Matching Real World Results

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Abstract
Comparing measured results to those generated by analyses is often not straightforward – like fitting a square peg into a round hole, it can be done, but the fit may leave something to be desired. In the mining industry, the grinding mill is one of the most critical pieces of process equipment. Large units, capable of processing in excess of 100,000 metric tons per day, require extensive analysis at the design stage to ensure they can meet the rigorous fatigue conditions. Various analytical techniques are used involving axisymmetric models with harmonic elements as well as complex 3D models that represent such non-axisymmetric features as bolted joints. This paper compares measurements obtained from strain gauges mounted on the machine in operation to results predicted by the various analytical techniques. It provides an excellent example of the practical issues involved in evaluating modeling assumptions and in using multiple sources of information to arrive at a design solution.

Introduction
The grinding mill is a large rotating cylinder used in the mining industry to reduce the size of rocks such that the valuable mineral ores can be liberated. Figure 1 shows a large 38’ diameter SAG (semi-autogenous grinding), 27,000 hp mill installation, one of the world’s largest. The mill is fed with a mixture of ore and water, and contains steel balls that help to crush the ore as the mill rotates. Large grinding mills, similar to the one in the photograph, are driven by a motor that wraps around the mill. The massive size, dynamic behavior of the rotating charge, and the large forces induced by the electric motor make quantification and mathematical modeling of the applied loads a challenge. Furthermore, the presence of bolted joints and heavy, non-structural components, such as segmented steel liners that protect the insides of the mill, also lend a degree of uncertainty to the analytical process.

Figure 1. SAG mill installation
All of the ore processed by the mine passes through the primary SAG mill. Grinding mill downtime is extremely expensive – as much as $250,000 per hour in a modern 100,000 metric tons per day operation. Therefore, it is imperative that the designs be conservative and able to withstand the millions of cycles of fatigue to which they are subject in year after year of constant, round-the-clock operation.

Historically, the stress analysis of the grinding mill has been performed with the use of two-dimensional finite element models that assumed an axisymmetric geometry – e.g. a cross section of the mill structure modeled as a volume of revolution. Recently, with the increase in available in-house computer power, three-dimensional finite elements models have been more commonly used, particularly to assess the behavior of the non-axisymmetric mill features including the behavior of bolted joints. These two different analytical approaches often give dissimilar results, with the analyst left wondering which to believe.

As part of its development program, FFE Minerals, with the collaboration of Farnell Thompson Applied Technologies Inc., has recently performed a program of field test measurements to quantify the performance of the latest generation wrap-around motor-driven, large grinding mills that it has supplied. Although the tests also included measurements of fluid film-bearing performance and mill/motor/foundation system dynamic behavior, it is the measurement of actual operating strains in a grinding mill structure and the comparison to analytically predicted values that is discussed in this paper.

The grinding mill structure consists of one-piece cast hollow trunnions, segmented cast conical heads, and a welded steel cylindrical shell, all bolted together and supported on fluid film bearings. As illustrated in Figure 2, the rotor poles are mounted to an extension of the head segments.

![Figure 2. Motor poles mounted on extension of head](image)

The objectives of the strain gauge test program were to measure strains on critical areas of the rotating structure for correlation with the predicted strains from the finite element analyses – specifically in the high stress areas of the trunnion, heads, and shell (locations A, C and D in Figure 3). A secondary objective was to verify the modeling assumptions concerning bolted joint behavior – specifically loading in the region where the head-to-head segment radial joints meet the trunnion-to-head circumferential bolted joint (location B in Figure 3).
Finite Element Analysis

As stated above, two separate types of analysis were performed for the correlation of the strain gauge results. The first was a repeat of the axisymmetric or harmonic analysis typically performed during the design phase of the mill, but with input parameters specifically corresponding to the actual test conditions. The second was a more detailed, 3D, nonlinear analysis, with bolts explicitly modeled, bolt pre-loads applied, and opening across bolted joints allowed through the use of contact elements. Although 3D analyses of this type are often performed in evaluating mill failures, this particular analysis was tailored to correlate the strain readings in the head-to-head segment bolts for design verification purposes. This also provided a unique opportunity to compare strain results from the two analytical techniques with field data.

2D Axisymmetric (Harmonic) Model

Given the general axisymmetric nature of the grinding mill design, an axisymmetric modeling technique provides an efficient method of modeling the mill structure. Harmonic analysis techniques have been used to apply the non-axisymmetric loading resulting from the charge pressure and the bearing reactions as well as the motor eccentric forces, and to interpret the resulting strains and displacements. The use of axisymmetric analysis using harmonic techniques to account for non-axisymmetric loading permits the element mesh to be defined on a cross section of the mill geometry. This in turn allows the structure to be represented by fewer elements than would be the case for a solid brick element model, or conversely, to be modeled in significantly more detail.

A special class of ANSYS elements, referred to as harmonic elements, have been formulated so as to model the axisymmetric nature of the structure, while allowing the non-axisymmetric charge and bearing reaction loads to be input as a Fourier series. Each term of the series, representing a different mode, is input as a separate load step. In essence, the model is solved for each load step, producing a set of displacements and strains that comprise a set of results in the form of Fourier coefficients. Final results are then obtained by evaluating and summing these resulting Fourier terms at the desired circumferential angle.
Harmonic elements used in the 2D models were PLANE25 elements. These elements are iso-parametric and provide a linear stress gradient across their surface. Figure 4 shows a typical 2D finite element model of a grinding mill.

Figure 4. Axisymmetric analysis model

Non-axisymmetric loading, such as pressure generated from the multi-pad fluid film bearings on the mill trunnion, or charge load in the mill, are defined as a series of harmonic functions (Fourier series). For example, a pressure load $P$ is given by:

$$P(\theta) = A_0 + A_1 \cos \theta + B_1 \sin \theta + A_2 \cos 2\theta + B_2 \sin 2\theta + A_3 \cos 3\theta + B_3 \sin 3\theta + \ldots$$

where each term of the series is defined as a separate load step. In general, a vertical plane of symmetry that includes the mill axis is assumed and the sine coefficients are not required. Figure 5 illustrates a typical pressure profile acting on the mill trunnion supported by four hydrostatically-lubricated fluid film-bearing pads. The blue line is the actual pressure profile, while the purple line represents the pressure approximation with a forty-two-term Fourier series.

Figure 5. Fourier series approximation of applied bearing pressure
**3D Model**

For the 3D analysis, the mill structure was modeled with 8-node SOLID45 elements. Due to the complexity of the bolted joint interface and the desire to keep models within manageable sizes on an in-house computer, three separate 3D models were used in the analysis. The first explicitly modeled the bolts on the head-to-trunnion and head-to-head segment joints but modeled the balance of the bolted joints as solid connections. This was used for correlation with strain gauges on the trunnion, head, and head segment-to-head segment bolts. One half of the mill was modeled – a vertical plane of symmetry perpendicular to the mill axis was assumed as illustrated in Figure 6.

![Figure 6. 3D model used for head bolt correlation](image)

The second and third 3D models explicitly represented the bolts on the head-to-shell joints – one with the rotor present to simulate the drive (feed) end, one without the rotor to simulate the non-drive (discharge) end. These models were used for correlation of the strain gauge results from the critical corner where the cylindrical shell plate is welded to the bolting flange. Only one quarter of the mill was modeled; vertical planes of symmetry perpendicular and parallel to the mill axis were assumed as illustrated in Figure 7.

![Figure 7. 3D model used for shell strain correlation](image)
Bolts were modeled using PIPE16 elements. The contact surface between the head and shell flanges was modeled using TARGE170 and CONTA173 elements. These elements prevent penetration but allow separation in the opposite direction while also providing sliding resistance of adjacent surfaces depending on the clamping load.

**Bolt Preload**

Special effort was made to accurately model the behavior of the bolts in the 3D model. Two “one-bolt joint” test models were created. The first was a coarse model using beam elements (PIPE16) for the bolt with the clamped members having a mesh density roughly equivalent to that in the 3D mill model – Figure 8. The beam elements representing the bolts were connected to the outer edges of the bolt holes with a series of constraint equations. The second explicitly modeled the bolts using brick elements (SOLID45) while using a much finer mesh for the clamped members – Figure 9. In the second bolted joint fine mesh model, the contact/target elements (CONTA173 and TARGE170) were also used in both the washer-to-flange and bolt head-to-washer interfaces. For both “one-bolt joint” models, the mill flanges were modeled using 8 node solid structural elements (SOLID45) and the flange-to-flange mating surfaces were modeled with contact/target elements (CONTA173 and TARGE170).

![Figure 8. One-bolt coarse mesh test model](image1)

![Figure 9. One-bolt fine mesh test model](image2)
The bolt pre-load was applied in a two-step process, somewhat similar to the PMESH command. The bolt elements were effectively disconnected midway along the bolt and opposing axial forces equivalent to the desired preload applied across the interface. The first solution determined the displacement across the interface generated by the forces; the second solution applied these displacements as constraint equations. In order to maintain the continuity of the bolt at the split, coupled degree of freedoms (CP) were used for the two transverse degrees of freedom (DOF), and the three rotational DOFs.

The bolt working stresses resulting from an external joint separating force applied on the joined members were evaluated for both test models. The results are presented in Figure 10, where it is demonstrated that the selected combination of constraint equations and mesh density in the coarse test model generated working stresses in the bolt similar to those derived from the detailed test model over a range of external forces applied across the joint. This good correlation gave a high degree of confidence in the predicted bolt behavior using beam elements in the global 3D models.

![Bolt Working Stress as a Function of External Load](image)

**Figure 10.** One-bolt test models – bolt working load as a function of external joint load

**Loading and Boundary Conditions**

Certain loads and boundary conditions must be imposed on the models in order to properly analyze the structure. For these analyses, they have all been defined so as to approximate the actual constraint conditions of the mill. Gravity generates a body force for all of the structural components. The charge is typically applied as hydrostatic charge pressure acting on the inner, wetted surface of the mill. The wrap-around stator generates both a uniform force and a sinusoidally-varying eccentric force around the circumference applied to the rotor poles. Several nodes on the trunnion have been constrained so as to prevent rigid body motion, but any reaction force has been negated through application of a pressure distribution on the journal surface simulating the pressures generated in the bearing oil film.

For the axisymmetric analysis, displacement constraints have been modeled in each of the first two load steps, referred to as the “rigid body modes.” The first load step, representing mode 0 of the Fourier series, included a displacement constraint in the axial (V) direction and the circumferential direction (Z) of one node on the bearing thrust face. The second load step, representing mode 1 of the Fourier series, contained displacement constraints in the circumferential (Z) direction for the central node on the trunnion journal surface. Note that for the higher termed series, displacement constraints are not required because the sinusoidal functions are self-balancing.
These degrees of freedom supported the mill for the initial run. Reaction forces from the constrained nodes were transformed into Fourier series approximations of the bearing pressure distributions, which were then applied for a second run. In the second, or final run, the reaction forces at all constrained nodes approached zero. Therefore the desired equilibrium between the mill and the charge weight and the bearing reaction pressure was obtained, effectively removing the displacement constraints and simulating the physical situation of solely a pressure acting on the mill trunnion journal surfaces.

**Strain Gauge Testing**

**Test Program**

The greatest challenge in any correlation test program is to separate and quantify the unknowns that are always present to varying degrees in real world machines. A significant portion of the test program was specifically aimed at evaluating critical assumptions made in the analysis of grinding mills.

- **Mesh density** – Most structural analysts are familiar with the relationship between mesh density and predicted stresses or strains. For this reason, the mesh density was carefully controlled in all critical regions of the models with mapped meshing. In both the 2D and 3D analyses, a finer mesh was used in critical areas to allow direct extraction of strains in regions close to geometric features causing stress concentration. In the 3D analysis a constant mapped mesh in the critical regions provided uniform strain gradients, especially in the circumferential direction.

- **Bearing reactions** – One of the most significant parameters measured during the mill testing was the bearing pad pressure. Separate pressure transducers were mounted on the oil inlet lines of the four pads on each of the two bearings that support the mill. As these pads depend on hydrostatic lubrication to provide the oil film, there is a direct relationship between the oil inlet pressure and the supported load. Given the geometry of the bearings, it was possible to derive an accurate measurement of the vertical and horizontal reaction forces at each bearing. These were then used to deduce the varying charge weight and motor forces present during each test.

- **Charge simulation** – Grinding mills typically operate in the 70-80% critical speed range. The dynamics of the cascading charge, including centrifugal and impact loads, are typically ignored in the mill analysis and an equivalent static charge is assumed. In order to quantify the dynamic effects, tests were repeated with the mill at rest, inching (turning at 1 rpm), and operating at normal operating speed (10 rpm). As an example of the difficulties involved in analyzing real world machines, the charge load was never static, varying between tests as water and ore feed were fed to the mill.

- **Motor forces** – The stator exerts a force on each rotor pole that is roughly proportional to the inverse of the distance across the air gap. As the mill (i.e. the rotor) and the stator are never exactly concentric, net forces act on the mill. Tests performed with the motor on and the motor off allowed quantification of these forces based on load derivations from the bearing pad pressures as described above.

- **Non-structural masses** – One of the most difficult components to analyze in a grinding mill is the structural contribution of the mill liners. These are large steel castings that protect the inside of the mill. A typical shell liner, of which there were two across the length of the mill and sixty around the circumference, weighs roughly two tons. These liners are modeled as non-structural mass – either by changing the density of the adjacent structural elements or, in the case of the 3D analysis, by using a surface effect element – SURF154 – an element that has density but no stiffness. In practice, these heavy liners contribute to the structural integrity of the shell especially in the initial period when the liners are new. However, from a design point of view, their structural integrity cannot be relied upon as they wear over time and occasionally break.

- **Bolted joints** – There are two issues of interest concerning bolted joints. The first involves predicting strains in the members adjacent to the bolts; the second involves predicting strains in the bolts themselves. In the axisymmetric analysis, all circumferential bolted joints were modeled...
as basically solid connections. However, certain nodes along the interface were “released” in the original design analysis to better simulate joint behavior. In the 3D analysis, the bolted connections were explicitly modeled to the extent discussed above. Bolt strains in the axisymmetric analysis were estimated by classical methodologies that rely on an extraction of the forces and moments acting across the flange interface (typically using the FSUM command). Forces and moments in the hoop direction are not explicitly available in an axisymmetric analysis, so must be estimated from elemental stresses.

**Gauge Locations**

There were severe limitations on the number of gauges that could be used to measure strain on the grinding mill. First, the time available to mount the gauges on the mill was limited to one eight-hour shift. Second, the number of channels that could be monitored for any given test was limited to nineteen by the data logging equipment. As such, it was decided to use primarily bi-axial gauges (instead of tri-axial) oriented in the axial and circumferential (hoop) direction. This would allow direct correlation of strain components, but not the determination of principal strains. Typically the axial or radial direction was designated “a” and the hoop direction was designated “b.”

The test program was broken into two parts – the first measured strains on the head and trunnion of the mill including the head-to-head segment bolts, the second measured strains on the shell. Eighteen individual gauges, plus an optical position indicator, were monitored in each test by the data logger that was attached to and rotated with the mill. The bearing pad pressures were monitored by a separate stationary system which shared the optical position indicator.

As illustrated in Figure 11, gauges R1, R2, R3, R5, R6, and R7 were mounted in a line measuring strains in the trunnion knuckle, on the trunnion flange OD and at the head thickness transition. The gauges were located approximately mid way between head splits (i.e. at 45 degrees to the head-to-head segment bolted joint).

![Figure 11. Gauges on trunnion and head](image)

Gauges B1, B2, and B3 were mounted on the three innermost head-to-head segment bolts as illustrated in Figure 12. For these locations the individual gauges “a” and “b” were located axially on either side of the bolt, 180 degrees from each other. During installation the bolts were oriented such that the two gauges were arranged in a line perpendicular to the conical inner surface of the heads. In this manner the gauges were able to measure axial strain and allow the separation of pure tension from bending-induced strain. Gauge “a” was located on the inner side with gauge “b” on the outer side. Gauge 4 was a uni-axial gauge mounted on the trunnion flange OD in line with the head-split joint.
Gauges R8, R9, R10, and R11 were mounted on the shell corner weld at feed (drive) end as illustrated in Figure 13. Gauges R13, R14, R15, and R16 were mirrored on the shell corner weld at the opposite, discharge (non-drive) end. Gauge 12 was located in the middle section of the feed-end shell can.

**Learning from the Correlation**

For the correlation, adjustments were made to the model parameters in order to reflect the test conditions. These include:

- increased cast head thickness to reflect measured thickness
- used best estimate of actual liner weights at the time of the test
- adjusted charge weight and motor forces to give observed bearing reactions
- adjusted bearing pressure distributions to reflect observed load sharing between pads

The average deviations between the strains predicted by the axisymmetric and 3D analyses and those measured during the “mill running” tests are presented in Table 1. Note that a positive deviation value reflects analyses predictions higher than test results.
Table 1. Average deviations between predicted and measured strains

<table>
<thead>
<tr>
<th></th>
<th>Axisymmetric</th>
<th>3D</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Trunnion and head</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Radial</td>
<td>-1%</td>
<td>-19%</td>
</tr>
<tr>
<td>Hoop</td>
<td>-6%</td>
<td>+5%</td>
</tr>
<tr>
<td><strong>Shell</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Axial</td>
<td>+41%</td>
<td>+89%</td>
</tr>
<tr>
<td>Hoop</td>
<td>+32%</td>
<td>+59%</td>
</tr>
</tbody>
</table>

The correlation in the trunnion and head is considered as good, especially for the axisymmetric analysis where the correlation is within 1% in the radial direction and 6% in the hoop direction. The correlation in the shell, for either the axisymmetric or 3D analysis, at first glance, would appear poor. However, the high average deviations on the shell are somewhat deceptive, as the analysis techniques have known conservatism – see the discussion below. The correlation results can be filtered to provide insight into critical mill design issues:

**Charge dynamic effects** – The correlation of the strain ranges measured during running and at rest and those predicted by the various FEA models for the trunnion and head are presented in Figure 14. The at rest test values are, in most cases, slightly higher than the running values, but this is mostly due to the higher charge levels and bearing reactions observed during those tests. The test results confirm the basic modeling assumption that static load conditions (charge assumed level, dynamic forces from the charge ignored) are representative of normal operation and can be used for design.

**Comparision of strain ranges on trunnion and head**
(At rest, running and analytical)

Figure 14. General correlation on the trunnion and head

**Bearing pad reactions** – A typical comparison of the measured strain profile to that predicted by the axisymmetric and 3D analyses, on a per revolution basis, for gauge R3a located in the trunnion
knuckle radius, is presented in Figure 15. The slight difference in curve shape between the analysis results and test measurements are because of the different assumptions in load sharing between pads. The 3D analysis assumed equal load sharing between pads; the axisymmetric analysis assumed 10% higher load on the inner pads and 10% lower load on the outer pads. The actual hydraulic pressures indicative of pad load sharing, as seen by the pressure transducers measuring oil pressure to the pads during the test, were 48.2, 51.1, 54.1, and 54.2 bar. Note that the actual pressures also reflect a net horizontal motor force, which is not accounted for in the analytical models.

![Figure 15. Per revolution comparison – predicted versus measured strains in trunnion knuckle radius – gauge 3a](image)

**Bolt loads** – The correlation on the bolts was performed using results from the 3D analysis where the fasteners were explicitly modeled, the preload was applied and contact elements were used to allow joint opening. The magnitude of strain range seen by the bolts correlates well with the 3D analysis results, as illustrated in Figure 16. However, when the measured strains are broken down into a mean and bending component for each bolt, a discrepancy in the load pattern between the predicted and measured results, on a per revolution basis, appears, as illustrated in Figures 17 and 18. It is hypothesized that this difference may be due to some residual joint opening that was noted during manufacture and installation. Of more importance, the measured results as well as those of the detailed 3D model show that there is load shedding from the head-to-head segment joint into the trunnion. The hoop strain measured on the trunnion flange OD by gauge 4 in line with the head-to-head segment split is higher than that measured by gauge 5, at the same location but 45 degrees from the splits. The implication of this is that forces extracted from a model that doesn’t include detailed evaluation of contact in the vicinity of a bolted “tee” configuration where a radial split meets a circumferential split are overestimated. The previously used analytical technique of extracting forces from either the solidly connected 3D model or elemental stresses from the axisymmetric model to predict bolt loads at this location is, therefore, not valid.

**Nominal strains** – As illustrated in Figure 19, the liners act as a bridge between the shell outer and inner flanges – effectively unloading the shell. Their structural contribution is more pronounced on the feed end where the liners are closest to the flange.
Figure 16. Strain range comparison in head-to-head segment bolts

Figure 17. Mean comparison in head-to-head segment bolts
Figure 18. Bending strain comparison in head-to-head segment bolts

Figure 19. Shell liner configuration

Figure 20 illustrates the correlation between the measured and predicted strains on the shell. The major analytical overestimation of strain, or expressed differently, unpredicted drop in measured strain, especially on the feed end shell gauges 8 through 12, is attributed to the structural contribution of the liners that is ignored in the analyses. Table 2 summarizes the differences at both ends of the mill for a “hot spot” strain calculation at the weld toe. As can be seen the prediction on the discharge end, where the liner effect is at a minimum, is quite good for the axisymmetric analysis – within 4%. On the feed end however, where the liners will have a significant influence, the lack of liner stiffness in the analysis is seen to significantly overestimate the strain in the critical shell weld. Note that during the test the shell liners were relatively new and at close to their original design thickness. Over time they will wear and, as was stated earlier, their structural contribution will diminish. As such, they have been ignored in the analysis, which has resulted in a known conservatism in the analytical results.
The design of the shell is governed by welding codes such as BS7608 where “hot spot” stresses at the toe of the fillet reinforced corner weld are extrapolated from the stress gradient approaching the geometric discontinuity caused by the weld. In an axisymmetric, linear, analysis, this “hot spot” stress can be affected to some degree by the modeling assumptions used for the adjacent bolted joint. Figure 21 illustrates the effect of releasing nodes on the inside of the discharge head-to-shell joint in the axisymmetric analysis to better match the strain gradient along the shell as measured by gauges 15 and 16. The effect of this release on the other end of the weld toe on the shell flange is represented by the predicted strain for gauge 13. Some compromise is necessary, but the measured strains have been used to define a radius of node release for the best correlation of analytical and measured results. This radius has in turn been used to calculate the angle “$\beta$” of an equivalent “cone of deformation” under the bolt washer face as illustrated in Figure 22.
Conclusions

The writers have been involved in the design of grinding mills for the last three decades. In that period they have had the opportunity to strain gauge a number of mills of different sizes and configurations. To some extent, there has always been some discrepancy between measured and predicted results. This has led to an evolution in analytical techniques to better reflect desired conservative design principles. The correlation on this particular grinding mill, because of its size, supporting bearing design and wrap-around motor, has led to a further evolution in this process.
Several specific conclusions may be drawn from this correlation exercise:

- The axisymmetric analysis method gives a better overall correlation with the strain gauge results than does 3D modeling. This confirms that the higher mesh density available using this technique more than compensates for the extra features available in 3D analysis, such as explicit modeling of bolts and specific application of non-axisymmetric loads. The axisymmetric analysis should continue to be the design tool of choice for the grinding mill structure, supplemented by 3D analysis of specific regions that are not amenable to the axisymmetric technique such as radial and axial bolted joint splits.

- The test results have confirmed the small differences between static and dynamic results – specifically, between readings taken with the mill operating and those taken with the mill at rest. This confirms the design assumption that the charge may be modeled as hydrostatic pressure from a horizontal fluid pool and dynamic action from the charge can be ignored.

- In general, the analytical results have accurately estimated strains in the trunnions, slightly underestimated strains in the heads, and overestimated strains in the shell. Comments on the causes for these differences have been provided, and include manufacturing variations, necessary modeling simplifications, as well as the effect of liners on the actual stiffness of the structure.

- Measured bolt fluctuating loads in the head-to-head joint have been shown to be significantly less than estimated by previous simplified methods. Although discrepancies between measured and predicted bolt loads have been attributed primarily to a lack of full contact in this joint during installation, this condition has been shown to have little effect on the magnitude of the bolt loads.

- A contact angle for an equivalent “cone of deformation” has been defined using the correlation of analytical and measured strains to provide guidance in modeling the bolted head to shell connection in an axisymmetric analysis.