The balancing of a centrifugal mill

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SYNOPSIS

A theory for the dynamic balancing of centrifugal mills is presented that takes account of the mass and shape of the charge in a gyrated grinding tube and the way in which the charge lags behind the centrifugal force vector. It predicts that a corresponding lag in the angular position of the counterweight centres of mass is necessary, and that the fractional filling should be increased if the ratio of gyration diameter to tube diameter is decreased. A worked example is given for a 1 MW centrifugal mill that was balanced adequately by the methods described.

SAMEVATTING

'n Teorie vir die dinamiese balansering van sentrifugale meule word gegee waar die massa en vorm van die lading in 'n omwentelende meul-silinder asook die manier waarop die lading agter die sentrifugale kragvektor agterbly in berekening gebring word. Dit voorspel dat 'n ooreenkomstige terugverstelling in die hoekposisie van die teengewigte nodig is en dat die fraksionele opvulling vermeerder moet word as die verhouding van die omwentelingdeursnee tot silinderdeursnee verminder word. 'n Uitgewerkte voorbeeld word gegee van 'n 1 MW sentrifugale meul. Dit was doeltreffend gebalanseer deur gebruik te maak van die metodes wat hier beskryf word.

Introduction

Centrifugal mills can be much smaller than conventional tube mills of comparable duty[1]. This fact makes possible the consideration of underground milling, which is important in the preparation of backfill, either by the comminution of waste or by the concentration of values from the reef. Such a concept has significant economic and operational potential[2].

The South African Chamber of Mines, in collaboration with Lurgi, has installed a centrifugal mill with a nominal rating of 1000 kW at Western Deep Levels Gold Mine. This is by far the most powerful mill of its type in the world, and embodies many novel features. In particular, out-of-balance forces generated when the mill is operated are neutralized by adjustable counterweights.

Balancing procedures for centrifugal mills in general are outlined, and the interdependence of percentage filling and mill configuration is discussed.

Nomenclature

- A Centre of gyration of the grinding tube
- B Centre of the grinding tube
- C Centre of mass of the charge
- M_1 Mass of one counterweight-adjusting piece (lead slug)
- M_1 Mass of gyrated plate and tube (acts at B, see Fig. 3)
- M_2 Mass of the charge (acts at C)
- M_3 Mass of one counterweight set (acts at T)
- R Centre of the eccentric bearing
- S Centre of the support or drive shaft
- T Centre of mass of the counterweight
- D Inside diameter of lined tube
- G $2AB$ = diameter of gyration of grinding tube
- Z $2BC$ = displacement diameter of M_2 from B
- H $2ST$ = diameter of rotation of centre of mass M_3 about S
- θ Complement of ABC = angle of lag
- ω Clockwise angular velocity of gyration (rad/s)
- μ Coefficient of friction
- ϕ Fractional filling
- F_1 Centrifugal force on M_1 (gyrated parts of fixed geometry)
- F_2 Force acting at the centre of mass of the charge M_2
- F_3 Centrifugal force exerted on the shaft by one counterweight set
- F_4 Additional force produced under dynamic conditions that cannot be gauged statically
- x Distance from S in x direction of adjusting piece centre of mass
- y Distance from S in y direction of adjusting piece centre of mass
- N Number of counterweight sets = number of shafts
- n Number of adjusting pieces per counterweight set.

The Mechanics of the Mill

Fig. 1 shows the main working parts of a typical small centrifugal mill. It consists of a horizontal grinding tube mounted in a triangular plate, which is gyrated in a vertical plane by means of three counterweighted, self-synchronized crankshafts. A two-shaft design in which the shafts are synchronized by means of cross-coupling gearboxes is used for larger mills such as the one at Western Deep Levels.

Fig. 2 (a, b, c, and d) shows the nature of the tumbling in a grinding tube at successive quarter-cycle intervals. Only one of the three counterweighted crankshafts is shown. It is pivoted at S and causes the corner, R, of the triangle to move about S in a circle diameter G and at a frequency ω. G is the diameter of gyration and ω is the clockwise angular velocity of rotation in radians per second. M_3 is the mass of the counterweight. Its centre of mass acts at T and moves about S in a circle diameter H and at a frequency ω. The other two crankshafts cause

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SA ISSN 0038 - 223X/$3.00 + 0.00.

the remaining corners to move in the same way so that the triangular plate gyrates. The charge tumbles around the mill as shown, and in this case a fractional filling of approximately 0,5 is illustrated. The mass of the mill without charge is M_1. Because the mill is symmetrical, the centre of mass acts at B, the centre of the tube. A is the centre of gyration with respect to B, which moves about A in a circle diameter G and at a frequency ω. The centre of mass of the charge, M_2, acts at C and moves in a circle diameter Z about B at a frequency ω; that is,

$2AB = G$, $2BC = Z$, and $\vec{AB} + \vec{BC} = \vec{AC}$, where the arrows indicate vectors.

If there were no friction between the charge and the wall of the mill, ABC would be a straight line. This, of course, is never the case. In the example in Fig. 2(a), the charge is shown with a lag angle θ of approximately 60°, i.e. the angle ABC is approximately 120°.

If the mill is not symmetrical, it is more difficult to dynamically balance the three-shaft mill and impossible to dynamically balance the two-shaft mill perfectly. A moderate degree of asymmetry, however, is not likely to cause any problems and will be discussed later.

Static Balancing of a Symmetrical Mill

With the grinding charge and all fixed accessories such as feeder and discharge in place, a static balance can easily be achieved by the rotation of the shafts until the counterweight centre lines are horizontal, and by the addition or subtraction of adjusting pieces symmetrically until the same force is required to move in either direction. As an illustration of the sensitivity of this method, it can be stated that a change of 1 per cent in the gyrated mass is readily detected at the counterweights.

Adjustments must also be made to give a good dynamic balance because of the additional forces produced by the movement of the charge about B when the mill is running. This necessitates not only some possible additional mass in line with the counterweights but invariably some lag in the angular position to compensate for the charge lag angle, θ, caused by friction between the charge and the tube wall.

The Forces Generated in a Symmetrical Mill

Fig. 3 illustrates the forces in a three-shaft mill that is gyrating at a frequency ω. Two perpendicular axes, x and y, are drawn for orientation of the direction of the force vectors with respect to the counterweight. The centre line of the counterweight is assumed to be in the y direction as shown, and the centre of the co-ordinate system is at S. Forces in the x and y directions are designated F_x and F_y.

This convention is used for the analysis throughout the paper. The analysis could have been carried out with the counterweight in any orientation, but the conclusions would be the same if an equivalent orientation of the axes were used.

It should be noted that A and S are fixed points in space. If the movement of any point around A or S is known, then the acceleration at that point can be calculated. For example, because B moves in a circle of radius $G/2$ and frequency ω round A, there is a force F_1 at B on M_1 that is given by

$$F_1 = \omega^2(AB)M_1 = \omega^2\frac{G}{2}M_1 \quad \ldots\ldots\ (1)$$

$$F_{1x} = 0 \quad \ldots\ldots\ (1a)$$

$$F_{1y} = -\omega^2\frac{G}{2}M_1 . \quad \ldots\ldots\ (1b)$$

Similarly, there is a force F_2 at C on M_2 that is given by

$$F_2 = \omega^2(AC)M_2 \quad \ldots\ldots\ (2)$$

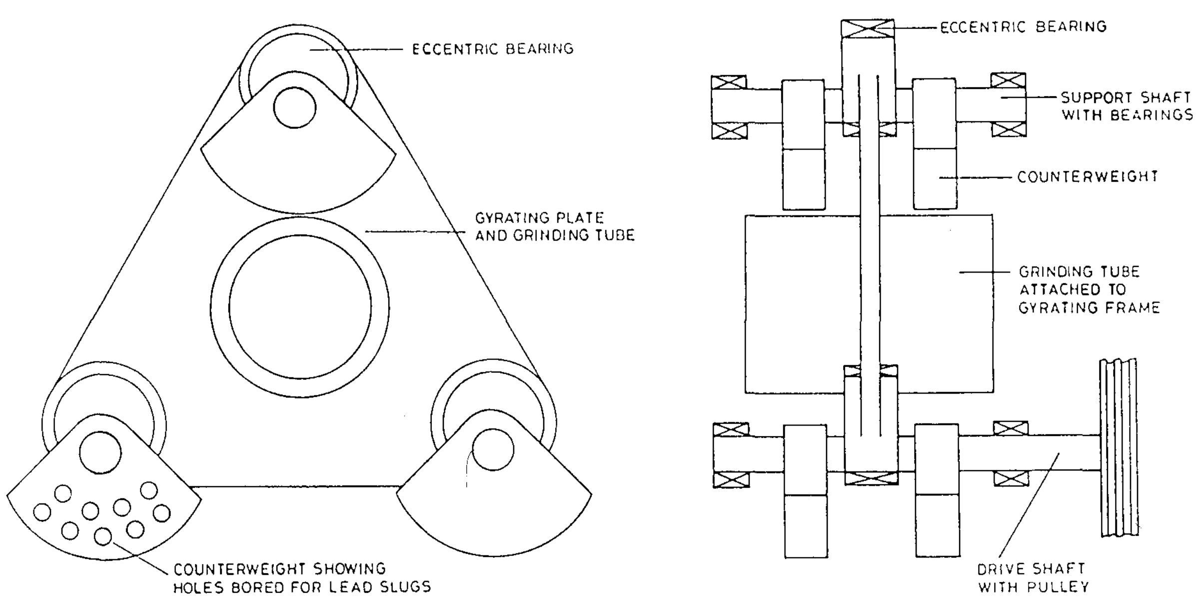


Fig. 1—Schematic views of a small centrifugal mill (the fixed frame is omitted)

$$F_{2x} = \omega^2 \frac{Z}{2} M_2 \sin \theta \quad \ldots\ldots\ldots\ldots \quad (2a)$$

$$F_{2y} = -\omega^2 M_2 (\frac{G}{2} + \frac{Z}{2} \cos \theta). \quad \ldots\ldots \quad (2b)$$

Finally, there is a force F_3 at S acting on M_3 given for each counterweight set by

$$F_3 = \omega^2 (ST) M_3 \quad \ldots\ldots\ldots\ldots \quad (3)$$

$$F_{3x} = 0 \quad \ldots\ldots\ldots\ldots \quad (3a)$$

$$F_{3y} = \omega^2 \frac{H}{2} M_3. \quad \ldots\ldots\ldots\ldots \quad (3b)$$

Neutralization of Forces That Cannot Be Balanced Statically

For static balance, both M_1 and M_2 work through B and pivot about S. M_3 works through T and also pivots about S. Therefore, the following quantities can be equated (on the assumption of three shafts):

$$3HM_3 = G(M_1 + M_2).$$

An examination of equations (1) to (3) by use of the above equality shows that F_{2x} and the second term of F_{2y} have not been neutralized. This dynamically induced force, which requires further adjustment of the counterweights, is designated F_4. Thus,

$$F_{4x} = \omega^2 \frac{Z}{2} M_2 \sin \theta \quad \ldots\ldots\ldots\ldots \quad (4a)$$

$$F_{4y} = -\omega^2 \frac{Z}{2} M_2 \cos \theta \quad \ldots\ldots\ldots\ldots \quad (4b)$$

A further complication is the fact that F_2 acts through C and not B, but, as shown in Fig. 4, F_2 acting through C is equivalent to F_2 acting through B plus a torque

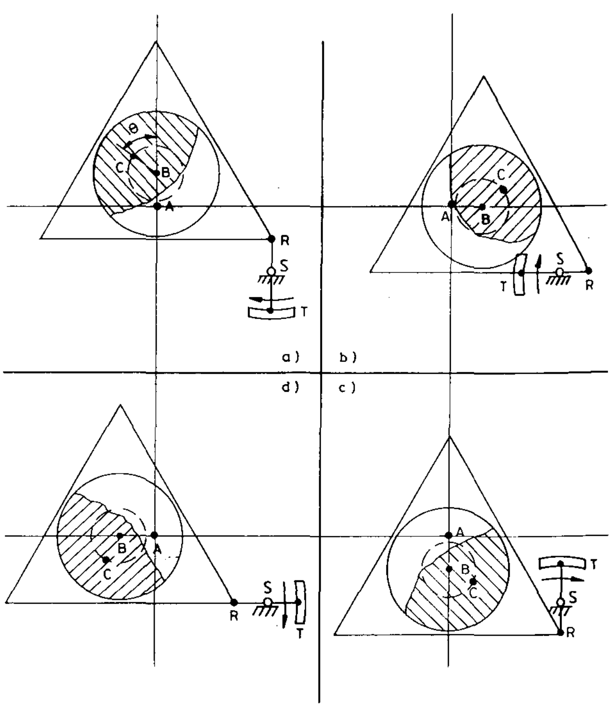


Fig. 2—Movement of the charge as the tube is gyrated about *A* by three cranks, one of which, *SR*, is shown rotating about pivot *S*

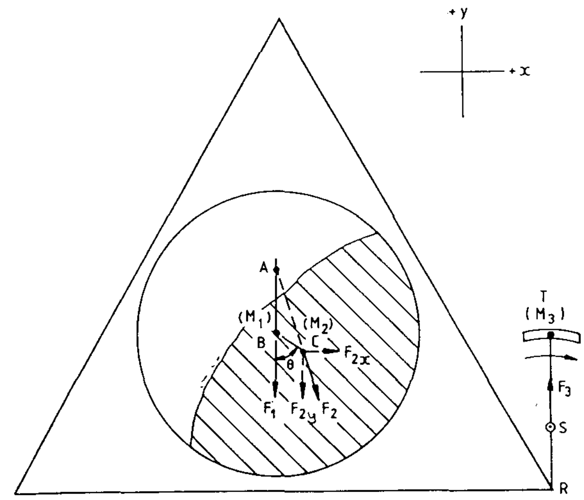


Fig. 3—Force F_1 acting at *B*, the centre of the grinding tube, and force F_2 acting at *C*, the centre of mass of the charge

about B as follows:

$$\text{Torque} = \frac{Z}{2} F_{2y} \sin \theta - \frac{Z}{2} F_{2x} \cos \theta.$$

This torque does no work because it is unable to rotate the gyrating plate against the synchronized crankshafts. It does introduce loads on the eccentric bearings, but the net effect is small if the distance between them is large compared with G and Z. From a balancing point of view, it can be ignored.

It is concluded therefore that F_4 also acts through B and is equivalent to three equal smaller forces, $\frac{1}{3}F_4$, at each of the three eccentric bearings as shown at R in Fig. 5. The force $\frac{1}{3}F_4$ at R can be transformed into a force $\frac{1}{3}F_4$ at S and a torque about S given by

$$\text{Torque} = -\tfrac{1}{3}(RS)F_{4x} = -\frac{G}{6}F_{4x}$$

$$= -\omega^2 \frac{ZG}{12} M_2 \sin \theta. \quad (5)$$

The work done in overcoming this torque in all three shafts is supplied by the drive motor, and can be applied to one shaft because the shafts are mechanically coupled by the gyrating plate. The power is given by

$$\text{Power} = \omega \frac{G}{2} F_{4x} = \omega^3 \frac{ZG}{4} M_2 \sin \theta. \quad \ldots \quad (6)$$

If the power is measured, F_{4x} can be calculated as follows:

$$F_{4x} = \text{power}/(\omega \frac{G}{2}). \quad \ldots\ldots\ldots\ldots \quad (7)$$

With the torque neutralized by the motor, the force $\frac{1}{3}F_{4x}$ on each shaft can be balanced by the introduction of a mass M at a distance x from S opposite F_{4x} so that $\omega^2 Mx = -F_{4x}$, i.e.

$$Mx = (-F_{4x})/\omega^2 = -\text{power}/(\omega^3 \frac{G}{2}); \quad (8)$$

similarly, $My = (-F_{4y})/\omega^2$, ... (9)

where Mx and My must be divided symmetrically between the three shafts.

This operation is done by the shifting and/or addition of lead slugs. In general, for balance, the following equations must hold for each counterweight set:

$$\sum_{i=1}^{n} M_1(x_2 - x_1)_1 = (-F_{4x})/(\omega^2 N) \quad . . . . . (10)$$

$$\sum_{i=1}^{n} M_1(y_2 - y_1)_1 = (-F_{4y})/(\omega^2 N), \quad . . . . (11)$$

where N is the number of counterweight sets, n is the number of lead slugs per set, M_1 is the mass of one lead slug, $x_2 - x_1$ is the shift in the x direction, and $y_2 - y_1$ is the shift in the y direction. New slugs can be considered to have been moved from S.

As a first approximation, F_{4x} and F_{4y} would be calculated from estimates of Z and θ by the use of equations (4a) and (4b). Once the mill is running, F_{4x} can be obtained more accurately from the power by the use of equation (7). F_{4y} can be determined by the adjustment of the in-line balance, with phase-indicating balancing equipment if necessary, and then by the calculation of Z and θ from equations (4a) and (4b). A later section indicates how first estimates of Z and θ can be obtained for various G/D, fillings, and charge types.

Asymmetrically Postioned Centre-of-Mass

If the centre of mass of the mill is not symmetrically positioned with respect to the bearings, either because the tube is offset or for any other reason, out-of-balance forces will be produced that are difficult to neutralize. The way in which this occurs for a two-shaft mill is shown in Fig. 6. Radial asymmetry of the gyrated mass is represented by a mass M at a distance d_1 above 0, the centre of the line joining the centres of the eccentric bearings as shown. The distance between these centres is d_2.

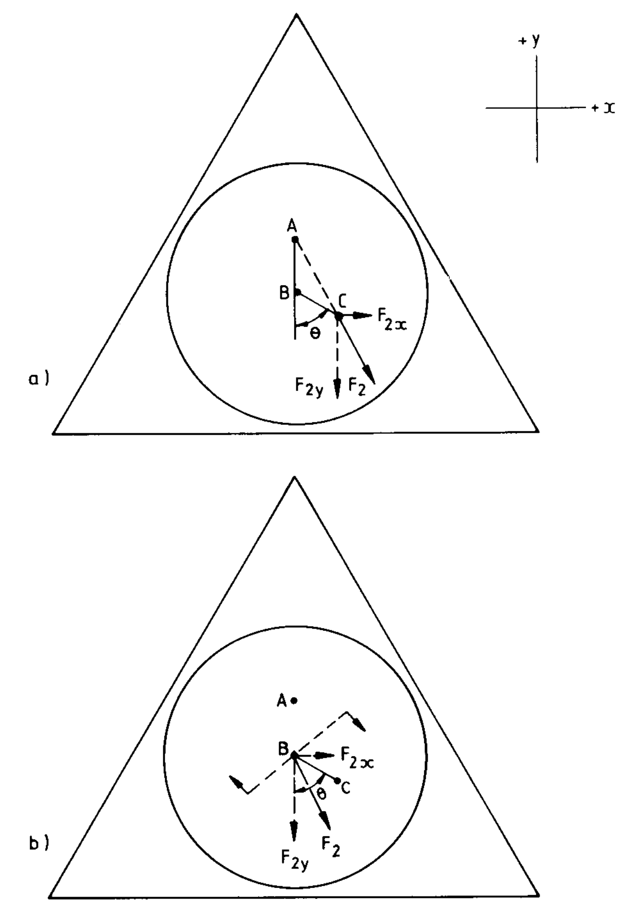


Fig. 4—(a) Force F_2 acting at C
(b) The torque and force F_2 at B, which is equivalent to (a)

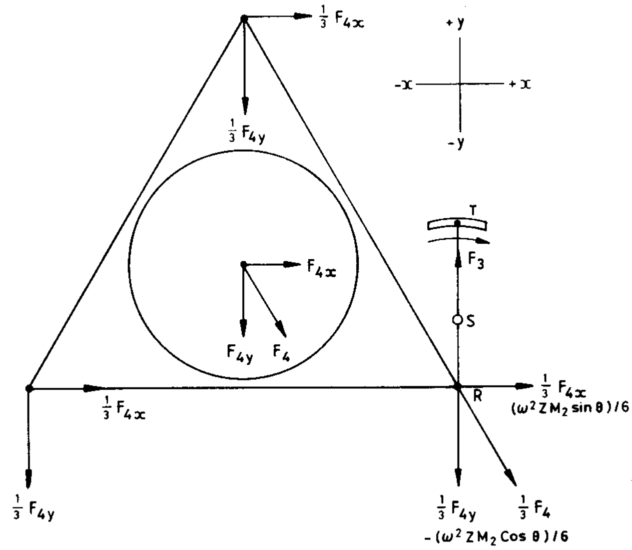


Fig. 5—The x and y components of F_4 transferred from B to the centre of the eccentric bearings as shown at R

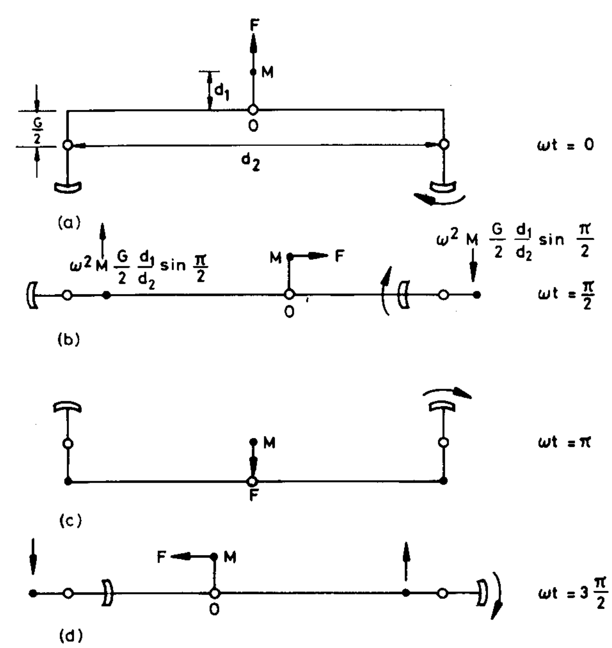


Fig. 6—The effect of radial mass asymmetry on the drive shafts

The force F on M generated by the movement of the mill is given by

$$\text{Force} = \omega^2 M \frac{G}{2}.$$

As F rotates, it creates a torque about 0 as follows:

$$\text{Torque} = \omega^2 M \frac{G}{2} d_1 \sin(\omega t).$$

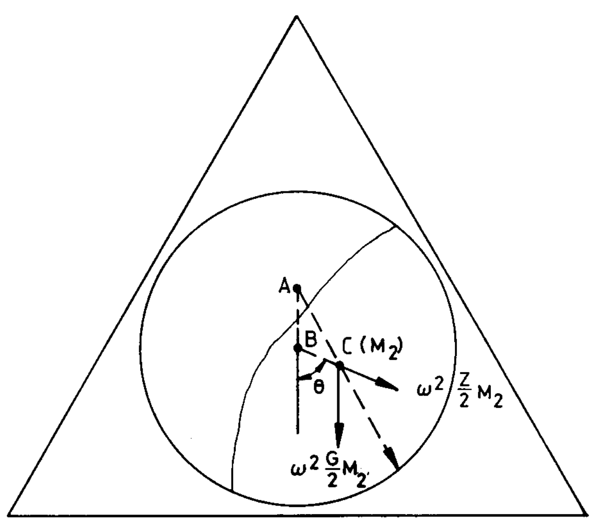
From Fig. 7(b) it can be seen that a positive moment is produced on the right-hand shaft, given by

$$\text{Moment} = \omega^2 M \frac{G^2}{4} \frac{d_1}{d_2} \sin^2(\omega t).$$

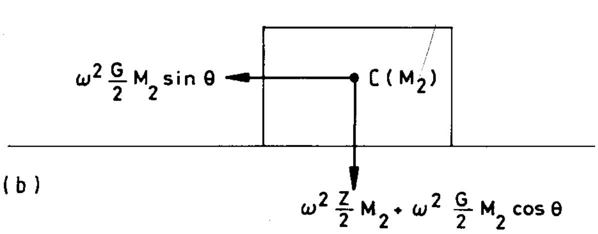
Similarly, a negative moment is produced on the left-hand shaft as follows:

$$\text{Moment} = -\omega^2 M \frac{G^2}{4} \frac{d_1}{d_2} \sin^2(\omega t).$$

The effect is that there are two pulses of acceleration on the right-hand shaft and two pulses of deceleration on the left-hand shaft per cycle, thus tending to desynchronize the shafts. To compensate for this, the driving torque to the two shafts has to vary cyclically. If the effect is significant, it is thought that a ripple on the power draw and an unpleasant level of vibration may by caused. It may be possible for this effect to be countered by the placing of weights on the frame below the bearing centre line, but this would cause an increase in the gyrated mass. Axial asymmetry can cause similar problems. This effect should, therefore, be borne in mind at the design stage.



(a)



(b)

Fig. 7—Development of the forces to enable the effect of friction to be assessed

Relationship between Filling, Angle of Lag, and G/D

The following hypothesis is presented to provide some insight into the mechanics of this type of milling and eliminate as much trial and error as possible in the balancing procedure.

In Fig. 7(a), the force on M_2 at C is given by $\omega^2(AC)M_2$ in the direction AC. This can be replaced by two forces: $\omega^2(BC)M_2$ in the direction BC, and $\omega^2(AB)M_2$ parallel to AB. Since $BC = Z/2$ and $AB = G/2$, these forces can be rewritten as $\omega^2 \frac{Z}{2} M_2$ and $\omega^2 \frac{G}{2} M_2$ respectively.

Now $\omega^2 \frac{G}{2} M_2$ can be replaced by $\omega^2 \frac{G}{2} M_2 \cos\theta$ in the direction BC and $\omega^2 \frac{Z}{2} M_2 \sin\theta$ at right angles to BC.

The arrangement is then analogous to Fig. 7(b). Whether M_2 will move or not depends on μ, the effective coefficient of friction, which arises because of the interaction between the charge and the wall. The effective value of μ will depend on many factors, for example whether internal friction or wall-to-charge friction dominates, the size and number of lifter bars, and the water content.

This limitation being borne in mind, the condition for movement is given by

$$\omega^2 \frac{G}{2} M_2 \sin\theta > \mu\, M_2 \omega^2 \left(\frac{Z}{2} + \frac{G}{2}\cos\theta\right). \quad \text{. . (12a)}$$

If, however, the left-hand side of equation (12a) is less than the right-hand side, movement of the charge will be intermittent and unstable because of the following sequence of events. Firstly, the charge sticks. Because the charge is now stationary with respect to the tube, the first term of the right-hand side disappears ($\omega = 0$ for that term only). The left-hand side is now greater than the right-hand side, and the charge starts to move again. The first term on the right-hand side immediately re-appears, the charge sticks again, and the sequence is repeated so that there is alternate surging and sticking.

If only the stable region is considered, equation (12a) can be divided by $\mu\omega^2 M_2$, thus simplifying the condition for stability to

$$\frac{Z}{G} \leq \frac{\sin\theta}{\mu} - \cos\theta, \quad \text{. . . . . (12b)}$$

but conclusions about the unstable region must not be drawn from equation (12b).

Plots of equation (12b) are given in Fig. 8 for different values of μ. For each value of μ, the curve passes through a maximum. By differentiation, the value of θ at which Z/G goes through a maximum is given by

$$\theta_m = 180° + \tan^{-1}\left(\frac{1}{\mu}\right). \quad \text{. . . . . (13)}$$

The importance of this is that, for a fixed G and μ, maximum Z defines the minimum filling for stable operation, and the charge will surge if the filling is less than this minimum. Because only positive values of Z/G and μ are possible, the limits of θ_m are between 90 and 180°.

is used, the filling must be above 20 per cent for stable operation,

(iii) at very small values of G/D as are used in vibration mills, the mill must be run essentially full to achieve stable movement of the charge.

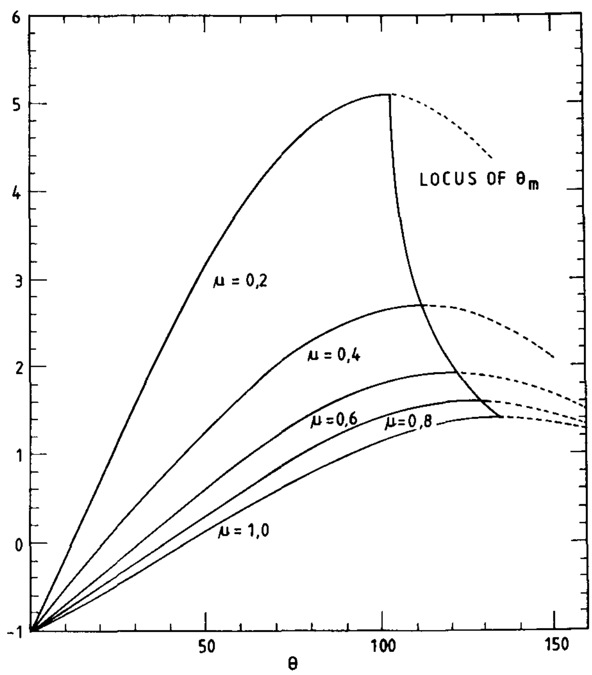


Fig. 8—Curves from which any one of θ, ϕ, or μ can be estimated if the others are known

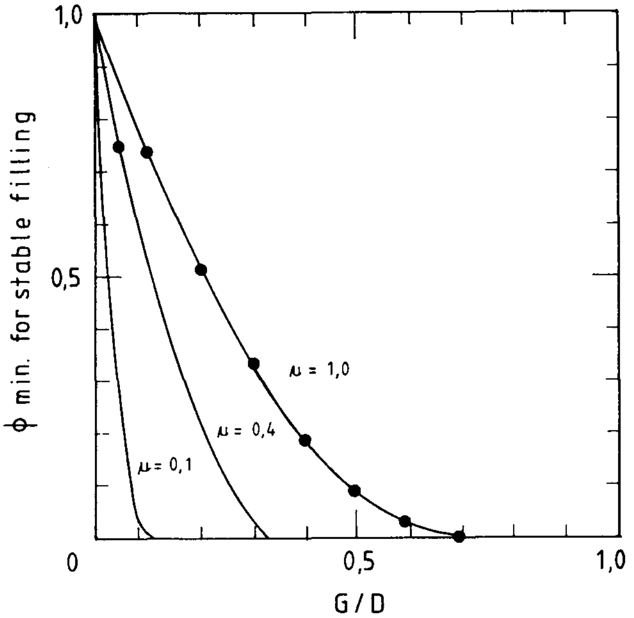


Fig. 9—Minimum filling for stable operation versus G/D for different values of μ

If Z/G is less than this maximum, there are two values of θ that solve equation (12b), namely θ_1 and θ_2. If θ_1 is the smaller angle and the charge is positioned somewhere between θ_1 and θ_2, equation (12a) indicates that the charge will accelerate towards θ_1. If it moves past θ_1, it will be decelerated and move back to θ_1. Because of this negative feedback, the stable position at which the charge will ride is θ_1.

The value of Z is not known. However, it is reasonable to assume that the charge is roughly circular in cross-section with the centre of mass acting at the centre of the circle so that Z is equal to the diameter of the tube minus the diameter of the charge. If the diameter of the charge is expressed in terms of the fractional filling, ϕ,

$$Z = D(1 - \sqrt{\phi}) \quad \ldots\ldots\ldots\ldots\ldots (14)$$

Substitution of this value for Z in equation (12b) leads to

$$\frac{1 - \sqrt{\phi}}{G/D} \leqq \frac{\sin \theta}{\mu} - \cos \theta, \quad \ldots\ldots\ldots (15)$$

which is a useful relationship between the various operating parameters and θ. For instance, one may enquire how the minimum filling, ϕ_{min}, varies with G/D and μ. The plot is shown in Fig. 9, from which it is apparent that

(i) the action is always stable above a G/D value of 0,7071 if μ does not exceed 1,

(ii) if μ has a 'worst case' value of 1 and a G/D of 0,4

Photographic evidence of this effect can be seen in Fig. 10. A tumbling charge consisting of 5 mm plastic beads was used without water, and anti-clockwise gyration was chosen. The arrows indicate the instantaneous position of the centrifugal vectors. The photographs were taken at random times. Unstable operation is characterized by the lag angle θ exceeding 135°. Note should be taken in particular of the unstable operation at 30 per cent filling and G/D values of 0,28 and 0,14 and at 50 per cent filling and a G/D of 0,14, and the relatively stable operation under all other conditions. Six lifter bars were used and a μ of approximately 1 would seem to be indicated.

The usefulness of these relationships can be illustrated further by the following example. The angle of lag, θ, is required so that a mill that has already been balanced statically can be balanced dynamically with two counterweighted shafts. It is assumed that G=0,4 m, D=1,0 m, ϕ = 0,5, μ = 1, and M_2 = 3000 kg. For a ϕ of 0,5, Z will be 0,3 m and Z/G will be 0,75. From Fig. 8, θ is 78°. From equations (4a) and (8), the right-angle dynamic out-of-balance correction is given by

$$Mx = -M_2\frac{Z}{2}\sin\theta = 440 \text{ kg}\cdot\text{m},$$

that is, the product of the mass and negative x coordinate for each of the two counterweights is 220 kg·m.

The in-line out-of-balance correction is given by

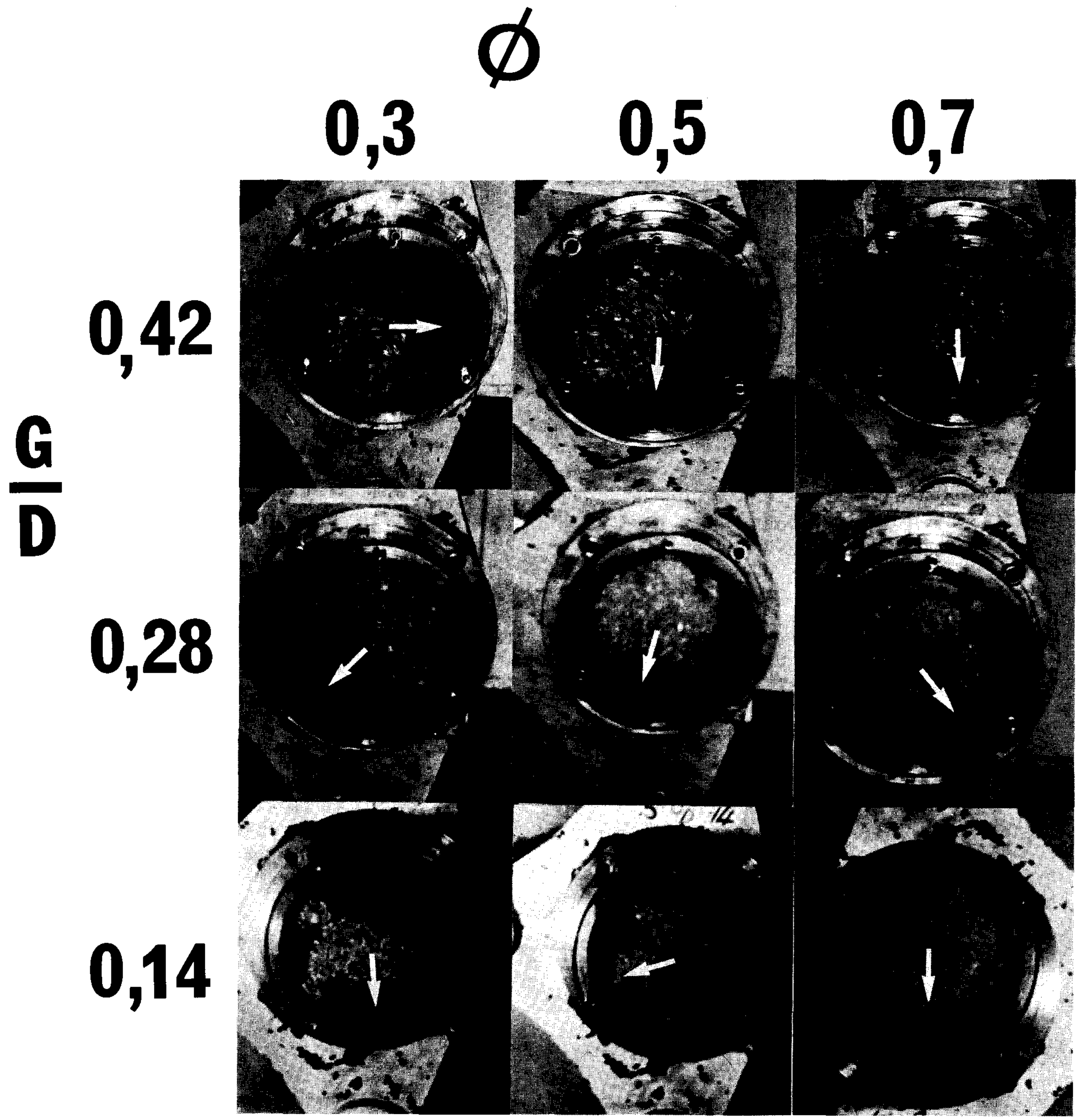


Fig. 10—The way in which the charge tumbles for different values of ϕ and G/D

$$My = M_2\frac{Z}{2}\cos\theta = 94 \text{ kg}\cdot\text{m}.$$

This is not a large in-line correction for a mill of this size and could probably be neglected. In general, if θ is close to 90°, no extra in-line balance is required.

Mx can be calculated from the power drawn by the mill. If it is assumed that the mill draws 1000 kW at 220 r/min,

$$Mx = -\text{power}/(\omega^3\frac{G}{2}) \quad \text{from equation (8)}$$

$= -409$ kg·m or -205 kg·m per counterweight.

Conclusion

The methods described for estimating the action of the charge in a mill are adequate for balancing purposes.

Good operation is achieved in spite of the approximations used.

Fig. 9 clearly shows the importance of increasing the filling as the ratio of gyration diameter to tube diameter is decreased. Photographic studies and vibration-mill practice have confirmed this conclusion, but further experimental work is necessary before the effect can be quantified and the implications assessed.

Acknowledgement

The permission of the Chamber of Mines of South Africa to publish this paper is gratefully acknowledged.

References

1. Bradley, A. A. Some principles of centrifugal milling. *Proc. Third European Symp. Comminution, Cannes.* 1971.
2. Lloyd, P. J. D. The potentia of integrated mining and extraction systems on the Witwatersrand. *Proc. Eleventh Commonwealth Min. Metall. Congress, Hong Kong.* London, The Institution of Mining and Metallurgy, 1979.

Mintek 50

The final circular and registration forms for the above Conference are now available from the address given below.

This Conference, which will be held in Johannesburg from 26th to 30th March, 1984, is being organized by the Council for Mineral Technology (Mintek) to mark its fiftieth year of existence, first as the Minerals Research Laboratory, then as the Government Metallurgical Laboratory and the National Institute for Metallurgy, and now as Mintek. The Conference will have as its theme Recent Advances in Mineral Science and will include a Symposium on International Mineral Policy. It will be followed by a week of technical excursions.

The technical sessions will have the following topics: Optimization of Ore-dressing Processes and Energy Saving; Treatment of Fines, Tailings, and Low-grade Ores; Innovations; Modelling Design and Control; Electrolytic Processes; Equipment and Modelling; Leaching Processes; Separation Processes; Process Development; Arc Furnaces and Direct Reduction; Plasma Technology; Plasma Processes; Plasma Facilities; Mineral and Material Science in Pyrometallurgy; Non-ferrous Pyrometallurgy; Applied Measurements; and Instrumental Analysis. The plenary speakers will be P. R. Jochens (Mintek), K. J. Reid (University of Minnesota, U.S.A.), D. S. Flett (Warren Spring Laboratory, England), A. L. Mular (University of British Columbia, Canada), J. E. Dutrizac (CANMET, Canada), A. J. Lynch (J. Kruttschnitt Mineral Research Centre, Australia), and R. L. Watters (National Bureau of Standards, U.S.A.).

During the Symposium on International Mineral Policy, which will be held on 28th March, 1984, a number of invited world authorities will deliver addresses.

Registration closes on 1st December, 1983. Enquiries should be directed to The Conference Secretary (C. 25), Mintek 50, Private Bag X3015, Randburg, 2125 South Africa. (Telephone: (011) 793-3511; telex 4-24867 SA.).

IPMI conference

The 8th International Conference of the International Precious Metals Institute will be held during the week of June 3rd to 7th, 1984, in Toronto, Canada. Toronto was the site of the IPMI 4th Conference, held in 1980.

Mr David Rose of Imperial Smelting and Refining has been selected as Conference Chairman. He requests all interested authors to submit a brief abstract and title for papers related to the manufacture, use, economics, metallurgy, refining, and mining of precious metals. Papers should be sent to Dr Thomas P. Mohide, Mineral Resources, Ontario Government, Room 4626, Whitney Block, Queen's Park, Toronto, Canada M7A 1W3 (416/965-1559), by 15th December, 1983.

IPMI announced plans to continue the very informative Mini-Exhibit initiated at the 7th Conference in San Francisco. This exhibit is available for IPMI members only, with Patron and Sustaining members receiving first choice of the limited spaces. Mr Rose promises that an outstanding technical and enjoyable social programme is being planned for the delegates and their spouses. Additional information can be requested from

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