Analysis of Fatigue Fracture and Strength Improvement of Backup Rolls

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Abstract. As an important part of the cold rolling mill, the backup rolls are always in service under complicated stress conditions. Once the roll fails, it will bring huge economic losses. In this paper, the problem of fracture failure of the backup roll neck position in an 1150mm six-roll reversible cold rolling mill of a company is analyzed. The stress distribution of the barrels under working condition is calculated by the finite element software ABAQUS, and the stress value at the dangerous section of the backup roll is obtained. The fatigue strength safety factor of the backup roll is obtained by analytical calculation. Combined with the actual production situation, from the perspective of structure and size design, improved schemes can be proposed to improve the fatigue strength of the backup roll. This analysis can provide reference for related enterprises.

1. Introduction
During rolling process, the backup rolls are subject to the combined effects of bending stress, shear stress, contact stress between the rolls, and residual stresses formed during the roller manufacturing process\cite{1}. Common forms of failure include roll surface wear, spalling, cracks, fracture, etc. Among them, the fracture of the backup roll not only increases the cost of production, but also greatly affects the normal production of the unit. Figure 1 shows the crack and fracture failure of a company's backup rolls.

![Figure 1. Crack and fracture of Backup roll (a) Crack (b) Fracture](image-url)
In recent years, Yan [2] calculated the bending stress of the backup roll neck and barrel by bending strength analytical calculation and finite element simulation to determine the weak point of the backup roll at the roll neck. Wang [3] analyzed the failure modes of the CSP production line backup roll, such as the crack of the barrel, the small drop; the shoulder; the cause of the crack at the arc of the roller neck, and proposed countermeasures. Sun [4] by comparing the common structures of several roll neck transitions, the final choice is to increase the radius of the arc to further reduce the stress level, thereby improving the bending strength of the backup roll of the cold rolling mill. Ouyang [5] through the analytical method to check the fatigue strength of high-speed finishing roller shaft, found that rolling force, rolling torque and steel temperature are the main factors affecting the fatigue strength of the roller. The fatigue strength of the roller shaft is improved by increasing the surface finish of the fracture portion of the roll and increasing the diameter of the fracture portion of the roll.

In this paper, the combination of finite element numerical simulation and analytical calculation is used to analyze the causes of fracture of the backup roll. Quantitative study of the relationship between different structures and dimensions and the safety factor of fatigue strength provides a reference for improving the fatigue strength of the backup rolls.

2. Analysis of the reasons for the fracture of the backup roll

2.1. Establishing a Finite Element Model of Roller System

There are many factors that cause roll fracture failure. It is confirmed that these rolls have no defects in the manufacturing process, and the working load is also smaller than the maximum rolling force. Therefore, the causes of roll fracture are analyzed from the viewpoint of structure and size design. In order to be closer to the actual force of the backup roll, half geometric model of the roller system is established during the simulation.

The structure of the backup roll is guaranteed to be the same as that given by the company. Part of the roll structure and key dimensions are shown in Figure 2. The center distance of the bearing is 1860mm. The connection between the barrel and the roll neck is a transition circle with a radius of 75mm. The structure of the work roll and the intermediate roll is simplified, and the barrel and roll neck size are retained. The key dimensions of each roll are shown in Table 1.

![Figure 2. Part of the structure of the backup roll](image)

| Roll Name   | Barrel Length(mm) | Barrel Diameter(mm) | Neck Diameter(mm) |
|-------------|-------------------|---------------------|-------------------|
| Work Roll   | 1150              | 310                 | 160               |
| Intermediate Roll | 1150              | 370                 | 190               |
| Backup Roll | 1050              | 950                 | 550               |

The backup roll material is the hypereutectoid alloy steel with Cr content of about 5%. The Cr5 steel has good hardenability and hardenability. The Young's modulus is 236,000 and the Poisson's ratio is 0.3.
The mesh is refined at the contact portion between the rollers and the portion where the dangerous section appears, that is, the transition portion between the roll body and the roll neck of the backup roll. The contact between the rollers is wedge-shaped mesh, and the rest is a hexahedral mesh. The mesh type is C3D8R. The overall roller meshing and local refinement are shown in Figure 3.

Figure 3. Mesh elements for the model (a) Whole structure of the model (b) Partial enlarged view of the roll neck (c) Partial enlarged view of the roller contact

Since the bending moment during the rolling of the six-high mill is mainly carried by the backup roll, when the strength of the backup roll is checked, it is considered to be subjected to the full rolling force. Set 75% of the maximum rolling force for the working load, that is, 9000KN. The backup roll sets a displacement constraint at the cross section of the bearing center. Both the work roll and the intermediate roll side are free ends.

2.2. Finite Element Simulation Calculation Results

Figure 4 shows the stress distribution obtained by the finite element analysis of the static load of the roller system. Figure 4 (a) is the stress distribution of the roll system, and Figure 4 (b) is the stress distribution at the top of the backup roll. In order to facilitate the observation of the deformation of the roller system, the deformation scale factor is 150. It is known from the calculation results that the stress concentration occurs at the root of the backup roll neck, which is the most dangerous point of the full roller system. When the rolling force is 9000KN, the maximum tensile stress at the root of the backup roll is 148.34 MPa.
2.3. Calculation of Safety Factor for Fatigue Strength of Backup Roll

When structure or size of the roll is improper, the stress concentration at the root of the roll neck may cause fatigue cracks. During the high-speed rotation of the rolling, the alternating load subjected to compressive stress and tensile stress acts for a long time, crack propagation will cause the roll to fracture. In this paper, the fatigue strength of the backup roll is checked by calculating the fatigue strength safety factor.

According to the formula (1) and combined with the finite element simulation calculation of the stress value at the neck of the backup roll, the fatigue strength safety factor can be estimated [6].

\[ n = \frac{\varepsilon_0 \beta \sigma_{\text{p}}}{K_{\sigma} \sigma_{\text{max}}} \]

(1)

\[ \sigma_{\text{p}} = 0.4 \sigma_b \]

(2)

\[ n = \frac{\varepsilon_0 \beta \sigma_{\text{p}}}{K_{\sigma} \sigma_{\text{max}}} \approx 1.38 \]

(3)

n–The safety factor; \( \sigma_{\text{p}} \)–The permanent limit of the standard sample material; \( \sigma_{\text{max}} \)–Maximum working stress; \( \varepsilon_0 \)–Dimensional coefficient; \( \beta \)–The surface quality coefficient, the surface of the backup roller is processed by grinding; \( K_{\sigma} \)–The stress concentration factor, the corresponding values of different roll sizes are different [7]. The values of these influence factors are shown in Table 2 [8].

| Table 2. Influence coefficient value |
|-------------------------------------|
| \( K_\sigma \) | \( \varepsilon_0 \) | \( \beta \) | \( \sigma_b \) (MPa) | \( \sigma_{\text{p}} \) (MPa) | \( \sigma_{\text{max}} \) (MPa) |
|-----------|-------|------|---------------|----------------|-----------------|
| 1.764     | 0.6   | 1    | 1500          | 600            | 148             |

Formula (3) is the safety factor of the roll according to the fatigue strength:

\[ n_1 = \frac{\varepsilon_0 \beta \sigma_{\text{p}}}{K_{\sigma} \sigma_{\text{max}}} = \frac{0.6 \times 1 \times 600}{1.764 \times 148} \approx 1.38 \]

When designing shaft parts, if the shaft diameter is greater than 200mm, the allowable safety factor according to the fatigue strength calculation should be between 1.8 and 2.5. However, the safety factor
of the support roll is 1.38, which is much lower than the allowable safety factor of 1.8. From this perspective, the roll is prone to fatigue failure.

3. Research on fatigue strength improvement of backup roll

3.1. Increasing the Fillet Radius

With reference to the design of shaft parts, this section studies three options for reducing stress concentration and improving fatigue strength by changing the structure at the transition between the roll body and the roll neck or increasing the size.

Backup roll has a compact structure, and the excessive radius of the fillet will change the existing structure too much. In order to keep the original roll structure as constant, the influence of the fillet radius at the roll neck on the stress distribution under the same rolling conditions is studied. In this paper, the fillet radius value is increased from 75mm to 85mm of the original structure.

The finite element calculation and analysis of the structure shows that the maximum stress at the root of the roll neck is 139.7MPa, and the stress concentration factor is 1.714. According to formula (1), the fatigue strength safety factor of the backup roll with a fillet radius of 85 mm is calculated:

$$n = \frac{\sigma_{\text{max}}}{\sigma_{\text{lim}}}$$

When the transition fillet radius is increased by 10 mm, the safety factor is increased by 8.7%, but the safety factor is still not achieve the allowable safety factor. Therefore, under the premise of keeping the existing structure and function unchanged, the method has limited improvement on the bearing capacity of the roll.

3.2. Increasing the Neck Diameter

The series of rolls with fatigue fracture, the roll neck diameter d is 550mm and the roll diameter D is 950mm, the ratio d/D is 0.58. In this part, the finite element calculation is carried out for the stress distribution of the barrel when the roll neck diameter d is 570mm, 590mm, that is, the d/D is 0.6, 0.62 and the transition fillet radius is 75mm and 85mm respectively. The corresponding maximum stress value and the calculated safety factor are shown in Table 3.

| d/D   | Fillet Radius(mm) | Stress Concentration Factor | $\sigma_{\text{max}}$ (Mpa) | Safety Factor |
|-------|-------------------|-----------------------------|-----------------------------|--------------|
| 570/950=0.6 | 75 | 1.772 | 134.7 | 1.51 |
| 590/950=0.62 | 75 | 1.797 | 118.8 | 1.68 |
| 570/950=0.6 | 85 | 1.702 | 126.9 | 1.67 |
| 590/950=0.62 | 85 | 1.712 | 113.2 | 1.85 |

It can be obtained from the calculation results that when the roll neck diameter is 590 mm and the transition fillet radius is 85 mm, the safety factor is greater than the allowable safety factor of 1.8. The fatigue strength condition is satisfied, and the roll neck is less prone to fatigue fracture.

The backup roll is mainly subjected to radial force during rolling. The bearing is usually selected from four-row cylindrical roller bearings. The outer dimensions of the bearing are large. Therefore, when designing the journal size, the bearing selection should also be considered. Through the above calculations, it is recommended to adopt a measure of appropriately increasing the diameter and the fillet radius of the backup roll neck to reduce the stress concentration at the root of the roll neck to improve the fatigue strength.
3.3. Changing the Structure of the Transition

In addition to the rounded structure, the barrel and the roll neck may be combined with a bevel and a rounded corner. Proper structural design can effectively increase the radial dimension of the transition, thereby reducing stress concentration. Figure 5 is a structural comparison diagram of two structures, wherein the structure 1 represents a rounded structure and the structure 2 is a structure with a bevelled and rounded corner.

In order to make as few changes as possible to the original structure, the fillet radius is 50mm, and the stress distribution at the top of the roll body is calculated by finite element calculation as shown in Fig. 6. The maximum stress value at the transition is 123.1 MPa and the safety factor is about 1.88, also greater than the allowable safety factor of 1.8.

![Figure 5. Structure comparison diagram](image)

![Figure 6. Stress distribution of structure 2](image)

It can be seen from the above analysis that a reasonable structure can effectively reduce the stress concentration and improve the fatigue strength of the roll.

4. Conclusion

In this paper, the finite element software is used to calculate the stress distribution of the support roller in which the company has fracture failure, and the safety factor calculated from the fatigue strength at the stress concentration. Then, three structural size designs that can improve the fatigue strength of the support rolls are discussed. The conclusions are as follows:

(1) If the rounded transition structure is still used, the fillet radius is increased by 5 mm and the safety factor is increased by 9%-12%. The diameter of the roll neck is increased by 20 mm, the maximum stress at the dangerous section can be reduced by 12 MPa to 15 MPa, and the safety factor
is increased by about 10%. The ratio d/D of the diameter of the journal journal d to the diameter D of the roll body should be greater than or equal to 0.62 to ensure sufficient fatigue strength.

(2) If the rounded corner structure of the transition is changed to a bevelled and rounded structure, the fillet radius is 50mm, and the angle of the bevel is 45°, the safety factor can be satisfied if the roll neck diameter is constant.

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