MILL ROLLER SHAFT FAILURES AND INVESTIGATIONS

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SUMMARY

The paper outlines the reasons for failures which occur in sugar mill roller shafts and describes how some of the causes of failure, such as corrosion and stress concentrations can be avoided. An explanation is given to show some of the features of the geometry of shafts and operation of mills which make a purely theoretical approach to an investigation of the conditions of stress in a shaft impractical.

Details are given of an experimental investigation to determine the levels of stress in a mill roller shaft under operating conditions. A method is described for affixing strain gauges to a shaft so that stress readings can be obtained over several months of crushing. Varying conditions of roll load were employed and measurements were made of the bending and shear stresses at the journal fillets at each end of the roll. In addition measurements were made of the torque at the drive end. Results obtained from the experiments are presented and discussed. It was found that an extremely significant factor affecting the magnitude of the bending stresses was the relative lifts of the drive and pinhole side of the roll.

Conclusions are that the shafts are highly stressed and recommendations are made for the design of shafts and modifications to mills.

INTRODUCTION

In Queensland, mills are worked hard and in 1962 it became apparent that the failure of mill shafts was an industry problem to which Sugar Research Institute should give attention.

The use of ultrasonic flaw detection had been tried without much success a few years earlier, but a more modern version of this equipment was inspected in 1962 and a searching test programme was undertaken by Sugar Research Institute officers using this much improved equipment. Techniques were developed which made it possible for an experienced operator to detect with confidence cracks in shafts and give a good indication of the magnitude of the defect.

A survey of hundreds of shafts in the industry was made and many of the shafts were found to be faulty. Some were condemned, some kept under observation, some transferred to lighter duty and in a few cases the cracks were machined out and the shaft repaired by welding. Many of the faults occurred in the journal fillets and these were verified using the magnetic particle inspection method of crack detection. Those occurring just inside the shell were not verified other than by employing various ultrasonic testing techniques such as testing from both ends of the shaft and employing angle probes.

The immediate problem of knowing the condition of available rollers was solved in this way but the longer range issues were to determine the reasons for the failures and eliminate these causes.
Attempts to calculate the stress at the journal fillet of a shaft introduce a number of unknowns upon which the bending moment, shear forces and torque depend. There is no reliable measure of the friction in the guides of the bearing, the distribution of load in the bearing is unknown, there is no information available on the position through which the resultant force acts. The twisting moment is almost certainly varying with the position of the muff coupling as well as the position and state of the gearing. The stress concentration at the fillet may be calculated but uses data available from work on smaller diameters.

Some of these may be considered to be second order quantities, but the bending moment assumptions, for example, can have such a large influence on the final stress value that no reliance could be placed on what might appear to be a reasonable estimate of the position of the resultant force at the bearing. Add to this the probable fluctuation of the dynamic case which must be considered with a mill in operation and calculations alone appear to be impracticable.

This concept of working with unknown quantities is nothing new to an engineering designer and it might be considered pedantic to worry over such details since some mill roller shafts have been in operation for years without the designer having access to full information. It is not pedantic for a mill manager to be concerned that a new shaft breaks within a few weeks of the mill starting date. This has happened on occasions and good reasons must be found for such failures which occur usually at the drive end journal inside fillet or just inside the shell landing.

CORROSION

An inspection of many shaft failures shows corrosion to be present and the appearance of the failures are typically fatigue failures. Corrosion fatigue failures are to be expected in this type of application and to minimise the danger it is appropriate to try to exclude the corrosive environment and to reduce the level at which the alternating and intermittently variable stresses are present during mill operation. The material from which the shafts are made must also be suspect and extensive physical and chemical tests would naturally be undertaken where premature failures occur.

The prevention of corrosion in general presents some problems but there were a large number of shaft faults found in the shell landings and it was evident that some penetration of juice with consequent pitting had occurred. This pitting and local corrosion can be effectively dealt with by providing a convenient rubbery seal at the end of the shell at the shaft shell interface. This is preferably done without mechanically constrained rings.

Investigations were carried out using various viscous liquids and pastes which are required to set and give a good bonding yet remain flexible. A test rig was designed to provide alternating stressing of fillets of these sealants in a bath of hot juice. The material Lastomeric hard which was the least expensive of those used in these tests proved to be the best. It has a heavy rubber base which is used in conjunction with a curing agent and this curing process takes approximately 14 hours.

A shaft which had been treated by applying a fillet of this material to prevent ingress of juice between shell and shaft was examined at reshelling and there was no evidence of pitting in the shell landing area.
STRESS CONCENTRATIONS

The fact that a number of comparatively new shafts have broken at the inside fillet of the journal at the drive end clearly indicates some inadequacy in design or material. Chemical analysis, physical testing, metallographic studies and macroscopic examination have been used to evaluate the condition of the metal. These tests have not generally shown any unsuspected properties or inadequacies of the material which would be likely to cause failure in service without severe stressing.

It is certain that a stress-raising effect occurs at the fillet of the journal and the stress concentration can be calculated. For many designs using a one inch fillet radius this stress concentration factor is approximately 1.7 and in accordance with Peterson's experimental evidence this value could be reduced by approximately 30% if the radius were increased to five inches.

In adopting this solution there is some danger since the bending moment increases linearly as successive positions along the shaft are considered towards the shell. At the same time removal of metal by increasing the radius reduces the section modulus which varies as the cube of the roll shaft diameter. The bending stress will depend upon the particular configuration of the bearings and shaft design and must be calculated for each case.

The unfortunate fact is that whereas the calculation to be performed is simple, doubtful assumptions must be made regarding the position of the line of action of the resultant force from the bearing pressure on the shaft. The distance of this resultant force from the cross section of shaft considered determines the lever arm which when multiplied by the magnitude of the force gives the bending moment at the cross section considered.

From these considerations and after assessing the range of possible bending moments, it became clear that it was highly desirable to obtain measurements of bending moments, shear stresses and torque on a mill shaft in operation.

METHOD

The shaft chosen for the experiments was the top roller shaft of the 5th mill of a five mill train. The top roller was hydraulically loaded.

Strain gauges were fixed to the shell landings at each end of the shaft. This location was necessary to prevent gauge damage during mill operation. The levels of stress at this location were related to the critical fillet locations by direct calibration of the shaft when stationary.

Fig. 1 shows the preparation of the shaft and equipment used to bring out the signals from the strain gauges. Holes A, B, C and D were drilled and a groove E was provided in the shell. These provided conduits through which wires from both pintle and drive end gauge positions could be brought to the pintle end of the shaft where slip rings were located.

Gauge protection was a major problem which was solved finally by the use of adhesives and mechanical techniques which preserved the gauges from damage in the extremely humid and high temperature ambient conditions.

The strain gauges were arranged in sets on the shaft to measure the bending,
shear, and torsional stresses on the drive end and the bending and shear stresses on the pintle end. Because the outputs of those gauge sets measuring bending and shear stresses were sinusoidal due to shaft rotation, it was necessary to arrange the angular positioning of the gauge sets so that each set could be sampled by the switching unit—strain gauge amplifier combination twice each shaft revolution when the output was a maximum. Roll lifts at both ends were measured with strain-gauge beams. The results were recorded on a digital voltmeter-printer combination.

EXPERIMENTAL RESULTS

The roll was calibrated for bending and shear stresses by applying hydraulic loads at fixed lever arms from the fillets. Gauges were positioned on the fillet to enable the stress concentration factor to be determined and to obtain a relationship between stress measured at the shell landing and the stress at the fillet. It was found that the stress concentration factor at the fillet was 1.39 compared with a theoretical value of 1.30 obtained from small diameter tests. This 7 per cent increase is in agreement with experiences of other workers using large diameter shafts. The torque calibration was achieved by applying weights to a 4.5-m lever arm rigidly attached to the roll.

The experiments to determine bending and shear stresses were performed at 16 combinations of drive and pintle loads giving top roll loads from $1.5 \times 10^6$ to $2.2 \times 10^6$ kg/m. Five replications were done for each combination. The period of each test was 10 min or approximately 30 revolutions.

The points shown in Fig. 2 to 7 represent the average values of the variables for the five replications.
The values obtained for pintle shear stress at the inside fillet of the pintle journal are shown in Fig. 2. It can be seen that pintle shear stress is independent of drive load (ram pressure x ram area) and is within 5% of the theoretical value determined from the geometry of the shaft and the applied pintle load. This close agreement is not surprising when the simple geometry of the pintle end bearing is considered.

The values of pintle bending stress at the journal inside fillet are shown in Fig. 3. The pintle bending stress is seen to be approximately 20% higher than the theoretical value calculated from the applied pintle load and the lever arm (distance between line of action of load and journal inside fillet) on the assumption that the load acts at the centre of the bearing. However the pintle lever arm was found to vary with difference in drive and pintle lifts as shown in Fig. 4. The pintle lever arm, on the assumption that the load acted at the centre of the bearing, was 25.4 cm. The high

(a) Pintle shear and pintle bending stress
Fig. 4. Effect of difference in drive and pintle lift on drive and lever arms.

Fig. 5. Effect of drive load and difference between drive load and pintle load on drive shear stress.

Fig. 6. Effect of drive load and difference between drive load and pintle load on difference between drive lift and pintle lift.
values of the experimentally determined lever arm account for the higher values of pintle bending stress.

(b) Drive shear and drive bending stress

The values obtained for drive shear stress at the inside fillet of the drive journal are shown in Fig. 5. The magnitude of the nominal drive shear stress

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\frac{\text{ram pressure} \times \text{ram area}}{\text{cross sectional area of shaft}}
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is higher than the theoretical value and is dependent on the value of pintle load as well as the value of drive load. It is higher for the greater difference between drive and pintle loads. This is because of the greater downward force exerted by the tail bar when the difference in lifts between the drive and pintle ends is greatest. The relationship between applied load and difference in lift is shown in Fig. 6.

The values obtained for drive bending stress at the inside fillet of the drive journal are shown in Fig. 7. The fact that the stresses are less than the theoretical values and depend on the difference between drive load and pintle load is due to the changes in lever arm that occur with changes in values of load difference, accompanying the difference in roll lifts (Fig. 4). When the theoretical values of drive bending stress were calculated using the experimentally determined values of lever arms, they agreed with the observed values to within ± 7%.

(c) Drive torque

It was found that torque on the top roll for set load conditions showed periodic fluctuations of ± 15% of the mean value. The frequency of these fluctuations was 18 per roll revolution, corresponding to the number of teeth on the pinion. This is evidence of uneven load sharing on gear teeth due principally to imperfections in gear geometry.
The results for average value of top roll torque are shown in Fig. 8. It can be seen that the scatter is appreciable.

![Fig. 8. Effect of top roll load on top roll torque.](image)

**CONCLUSIONS**

It has been demonstrated that it is possible to employ strain gauges to measure the stresses occurring in a mill shaft in operation and to have these remain operative during several months of crushing.

The value of the pintle lever arm has been shown to be a significant factor affecting the magnitude of the stresses at the pintle journal inside fillet. Because the magnitude of the observed stresses is very close to the endurance limit of the usual shaft material, it is essential that this lever arm at all times be kept to a minimum. This can be done by either limiting the value of difference in lift between drive end and pintle end or alternatively providing a better type of self aligning bearing.

The difference in roll lift has a reverse effect on the lever arm at the drive side to the effect it has on the lever arm at the pintle side. It is fortunate that the large differences in roll lift which give rise to the higher values of drive shear stress also give rise to the lower values of lever arm, thus lowering the values of drive bending stress. If, however the shaft is to operate with minimum difference between drive and pintle lifts for pintle bending stress considerations, it is essential that the extra shear force applied to the shaft by the rigid tailbar coupling to the prime mover be kept to a minimum. This can be done by increasing the flexibility of the tailbar by increasing its length, or where this is not practicable by providing a type of coupling that can accommodate misalignment of the prime mover output shaft and the top roller.

The general level of stress due to bending moment is high and unless design
modifications as suggested above are introduced the conventional shaft should be increased in diameter or a better class of steel used in the shafts.

Ultrasonic testing has proved to be a valuable inspection tool for use during the crushing season, as well as during the non-crushing season, to detect and observe the progress of flaws in a roller shaft. In this way shaft breakages in service can be minimised and a true assessment of the condition of mill shafts and spares can be maintained.

ACKNOWLEDGEMENTS

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REFERENCES


Discussion

P. A. SHAFFORD: Asked for explanation on shaft sealing. His company uses mechanically restrained O-rings. Dr. Allen and Mr. Seaford exchanged views by means of diagrams.

PATTUDD: What diameter of shaft used in experiments?

ALLEN: 19". Is annealing of shaft usual practice in Australia?

S. O. CLARKE: No annealing of shafts in Australia.

R. C. TURNER: Do fractures occur in Australia in positions other than at the fillets? In South Africa the majority of fractures occur in the center of the top roll or just inside the shell of the front or discharge roller. The only fillet failure known to occur was due to the jamming of the hydraulics and subsequent cocking of the shaft.

ALLEN: Failures in Australia occur mainly at fillets on both drive and pintle ends of shafts, but also close to the end of the shell underneath the shell.

A. S. HONEY: Deterioration in roller surface condition can cause rollers to slip on the feed. Quite large variations in fibre rate can result. The deterioration of surface may be detected by erratic variations in turbine nozzle bowl pressure.

ALLEN: Agreed. Similar indication of deterioration of surface is obtained with pressure chute monitoring equipment.

G. ASKE: At Unifolosi breakage in top rollers always occurs on the gear side adjacent to the shell. More wear was found on top bearing on gear-side, which caused "cocking". The use of packing to take up the wear has eliminated breakages.

J. CARMICHAEL: The importance of ductility in roller shafts has not been mentioned and should be considered as well as tensile strength, shear factors, etc.

ALLEN: Agreed and mentioned similar requirements were evident in experience with truck axles.

A. JANS: Did you find most breakages at the fillets of the shaft on the pinion side or at the pintle side?

ALLEN: Breakages occurred almost equally at drive and pintle ends and also just inside the shell.

E. S. BRAGA: Is there any data available on the reliability of welding broken shafts? Is welding of broken shafts possible?

ALLEN: At ISSCT congress in Mauritius 1962, R. Hughes of Hawaii reported having successfully welded shafts at centre making two broken shaft into one. Also major repair one and half inch deep groove built up by welding in Australia successfully.

W. R. CRAWFORD: Commented on the scoring of metal at the fillets and the large stress concentrations occurring due to this.