POSSIBLE CAUSES OF RECENT ROLL SHAFT FAILURES IN SOUTH AFRICAN SUGAR MILLS

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ABSTRACT
Following a survey of one hundred roll shaft breakages which have occurred at eight selected sugar mills in Natal since 1979 it became clear that many of the failures could have been avoided by correct specification, manufacture, maintenance and care during operation. Calculations of shaft stresses and fatigue stress concentration factors were carried out to determine if present shaft design, machining practices, material specifications and mill maintenance procedures are satisfactory and whether or not they can be improved. The feasibility of using adhesive to fix the shell to the shaft is discussed and some comment on specifications for manufacture is given.

INTRODUCTION
In recent years there have been many roll shaft failures at South African sugar mills which have prompted the Sugar Milling Research Institute (SMRI) to take a closer look at possible causes and to make some recommendations to reduce the frequency of failure. A survey of roll shaft failures was undertaken to establish the magnitude of the problem and to establish the most common causes of failure. The results of this survey indicate that there is an average of one failure per mill per season. If it is assumed that these failures could have been avoided, there is a potential saving of up to US$ 25 000 for each mill every year, with the expenditure of very little effort.

THE SURVEY
Eight mills were asked to provide details of all roll shaft failures which had occurred since 1979. Unfortunately not all of these mills keep comprehensive records of all failures, but the survey nevertheless revealed the following:

a) Shaft failures in service occur more frequently when the roll is being used as a top roll. Top roll failures represented 68.5% of the total.
b) The most likely position for a break to occur is at the inner fillet radius on the drive side of the roll. The frequency of this occurrence was 43.6%.
c) The next most likely position for a break to occur is at or near the drive side end of the shell. The combined frequency of this occurrence was 27.6%.
d) The average age of a shaft which fails in service is 5.5 seasons.

A diagram of the frequency of failure at different points along the shaft is given in Fig 1.

FAILURE INVESTIGATIONS
Several shaft failures have been investigated in detail in recent years. They have several common features:

a) The fracture always has the appearance of a fatigue failure indicated by the characteristic "clamshell" lines from the point of the initial crack followed by parallel failure lines. There is always a relatively small brittle failure area at the centre of the shaft where final fracture takes place.
b) The initial crack usually follows a line at 90 degrees to the shaft axis which indicates that the direction of the primary stress is mainly due to bending of the shaft and not to torsion.

c) The initial stress raiser is seldom evident because of subsequent surface damage in the vicinity of the fracture. However, in most cases the evidence suggests the following stress raisers to be responsible:
- Fretting and pitting corrosion
- Surface defects such as welding inclusions
- Deep machining marks or scratches
- Poor blending of fillet radius into journal
- Wear grooves at or close to fillet radius

d) In all cases in which the shaft material was analysed it was found to be within specification.

**DISCUSSION OF STRESS CONCENTRATION**

The levels of stress which have been calculated (see Appendix 1) and those which have been measured in Australia (Cullen') are somewhat below the yield point of the steel and are therefore not high enough in themselves to cause failure of the shafts. The mechanism of failure is therefore always that of fatigue, which requires a point from which the failure is initiated.
This point is usually that at which the stress is intensified by physical factors which are discussed below:

**Fillet Radius**

Because of the prevalence of failure at the fillet radius on shafts in recent years, the SMRI commissioned the National Mechanical Engineering Research Institute to carry out a finite element analysis of the stresses in the region of the fillet radius of a typical sugar mill roll. The result of this analysis revealed that the stress concentration factor at the point where the bearing journal meets the fillet radius could be reduced from 1.93 to 1.30 by increasing the radius from 30 mm to 180 mm. This represents an improvement of only 33% in stress concentration factor. Nevertheless it is strongly recommended that the largest radius which can be accommodated by the roll geometry be used in every case, especially for the fillet on the drive side of the roll.

**Wear Grooves and Surface Finish**

The surface finish of the fillet and the adjacent journal is of far greater importance to stress concentration than the size of the radius. There is often a sharp change in section at the point of runout of the radius with the bearing journal, which would almost certainly initiate a fatigue failure. Another frequent source of fatigue failure is grooving of the journal or fillet radius by grit trapped in the bearing. To place this in perspective, consider as an example a groove which has a root radius of 0.5 mm. This would cause the stress concentration factor to increase to about 4.0, which is considered by Juvinall\(^2\) to be the maximum that any physical discontinuity can cause. If this is compared to the figures calculated for different radii quoted above it may be concluded that such a groove would almost certainly cause fatigue failure even at very low stress levels.

**Shrinkage Stresses**

There is also a significant stress concentration factor caused by the shrinkage of the shell to the shaft. The effect of this is clearly illustrated by the experiments quoted by Peterson\(^4\), in which stress concentration factors up to 3.8 were obtained in various cases. The results are slightly confused by the presence of fretting corrosion which is discussed in more detail below.

**Fretting Corrosion**

This phenomenon is responsible for a large number of fatigue failures which make it almost as important as defects associated with the fillet radius. The number of failures which occurred at the edge of the shell would indicate that fretting, and the corrosion which follows, should be given more attention. The mechanism of fretting is not clearly understood but would appear to be caused by the "pumping" effect of the microscopic movement which inevitably takes place between the shaft and the shell. This allows entry of juice into the very narrow space between shaft and shell where crevice corrosion can easily cause severe pitting. The oxides which result from the corrosion are compressed and become finely powdered which is a clear indication of this type of corrosion. The most effective cure for fretting corrosion is to seal the joint between the shaft and shell with a flexible adhesive. Tests on various adhesives for this purpose, carried out in Australia, indicated that it was feasible to seal this joint with proprietary compounds (Cullen\(^1\)).

**Surface Defects**

The original casting of the ingot from which the shaft forging is manufactured could
contain inclusions and centre line shrinkage. If these inclusions are near the surface of the forging, they could be a cause of fatigue failure. However, none of the failures which have been carefully investigated recently has revealed any such surface defects. In one investigation such inclusions were found, but they were below the surface and were therefore not regarded as critical.

A more frequent type of surface defect is that which results from welding on the surface, such as when the journal of a roll is built up to restore its diameter. At least one roll shaft breakage of those investigated recently was found to have a few small weld slag inclusions in the middle of the shell landing, as if a weld repair had been carried out prior to shrink fitting the shell. It was fairly clear in this case that the source of the fatigue crack was one of these slag inclusions. These repairs must be very carefully carried out to avoid trapping slag and scale in the weld area, and adequate stress relieving must always follow such repair. Surface inclusions should be gouged out and the resulting depression should be carefully polished to remove all stress concentrations.

Residual Stresses

Machining either by cutting or grinding always induces surface stresses which could become sources of fatigue cracks. This is difficult to avoid or rectify, except perhaps by ensuring that the final cuts on the lathe are as light as possible.

There have been rare occasions when mechanical surface damage has been the cause of fatigue failure. It is fairly easy to see such damage which can usually be repaired by gouging out the damage and polishing the resulting depression. It could also be repaired by welding and subsequent stress relieving if it is considered to be sufficiently serious.

DISCUSSION OF SHAFT STRESSES

A calculation of the stresses which can be expected in a typical sugar mill roll is given in Appendix 1. It is assumed that the applied hydraulic load and torque are both steady, but at their maximum levels, eg at stalling point on the turbine.

The highest combined stress on a roll on which the shell is well fitted is 127 MPa and occurs at the drive end fillet radius. Although this stress is still well below the yield stress of the shaft which is specified at 275 MPa, it is slightly higher than the endurance limit which is estimated at 123 MPa for this type of steel.

It should be noted that the shaft/shell combination is far stronger than the shaft on its own. This is borne out by many failures which have occurred in which the shell has fractured first, followed very quickly by the failure of the shaft. There are many reasons why the shell could fail, particularly when it is appreciated that the material, being cast iron, is brittle and unable to withstand tensile stress.

A common cause of shell failure is poor quality control during shrink fitting, either through excessive interference fit or uneven cooling.

Direct measurements of stresses on a roll were carried out in Australia (Cullen1) using strain gauges in 1967. These stresses were found to vary up to 110 MPa. The factor which had the greatest effect on the magnitude of these stresses was the value of the pintle lever arm, which is affected by the alignment of the bearing. Another significant factor was the difference in roll lift which in turn is affected by the shear force applied to the shaft by the rigid tailbar coupling. This effect could be much reduced by using a longer tailbar or by improving the flexibility of the tailbar coupling.

ADHESIVE BONDING OF SHELL TO SHAFT

In 1970 the engineers at Mount Edgecombe developed a method of bonding the shell
to the shaft using an epoxy resin adhesive. The method was subsequently patented and some trials were carried out at the mill with limited success. In order to apply some scientific background to this idea, the SMRI investigated various adhesives and tested the shear strength of the most suitable one. The required shear strength for a typical mill roll has been determined by calculation to be 12.0 MPa to avoid a separation of the shell from the shaft when under load. Tests carried out on the shear strength of an adhesive using a steel bar and cast iron collar proved that the shear strength obtainable with adhesive up to 2 mm thick was in excess of 28 MPa (Lawrence). It is important that the adhesive remain moderately flexible after curing to allow for relative movement between the shell and shaft. The method is considered to be quite feasible, and the procedure has been discussed with a local manufacturer who considers that the technique could simplify the manufacture of rolls to a great extent. The advantages of using an adhesive instead of a shrink fit are seen to be as follows:

- No shrinkage stresses in shell
- No stress concentration at edge of shell
- Sealing against entry of juice between shaft and shell
- Fretting corrosion can be prevented
- Shell can be removed without damaging the shaft

ROLL SPECIFICATIONS

It is strongly recommended that every order for a new shaft or reshell be accompanied by a rigorous specification which should cover shaft and shell materials, shaft preparation and dimensional tolerances to ensure that the possible causes of failure discussed in this paper are eliminated. Some detail should be given in the specification of heat treatment, surface finishes and shrink fit tolerances.

CONCLUSION

There are many external causes of shaft failure which can be eliminated by changes in mill design and operation, such as an improvement in the tailbar coupling, and limitation of the hydraulic loading.

It should also be possible to use expensive alloy steel or sophisticated surface treatments to reduce the danger of fatigue failure, but in the final analysis it is evident that the major causes of failure originate on the surface of the roll shaft. Care of the surface of the shaft during operation and maintenance can therefore be rewarded by a much longer life for the shafts.

One of the areas in which this care can be applied to great effect is in adequate planning of the roll repair and reshell programme for the annual off crop. Whenever those involved in repair and machining are pressed for time, mistakes which escape notice until a failure occurs can easily be made. Provided all the precautions enumerated in this paper are carefully observed there is no reason why every roll shaft should not give a minimum life of 10 million tons of cane.

REFERENCES

Figure 2. Stress diagrams for a typical sugar mill top roll shaft.
APPENDIX 1 - CALCULATION OF STRESSES IN A TYPICAL ROLL SHAFT

Assumptions:
- Load on bearings: 3000 kN
- Distance of point of application of bearing load from inner fillet: 400 mm
- Shaft diameters:
  - bearing journal: 500 mm
  - on shell landing: 600 mm
- Maximum torque on mill is twice running torque: 2 × 1500 kNm
- Torque on top roll is 50% of total mill torque: 1500 kNm

The shearing force, bending moment, torque and combined stress at each significant point along the shaft have been calculated and are shown in Fig 2.

CAUSES POSSIBLES DES RUPTURES RÉCENTES D'ARBRES DE CYLINDRES DE MOULINS SUD-AFRICAINS

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Mots clés: Fatigue; arbres de cylindres, concentration de tensions

EXTRAIT

A la suite d’un examen d’une centaine de ruptures d’arbres de cylindres survenus depuis 1979 dans huit usines sélectionnées au Natal, il ressort que beaucoup de ces fractures auraient pu être évitées grâce à une spécification, une fabrication, un entretien et un soin opérationnel corrects.

Des calculs de tension des arbres, et les facteurs de concentration des tensions de fatigue ont été entrepris pour déterminer si la conception actuelle des arbres, les méthodes d’usinage, les caractéristiques des matériaux, et les procédures d’entretien sont satisfaissants et s’ils peuvent être améliorés.

La possibilité de fixer la coque à l’arbre en employant de la glue est aussi discutée, et quelques commentaires sur les spécifications de fabrication sont donnés.

CAUSAS POSIBLES DE RECENTES RUPTURAS DE EJES DE MOLINOS SUR AFRICANOS

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RESUMEN

Por observación de una centena de rupturas de ejes en ocho centrales azucareras escogidas en Natal, después de 1979 se quedó claro que muchas quiebras pudieron haber sido evitadas con correctas especificaciones, manufactura, manutención y cuidados durante la operación.

Cálculos de tensión en los ejes y de concentración de factores de tensión de fatiga fueron hechos para determinar si el diseño, el acabamiento de los ejes, las especificaciop-
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Las del material y la manutención de los molinos son satisfactorias y si necesitan ser mejoradas.
También se discute la posibilidad del uso de adhesivos para fijar el rollo al eje y se comenta sobre las especificaciones para la manufactura.