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FAILURE ANALYSIS OF BALL MILL GEARS

by

Chris Meimaris, Managing Director, EAnD, Brisbane, QLD, Australia
Michael Duncan, Associate, EAnD, Brisbane, QLD, Australia
Leigh Cox, Engineering Manager, Cadia Hill Gold Mine, NSW, Australia

ABSTRACT

The pinion and girth gears of two 9 MW ball mills exhibited severe scoring within five months of commissioning. Large temperature fluctuations in the gear set were noted in the months prior to the failure being discovered. The damage occurred in a three-week period on both mills. This paper presents the investigation undertaken to determine the cause of the failure. Lubrication failure, torsional vibrations, alignment and gear stiffness were all considered. It was found that the differential stiffness across the face of the girth gears due to casting and structural features, together with sudden power draw changes were the proximal causes of failure.

INTRODUCTION

The Cadia Gold Mine grinding circuit was commissioned in 1998. The circuit consists of a 40 ft, 20 MW SAG mill and two twin-pinion 22 ft, 9 MW ball mills. At the time of commissioning, both the ball and SAG mills were the largest mills in the world. During commissioning, both pinions gears on both ball mills exhibited excessive excursions in the thermal gradients across the pinion. The incidence of these excursions seemed random. Commissioning reports show that the thermal gradient

sometimes exceeded 50 °C (90 °F). Thermal gradients are often a sign of pinion to gear misalignment. Attempts to reduce the thermal gradient by realigning the pinions proved unsuccessful. The commissioning team reported that the temperature gradients could only be reduced to acceptable levels after the mills were shut down long enough for temperature of the pinions to reduce to ambient. When the mills were restarted, the temperature gradient would be within the manufacturer's normal range. This "normal" gradient would then continue for many hours or days and then, without any warning, an excursion in thermal gradient would recur. This situation persisted but was finally thought to be under control at the end of commissioning. Soon after handover, the thermal excursions resumed. In September 1998, an inspection revealed that the gears and pinions on both mills were severely scored. The damage on one gear was slightly worse than the other but in both cases the damage was very similar. A photograph of one of the damaged pinions is shown in Figure 1.

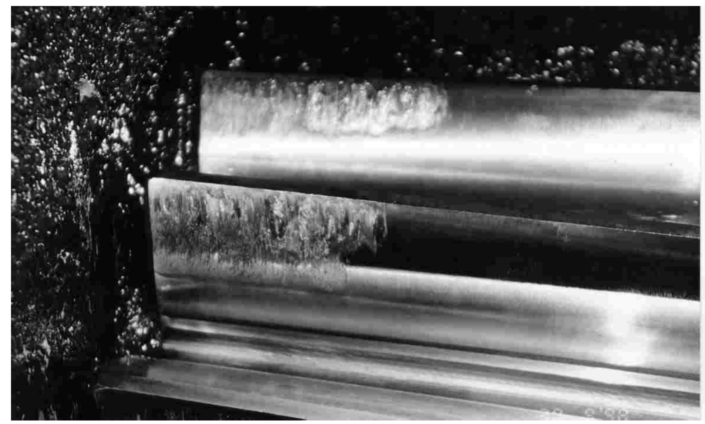

Fig. 1 – Pinion Tooth Damage

The investigation to determine the root cause of the gear damage is described in this paper. Drive-train dynamics, pinion forces, spent gear lubricant build-up, gear lubrication viscosity, operational history and gear design were considered as possible causes of the damage. Each of these causes will be discussed in the following sections. Thermal and coupled thermo-structural finite element analyses of the gears were also undertaken to assist in determining the cause of the failure. The results from these investigations are also presented. The basis of determining the root cause of a failure is then defined and applied to the present gear damage analysis.

ANALYSIS OF GEAR DAMAGE

The damage to the gears was very localised. It was limited to a region within 100 mm of the ends of the gear teeth. The damage was typical of scoring caused by the metal-to-metal contact due to lubrication breakdown. However, the lubricants used on the gears had been used in the past on other, more highly loaded gear sets by the Cadia engineering staff without any problem. It was necessary to determine why the lubricant broke down so that further damage could be avoided. Some of the possible causes considered during the root cause analysis are presented below.

Misalignment

Misalignment between mating gears results in a thermal gradient across the gear face. This gradient can be measured while the gears are operating and it is often used to determine the severity of misalignment. Initially, it was thought that misalignment was the cause for the temperature gradients, although this would not explain the sudden increase in thermal gradient that was typical of the temperature excursions observed during commissioning.

Commissioning engineers who observed the thermal excursions attempted to stop the temperature rises by stopping the mill and realigning both pinions according to correlations of temperature gradient and misalignment supplied by the gear manufacturer. The mill would then be restarted after shimming the pinions but the thermal gradient would continue to rise in the same sense (direction) despite the change in pinion alignment. It was found that the only way to stop the thermal gradient and overall temperatures rising was to stop the mills and allow the pinions to return to ambient temperature. Upon restarting of the mill, the pinions temperature profiles would show a severe thermal gradient in the opposite sense to the gradient that instigated the change in alignment, that is, the pinions were now misaligned in the opposite direction. The engineers than had to remove *all* the shims that they had added to the pinions during the realignment before the gradient would return to acceptable levels. Thus the pinion alignment would be need to be set to the position that it was prior to the thermal excursion having occurred so that the gears could run properly. After having performed this procedure several times, the engineers decided to eliminate the re-shimming step when a thermal excursion occurred. They shut down the mill and waited until the pinions reached ambient temperature and then restarted. Upon restart, the gears would operate with satisfactory thermal gradients until the next thermal excursion.

The behaviour of the gears was not typical of misalignment. Re-shimming the gears should have eliminated the thermal gradient if misalignment was the cause. Therefore, misalignment was rejected as a contributory cause of the damage.

Drive train dynamics

The next possible cause considered was torsional vibration. Torsional vibration can cause premature gear damage [1]. It was decided to measure the torque in the pinions in order to assess torsional vibration may have caused the damage.

Strain gauges were attached to both pinions shafts on both ball mill gears and the torsion in the shafts recorded. The maximum torque measured during steady state operation was 210 kNm. This was within the design torque for the gears. A frequency spectrum of the torque signals showed that the principal frequencies were the pinion running speed, the third harmonic of this speed and the tooth mesh frequency and that the magnitudes of these frequency components were typical of most twin pinion ball mills.

Torsional measurements were undertaken several times during the 12 months after the damage was observed. Results proved to be very consistent between tests. In one case, the dynamic torques were measured for a period of three weeks to determine if excursions in the torque were occurring that may not have been picked up by the short-term measurements. Results from these tests are showed that the torque in the pinions was very steady over the three weeks of measurement. The only variations in torque fluctuation amplitudes were those related to changes in the power drawn by the mill. The results from the long-term tests were generally identical to those obtained from the short-term tests over the previous twelve months. This showed that the mill drives were very stable.

Only one set of torsional tests showed any evidence of unsteady torsional behaviour. This set of data was from a test that was performed approximately two weeks after the gear damage was first observed. The torsional signal recorded during these tests exhibited spikes at random times as shown in Figure 2. The signal showed an increase in torque once per revolution of the pinion for a short time and then the torque signal returned to steady state. This behaviour was not noticed in any of the tests taken after this time. It is possible that these torque spikes may have been evidence that metal was being removed from the gears during operation.

The generally unremarkable nature of the results obtained from the torsional measurements indicated that torque fluctuations in the drive train could be rejected as a contributory cause of the gear damage.

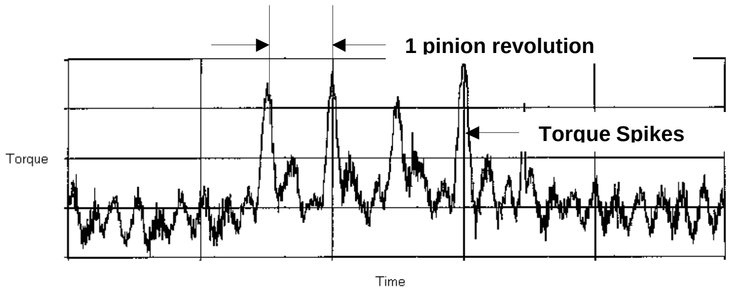

Fig. 2 –Spikes in Pinion Torsional Vibration Signal

Spent lubricant build-up

The gear manufacturer observed spent gear lubricant build-up during a site visit and considered this a possible cause of the gear damage. The build-up occurred as spent lubricant deposited on the sides of the gear guards. The deposit thickness increased until the side of the gears rubbed against the spent lubricant. Lubricants such as “blackjack” used on open gearing tend to cake and harden once deposited. The temperature of a gear rubbing against such a deposit may increase. However, the synthetic lubricants used at Cadia (Ceplattyn KG10 HMF1000 and KG10 HMF2500) do not harden and cake in the manner that “blackjack” does. They remain as soft greases and hence frictional energy build-up is negligible. Furthermore, spent lubricant was noted on both sides of the gears but the damage was confined to one side only. The spent lubricant argument was not considered a plausible mechanism for causing the damage.

Gear lubricant viscosity

Another possible cause for the damage was lubricant selection. The argument was that the viscosity of the lubricant was insufficient to prevent metal-to-metal contact. Calculations were undertaken by a specialist consultant to assess the suitability of the lubricant used in and on the gears. The manufacturer had

thought that Ceplattyn KG10 HMF1000 was used on both the gears of both mills and that after the damage, the lubricant was “upgraded” to KG10 HMF2500, a lubricant with higher viscosity. The lambda factor for the gears was determined for each of these lubricants as follows:

- HMF1000: $\lambda = 1.0$
- HMF2500: $\lambda = 1.75$.

The lambda factor is the ratio of the EHD film thickness to the surface roughness. A factor of greater than 1.0 will ensure that metal-to-metal contact will not occur. Allowing for a safety factor, it was stated that a lambda factor of 1.5 should be sufficient for open gears. Thus HMF1000 was considered inadequate but the HMF2500 lubricant was considered suitable for the installation. The manufacturer then thought that the gears failed due to inadequate lubricant viscosity, ie, the HMF1000 was an inappropriate lubricant.

This viscosity argument was flawed for the following reasons:

1. Firstly, the assumption that the gears on both mills were operating using HMF1000 when the damage occurred was incorrect. The mine had changed one of the mills to HMF2500 to assess the effect of lubricant viscosity on the temperature excursions. At the time when the lubricant was changed to HMF2500, there was no damage on either set of gears.

2. The damage on the gears with the HMF2500 lubricant was significantly worse than the gears with HMF1000. If the viscosity argument was correct, no damage should have been observed on the gears operating with HMF2500.

3. The lambda factor for HMF2500 was shown to be adequate and thus the viscosity of this lubricant was sufficient for the imposed duty.

4. Other, more highly loaded twin pinion drives (up to 12.5 MW) have run with HMF1000 very successfully for some years. Cadia engineering personnel had used this product in the past and under some extreme conditions, which were sufficient to cause tooth cracking, yet no lubrication failure had occurred.

Inadequate lubricant *viscosity* was rejected as the root cause of gear failure.

Gear support structure design

The damage on one set of gears on one ball mill was on the drive side whereas the damage on the other ball mill gears was on the non-drive side. Pinion temperature measurements showed that the temperatures were consistently higher on the ends of the pinions at which the damage occurred despite many attempts to realign the gears so that there would be no end-to-end temperature gradient.

The mills are mirror images of each other and effectively identical. However, after some detailed inspection, it was found that the girth gears had not been set up the same on both mills. The gear cope on one mill was set up on the drive end of the gear but on the other mill, gear had been rotated and the cope was set up on the non-drive end. The gear damage coincided with the cope side of each gear. It was decided to investigate if the gear casting features could be a contributory cause of the damage.

It was thought that the gear deflection due to radial and tangential load may be affected by the differential stiffness across the gear face and that this caused preferential loading on the ends of the gear where the damage occurred. Falk [2] and Merrit [3] indicate that the alignment of a pair of mating gears is determined, in part, by the deflections of the gears and their supporting shafts and structures; this encouraged further investigation.

The approximate girth gear profile is shown in Figure 3 page II-91. The gear has a Y-shaped supporting structure and the cope end of the gear is almost twice as thick as the drag end. When a pinion applies a torque to the gear, the differential stiffness of the gear will result in a higher load being taken by the ends of the gear opposite the webs due to the greater stiffness of the gear in this region. Furthermore, the cope end of the gear will attract more load than the drag side, as the cope side is the stiffer. Therefore, if a uniform stiffness pinion was used to apply load to the gear face, the contact load would be greatest near the web on the cope side of the gear. Temperature would also be greatest at that end and the gears would appear to be out of alignment even if their mechanical alignment was perfect.

Pinion temperature profiles over a one-year period from July 99 were interrogated to determine if there was any evidence of preferential load in the gear sets. The results obtained from Ball Mill 1 are shown in Figure 4 p. II-92. In this figure, the temperatures across the face of the pinions at five discrete locations are plotted against time. It can be seen that both pinions showed consistently higher temperatures on the cope end corresponding to the non-drive end of the girth gear. The maximum

temperature usually occurred at the NDC location inboard of the damaged area of the pinions. If the differential stiffness theory is correct, then it would be expected that the maximum temperature would occur at the ND location of the pinion, ie, at the end of the tooth where the damage occurred. However, the temperature records used in generating Figure 4 were all recorded *after* the damage to the gears had occurred. Material had been removed from the gears both by the damage and by the linishing of the gears performed as part of the pinion repairs. Thus, the contact force at the damaged sections of the pinions would be lower than it would have been prior to the damage having occurred. This explains why the maximum temperature generally occurs inboard from the non-drive end for Ball Mill 1. A similar analysis of gear temperatures was performed for Ball Mill 2. Results were almost identical except that highest temperatures were at the DC location inboard of the gear cope.

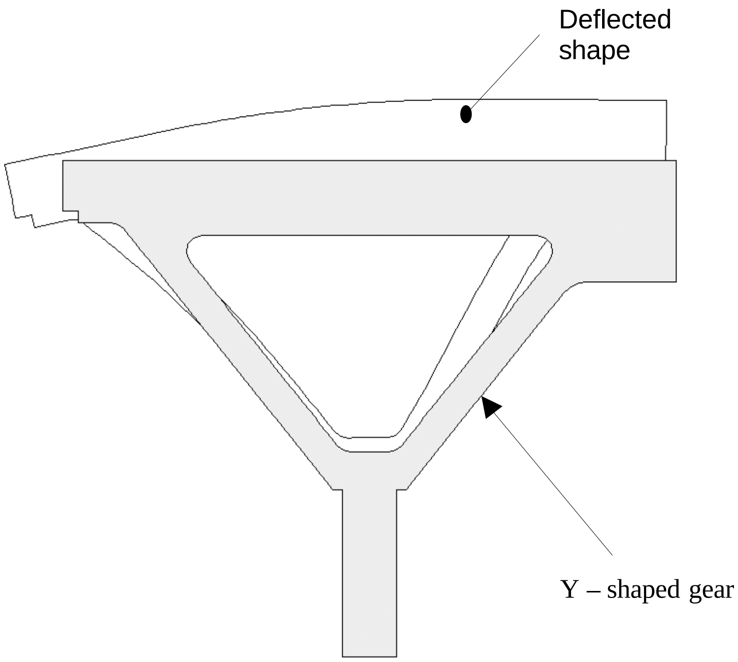

Fig. 3 – Gear Cross Section and Deflected Shape due to Uniform Thermal Input

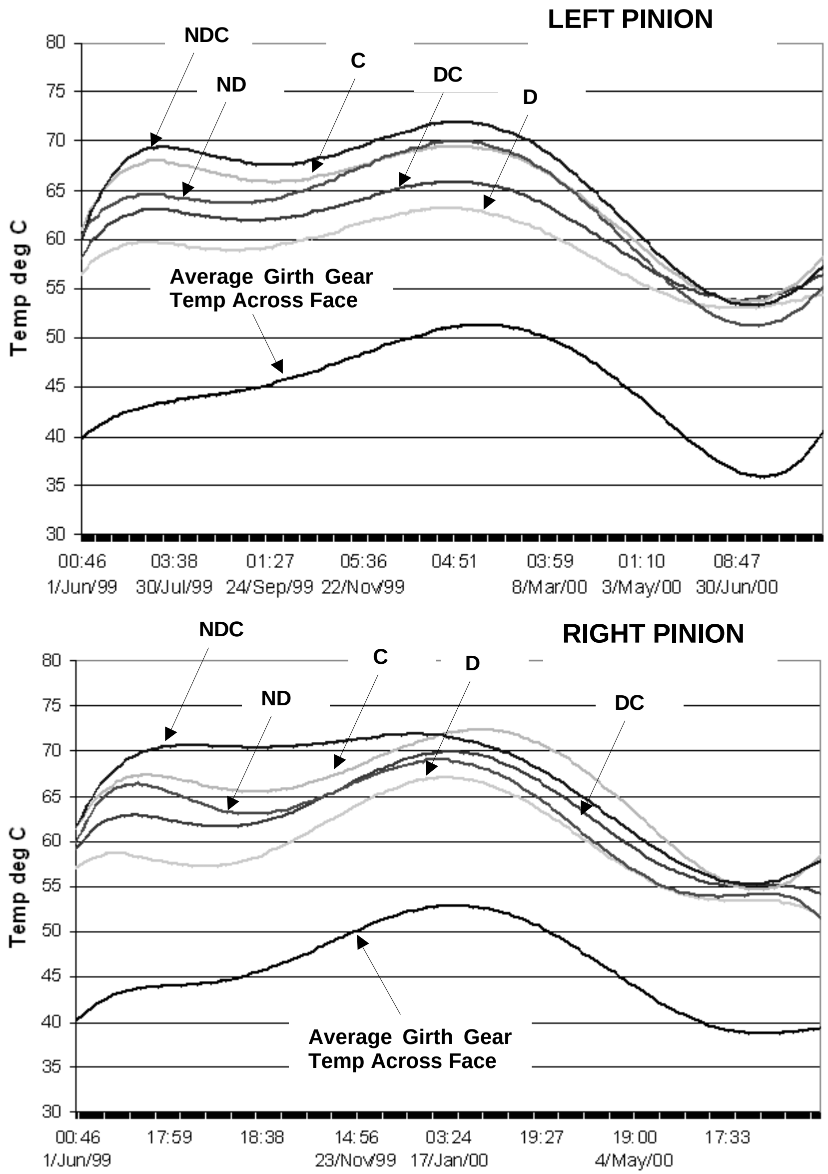

Fig. 4 – Mill Pinion Temperatures Over 12 Months
Legend – measurement locations on pinion tooth surface
D – drive end, ND – non-drive end, C – centre
DC – half way between D & C locations
NDC – half way between ND & C locations

The analysis of the gear temperatures does not explain the random temperature excursions that were observed. The mine's engineers had noted that the temperature excursions had occurred when there was a step change in the mill power draw. This often occurred when operators kibble loaded balls into the mills rather than using the ball charging mechanism provided or when the mill was ground out. These relatively fast changes in the power draw result in similar increases in the gear to pinion contact load. The increase in load would also result in an increase in thermal energy into the girth gear. As the contact load was highest at the cope end of the gear, it was reasoned that the temperature rise due to the increased load would also be greater at the cope end of the gear. The temperature excursions could then be explained by thermal ratcheting as follows:

(a) The load in the mill would increase relatively quickly due to an external influence such as kibble loading or grind out.

(b) The energy input into the gears would rise as a function of the load increase.

(c) The girth gear would then expand but the expansion would be non-uniform with the greatest expansion occurring at the cope end where the temperatures were higher.

(d) This non-uniform expansion would result in an increase in the load at the cope end and thus the temperature at that location would rise fuelling further increase in contact load.

(e) The load increased to a point where the lubricant film failed and scoring due to metal-to-metal contact occurred.

It was considered that thermal ratcheting described in items (a) to (e) was the most likely mechanism that caused the failure of the gears. However, it was not certain how the girth gears would expand under thermal load – item (c) – and if the expansion of these gears would result in an increase in the contact load – item (d). Finite element analysis was used to assess these issues as described in the next section.

FINITE ELEMENT ANALYSIS OF THE GIRTH GEARS

Two finite element analyses were performed to determine the behaviour of the girth gears under load. The aim of the first analysis was to determine how the girth gears expanded due to a thermal energy input.

The aim of the second analysis was to assess how the casting features (the cope and drag) and the Y-shaped structure affected load distribution across the gear face and how thermal expansion of the gear would change this load distribution.

Thermal analysis

A simple analysis of the girth gears was devised to assess the thermal behaviour of the girth gears. The gear was considered as an axisymmetric structure as shown in Figure 3 page II-91. A *uniform* thermal energy input was applied across the face of the gear. The deflected shape of the gear determined from the model is also shown in Figure 3. It can be seen that the cope end of the gear deflects more than the drag end. This result was sufficient to show that the assumption in item (c) in the previous section was correct, ie, that the girth gear would preferentially expand at the cope end.

Contact analysis

The analysis of the contact stresses was more difficult. Modelling the entire girth gear, the pinions, the interaction between the gear and pinion teeth and the supporting structures was considered too expensive, thus an idealised analysis was developed. The model used for the analysis considered only a single sector of the girth gear between the web gussets as shown in Figure 5. The sector comprised of a the gear bolting flange, the two "Y" webs and lightening holes, the gussets either side of the lightening holes and an idealized tooth at the centre of the sector. The different thickness in the gear due to the casting cope and drag were included. The model was constrained to simulate the contact of a pinion with the tooth on the gear taking care not to violate the rules of 3D-elasticity. Bending stiffness of the pinion between its bearings was neglected for this analysis.

It was not possible to model the helical form of the tooth as part of this analysis due to budget constraints. The model simulated a spur gear more closely than a helical gear however, this simplification was considered adequate to demonstrate the effects of flange thickness on stiffness and contact force distribution.

Two analyses were run to determine the effects of thermal expansion on the contact forces. In the first, no thermal input was applied to the gear; the contact force distribution was determined due to the stiffness of the gear and the applied constraints only. In the second, a *uniform* energy input was applied to the gear face and heat was convected away from the gear from certain surfaces. The magnitudes of the thermal input and

the convection coefficients were chosen so that calculated temperatures matched the temperature distribution measured onsite.

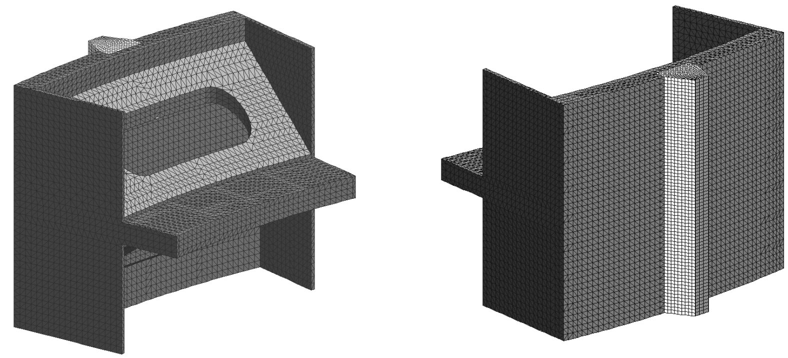

Fig. 5 – Model of Gear Sector

Results: The calculated contact force distributions as a function of location across the tooth face *for no heat input* is shown in Figure 6; results have been normalised with respect to the maximum calculated contact stress. The full line curve represents the contact stress in the gear tooth and the dashed curve represents the average load on the tooth (the integral of the total curve divided by the width of the tooth). The load at the cope side is 13% greater than the drag side load and the peak load is 39% greater than the average load. The results obtained from the model when heat input was included are shown in Figure 7. The total load on the cope side is 25% greater than the drag side load and the peak load is 44% greater than the average load.

The results in Figures 6 and 7 indicate that the additional thickness of the cope results in locally higher contact loads on the cope side. Furthermore, thermal expansion of the gear results in an increase in the load on the cope side the gear relative to the drag side. The *uniform* thermal energy input used in this analysis resulted in an increase in peak load from 139% to 144% of average tooth load at the cope end of the gear whereas the load at the drag end reduced from 86% to 81% of the mean load.

It is interesting to note that the contact force distributions in Figures 6 and 7 show a drop off in force near the ends of the tooth. Initially, this was thought to be a function of the constraints and idealisations used in the analysis. However, further analysis showed that the effect was due to a plane stress zone that extends into the gear face for a short distance from the free surfaces. Capturing the extent of this plane stress zone requires a high mesh density. The final mesh density used in the model was determined after several steps of mesh refinement until the plane stress zone thickness converged. Coarser models resulted in the much larger plane stress zones and the resultant peak contact forces were reduced. As always, convergence tests in finite element analysis are necessary to ensure the validity of results.

Further bias of the contact forces towards the cope end would be obtained from a thermal energy input that was a function of the contact load rather than being uniform.

The analysis of the contact forces assumed that the contact would occur at the pitch circle diameter. However, as the girth gear and pinion expand, the contact points move further towards the roots of the teeth. This effect occurs more at the cope side of the gear than the drag due to the differential thermal expansion of the girth gear. The analysis was modified to account for the tooth contact location. Results showed a further increase in the contact force taken at the cope side of the gear. This result is consistent with the thermal ratcheting behaviour described in the previous section.

Discussion: The analysis presented in this section is approximate. The helical form of the tooth could not be modelled and the interaction of the pinion and gear tooth was simulated using constraints rather than modelling the actual meshing. However, the analysis is instructive in that it shows that the structure supporting the gear teeth in these particular girth gears will tend to bias load towards one side of the gear and thus simulate misalignment where none exists.

The effects of crown and profile changes to the ideal helical tooth profile that gear manufacturers often use have not been considered. These modifications are used in part to redistribute the load on the gears more towards the centre of the flank and to improve the misalignment resistance of the gears. However, contact between a Y-shaped gear and a pinion supported between bearings will result in the stiffest parts of the gear adjacent to the webs of the Y structure meshing with the stiffest part of the pinion near the bearings. Whilst modifications to the pinion tooth profile can improve the contact force distribution, it cannot be made uniform, particularly if significant stiffness variations exist between the two sides of the gear due to casting features.

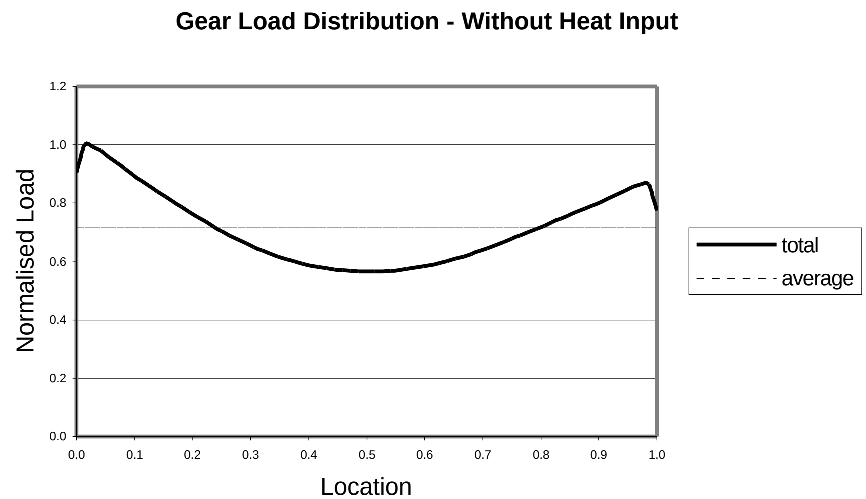

Fig. 6 – Normalised Load Distribution Without Heat Input

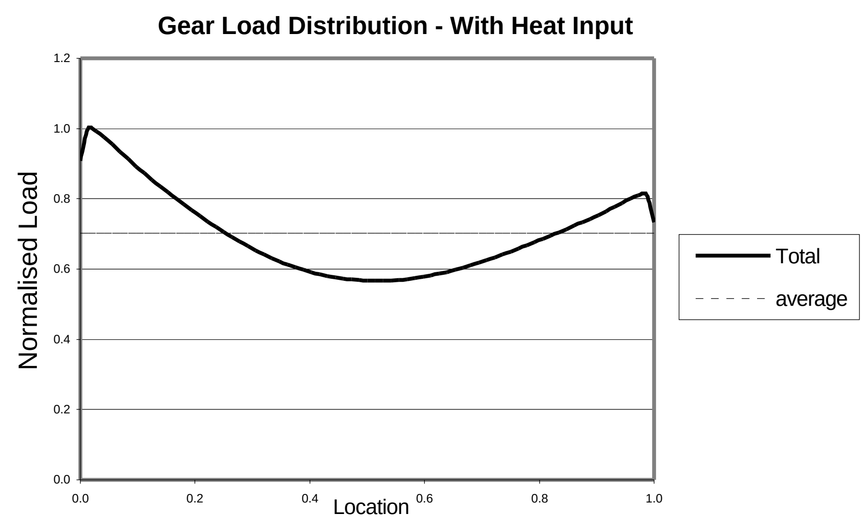

Fig. 7 – Normalised Load Distribution With Heat Input

ROOT CAUSE ANALYSIS

An investigation of various possible causes of the gear damage shown in Figure 1 has been presented. The ultimate aim of the investigation was to the root cause of the damage. In this study, ***root*** cause was defined as the one contributing cause that, if eliminated, would also have eliminated the failure, all other things being equal. It is important to determine the root cause in order to ensure that remedial measures address the fundamental cause of a problem. This definition focuses on causes that can be "engineered out" of the system.

The analysis of the gear damage has considered several possible contributory causes, namely: misalignment, drive train torsional dynamics, spent lubricant build-up, lube oil viscosity and gear blank design. Of these possible causes, only lubricant viscosity and gear blank design could have contributed to the failure. It is clear that metal-to-metal contact must have occurred and thus the lubrication film had broke down at the instant when the damage occurred. However, the worst damage occurred in the ball mill gears operating with the lubricant with the higher viscosity, KG10-HMF2500. Furthermore, the lambda factor for this lubricant and gears was more than sufficient to prevent failure in ordinary circumstances[1]. The analysis of the load distribution in the gears shows that the stiffness distribution of the Y-shaped structure supporting the girth gear teeth resulted in a bias of the contact load towards the cope end of the girth gears. Thermal data trends over at least one year of operation showed consistently higher temperatures in the pinions at the ends corresponding to the gear cope supporting the analysis results. It is unclear if any lubricant could have prevented the failure. However, a change to the form of the gear from a Y to a T shape or changes to the detailed shape of the Y gear that produce a more uniform load distribution would have prevented the failure.

The root cause of the damage to the gears was the differential stiffness of the girth gear caused by casting features and the Y-shape of the support structure. The structural features in combination with an external perturbation (kibble loading of balls into the mill or mill grind-outs) initiated thermal ratchetting that lead to the damage.

[1] Run-in lubricants can improve the lambda factor by improving surface finish. Use of these lubricants was not permitted by the gear manufacturer for the present gears.

CONCLUSIONS

The analysis presented in this paper was for a specific set of gears at Cadia Hill Gold Mine. Much of the analysis can be applied to other gears however, it does not follow that all Y-shaped gears with a cope and drag and similar power draw will suffer the same damage as the Cadia gears.

It is noted that the gears exhibited thermal excursions after the damage was detected. These excursions were not as severe as those observed in the first few months after commissioning and they were not as frequent. Run-in lubricant was used after the failure and kibble loading was eliminated. These changes in combination with a localised change in the stiffness of the gear due to the material removed by the scoring damage noticeably reduced the occurrence of the temperature excursions.

It is interesting to note that there was damage to the girth gears on the unloaded face of the gear teeth. This was at the dedendum of the teeth and the location of the damage corresponded to the tips of the mating pinion teeth. Checks of commissioning backlash records showed that backlash had reduced to almost zero after thermal excursions. The damage to the unloaded faces of the gear teeth may have been caused by loss of backlash.

A novel finite element analysis of the gear structure has been presented in this paper. It is applicable to all open gear types, Y and T shaped and can be used to assess the uniformity of load on gear teeth. Improvements to the method such as coupling the pinion stiffness to the model, including profile and crown effects would be useful.

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