearance reduction caused by temperature difference, and then adding the clearance increase caused by the bearing load. As a result, the rolling bearing is able to work within an appropriate clearance range to ensure the normal operation. According to available reports, the rolling bearing could run to the longest life when the clearance is zero or slightly positive [6]. Therefore, when it comes to the modeling dimensions of the rings and the roller of the bearing, it is assumed that the bearing working clearance is zero, and the ideal working condition is met with this assumption.

4. Finite element analysis process

4.1. Importing the finite element model

The 3D large-scale finite element simulation software ANSYS-Workbench have good compatibility with SolidWorks. Therefore, the support system assembly model built by SolidWorks can be directly imported into ANSYS-Workbench. The materials’ mechanical properties of each part are shown in Tab. 1.

| Name         | Material | Density(g/(cm)$^3$) | Elastic modulus(Gpa) | Poisson ratio |
|--------------|----------|---------------------|----------------------|--------------|
| Roll neck    | 9Cr2Mo   | 7.8                 | 236                  | 0.3          |
| Bearing      | GCr15    | 7.8                 | 210                  | 0.3          |
| Bearing block| 45       | 7.8                 | 210                  | 0.3          |

The mesh of model is refined at the inner and outer ring contact surfaces of the cylindrical bearing to obtain higher calculation accuracy. The model consists of 7 parts, 123 bodies, and a total of 266,317 units. The overall mesh is shown in Fig. 2, and the mesh refinement model at the bearing contact surfaces is shown in Fig. 3.
4.2. Setting of contact conditions
The contact conditions for contact surfaces of the assembly are set as follows:

(1) Since the roll neck and the inner ring of the cylindrical bearing are tightly fitted, the inner ring and the roll neck are set bonded together by the bonded contact condition so that they do not generate relative displacement.

(2) The outer ring of the bearing is built integrated with the bearing block, so we only consider the contact condition of the rollers and the outer ring raceway. Since the roller surface and the inner race of the outer ring are frictional and can be separated from each other, the frictional contact condition is applied. The cylindrical surface of the cylindrical roller is set as the contact surface, the target surface is set as the outer ring raceway, and the contact friction coefficient is set to 0.05 by the material and lubricant conditions. It is not easy to cause pathological conditions when the model is calculated by Augmented Lagrange formulation, and the normal contact stiffness is set by program control for an appropriate result.

(3) Since the bearing block could not be bent under bending force for its axial constraint, a deflection angle between the inner and outer ring of the cylindrical bearing would be generated. To simplify the model, referring to the assumption of GX Shen about the radial eccentric loading analysis by the boundary element method [7], the rollers are assumed to be plate units with infinite bending stiffness, and these plate units are considered fixed on the inner race. Similarly, it is assumed in this paper that the rollers are bonded to the inner ring, and that does not influence the results of the static analysis of the roll neck support system.

4.3. Load and constraints settings
During the static analysis of the support system model, the big end of the cantilever beam is fixedly constrained. The displacement constraint is applied to the bearing block so that it has only the freedom of movement in the radial direction. In other words, it can only move in one direction with the bending force. An axial displacement constraint is applied to the inner ring of the bearing, and a bending force is applied to the lugs supporting the bending cylinder on both sides of the bearing block. Turn on the large deflection switch in the analysis setting, set the number of load steps to 5, and divide the bending force of 550KN into 5 equal parts. After the load and constraints are applied, the support system is solved and post-processed.

5. Post-processing and result analysis

5.1. Analysis of roll neck deflection of support system
A nonlinear iterative calculation is performed on the total deformation of the model and the equivalent contact stresses of the bearing. The results turn out to be of good convergence property. The total
deformation of the roll neck is shown below in Fig. 4. In Workbench the path setting can be used to obtain the deflection curve to calculate the deflection and deflection angle at any point on the roll neck. Thereby, the bending deflection of the end of the third stepped shaft from the right hand of the roll neck model can be obtained. In order to verify the superiority of building an integrated model, this paper also calculates the deflection using the load tool Bearing Load in ANSYS-Workbench which named simulation 1. Compare the calculation results with the actual integrated bearing model deformation named simulation 2. The results of simulation 1 and 2 were compared with those calculated by Mohr integration, and the comparison was shown in Tab. 2.

![Figure 4. Roll neck bending deformation nephogram](image)

Table 2. Comparison of theoretical calculation with simulation results of deflection

| Method            | Deflection | Deflection angle |
|-------------------|------------|------------------|
| Theoretical calculation | 0.283 mm   | 2.2'             |
| Simulation 1      | 0.341 mm   | 4.0'             |
| Simulation 2      | 0.273 mm   | 3.1'             |
| Percentage difference 1 | 20%        | 45%              |
| Percentage difference 2 | 4%         | 29%              |

It can be seen from the table that the deflection angle in three calculating methods are all greater than 2', which further indicates that the roll neck deflection angle at the cylindrical bearing position is bigger than its allowable limit, which seriously affects the fatigue life of the bearing. Based on the comparison, it is found that the difference from the theoretical calculation is smaller. That is to say, the results of the real integrated bearing model is more desirable than those with the load tool Bearing Load in ANSYS-Workbench. This demonstrates the accuracy of the results by building an integrated actual model as well as the feasibility and reliability by using ANSYS-Workbench.

5.2. Analysis of contact stress of support system

The post-processing module of ANSYS-Workbench presented the equivalent von-Mises contact stress nephogram of the model. Fig. 5 displayed the contact stress distribution of the four-row cylindrical rollers with the outer ring of the cylindrical bearing. Where the row of rollers closest to the fixed end of the roll neck is the first row, and the row farthest away from the fixed end is the fourth row.

5.2.1. Contact stress distribution of roller and outer ring. By observing the contact stress values of the rollers with the inner and outer ring, it is found that the stress distribution of the four row rollers is
eccentric in both axial and circumferential direction. Moreover, the contact stress on the rollers and the outer ring of the bearing is greater than that on the inner ring. For the fatigue life and damage analysis of the rollers, it is necessary to observe the roller that suffered the biggest contact stress. Therefore, we only research the contact state of the four-row cylindrical rollers with the outer ring in this paper.

![Four-row cylindrical rollers contact stress distribution nephogram](image1)

**Figure 5.** Four-row cylindrical rollers contact stress distribution nephogram

![Four-row rollers and outer ring contact stress distribution diagram](image2)

**Figure 6.** Four-row rollers and outer ring contact stress distribution diagram

To analyze the contact stress values of the four-row rollers more clearly, four points are taken along the axis of the roller’s contact area for stress value, and then the contact stress values is utilized by the software ORIGIN to draw the three-dimensional stress distribution diagram shown in Fig. 6. It can be seen that stress of the rollers in the center position of the cylindrical bearing is not the largest, and there are obvious peaks at the two sides of the center. When the contact stress is 0, it means that the rolling element and the rings do not actually have effective contact, and the rollers do not bear load. The contact stress of first row of the four-row cylindrical roller is the largest, the second row is about half of the first
row, and the third row and the fourth row are subjected to smaller contact stress. That is to say, the four-row cylindrical roller has an obvious unbalanced stress distribution, and the gap between the four rows of rollers and the outer ring is also gradually increased from the position of the first row to the fourth row. It is this uneven gap that causes the eccentric load, which would aggravate the damage of the bearing together with the high temperature and poor lubrication conditions. The uniform load distribution can be realized by improving the self-aligning structure of the bearing block.

5.2.2. Distribution of stress along the circumference of rollers. The analysis of the distribution nephogram shows that the maximum stress is on the roller edge contact with the outer ring. The stress value is introduced into ORIGIN to generate a line graph as shown in Fig. 7. The actual stress distribution of the first row is shown in Fig. 8. It can be seen that the contact stress distribution curve of each row of cylindrical rollers is M-shaped, and particularly, the first row of rollers peaked on both sides with a maximum value of 994Mpa, which is smaller than the general ultimate load capacity 3000Mpa.

When the bending force is applied to the bearing through the bearing block, and the maximum load-bearing area of the bearing block under heavy load is the thin whole section on the center line, which is shown in Fig. 9. This thin whole section is deflected due to insufficient rigidity, which in turn causes the reaction force of the bearing and the bearing block, so that the radial load is redistributed to the region of the larger stiffness around the bearing block hole, which leads to the M-shaped stress distribution curve [7]. According to the analysis results, it is possible to increase the rigidity of the bearing block by improving the structural shape of the bearing block, thereby improving the working condition of the rollers and achieving a uniform distribution of contact stress.
5.2.3. **Contact stress distribution of single roller.** It can be seen from the stress distribution of the first row of roller contacts that there is obvious stress concentrating phenomenon in rollers, and the contact stress on both sides is greater than that in the middle. In other words, there is a side contact effect as shown in Fig. 10. The side contact effect can be improved by changing the bus bar or crown shape of the roller to achieve a uniform load.

It can be seen from the section stress contour map of the roller that the maximum Mises stress of the roller occurs in the subsurface of the contact area between the roller and the outer ring, as shown in Fig. 11. Due to the stress concentration phenomenon on the roller, it is easy to cause fatigue failure during its movement, which will lead to internal crack initiation and decreasing the rolling fatigue life.

5.2.4. **Contact stress distribution of bearing rings.** The contact stress distribution of the inner and outer ring are shown in Fig. 12 and Fig. 13. It can be seen that the stress distribution regularities are consistent with the stress distribution state of the rollers before.
6. Conclusion

In this paper, the finite element method is used to conduct static analysis on the integrated model of the roller neck support system, exploring the bending deformation of the roller neck and the stress distribution of the four-row cylindrical roller bearing. The main results are as follows:

(1) After the modeling of the integrated supporting system by SolidWorks, the finite element software ANSYS-Workbench was used to simulate the influence of roll bending force on various parts of roll neck. The difference between the bending deflection of the roll neck calculated by Mohr integration and that obtained by post-processing analysis is only 4% which is in a reasonable scope, and that verified the feasibility and reliability of the integrated bearing system build by FEM. On the radial bearing, the bending deflection angle of the roll neck under 550KN bending roller force is greater than its allowable value, which will seriously decrease the fatigue life of bearing.

(2) Importing the equivalent Mises contact stress values between the rollers and outer rings of the four-row cylindrical roller bearing, and drawing the stress distribution diagram by ORIGIN, it can be clearly observed that the bearing has obviously unbalanced load distribution. The contact stress of the rollers in the first row is the largest, about twice that of the second row, while the stress in the third and fourth rows is smaller. This unbalanced distribution is attributable to the hyperstatic structure caused by the axial baffle restraint of the bearing block, which in turn generate the uneven gap between the inner and outer ring of the bearing. Coupled with high temperature and poor lubrication conditions, it will aggravate the damage of the bearing. The uniform load distribution can be realized by improving the self-aligning structure of the bearing block.

(3) The M-shaped stress distribution state in single-row rollers is attributable to the large deformation caused by insufficient stiffness of the thin-hole in bearing block. The roller has side contact effect along the axial direction. By improving the roller crowning and the structural stiffness of the bearing block, the radial load between the four rows of cylindrical roller bearings can be distributed uniformly and the damage of the roller bearing can be improved.

To conclude, the three-dimensional finite element method is used to solve the mechanical analysis of the supporting system of the roll neck, hoping to provide some technical support and reference for subsequent scholars.

Acknowledgments

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