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FAILURE ANALYSIS OF THE GIRTH GEAR OF AN INDUSTRIAL BALL MILL

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ABSTRACT

This paper describes the major activities carried out in the process of failure analysis of the girth-gear of an industrial ball mill. Three dimensional finite element modeling and analysis of the mill were performed under various loadings including: charge weight, dynamic loading due to the charge motion, centrifugal forces imposed by partial rotation of the charge, and the driving force imposed by the pinion. The results of the stress analysis conformed to various characteristics of the existing cracks. It was concluded that the charge weight and the dynamic effects due to the charge motion inside the mill were the main causes of the stress built-up in the gear, and the contribution of the forces imposed by the pinion was only 20 percent. Finally, semi-elliptical cracks were modeled and grown through a variable stress field and the crack driving forces were calculated. The calculated crack growth rates were compared with those monitored over a period of 21 months for real cracks.

KEYWORDS: Failure Analysis, Girth Gear, Ball Mill, Stress Analysis, Fatigue, Life Assessment, Finite Element.

INTRODUCTION

The Girth-Gears of industrial Ball-Mills are up to 12 meters in diameter and over 90 tones in weight, with a manufacturing cost exceeding $500,000 (see Fig.1a). These types of gears are expected to have fatigue lives of 20 years and more. In this case history, within the first two years of operation, a few cracks initiated from certain locations between the gussets and the gear flange, and propagated towards the lightening holes, as shown in Fig.1b. For some cracks, the propagation paths were almost identical on both sides of the gear, while there were several cases of crack propagation on one side. Since the premature occurrence of several similar cracks in certain locations could be interpreted as the possibility of a faulty design, it was decided to perform a complete stress analysis of the mill using the finite element method. In similar analyses reported in the literature the main charge with a flat surface has been considered as a major source of loading [1,2]. Accordingly, due to the symmetry of geometry and loading, only one half of the mill has been analyzed. In our analysis we considered various sources of loadings including the charge weight with a slanted surface, which required a complete 3-D modeling and analysis. The analysis results clearly revealed the cause of failure, i.e., high stress built-up in specific locations adjacent to the gear flange, and conformed to various characteristics of the existing cracks, including their propagation paths. This paper also describes the assessment of the remaining life of the gear through modeling of crack growth in the high-stress region. In these analyses, semi-elliptical cracks were modeled and grown through a variable stress field, and the crack driving forces were calculated. The calculated crack growth rates were compared with those monitored over a period of 21 months for real cracks.

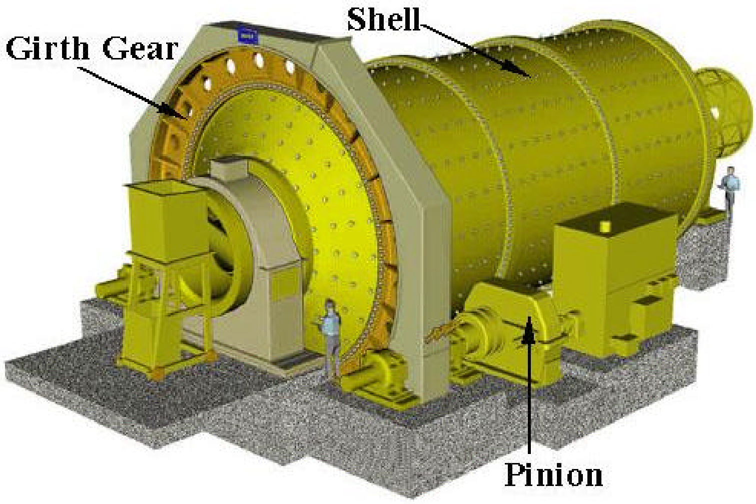

Figure 1: **a)** Schematic of an Industrial ball mill [5]. **b)** A typical crack in the rear side of the gear.

STRESS ANALYSIS OF THE GEAR

The modeling and analysis were performed using the LUSAS software. Figure 2a shows the three-dimensional finite element model of the mill, comprising of 8-noded brick elements. Figure 2b is a schematic presentation of the modeling of the charge motion inside the mill, which has been developed using the information presented in reference [3]. In general, the loadings included: charge weight, dynamic loading due to the charge motion, centrifugal forces imposed by partial rotation of the charge, and the driving force imposed by the pinion. The model was restrained in all directions at the location of the front trunnion (the gear side trunnion) while the rear trunnion was free to move only in the axial direction.

ANALYSES RESULTS

Figure 3 shows the distribution of the maximum principal stress component, S1, in the front face of the gear. The maximum amount of these stresses, 120MPa, occurs between the flat end of the gussets and the flange, at the location of 6 dclock. The distribution of this stress component conforms to the actual crack paths observed on the cracked gear (see Fig.1b).

The distribution of the minimum principle stress component, S3, in the front face of the gear showed a minimum amount of -65MPa, which occurred at the same region, but at a different location, i.e. at 4 o'clock. In fact the deflection of the shell, under the forces imposed by the charge motion, creates bending moments at different locations of the gear flange. These local bending moments cause tensile stresses at one side and compressive stresses at the opposite side of the critical region. As every point at the critical region around the mill, passes through these locations during the mill revolution, a cyclic loading is imposed which can lead to fatigue crack initiation and growth in this region. It is interesting to note that the charge weight and dynamic effects due to the charge motion inside the mill were the main causes of the stress built-up in the gear, and the contribution of the forces imposed by the pinion was only 20 percent.

Figures 4 shows the distribution of the maximum principal stress component, S1, in the rear side of the gear. It is clear that the stress levels are generally lower than the front side. However, the existence of two high tensile stress locations can create two loading cycles for every mill rotation.

Based on the results of the above analysis and considering the mechanical properties of the gear material shown in table 1, it is clear that the stress levels at the critical regions on the both sides of the gear are sufficient for fatigue crack initiation and growth in these regions. The results obtained from the above

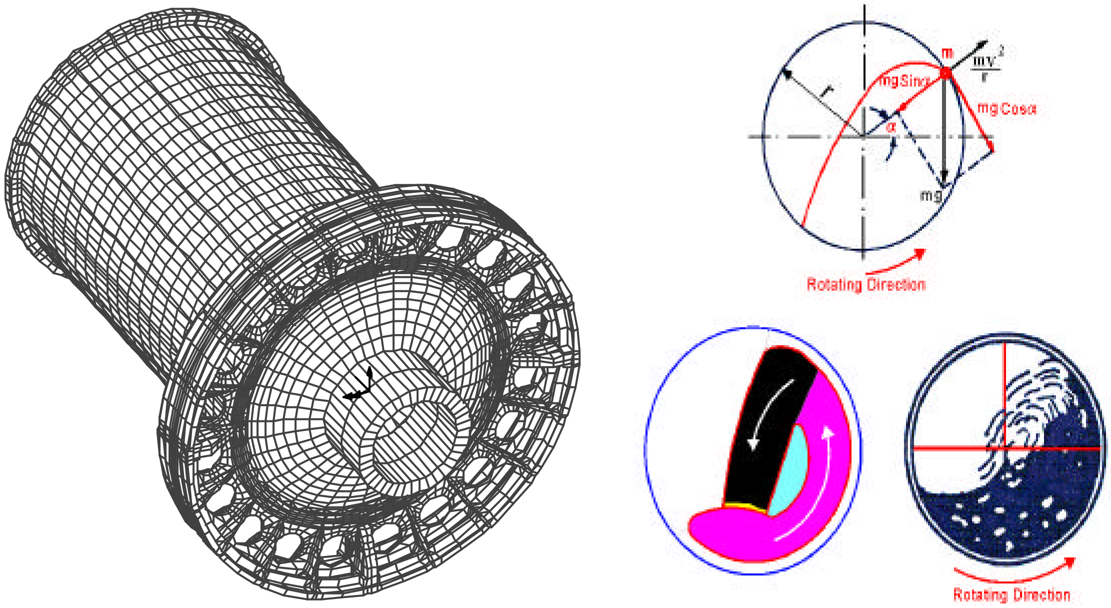

Figure 2: **a)** 3-D finite element model of the mill. **b)** Modeling of charge motion inside the mill.

analyses support the initial assumption of the cause of the crack initiation, and provide significant clues to the crack growth pattern and the shape of the final crack front. Based on the results of the above analyses, the remaining life of the gears were calculated using a Fracture Mechanics approach.

TABLE 1
Mechanical Properties of the Gear Material

Gear Material : Cast Steel (air quenched and tempered) C=0.35, Mn=1.5, Si=0.5, Mo=0.4	
Hardness (Brinell) : 240,	Tensile Strength : 800 MPa
Fatigue Endurance Limit (theoretical) : 300 MPa	
Endurance Limit (after surface, size, and other corrections) : 75 MPa	
Paris Law Coefficient, C : 1.2E-11	Paris Law Exponent, m : 2.7

LIFE ASSESSMENT OF THE GEAR

In order to assess the remaining life of the gear a failure criterion was required. The usual failure criterion in fracture mechanics is the onset of unstable crack growth. In this case, however, the stress built-up in the critical region was caused by deformation of the mill shell, which is in fact a displacement-controlled condition, where crack growth can give rise to stress relaxation. Hence, it was concluded that the cyclic crack growth would continue as far as the lightening holes. Accordingly, the remaining life was defined as the time required for the cyclic growth of the existing cracks up to the lightening holes, and was calculated by numerical integration of the Paris equation:

$$\frac{da}{dN} = C.(\Delta K)^m$$

Detailed investigation of crack growth in a variable stress field is time consuming, involving finite element determination of stress intensity factors (SIF) for a range of crack sizes. Hence, we started with an approximate method using the available SIF calibrations of semi-elliptical cracks, along with proper adjustments of the coefficients for the problem in hand. Figure 5 shows a comparison between the predicted crack growths using the above method with those measured over a period of 21 months on the real gear. It is clear that the calculated lives are significantly lower than the real life.

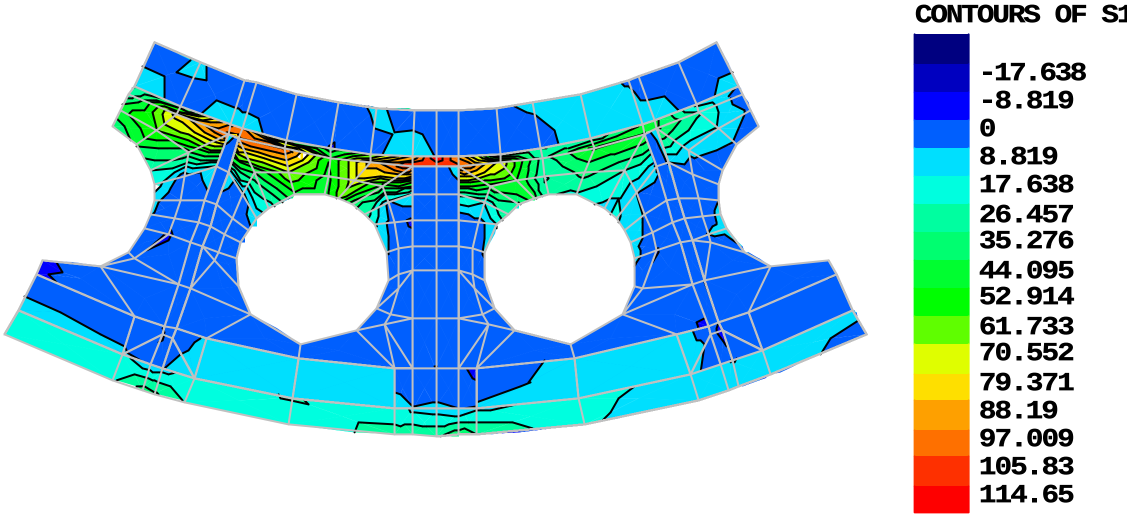

Figure 3: Contours of maximum principal stress component in the front side of the gear.

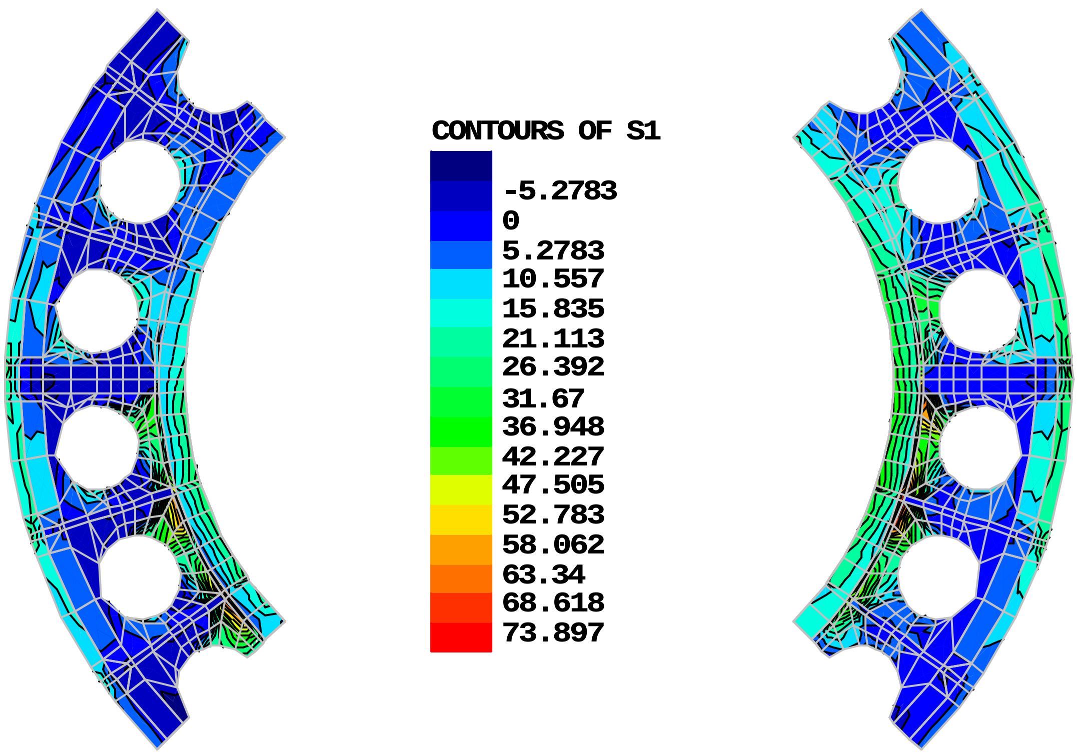

Figure 4: Contours of maximum principal stress component in the rear side of the gear.

In the next step, a numerical SIF calibration was obtained using the finite element method. Several procedures are available for numerical evaluation of stress intensity factors [4]. In this analysis we used stress and displacement point matching methods because of simplicity and consistency with our original model. As depicted in Fig.5, the calculated lives using the SIF calibration obtained from finite element analysis provided a better estimation of the real life. However, the fatigue lives predicted by both methods were quite shorter than the real life.

CONCLUSIONS

The problem of initiation and propagation of fatigue cracks in the girth gear of the ball mill under consideration was attributed to the existence of relatively high stress gradients at critical regions between the gussets and the flange. A 3-D finite element analysis of the mill revealed that the charge weight and dynamic effects due to the charge motion inside the mill were the main causes of the stress built-up in the gear, and the contribution of the forces imposed by the pinion was only 20 percent. The analysis results indicated that the stress built-up in the critical region was caused by deformation of the shell, which is in fact a displacement-controlled condition, where crack growth can give rise to stress relaxation. Hence, it was concluded that under normal operating conditions the crack growth would never become unstable and the cyclic crack growth would continue as far as the lightening holes. Accordingly, the remaining life, defined as the time required for the cyclic growth of the existing cracks up to the lightening holes, was calculated by numerical integration of the Paris equation. The calculated crack growths using an adjusted SIF calibration for semi-elliptical surface cracks were significantly higher than those measured over a period of 21 months on the real gear. Although the calculated lives using the SIF calibration obtained from finite element analysis provided a closer estimation of the real life, both methods overestimated the actual crack growth rates. This can be attributed to the inexact nature of our modeling and numerical analysis and/or the retardation phenomena like fatigue crack closure, which are operative in reality but were not considered in our analyses.

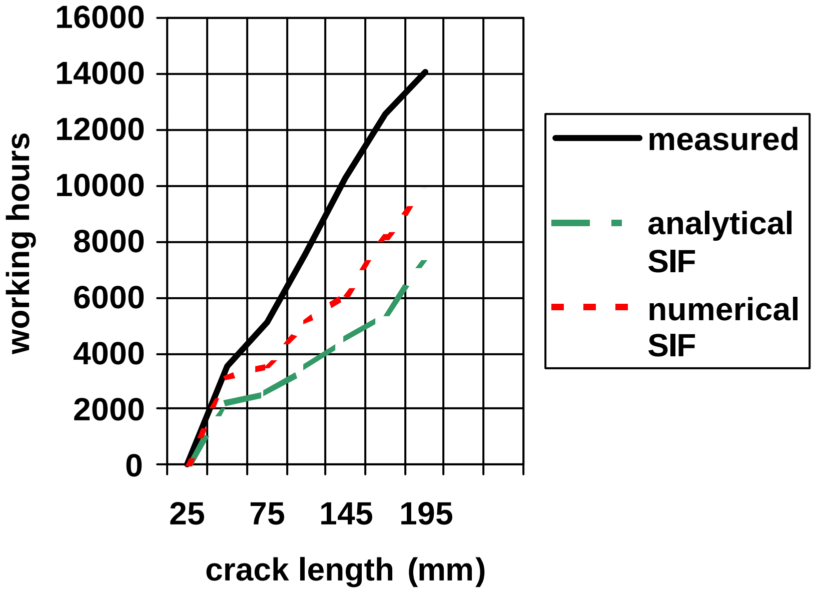

Figure 5: Measured and predicted crack lengths versus the working hours of the mill.

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