ANALYSIS OF THE CAUSES OF RECENT ROLL SHAFT FAILURES IN NATAL SUGAR MILLS

By M. J. REID

Sugar Milling Research Institute

Abstract

Details of 93 roll shaft breakages which have occurred at eight selected sugar mills in Natal since 1979 have been collated and analysed to determine the most likely causes of failure.

Theoretical analyses of shaft stresses and fatigue stress concentration factors have been carried out to determine whether present shaft design, machining practices, material specifications and shell assembly techniques are satisfactory and whether they can be improved.

The feasibility of using adhesive to fix the shell to the shaft is discussed and some recommendations to users and manufacturers on roll shaft and shell specification, design, assembly and operation are given.

Introduction

In recent years there have been many roll shaft failures at South African sugar mills which could have been avoided with the proper care and attention to detail on the part of the mill engineer and/or the roll manufacturer. A survey of roll shaft failures was undertaken to establish the magnitude of the problem and to obtain an idea of 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 R40 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 breakages which occured since 1979. Unfortunately not all of these mills keep comprehensive records of all breakages, but the survey has nevertheless revealed some interesting facts which are as follows:

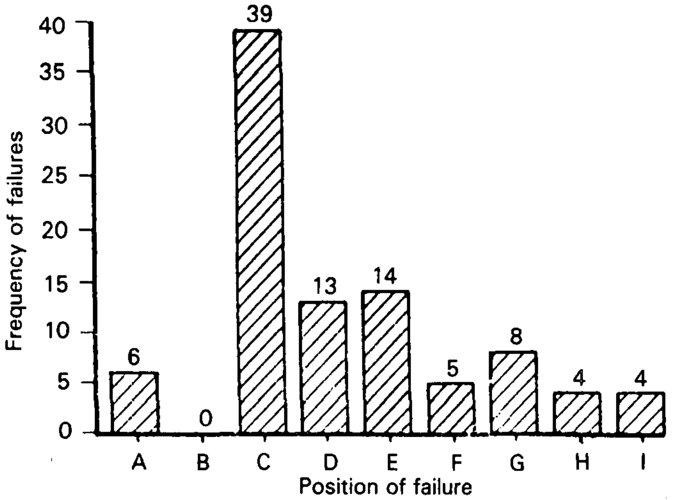

FIGURE 1 Frequency of failures at each location on roll shaft. (See figure 2 in appendix for position key.)

(a) Breakages of shafts in service occur more frequently when the roll is being used as a top roll. The percentage of top roll failures is 66% 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 42%.

(c) The next most likely position for a break to occur is at or near the drive side end of the shell. The frequency of this occurrence was 29%.

(d) The average age of a shaft which fails in service is 5,6 seasons.

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

Failure investigations

A number of shaft failures have been investigated in detail in recent years, and the results have several common features which are worth enumerating:

(a) The fracture always has the appearance of a fatigue failure because of the characteristic "clamshell" lines from the point of the initial crack followed by parallel failure lines similar to the growth rings on a tree. 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 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 in Appendix B and those which have been measured by the SRI (Cullen[1]) 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 almost always that at which the stress is intensified by physical factors some of which are listed below:

Fillet radius not large enough or poorly blended into journal surface

Wear grooves and poor surface finish
Shrinkage stresses
Fretting corrosion
Surface defects caused by inclusions or welding
Residual stress from machining operations or mechanical damage

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 factors due to various radii were as follows:

Radius	S.C. Factors
30 mm inner and 20 mm outer	1,93 and 1,25
180 mm	1,30
25 mm inner combined with a mixed radius outer fillet of 40 mm and 180 mm	1,93 and 1,45
Elliptic inner and outer	1,73 and 1,12

It can be seen from these results that the larger the radius the smaller the stress concentration factor. However the effect is not as marked as the increase in radius would lead one to expect. For example, an increase from 30 mm to 180 mm gives only a 33% improvement in stress concentration factor. Nevertheless it is recommended that the largest radius which can be accommodated by the roll geometry be used in every case especially for the inner 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 could be a serious source of stress concentration. Another frequent source of fatigue failure is grooving of the journal or fillet radius by grit trapped in the bearing. To put this into perspective consider as an example a groove which has a root radius of 0,5 mm. This would have a stress concentration factor of 14,5 which compared to the figures calculated for different radii quoted above would almost certainly cause fatigue failure even at very low stress levels. The calculation of this factor is given in Appendix A.

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 of up to 3,8 were obtained in various cases. The results are slightly confused by the presence of fretting corrosion which will be discussed in more detail below.

Fretting corrosion

This phenomenon is perhaps even more important than the fatigue effect of a small fillet radius. The number of failures which occurred at the edge of the shell indicate that fretting and the corrosion which follows could well be the major problem requiring attention. The mechanism of fretting is not very clearly understood in the literature, but appears 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 carried out by the SRI in Australia showed that an adhesive called Lastomeric Hard was the most effective for this purpose.[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 so far 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 that in this case 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 surfaces 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 should be 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 B. 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. In the case where the shell is loose on the shaft the stresses rise to much higher levels, reaching a combined stress of 159 MPa at the centre of the shaft. Although these stresses are still well below the yield stress of the shaft which is specified at 275 MPa, they are above 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 born 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 measurement of stresses on a roll were carried out in 1967 by the the SRI[1] using strain gauges. 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 (see Appendix B) 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 up to 2 mm thickness of adhesive was in excess of 28 MPa.[3] The method is therefore considered to be quite feasible, and the procedure has been discussed with a local manufacturer who considers that the technique could greatly simplify the manufacture of rolls.

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

The next step in this experiment is to persuade a mill to attempt adhesive bonding on one roll in order to test the theory and to discover any pitfalls which may arise in the assembly procedure.

Roll specifications

A recommended specification is provided in Appendix C. Shaft and shell materials, shaft preparation and dimensional tolerances are covered by the specification to ensure that the possible causes of failure discussed in this paper are eliminated. The specification should not be regarded as comprehensive. When applied by a mill it should include more detail on surface finishes, dimensional tolerances, roller grooving angles and accumulative pitch error allowances and any other details to suit individual mill requirements. Some discussion of the materials of this specification is necessary.

The mild steel forged shaft made from "28/33 ton" steel which is similar to 070M20 in BS970 has been in use for many years and with careful design and operation can provide an adequate life. Other shaft materials have been tried. At Sezela some shafts were made from cast steel with no improvement over the mild steel forging, and high tensile steel has been used at the same mill, with results which indicate an increase in fatigue failure ascribed to the lower notch sensitivity of the steel.

The cast iron used for the shell is a fairly high tensile grade having a tensile strength of 300 MPa. Cast iron still appears to be the best material for a roll shell because of its low cost and ease of manufacture and assembly. In regard to its effect on the failure of shafts there is no reason for any change in this specification.

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. But in the final analysis, it is evident that the major causes of failure originate on the surface of the roll shaft. "Tender Loving Care" of the surface of the shaft can therefore be rewarded by a much longer life for the shafts.

One of the areas in which this care can be applied with great effect is in adequate planning of the roll repair and reshell programme for the annual off crop. Whenever the persons 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

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3. Lawrence, AG (1985). Adhesive bonding of roll shells to shafts Part 2, SMRI Internal Report No. 2/85.
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5. Timoshenko, S (1960). Strength of materials Part II. D van Nostrand, Princeton, New Jersey, 326.

APPENDIX A

Stress concentration due to a small groove

When a shaft containing a circumferential groove is subjected to bending, the stress concentration factor may be given by the formula (Timoshenko[5]):

$$k_t = \frac{3}{4}\sqrt{\frac{d}{2r}}$$

where d = diameter of shaft

r = radius at bottom of groove

This value of k_t must be reduced by a factor to allow for the ductility of the material of the shaft. The equation below gives k_f the stress concentration factor due to a small groove as a function of the factor k_t due to a groove of radius r (Juvinall[2]):

$$k_f = 1 + \frac{(k_t - 1)}{1 + \frac{\sqrt{a}}{\sqrt{r}}}$$

where $\sqrt{a}$ is a factor which depends on the ductility of the material, and is roughly proportional to the grain size. From Figure 13.20 in Juvinall[2] this factor is 0,12 for 28 ton steel.

Thus, given a shaft of 500 mm diameter, with a groove of 0,5 mm radius the value of k_f is calculated to be 14,48.

APPENDIX B

Calculation of stresses in a typical roll shaft

Assumptions:	
Load on bearings	3 000 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 × 1 500 kNm
Torque on top roll is 50% of total mill torque	1 500 kNm

The shearing force, bending moment, bending stress, torsion stress and combined stress at each significant point along the shaft are calculated and shown in Table 1 and Figure 2.

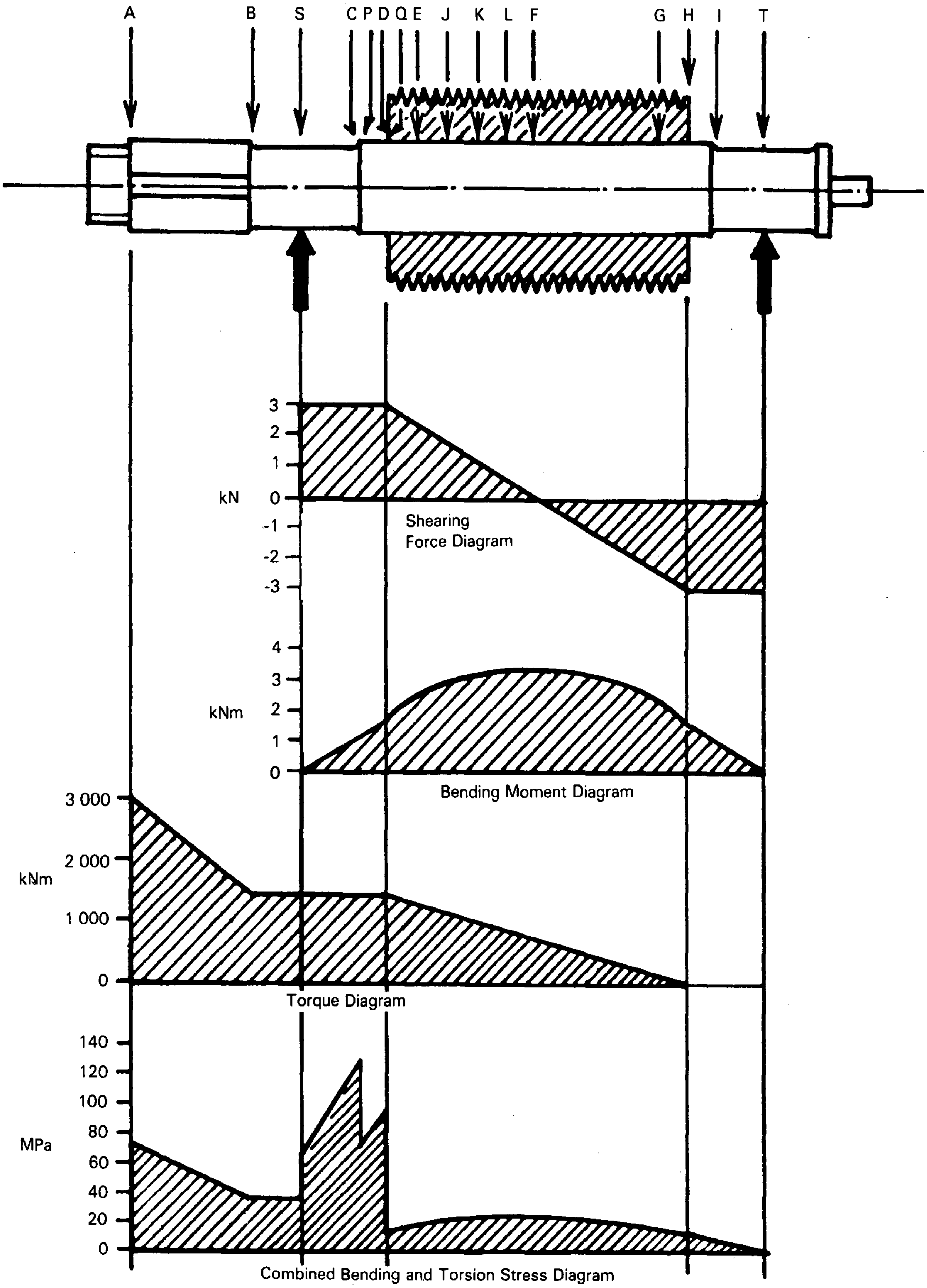

FIGURE 2 Stress diagrams for a typical sugar mill top roll shaft.

Table 1

Roll shaft stress calculation

	S	C	P	D	Q	E	J	K	L	F	H
Distance from support m	0.00	0.40	0.40	0.58	0.58	0.80	1.00	1.20	1.40	1.65	2.72
Shearing force kN	3.00	3.00	3.00	3.00	3.00	2.38	1.82	1.26	0.70	0.00	-3.00
Bending moment kNm	0.00	1.20	1.20	1.74	1.74	2.33	2.75	3.06	3.25	3.34	1.74
Torque kNm	1500.00	1500.00	1500.00	1500.00	1500.00	1345.79	1205.61	1065.42	925.23	750.00	0.00
WITH SHELL											
Shaft diameter m	0.50	0.50	0.60	0.60	1.14	1.14	1.14	1.14	1.14	1.14	1.14
Section modulus m^3	0.0123	0.0123	0.0212	0.0212	0.1454	0.1454	0.1454	0.1454	0.1454	0.1454	0.1454
Bending stress Mpa	0.00	97.63	56.50	81.96	11.95	16.01	18.90	21.02	22.37	22.97	11.95
Torsion stress Mpa	61.12	61.12	35.37	35.37	5.16	4.63	4.14	3.66	3.18	2.58	0.00
Combined stress Mpa	61.12	127.03	73.51	95.12	13.87	17.25	19.77	21.64	22.81	23.26	11.95
WITHOUT SHELL											
Shaft diameter m	0.50	0.50	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60
Section modulus m^3	0.0123	0.0123	0.0212	0.0212	0.0212	0.0212	0.0212	0.0212	0.0212	0.0212	0.0212
Bending stress Mpa	0.00	97.63	56.50	81.96	81.96	109.83	129.64	144.17	153.41	157.56	81.96
Torsion stress Mpa	61.12	61.12	35.37	35.37	35.37	31.73	28.43	25.12	21.82	17.68	0.00
Combined stress Mpa	61.12	127.03	73.51	95.12	95.12	118.34	135.60	148.42	156.45	159.52	81.96
Stress difference due to shell Mpa	0.00	0.00	0.00	0.00	81.25	101.09	115.83	126.78	133.64	136.26	70.01
STRESS AT SHAFT SURFACE WITH SHELL											
Bending stress Mpa	0.00	97.63	56.50	81.96	6.29	8.43	9.95	11.06	11.77	12.09	6.29
Torsion stress Mpa	61.12	61.12	35.37	35.37	2.71	2.44	2.18	1.93	1.67	1.36	0.00
Combined stress Mpa	61.12	127.03	73.51	95.12	7.30	9.08	10.41	11.39	12.01	12.24	6.29

APPENDIX C

Specification for sugar mill rolls

This specification covers the material, dimensions and quality control applied to roll shafts and shells made of cast iron and shrunk onto steel shafts to be used in sugar mills.

Note: The inspector refers to an inspection authority appointed by the customer.

Materials

Shaft

Mild steel conforming to the specification ISCOR steel SS 10/101 entitled "28/33 Ton Normalised Forged Steel Shafting". Forging to be double normalised.

Shell

Close grained cast iron generally conforming to BS 1452/1977 grade 300. Casting to be homogeneous and free from blowholes and cracks.

Removal of old shell and preparation of used shaft for reshelling

The old shell must be removed in such a way that damage to the shaft is kept to a minimum.

The shaft is to be examined magnetically and ultrasonically by the inspector and any defects are to be reported immediately to the mill engineer.

Building up and machining of worn journals is to be carried out according to the instructions on each order.

The shaft must be stress relieved by holding at a temperature between 580 and 620 degrees Celsius for a period of two hours plus fifteen minutes for every 25 mm of greatest shaft diameter over 50 mm. The shaft must be well supported over its entire length during this stress relieving operation. If any welding has been carried out on the shaft the stress relieving must take place after such welding has been completed.

The shell landing must be machined to a surface finish better than 3,2 micro metres (125 micro inches) over its entire length and any discontinuities in diameter must be blended from one diameter into the other with a minimum radius of 75 mm and polished to a finish better than 3,2 micro metres.

Journals and fillet radii are to be examined carefully for grooves or scratches which may cause stress concentration. Such grooves are to be removed by remachining.

The surface finish on all fillet radii is to be better than 1,6 micro metres.

Dimensional Tolerances

Interference fit between the inside diameter of the shell and the landing diameter of the shaft shall be between 0,0004 and 0,0006 multiplied by the shaft landing diameter, measured when both the shell and the shaft are at ambient temperature.

Tolerances in taper and ovality on the shaft landing and in the bore of the shell shall be such that the interference fit will nowhere exceed the above limits.

If the shell and shaft are stepped to facilitate fitting during shrinkage the clear distance between the step on the shell and the step on the shaft in their final position shall be not more than 50 mm.

Facilities for the inspector

If requested by the mill engineer the shaft and shell are to be made available to the inspector during all stages of manufacture, and in particular the shaft and shell shall be available prior to the shrinking operation while both are at ambient temperature.