The control of autogenous and semi-autogenous mills: the relationship between measurements of bearing pressure and parameters describing the mill load

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SYNOPSIS

The capacity of grinding mills can be maximized by use of measurements of mill power and bearing pressures. The criterion is as follows. If the derivatives with respect to time of the power and bearing-pressure signals have the same sign, the volume of the mill load is smaller than the optimum value; if they have different signs, the volume of the mill load is greater than the optimum value. Few attempts have been made to use measurements of bearing pressure in the estimation of absolute values for the parameters that define the state of the mill load (e.g, mass, volume, and dynamic angle of repose) because the bearing-pressure signal is a complex function of these and other parameters. This paper gives a mathematical model that relates bearing pressures to these parameters. The theoretical values obtained from this model are compared with experimental values, and it is concluded that the model constitutes a good description of experimental reality. If accurate measurements of the load supported by each bearing are available, the model can be used in the estimation of the mass of the load in the mill. If independent estimates of the volume of the load or the dynamic angle of repose are available, the other parameter can be estimated from measurements of the bearing loads.

SAMEVATTING

Metings van die meuldrywing en laerdrukke kan gebruik word om die drywing van breekmeule te maksimeer. Die maatstaf is soos volg. As die afgeleide tydwaardes van die drywing- en laerdrukseine dieselfde teken het, is die volume van die meullading kleiner as die optimale waarde; as hulle verskillende tekens het, is die volume van die meullading groter as die optimale waarde. Daar is maar min pogings aangewend om metings van die laerdruk te gebruik om absolute waardes vir die parameters wat die toestand van die meullading (bv. massa, volume en dinamiese rushoek) bepaal, te beraam omdat die laerdruksein 'n ingewikkelde funksie van hierdie en ander parameters is. Hierdie referaat gee 'n wiskundige model wat laerdrukke met hierdie parameters in verband bring. Die teoretiese waardes wat van hierdie model verkry is, word met die eksperimentele waardes vergelyk en die gevolgtrekking is dat die model 'n goeie beskrywing van die eksperimentele werklikheid is. As daar akkurate metings van die las wat op elke laer rus, beskikbaar is, kan die model gebruik word vir die beraming van die massa van die lading in die meul. As daar onafhanklike ramings van die volume van die lading of die dinamiese rushoek beskikbaar is, kan die ander parameter aan die hand van metings van die laerlaste beraam word.

Introduction

A number of authors[1-4] have reported the use of measurements of power and bearing pressure to control the feed rate to a mill so that the power drawn by the mill is at a maximum. The following criteria are used:

(1) if the power and bearing pressure are increasing, the volume of the mill load is smaller than the optimum value and the feed rate can be increased, and

(2) if the power is decreasing and the bearing pressure increasing, the mill is overloaded and the feed rate must be reduced until both the power and the bearing pressure decrease.

Except for Oswald and Ziegler[1], who used measurements of bearing pressures to control the inventory of ball mills, none of the authors already mentioned[2-4] reports any attempt to use the bearing-pressure signal to obtain quantitative estimates of the mass or volume of the mill load. The reason cited is that a given bearing pressure does not correspond to a given load mass. Bearing pressure is, in fact, a function of many variables in addition to the mass of the load, e.g., volume of the load, the power drawn by the mill, the angle of repose of the load (which, in turn, is a function of the speed of rotation of the mill, the amount of wear on the liners, the size distribution of the load, and the viscosity of the pulp), the amount of ore packed in the liners, the design of the bearings, the oil temperature, and so on. Therefore, as indicated above, only the direction of change of the bearing-pressure signal is taken into account, on the assumption that this is a good indication of the direction of change in the load mass.

The question arises as to whether it is possible – at least in principle – for better use to be made of bearing-pressure measurements for the identification of load parameters such as mass, volume, and angle of repose. These three variables are chosen because they give a comprehensive definition of the conditions of the load. If the mass and the volume of the load are known, its density can be calculated. This, in turn, can be related to the mass of steel or pulp, or both, in the load, the voidage, and so on. The maximum capacity is generally reached at a load volume of 45 to 50 per cent. The angle of repose provides one with an estimate of the ability of the load to absorb power, and is related to the relative amounts of medium and pulp in the load, and the viscosity of the pulp. There appear to be no publications giving a detailed analysis of bearing pressure as affected by these variables. It is the purpose of this paper to provide such an analysis – an analysis that will make it clearer to workers in this field why, at times, the bearing-pressure signal exhibits apparently anomalous behaviour, and that will provide information enabling the best use to be made of this signal in the control of mills.

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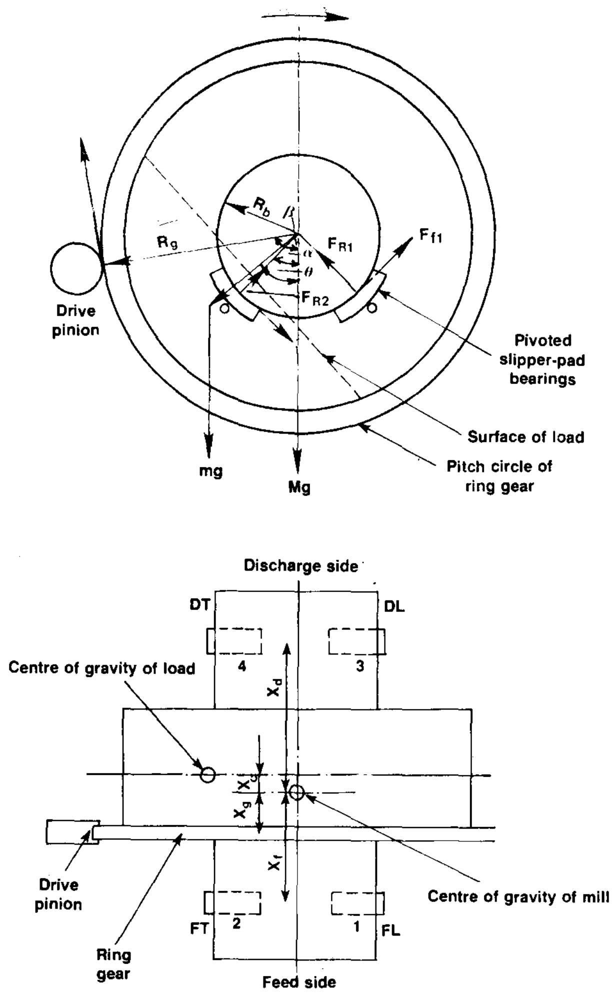

Fig. 1—Schematic diagram of the NIM Aerofall mill

TABLE I

VALUES OF THE PARAMETERS USED IN THE SIMULATIONS
(see also Fig. 1)

Diameter of mill inside lifter base	D	= 1,7m
Radius of ring-gear pitch circle	R_g	= 1,06m
Radius of trunnion-bearing surface	R_b	= 0,294m
Length of mill inside liners	L	= 0,505m
Distance between centre of gravity of the mill and feed bearings	x_f	= 0,542m
Distance between centre of gravity of the mill and discharge bearings	x_d	= 0,627m
Distance between centre of gravity of the mill and mill centre	x_c	= 0,043m
Distance between centre of gravity of the mill and ring gear	x_g	= 0,32m
Angle of displacement of mill bearing	α	= 45°
Angle of displacement of drive pinion	β	= 80,8°
Dynamic angle of repose of the load	θ	= 47,5°
Mass of mill plus ring gear	M	= 4000 kg
Surface area of one bearing	A	= 0,0406 m²

Modifications were incorporated in the 1,7m Aerofall pilot mill at the National Institute for Metallurgy (NIM) to permit measurements of the bearing pressure and the temperature of the oil on each of the four slipper-pad bearings that support the mill. The results obtained in various experiments were then compared with those obtained from a theoretical model of the system.

The Generation of Estimates of Bearing-oil Pressures

The NIM Aerofall mill (Fig. 1) is supported on four slipper-pad bearings, and is driven by a pinion acting on a gear ring near the feed side of the mill. R_g is the radius of the gear ring, R_b is the radius of the trunnion bearing, and R_L is the distance between the centre of gravity of the load and the mill axis. The bearings, pinion gear, and dynamic load are displaced at angles α, β, and θ, respectively, from an imaginary vertical line through the mill axis. F_g is the force exerted on the mill by the pinion, F_{Ri} are the radial forces exerted by each bearing on the bearing surfaces, and F_{fi} are the frictional components. M and m are the mass of the mill and of the load respectively. The heavy ring gear near the feed trunnion ensues that the centre of gravity of the mill (the shell and trunnion bearings excluding the load) is near the feed

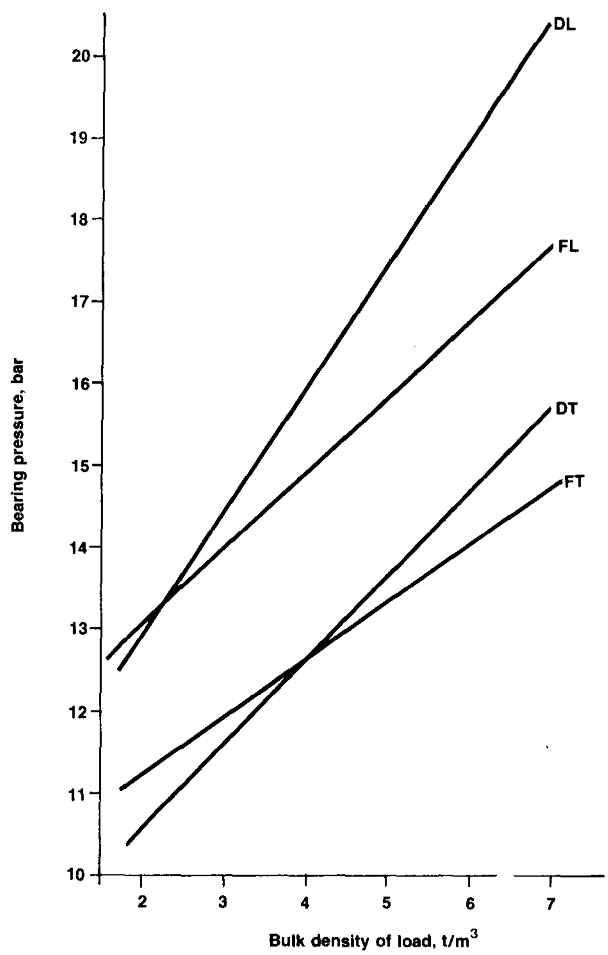

Fig. 2—Effect of the bulk density of the load on bearing pressures

side of the mill. The values of the parameters shown in Fig. 1 are listed in Table I.

Vertical and horizontal force balances yield the following equations:

Vertical: $F_g.\sin\beta + \sum_{1} F_{Ri} . \cos a +$

$$(F_{f1} + F_{f3} - F_{f2} - F_{f4})\sin a = (M+m)_g \quad . (1)$$

Horizontal: $F_g.\cos\beta + (F_{R1} + F_{R3} - F_{R4})\sin a =$

$$\Sigma_1 F_{fi} . \cos a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Moments about the centre of gravity of the mill in a vertical plane containing the axis yield

$$\{x_d\,(F_{R4} + F_{R3}) - x_F(F_{R1} + F_{R2})\}\cos a +$$
$$\{x_d(F_{f3} - F_{f4}) - x_f(F_{f1} - F_{f2})\}\sin a$$
$$= x_g.F_g.\sin\beta + m.g.x_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Moments about the centre of gravity of the mill in a horizontal plane containing the axis yield

$$\{x_f(F_{f1} + F_{f2}) - x_d(F_{f3} + F_{f4})\}\cos a +$$
$$\{x_f(F_{R1} - F_{R2}) + x_d(F_{R3} - F_{R4})\}\sin a = x_g.F_g.\cos\beta \quad . (4)$$

Finally, angular moments about the axis of the mill yield

$$R_g.F_g = R_b.\sum_{1} F_{fi} + m.g.R_L.\sin\theta \quad . \quad . \quad . \quad . \quad . \quad (5)$$

In practice, it is sometimes not possible for a measurement of the F_{Ri} to be obtained, since the bearing pressure is distributed non-uniformly over the bearing surface and only the measurement of pressure at a single point is practical. However, if it is assumed that F_{Ri} is proportional to the bearing pressure and to the surface area of the bearing, A, then

$$F_{Ri} = \psi_i.A.P_i.$$

The frictional forces F_{fi} are complex functions of bearing design, oil viscosity and therefore temperature, thickness of oil film, and other factors. It is assumed simply that they are proportional to the corresponding F_{Ri}. Then

$$F_{fi} = \mu.F_{Ri} = \mu.\psi_i.A.P_i. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where μ is the friction coefficient.

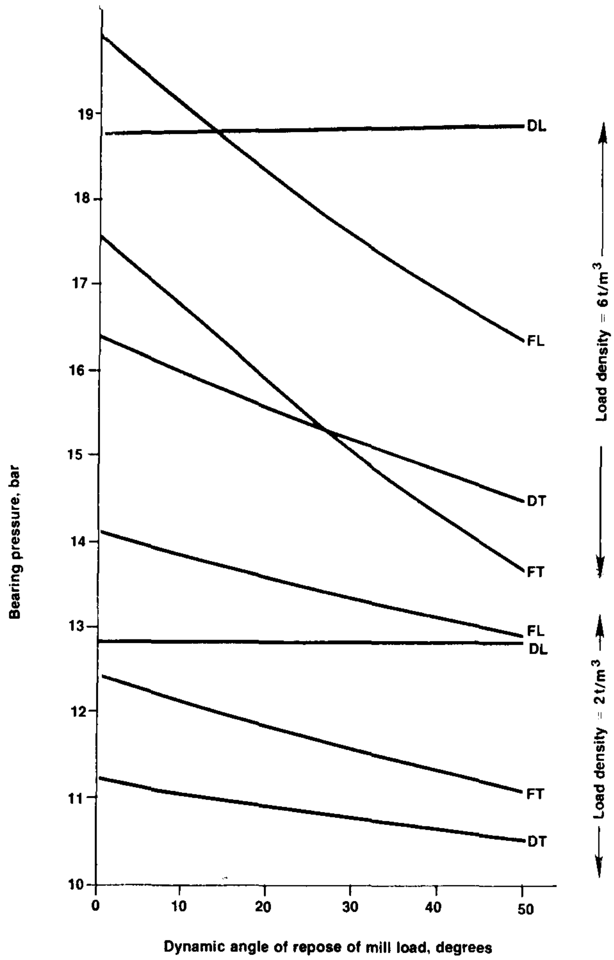

Fig. 3—Effect of the dynamic angle of repose of the mill load on bearing pressures for load bulk densities of 2,0 and 6,0 t/m³

TABLE II

RESULTS OF THE SIMULATION OF BEARING-OIL PRESSURE

No.	Parameters varied and conditions observed in the simulation	Fig.	Definition of bearing-pressure sensitivity	Bearing-pressure sensitivity for bearings			
				FL	FT	DL	DT
1	Load density varied from 2,0 to 6,0 t/m³ at 50% filling of mill volume	2	$\delta P/\delta\rho$ (bar.m³/t at 50% filling of mill volume)	0,918	0,70	1,51	1,01
2	Dynamic angle of repose of mill load varied from 0 to 50° at 50% filling of mill volume for bulk densities of 2,0 and 6,0 t/m³	3	$\delta P/\delta\theta$ (bar/degree at 50% filling of mill and 45° angle of repose)	$\rho = 2,0$ 0,019 $\rho = 6,0$ 0,057	0,021 0,062	0,0 0,0	0,02 0,031
3	Mill load (%) in liners varied from 0 to 20% at 50% filling of mill volume for a bulk density of 5,0 t/m³	4	$\delta P/\delta y$ (bar/% of mill load in liners at 50% filling of mill)	0,017	0,025	0,006	0,005
4 (*a*)	Axial load varied from 0 to 1000 kg. Ordinate in Fig. 5 is volume % for an equivalent mass of rock assuming bulk density = 2,0 t/m³	5	$\delta V/\delta P$ (% filling of mill per bar)	17,6	20,0	20,1	23,0
4 (*b*)	Normal dynamic load varying from 0 to 1 100 kg. Bulk density = 2,0 t/m³	5	$\delta V/\delta P$ (% filling of mill per bar at 50% filling of mill volume)	18,8	21,3	18,5	21,3
5	Bearing friction forces doubled by doubling of the bearing friction coefficient. Load of bulk density 2,0 varying from 0 to 1100 kg	—	$\delta P/\delta\mu$ (bar per % change in bearing-friction coefficient)	0,0036	0,0087	0,0049	0,007

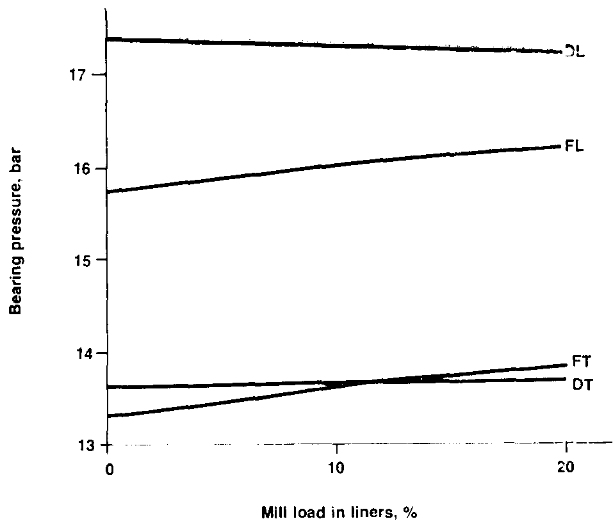

Fig. 4—Effect of the liner packing on bearing oil pressures

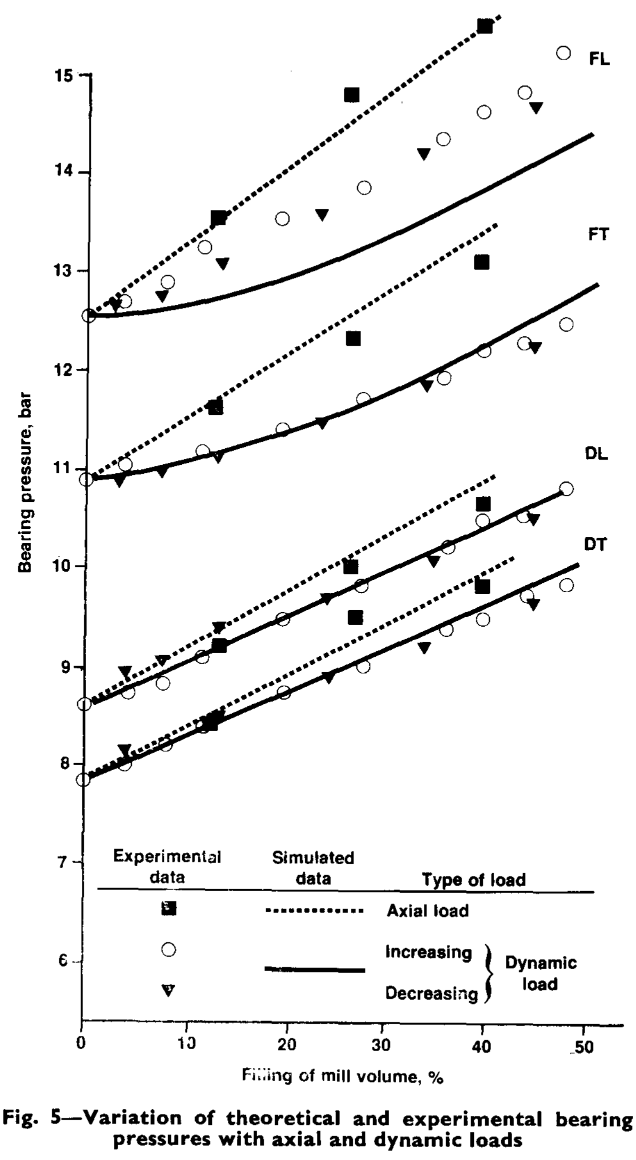

Fig. 5—Variation of theoretical and experimental bearing pressures with axial and dynamic loads

A value is given to ψ_1 to ensure that the simulated bearing pressure at zero load is equal to the value obtained experimentally for each bearing. The use of equation (5) with $m = 0$ and estimates of power absorbed by the bearings at zero load conditions yields a value for μ.

It is easily shown that R_L, the distance between the centre of gravity of the load and the axis of the mill, can be obtained theoretically from a knowledge of the value of the radius of the mill, R, and the volume of the load, V_L:

$$R_L = 2L(R^2 - H^2)^{3/2}/3V_L,$$

where H is the perpendicular distance between the planar surface of the static load and the axis of the mill, and is easily measured or related analytically to V_L.

From measurements of power drawn by the mill as a function of load volume and density, it was shown that, for the experiments described in this paper, the dynamic angle of repose, θ, is 47,5°C.

Equation (5) can then be used to relate F_g to the load parameters and the bearing pressures. This expression for F_g is substituted into equations (1) to (4), and these equations, in turn, can be rearranged to yield the following matrix equation in terms of the bearing pressure vector $\overline{P}$:

$$\overline{\overline{Y}}.\overline{P}. = \overline{Z} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

Once the values for the mass, m, and bulk density, ρ, of the mill load have been specified, these equations and assumptions allow all the elements in $\overline{\overline{Y}}$ and $\overline{Z}$ to be evaluated. The matrix equation can then be solved for $\overline{P}$.

Prediction of Bearing-oil Pressures

The model permits the prediction of the pressure on each of the four bearings as a function of the mass of the mill load, its dynamic angle of repose and bulk density, the mass and dimensions of the mill, and the friction forces on the bearings. For convenience, the bearings are referred to as follows: FL is the 'feed-leading' bearing (i.e., the lead bearing on the feed trunnion), and FT is the 'feed-trailing' bearing, DL the 'discharge-leading' bearing, and DT the 'discharge-trailing' bearing. Simulations of the behaviour of bearing-oil pressure are listed in Table II, which also gives quantitative measures of the effect of the parameters varied in the simulations. These are illustrated in Figs. 2 to 5.

The following points should be noted.

(*a*) The variation in load density has more effect on the responses of the DL and DT bearings than on those of the FL and FT bearings, since, with an increase in the density of the load, the centre of gravity of the mill-load combination moves towards the discharge side of the mill, and there is greater input of power (and therefore lifting force) near the feed side of the mill.

(*b*) The dynamic angle of repose influences the amount of power drawn, and hence the lifting force applied by the pinion, with the result that the responses of the FL and FT bearings are greater than those of the DL and DT bearings. In particular, the response of the DL bearing is negligible.

(*c*) The packing of the liners with ore influences the pressure responses of the DL and DT bearings much less than it affects those of the FL and FT bearings, because, as the volume of ore packed in the liners increases, the mill volume decreases and so does the power, resulting in an increase in the pressures on the bearings closest to the drive pinion.

(*d*) There is a difference between the responses to the axial and the dynamic load pressure because, for the dynamic load pressure, the increased power input counteracts part of the mass of the load, and the bearing pressures at a given load are decreased. This is more marked on the FL and FT bearings because they are closest to the drive pinion. The DL and DT bearings show an almost linear response to variations in dynamic load.

(*e*) The effect of an increase in the friction coefficient of the bearings is an increase in the responses of the lead bearings and a decrease in the responses of the trailing bearings by a value that is independent of the load in the mill, i.e. a change in friction coefficient simply displaces the response by a constant value.

If a single measurement of bearing pressure is to be used in an attempt to control the load volume, the signal from either the DL or the DT bearings should be used because of their almost linear load response, low sensitivity to angle of repose, and negligible dependence on the amount of material packed in the liners. However, the effect of bulk density on these signals is larger than that on the FL and FT signals.

Before an attempt is made to use this simulator for the design of more rational control strategies, consideration is given to the question of how well the simulator describes experimental reality.

Experimental Measurement of Bearing-oil Pressure

Trumpler[5] discusses the distribution of bearing pressure across the surface of a bearing pad. By use of his equations, it can be shown that the maximum pressure occurs at a point that is displaced from the leading edge of the pad by about 60 per cent of the length of the pad. The slipper pads of the mill were accordingly drilled and tapped for the measurement of bearing pressure at that point (Fig. 6). A temperature-sensitive thermistor was mounted on the trailing edge of the bearing so that the temperature of the oil leaving the bearing could be monitored. The pressure ducts from each bearing entered a manifold and, when one of four taps was opened, the pressure on a bearing could be measured. A Kyowa PG20KU pressure transducer produced an electronic pressure signal, and this, together with bearing-temperature signals, was recorded on a flatbed recorder. Thermometers located on the plant and a pressure gauge on the pressure line were used in the calibration of the signals.

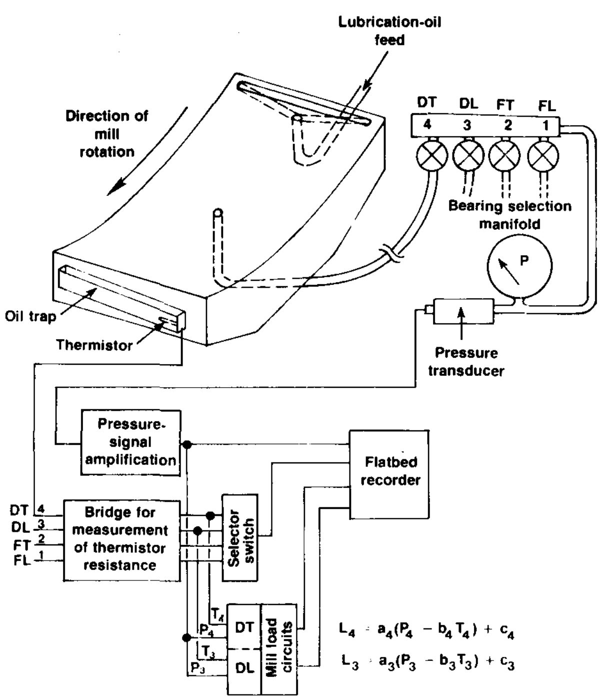

Fig. 6—Schematic representation of the system for the measurement of bearing temperature and pressure, and for the production of a load signal as a function of these two variables

The first experiment involved a determination of the effect of axial load on the oil pressure of each bearing (as a function of the oil temperature). Six lead weights of 166 kg each were cast, which could be mounted on a bar concentric with the axis of the mill. Figs. 7 and 8 show the variation of bearing pressure with bearing temperature for the FL and FT bearings and the DL and DT bearings respectively. The four loci are for axial loads of 0 kg, 333 kg, 666 kg, and 1000 kg respectively. The anomalous behaviour of the signals, particularly for the DL and DT bearings, indicates that other variables may also have an effect on the oil pressure of the bearing. One such variable would be the temperature of the oil pumped to the bearings; another cannot be quantified and relates to the design of the bearing itself. Each bearing is pivoted on a ball bearing, which allows it to alter its position in response to imperfections in the trunnion surface. The quality of the engineering of the bearing could have a major effect on the bearing-oil pressure.

In the second experiment, the mill (with discharge ports closed) was run for a few hours without any load, and the bearing pressures and temperatures were monitored (Fig. 9). When these variables had almost reached steady state (i.e., when the heat generated by friction was equal to the heat lost to the atmosphere by the mill), batches of a pyroxenite ore of a known mass were fed to the mill. Pressures, temperatures, and mill power were measured after each addition of ore. When 1100 kg of ore had been added, the covers of the discharge ports were removed, and the product discharged was collected in drums at recorded times so that the variation in the mass of load with time was known. Previous measurements of the bulk density of the ore allowed the percentage volume occupied by the load to be calculated. The bearing pressures for each bearing are plotted in Fig. 5 as a function of the percentage volume for both increasing and decreasing load conditions. Also plotted in Fig. 5 are the pressures caused by the imposition of axial loads (at a bearing temperature of 30°C, which were normalized so that the pressures at zero axial load were equal to the pressures measured for each bearing before the batch addition of ore was started).

The model described earlier was used to simulate the results obtained from these experiments. The parameter ψ_1 for each bearing was adjusted so that the simulated and experimental bearing pressures for the empty mill coincided. No other attempt was made to fit the model to the results. The correspondence between theory and practice is substantial, except for the variations of the FL pressure with dynamic load volume, where the experimental values are substantially higher than the theoretical values. One could speculate on the reasons for this, but it seems that bearing design and adjustment are the most likely causes. However, with this exception noted, the high degree of correlation between the theoretical and the experimental values allows one to conclude, with a certain amount of confidence, that the model constitutes a good description of the dynamics of the behaviour of bearing-pressure signals.

In the third experiment, an attempt was made to infer the mass of the mill load from measurements of bearing temperature and pressure. Information from the second experiment was used in the design of an electronic circuit that would account for the effect of bearing temperature on bearing pressure. On the assumption that this is a linear effect for small changes in temperature, and that, at any given temperature, the mass of the load is proportional to bearing pressure (as has been shown both theoretically and experimentally for the DL and DT bearings), an equation of the following form describes the dependence of load mass on bearing-pressure and temperature signals:

$$m = a(P - b.T) + c, \quad \quad (8)$$

where a, b, and c are constants, and m, P, and T represent the mass of the load, the bearing pressure, and the temperature respectively.

The design of the two circuits is elementary, and will not be discussed here. Each of the circuits was connected to one of the signal-source pairs for the measurement of the temperature and pressure of the DL and DT bearings respectively (Fig. 6). The resistors that determined the values of gain b were adjusted for each circuit so that the output load signals did not change while the mill was being warmed up under zero load conditions. Gains a were adjusted until the slopes of the two curves for

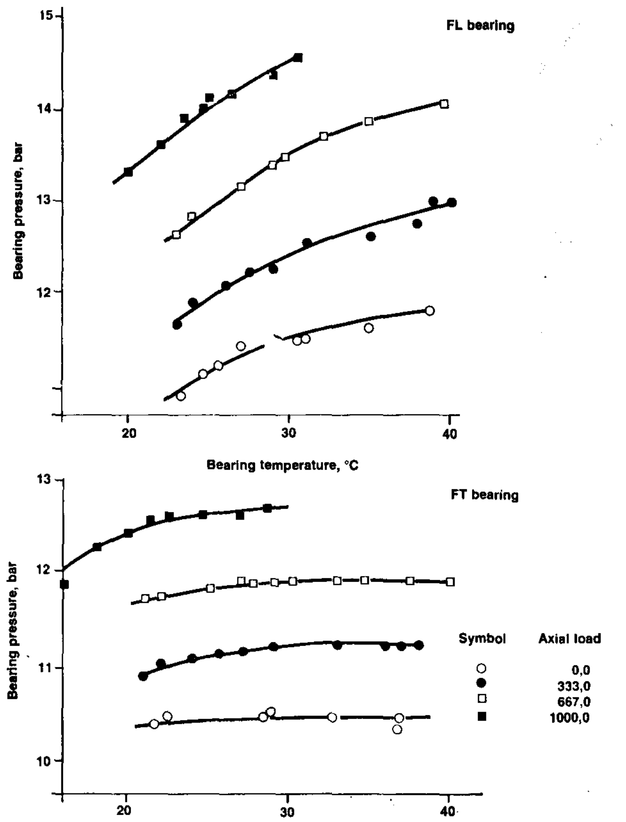

Fig. 7—Measurements of bearing-oil pressure as affected by axial load and bearing-oil temperature for FL and FT bearings

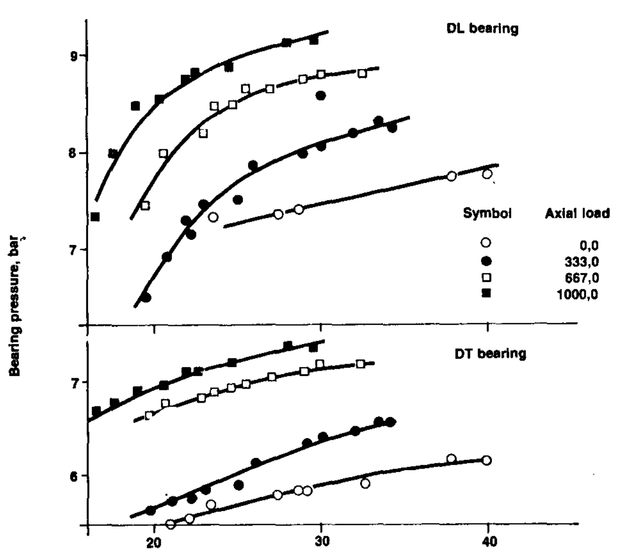

Fig. 8—Measurements of bearing-oil pressure as affected by axial load and bearing-oil temperature for DL and DT bearings

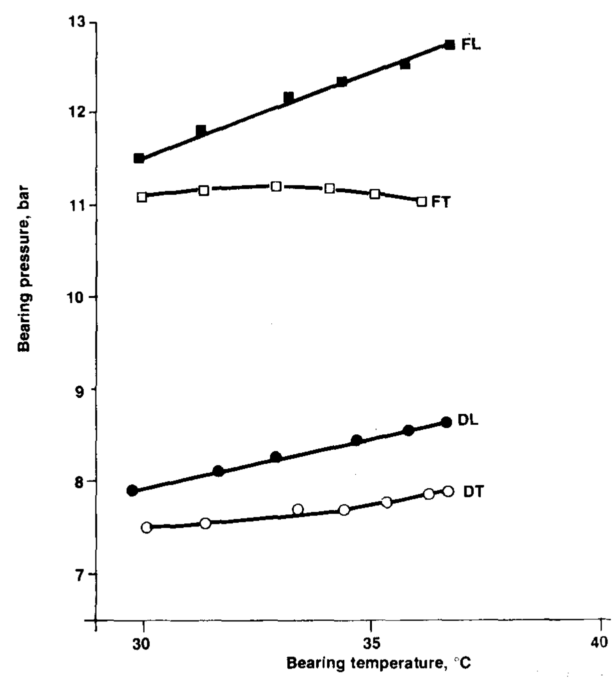

Fig. 9—Variation of bearing-oil pressure with bearing-oil temperature for all the bearings under zero dynamic load conditions

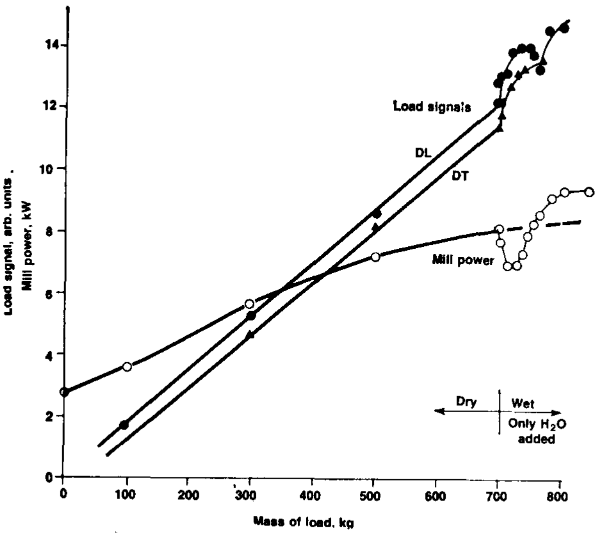

Fig. 10—Variation of mill power and load signals with mass of load (dry and wet)

load signal versus load mass were the same. Weighed batches of ore (in this instance Witwatersrand quartzite) were added to the mill, and the responses were recorded as before. After 1 000 kg had been added, the mill was allowed to run for 10 minutes, resulting in the production that was approximately 10 per cent fine material smaller than 13,5 mm. (This amount was estimated from a knowledge of the grindability of this ore.) Weighed amounts of water were than added to the mill. The power and load signals recorded are shown in Fig. 10 as a function of the total mass of the load. The following points are noteworthy.

(i) The drop in power resulting from the addition of up to 26 kg of water is probably due to the packing of damp material in the liners. The corresponding increase in load signal can be explained qualitatively if it is postulated that the decrease in the power drawn implies that less of the mass of the mill and its contents is supported by the drive pinion so that each bearing must support more of the mass Mill power should therefore have been included as a variable in equation (8).

(ii) As more water is added, the damp material adhering to the liners and the sides of the mill is washed off and again becomes part of the dynamic charge of the mill, resulting in an increase in power. The increased mass of the charge contributes to the increase in the load signals.

(iii) As even more water is added, the bulk density of the load in the mill increases to a value above that of the dry load, and the power drawn rises to values above the extrapolated curve for dry load versus power. The anomalous behaviour of the load signals cannot be easily explained. It could be the result of a complex interaction between bulk density, dynamic angle of repose, and degree of liner packing. In this case, the inclusion of mill power as a variable in equation (8) would not have improved the ability of the system to provide an accurate estimate of load mass.

The question arises as to whether a more sophisticated approach would be likely to yield better results, and whether the model and measurements of power and of all four pairs of bearing pressure and temperature could be used in the estimation of the mass, volume, and angle of repose of the load. Before this question can be answered, it is necessary to ensure that there is an unambiguous relationship between the bearing pressure and the load supported by the bearing. This is not possible for hydrodynamic bearings such as those used on the NIMmill, because, as Trumpler[5] has shown, the pressure for such bearings is distributed non-uniformly over the surface of the bearing, and this distribution is a complex function of the speed of the trunnion bearing, the thickness and viscosity of the oil film, and the design of the bearing. However, the use of hydrostatic bearings (e.g., the bearing marketed by SKF[6] and used on a number of mills in Sweden) can eliminate this problem. An even better solution is the support of each bearing on a load beam that gives an extremely accurate measurement of the load supported by the bearing. (This has been done successfully on some South African mills.) In this case, all the variables in equation (1) – except the frictional forces F_{fi} and the combined mass of the mill and load $(M+m)$ – are known. The frictional forces are proportional to the reactions for well-engineered and adjusted bearings, and are generally small (about one-fortieth of the reaction values for the bearings if 5 per cent of the power is used to overcome friction). In any event, these forces tend to counteract each other, as an inspection of equation (1) reveals. Thus, equation (1) provides an accurate estimate of the total mass of the mill plus the load $(M+m)$, and m can be calculated if M is known. (M is a slowly varying function of the degree of wear and packing of the liners, and should easily be correlated with the number of hours milled since the most recent replacement of the liners.)

The effects of the angle of repose, θ, and the volume of the load, V_L, are incorporated in the model by equation (5), as follows:

$$R_g.F_g = R_b.\sum_i.F_{fi} + mgR_L.\sin\theta \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

Angle θ appears explicitly, and V_L is a complex function of R_L, the distance between the centre of gravity of the load and the axis of the mill. This equation is applicable only for non-cataracting loads, i.e., for mill speeds less than approximately 75 per cent of the critical speed. Angle θ and R_L do not appear in any of the other equations that define the model, and they appear as the product $R_L.\sin\theta$ in equation (5). Hence, independent estimates of R_L and θ cannot be obtained from measurements of bearing pressure alone; only the product $R_L.\sin\theta$ can be estimated. If an independent estimate of V_L (e.g., with the radioactive-isotope method of Geizenblazen *et al.*[7]) or of θ is available, then the value of the other parameter can be estimated from measurements of bearing reactions.

Conclusions

The model of the dynamics of a rotating grinding mill revealed the complex dependence of the load supported by each bearing on the mass, bulk density, and angle of repose of the load, and on the mass of ore packed in the liners and the friction forces on the bearings. The model predicts experimental trends fairly accurately.

The mass of the load in the mill can be estimated if accurate measurements of the power and the load supported by each bearing are available. The accuracy of this estimate depends only on the accuracy of the measurements and on the accuracy of the estimates made of the masses of the mill, the liners, and the ore packed in the liners.

The volume and dynamic angle of repose of the load cannot be estimated accurately from measurements of power and bearing load alone. If an independent estimate of one of these variables is available, the other can be estimated. This is possible only for non-cataracting loads, i.e. for mill speeds less than 75 per cent of the critical speed; for higher speeds, empirical corrections would have to be applied.

The results given in this paper will be of quantitative value to plant designers who intend using bearings that provide an accurate measurement of the load supported by the bearing (e.g., hydrostatic bearings, or bearings supported on load cells.) If these bearings are not used, other methods such as those described by Krogh[4] must be used to control the performance of a mill.

Acknowledgement

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SAIMM diary

2nd to 6th February 1981 **9th to 13th February 1981** **16th to 20th February 1981** University of the Witwatersrand	Vacation Schools on Increased Underground Extraction of Coal
23rd April 1981 Kelvin House	Colloquium on Wear and Abrasion in Industry in collaboration with the Institution of Metallurgists (South African Branch)
July 1981 University of Pretoria	Refresher course on the Heat Treating of Steel — Theory and Practice (organized by the SAIMM Material Engineering Division)
July 1981 National Institute of Metallurgy	Vacation School on Uranium Ore Processing — Extraction of Uranium
August/September 1981 Venue to be announced	Vacation School on Mining, Finance and Taxation Course Leader: Professor B. Mackenzie, Canada
26th April to 2nd May, 1982	Pre-congress tours
3rd to 7th May, 1982 Carlton Hotel, Johannesburg	Twelfth Mining and Metallurgical Congress in collaboration with the Geological Society of South Africa
10th to 23rd May, 1982	Post-congress tours