The structural design of large grinding mills, with reference to shell-mounted bearings

by D. A. FENTON*, P. Eng. (Visitor)

SYNOPSIS

The paper outlines the advantages of shell-mounted grinding mills over mills having a conventional trunnion-supported shell.

The growing use of larger mills in the past decade has led to a significant number of mechanical and structural failures, the most common being those at the junction of the head and the shell. Larger mill diameters favour the use of shell-mounted bearings.

The elimination of castings, which have an inherent weakness to fracture, makes the shell-supported mill lighter and stronger than a conventional mill of the same shell thickness.

SAMEVATTING

Die referaat behandel in hooftrekke die voordele van rompgemonteerde breekmeule bo meule met 'n konvensionele romp wat deur 'n dratap gesteun word.

Die toenemende gebruik van groter meule gedurende die afgelope dekade het 'n beduidende aantal meganiese en strukturele falings tot gevolg gehad, meestal by die aansluiting van die kop by die romp. Groter meuldiameters bevorder die gebruik van rompgemonteerde laers.

Die uitskakeling van gietstukke wat 'n inherente neiging het om te breek, maak die rompgesteunde meul ligter en sterker as 'n konvensionele meul met dieselfde rompdikte.

Introduction

The diameter and sizes of grinding mills have increased steadily in the past few years. Many small mills have run for long periods of time with low rates of failure, but there have been a significant number of failures in mill shells and heads, some of which have been analysed by the author. Many structural failures have occurred, but they have not been well documented: data on high-cycle fatigue is sparse; scale effect can be truly represented only by a full-size machine; and long time to failure, changes in the conditions of the charge, liner wear, ball or rod size, and thermal effects make documentation difficult. In wet-milling, stress corrosion is a significant factor in the fatigue life of the machine.

Design of a Conventional Mill

The head of a conventional mill is illustrated in Fig. 1. Off-setting of the head–shell junction from the bearing reaction means that fully reversed rotating bending stresses are induced in the head and head–shell junction.

The dotted curve represents the neutral axis N–A of the section resisting the rotating bending stresses. The value of K shown in the small-scale diagram represents the ratio of minimum stress divided by maximum stress, and $K=-1$ represents the worst conditions for fatigue damage. This situation is inherent in the trunnion-mounted ball mill.

Membrane stresses are illustrated in Fig. 2. These stresses are high when compared with the rotating bending stresses in Fig. 1, but they are unidirectional in nature and are not so damaging as a fully reversed stress.

In Figure 2a, the overall bending stresses are combined; that is, the rotating bending due to the offset of bearing reaction in Fig. 1 is combined with the bending of the shell between the bearing supports, behaving as a short beam. The local stress due to the impact of the charge to centrifugal and inertia forces is shown in Fig. 2b.

The nature of these stresses is a local curved-plate bending stress, the distribution of which is illustrated in Fig. 3. The value $K=0$ ratio of minimum stress to maximum stress gives these stresses a unidirectional property that is not so damaging in fatigue as is the fully reversed stress represented by $K=-1$. The magnitude of the local stress, however, can be much greater than the overall stress and must be accounted for in large-diameter mills with short lengths (length-to-diameter ratios of less than 2).

When the above stresses are combined, the effect is to produce concentrations of stress near the junction of the head and shell, which have resulted in many failures in mills. The thickness distribution is also a factor of head failures. The evaluation of correct thickness distribution is complex owing to the conflicting requirements of fatigue (to bring the stress level down) and fracture criteria, which require the thickness of the cast material to be a minimum (owing to the propagation of cracks).

The following are disadvantages of conventional mill design:

(a) heavy head sections, susceptible to fractures or fatigue failures, involving high cost,

(b) high concentrations of stress where the rigid head joins the compliant shell,

(c) restriction of relatively small trunnion diameters, reducing the throughput of the mill, and

(d) relatively high frictional losses in hydrostatic sleeve bearings, representing some 7 to 9 per cent of the mill horsepower supply.

Item (d), the hydrostatic sleeve bearing, requires sustained high pressure to maintain lift and so retain the oil film. The power required to maintain the high pressure is about 4 per cent of the mill horsepower supply. The frictional characteristics of the long ski-like shoe, and the fact that the motion of the journal opposes the pumping action of the lifting fluid along one half of the shoe length, give rise to boundary friction of between

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3 and 5 per cent of the mill horsepower supply. In total, the loss of horsepower is between 7 and 9 per cent of the total horsepower. As mills increase in size, the deflection of the journal causes problems in the oil-film wedge (oil film is wiped), leading to higher pumping capacities to maintain lift.

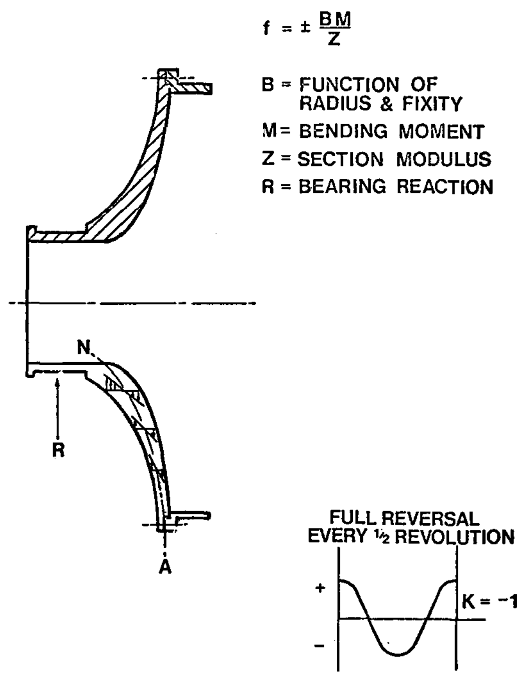

Fig. 1—Rotating bending stresses in the head of a conventional mill

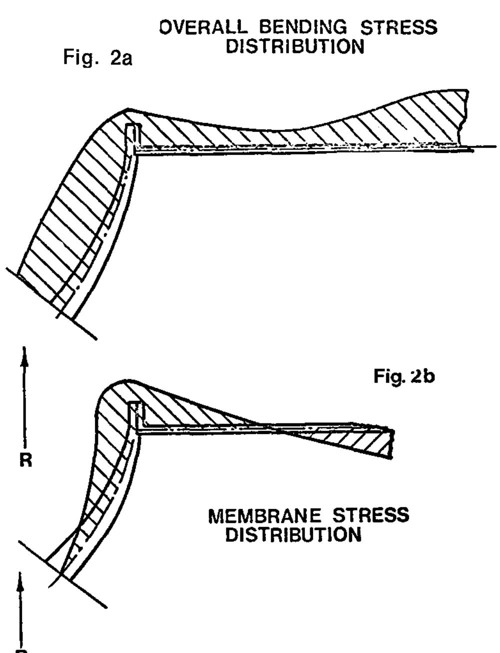

Fig. 2—Distribution of stress in the bearings of a conventional mill

Shell-supported Ball Mills

The shell-supported ball mill and the conventional mill are illustrated in Fig. 3.

The function W is complex in nature and depends on the following:

(a) the mass of the mill shell,
(b) the mass of the lining (a variable subject to wear),
(c) the ball or rod size (affecting the impact pressure),
(d) inertia and impact pressure (a function of liner wear),
(e) disposition of the charge along the mill,
(f) a variable impact zone as the mill rotates, and
(g) scale effect.

The function C is a dimensionless constant and varies with the geometry and constraint of the shell section. The value of C varies between 0,4 and 0,75.

It is evident that the shell-supported mill requires less space than does the conventional mill, and the distance between the bearing reactions is less than that of a conventional mill with the same working length. Thus, the bending moment and rotating bending stresses are less for the shell-supported mill of identical shell thickness.

The distribution of membrane and rotating bending stress is shown in Fig. 4. The dotted curves reflect the effect of reinforcing round manholes and the local thickening of the shell due to stiffness requirements of the journal. The thickening of the shell reduces the membrane stresses and improves the fatigue life of the structure.

Fig. 5 illustrates the stresses and geometry of the journal. The bending stresses refer to circumferential bending,

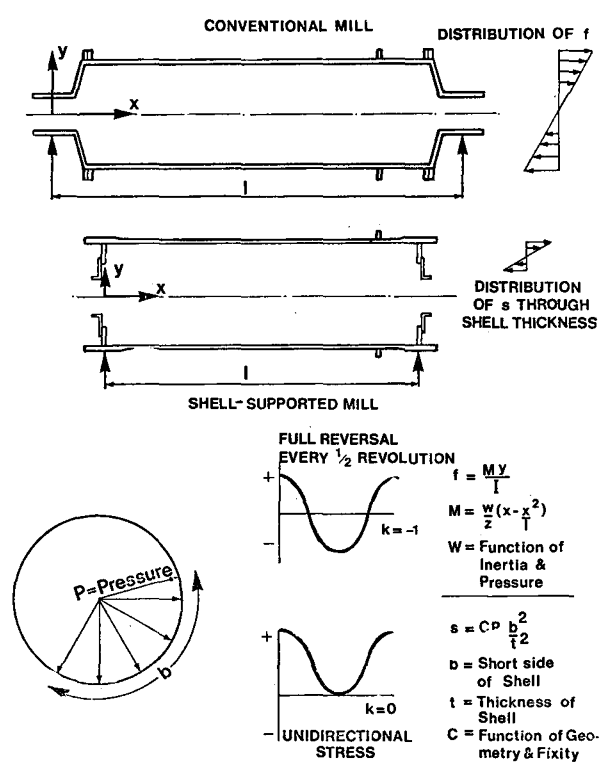

Fig. 3—Stresses in a conventional mill and a shell-supported mill

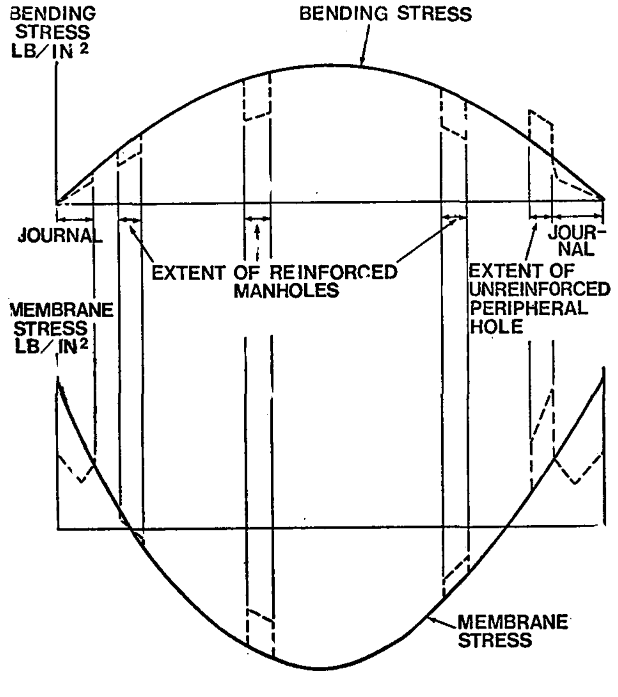

Fig. 4—Distribution of overall bending stress and membrane stress in a shell-supported mill

and the distribution of these is shown. Fatigue considerations are the criteria for the design. The bending stresses are fully reversed twice in each revolution of the mill. The shear stresses also vary round the journal circumference, and the local distribution is illustrated. *A2* and *T2* refer to the effective length of the journal and its thickness.

A4 and *T4* refer to the depth and thickness of the vestigial head. The torsional shears are generated by the thrust force *F* and the offset of the shoe pressure from the shear centre of the structure. *N–A* on the diagram refers to the neutral axis of the curved beam.

The deflection of the journal is compensated for by the rotation of the shoes to conform to the motion of the journal, and journal deflections will not interfere with the ~~thickness~~ of the oil film.

It should be noted that the closeness of the welds to the neutral axis of the curved beam is an advantageous design feature. When the curved beam is analyzed for torsional and flexural shears, particular attention being paid to the analysis of the welds and shell structure for fatigue design, the following are apparent. The rigid constraint requirement for hydrostatic bearings is eliminated since the bearing shoes are radially adjustable, and benefit is derived from the fabricated sections. Local bending of the shell must be allowed for, and it is desirable for a gear flange to be mounted where the membrane stresses are low.

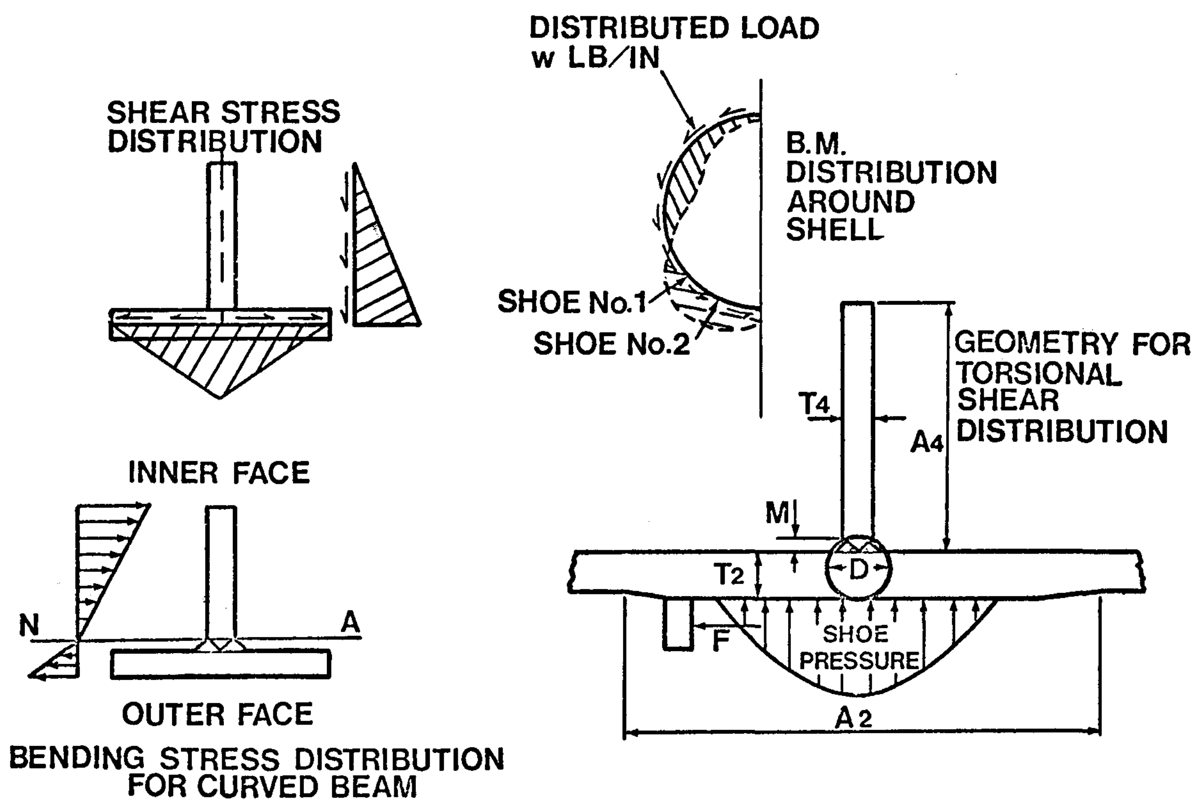

Fig. 5—The stresses and geometry of the journal in a shell-supported mill

The analysis shows the advantages of the shell-supported ball mill to be as follows:

(a) relatively light head structure,
(b) low stress levels at previous critical sections of the shell,
(c) possibility of large openings in the feed and discharge end of the mill,
(d) significant space saving feature due to elimination of trunnions, and
(e) very low frictional losses in the hydrodynamic bearings, representing 1 to 1,5 per cent of the mill horsepower supply.

Fatigue in Shell Design

The stress reversibility factor and typical fatigue stress are illustrated in Fig. 6. The full line represents 2×10^6 cycles, and the dotted curve represents limiting stresses for twenty years' continuous running for a typical ball mill 6 m in diameter. In the determination of safe levels of working stress, a number of factors are considered, including stress raisers, type of stress, torsional and flexural shears, rotating bending stress, membrane stress, and combinations of these.

The statistical nature of the fatigue data is significant. A wide scatter of results means that, for long life and to ensure low probability of failure, the maximum stresses permitted should not be based on a median value of the data but rather on the lower limit of the scatter.

Typically, $1,4 \times 10^8$ cycles for twenty years' continuous running of a 6 m-diameter ball mill is a measure of the number of cycles required. Many mills have failed in less than five years of operation, some of them because of weakness in the design features mentioned.

Pivoted-shoe Hydrodynamic Bearings

The Aerofall bearing utilizes four pivoted hydrodynamic shoes, the short discrete lengths of shoe permitting true hydrodynamic action. The shoes are supported on spherical pivots in such a manner that they can rock freely in all directions and are radially adjustable to conform to the shape of the journal. The larger the diameter of the mill the better are the hydrodynamic characteristics. These features make the shell-supported mill a reality.

The computer facility at Aerofall has made rapid optimizations possible for shell structure, bearing design, mass analysis, and inertia characteristics so that motors can be matched to the mill.

Fig. 7 illustrates, in carpet form, the distribution of shoe-aspect ratio for various pressures and shoe lengths. The data were obtained from the mill inertia characteristics, and the viscosity, kinematic, and thermal conditions of the oil. The aspect ratio is given by B/L, and the minimum thickness of the oil film is denoted by H_o.

The limit in pressure on the shoe is consistent with the average bearing pressures to sustain an oil film. The limit in shoe length is the physical limitation of the bearing shoes consistent with the geometry of the bearing.

It is seen that there is a reduction in minimum thickness of oil film with decreasing values of B/L. The choice of a nearby square shoe with a relatively thick minimum oil film is desirable. A long ski-type of shoe,

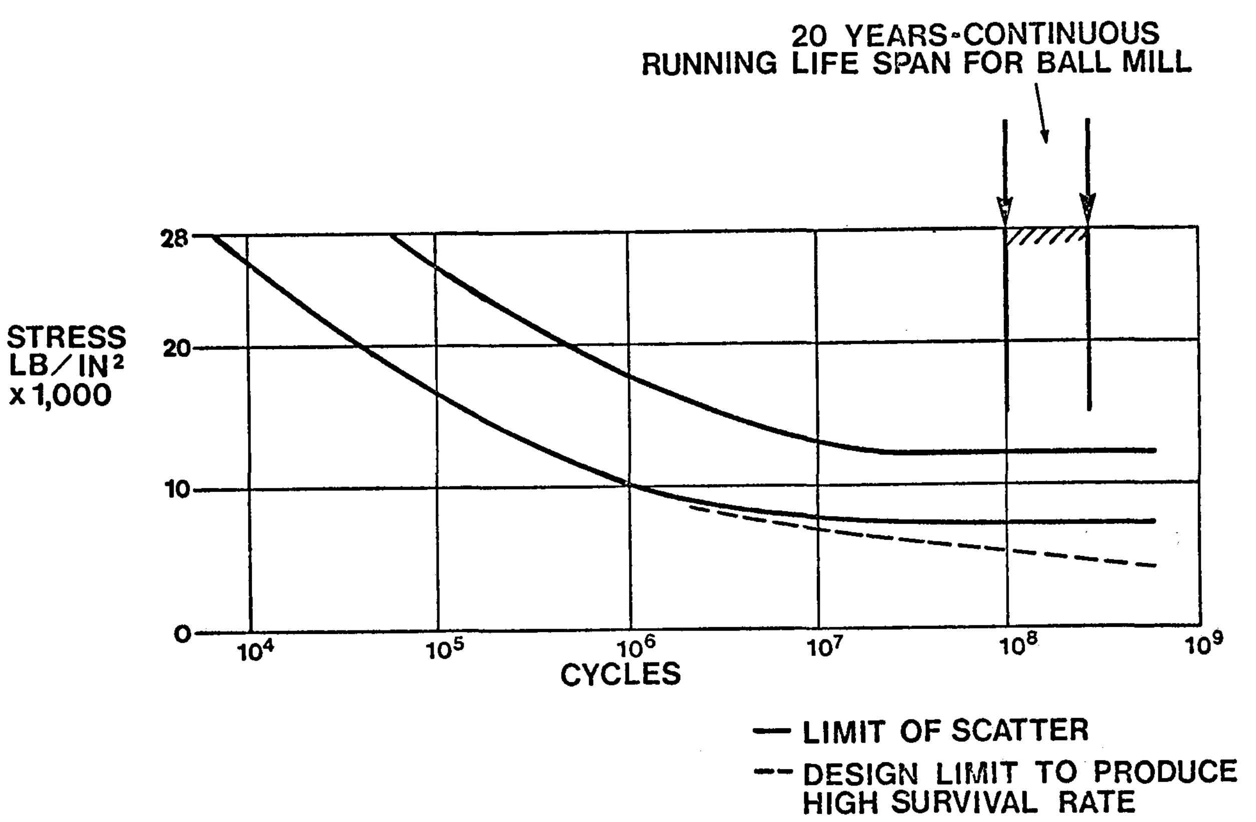

Fig. 6—Stress versus number of cycles for mild steel

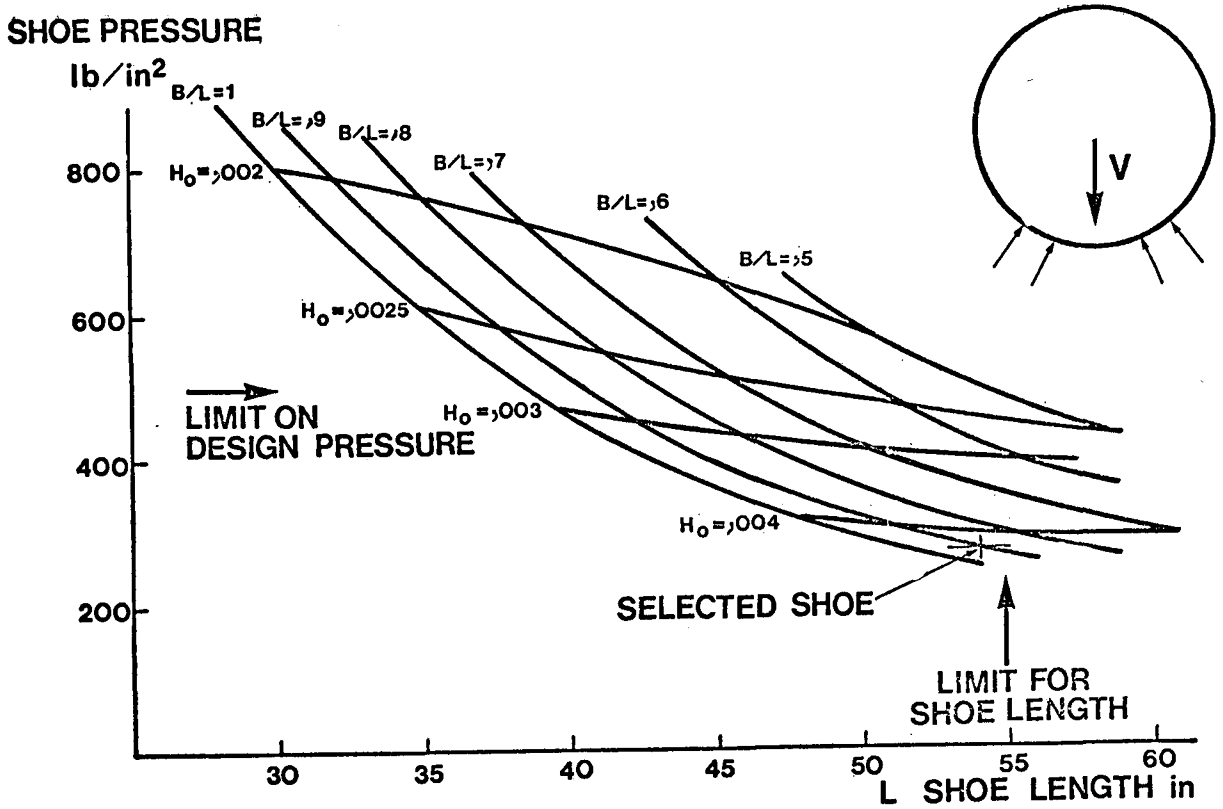

Fig. 7—The distribution of shoe-aspect ratio

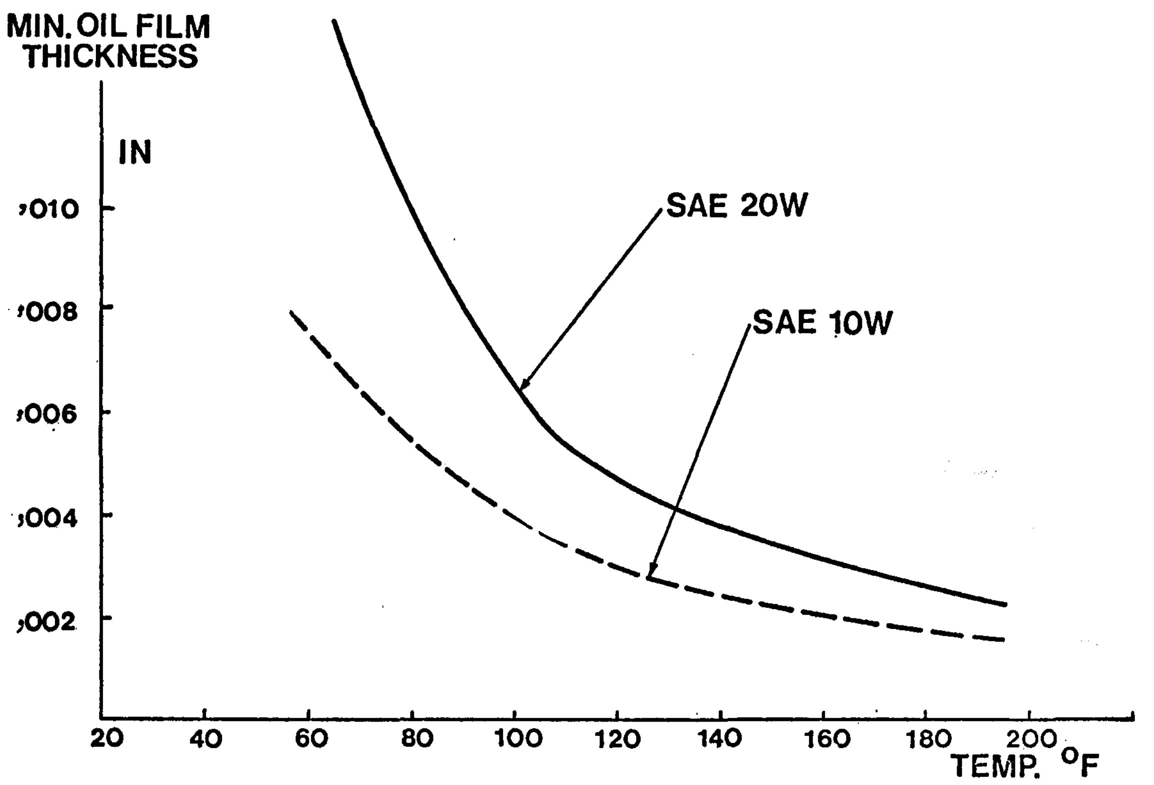

Fig. 8—The effect of temperature on the minimum thickness of oil film

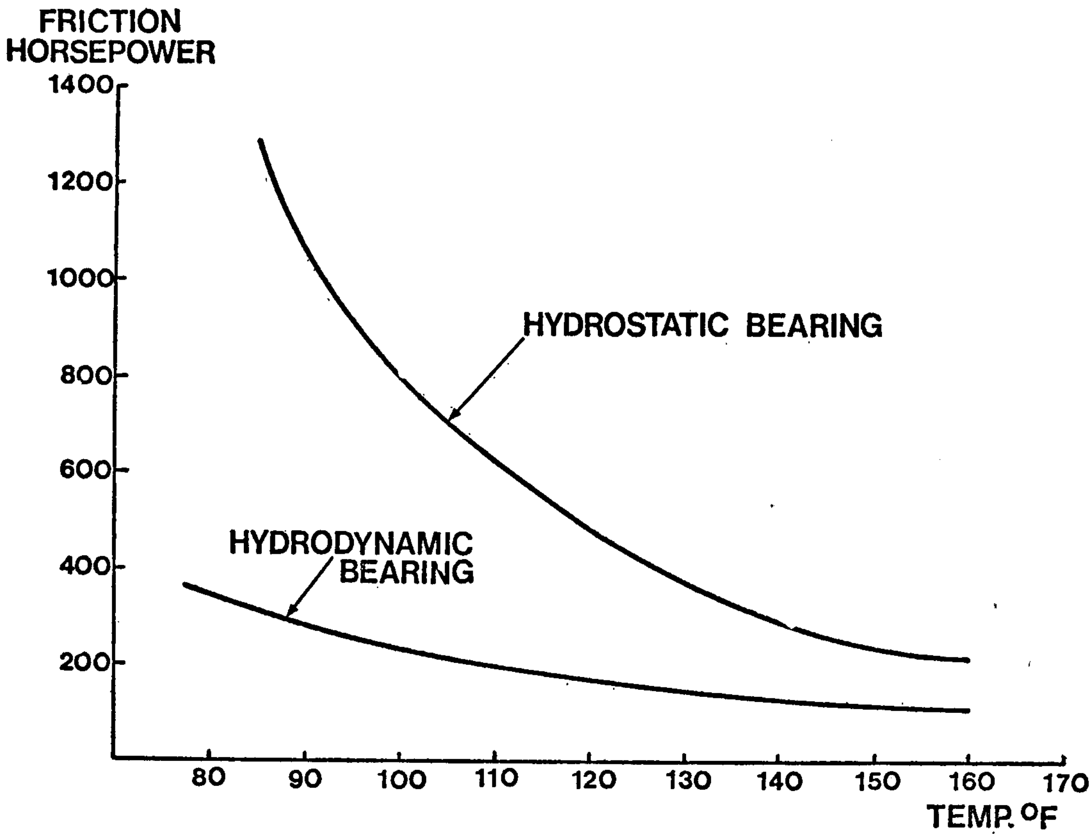

Fig. 9—Friction characteristics of hydrostatic and hydro- dynamic bearings

characterized by $B/L \leqslant 0{,}5$, would give low oil-film thicknesses and greater possibility of film breakdown and high boundary friction. This would be characteristic of a hydrostatic sleeve bearing with heavy load/diameter.

Fig. 8 shows the effect of temperature on the minimum oil-film thickness. Thermal balance must be obtained by cooling methods if shell temperatures reach high values.

Fig. 9 illustrates the friction characteristics of the hydrostatic and hydrodynamic bearings for a large-diameter ball mill. Long ski-type shoes that have hydrodynamic action have friction characteristics between the two extremes. This is because the shoe efficiency is less than that of a hydrodynamic bearing and a higher leakage rate. Reference to Fig. 7 shows the limit of such shoes when applied to shells of large diameter.

Conclusion

The conflicting requirements of fatigue stress versus cycles to failure, and the growth of fracture cracks versus cycles to failure, pose a problem to the designer. By the elimination of one of the requirements, e.g., the fracture criteria (elimination of thick heavy castings with relatively small trunnions and inherent stress concentration factors), the designer of shell-supported ball mills has one less possible failure to consider at the design stage.

Statistical data on the high-cycle fatigue of mild-steel structures are very sparse at the present time. Small-scale fatigue tests are not representative but can be used as a guide, particularly with the range of the scatter band. However, field failures of full-scale structures are the best guide to representative stresses in design.

Failures in large grinding mills have happened frequently. Although defects in materials and workmanship can be blamed for some of the failures, many of the features contributing to failures are incorporated at the design stage.

The hydrodynamic pivoted-shoe bearing has significant advantages structurally, as well as in utility, over the hydrostatic sleeve bearing, or over small-diameter bearings with long ski-type shoes that have hydrodynamic characteristics. The capital cost of the hydrodynamic bearing is secondary after a few years running.

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Discussion of the previous paper

E. N. H. MOLLOY*

The first 22-foot-diameter Aerofall mill was installed at Mangula in 1957. This was fitted with cast-iron trunnions located to the mill shell with 2-inch-diameter mild-steel bolts. The cast-iron trunnion on the discharge side started to crack in 1960, and this was accompanied by frequent breakages of the 2-inch-diameter steel bolts. This occurred thirty-seven months after the commissioning of the mill.

The second 22-foot-diameter Aerofall mill, installed in 1959, was fitted with the same trunnions and the same size bolts as those used in the first mill. The discharge trunnion developed cracks late in 1960, seventeen months after commissioning. As with the first mill, great difficulties were experienced with frequent breakages of the 2-inch-diameter bolts.

Two changes were made in the design of the mills. New discharge trunnions made of cast steel to specification ASTM/A27/65/35 were fitted, and the 2-inch-diameter mild-steel bolts were replaced with high-tensile bolts of 2¼ inch diameter. The cast-steel trunnions have stood up well during the past thirteen years, and failure of the bolts has been negligible. The trunnions are mounted on Michell-type bearings, which have proved very efficient.

Comparisons with plants handling equivalent tonnages have shown that the cost of dry grinding in the Aerofall mills at Mangula is lower than that at concentrators using conventional crushing and grinding equipment.

*The Messina (Tvl) Development Company, Johannesburg.

O.F.S. Branch

Minutes of the General Meeting held at the Welkom Club on Wednesday, 2nd June, 1976, at 4.00 p.m.

Mr G. J. C. Young (Chairman of the O.F.S. Branch) was in the Chair. Also present were:

Two Fellows Messrs Z. J. Lombard (Committee Member), D. A. Smith (Committee Member).
Two Members Messrs R. W. Impey, D. I. Watson.
Four Associates Messrs A. J. Johansen, W. F. de Lange, J. Scott, C. P. Visser.
Two Graduates Messrs A. P. S. Howard, P. S. Wentworth.
One Student Mr D. A. Arnold.
Thirty-two Visitors.
Total Present Forty-four.
Mr Young declared the Meeting open and extended a welcome to the members and visitors present.

Apologies

Apologies for non-attendance were received from B. J. Drysdale, J. Lorenzon, A. N. Shand, and L. Vorster.

Minutes of Previous General Meeting

The Minutes of the General Meeting held on the 11th February, 1976, were taken as read, and their adoption, proposed by Mr D. A. Smith and seconded by Mr Z. J. Lombard, was carried.

General Business

Mr Young reported that Mr R. R. Perkin, Honorary Secretary of the O.F.S. Branch for the past two years, had recently been transferred to Johannesburg and had had to resign his office. On behalf of the Committee, Mr Young expressed his thanks to Mr Perkin for the hard work that he had put into the local branch. Mr A. R. Godfrey had been asked by the Chairman to perform the duties of Honorary Secretary until such time as a new Honorary Secretary could be appointed.

Talks on Nuclear Instrumentation

Mr Young introduced Mr J. W. Tongs and Mr J. G. Barnard of Texas Nuclear (S.A.) (Pty) Ltd, and called on them to give a talk on the uses of nuclear instrumentation in industry.

Mr Tongs began by outlining the basics of radiation and nuclear energy, and how these were utilized in nuclear mass meters and density meters. Mr Barnard spoke about the problems involved in weighing and in the calibration of mass meters. He outlined the way in which many of these problems could be overcome with the use of nuclear instruments. A demonstration of the important features of the Texas Nuclear meter concluded the presentation, which attracted a lively discussion.

Mr Young thanked Mr Tongs and Mr Barnard on behalf of those present for their very interesting presentation.

Closure

The Chairman thanked members and visitors for their attendance, and declared the Meeting closed at 7.00 p.m. He thanked Texas Nuclear (S.A.) (Pty) Ltd for very kindly providing the refreshments served after the meeting.