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AN EXPERIMENTAL INVESTIGATION INTO TORSIONAL VIBRATION IN BALL MILLS

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ABSTRACT

Torsional vibration in ball mills can be a serious problem - even leading to failure. In the design of such mills it is therefore important to be able to design them so that such vibration will not be a problem. It is thus desirable to be able to model the torsional vibration of a rotating system which includes a ball mill. The major difficulty with modelling torsional vibration is the low level of damping in rotating systems and insufficient information about its magnitude. This paper describes an experimental investigation of the torsional vibration of a ball mill with a particular emphasis on the damping of the mill itself. It was considered possible that the mill could be a significant source of energy dissipation for torsional vibration. A small batch ball mill was investigated using a servo motor as both the drive and a torsional exciter. The main parameters of the rig were varied and a systematic series of tests was performed for each combination of parameters. The inertia and damping of the mill were studied and the results are presented.

1. INTRODUCTION

Tube and ball mills are the principal items of equipment in many mining operations and cement plants. It is essential for the profitability of mining operations that this equipment can run continuously with minimal time set aside for maintenance. If the drive system is chosen incorrectly, or the design of components is incorrect, large torques caused by the dynamic response of the system may lead to failure of the mechanical components.

At the design stage, torsional vibrational problems can be minimised if an accurate model of the system can be produced. The estimation of damping in the mill is then an important parameter. When a torsional excitation frequency approaches the natural frequency of a system, the damping will have a great effect on the dynamic response of the system. With the equipment available today it is possible to accurately measure the torsional response of a ball mill. From such results it possible to confirm whether it is suitable to model a mill as a simple inertia with damping or whether a more complex model is necessary.

To determine whether a simple inertia plus damping model is adequate, a number of related questions need to be answered. These include:

(1) When the ball mill is empty, is it appropriate to model the mill as a simple inertia? Or, for example, should the mill be modelled as a flexible tube with an inertia at each end?

(2) When media are placed inside the mill, do the media change the effective inertia of the ball mill?

(3) When media are placed inside the mill, does this change the damping of the ball mill?

The modelling of a ball mill and drive train system using the systems approach has been described in a paper presented by Derry and Stone [1]. The accuracy of this model was limited due to the lack of knowledge of certain subsystems. These included the torsional stiffness and damping from the field of electric motors, the stiffness and damping of flexible couplings and the characteristics of a loaded ball mill drum. Since that paper was published there have been further publications, including a paper on flexible couplings [2] and one on the torsional stiffness and damping from the field of a three phase servo-drive [3]. It is the purpose of this paper to report on an investigation [4] into the torsional characteristics of a loaded ball mill.

In other previous work Mayer [5] discussed the stiffness and damping characteristics of various electric motors and components used in the drive systems of tube and ball mills and Zhiqian et al [6] concentrated on the damping effect of media in a tube mill. In their investigation it was assumed that the media in the mill did not add inertia to the system.

2. EXPERIMENTAL RIG

2.1 Design of the Experimental Rig

The Department of Mechanical and Material Engineering at UWA had a batch ball mill which was available for the tests. This batch ball mill was adapted to suit the experimental equipment. The rig consisted of an electric motor, a 20:1 gearbox, drive shaft and a 0.3 m diameter by 0.3 m long drum. Various alterations to the batch ball mill were required so that it could be used for the experiment. These included:

(1) Replacing the existing electric motor with the servo-motor and improving access for laser torsional vibrometer measurements.

(2) Re-machining of the drum shaft and the fitting of strain gauges and a telemetry system.

Torsional excitation of the rig under working conditions was by means of an a.c. servo motor. The input voltage to the electrical system was used for analysis and was calibrated. Shaft angular velocity was measured using a Dantec laser torsional vibrometer (LTV).

A diagrammatic layout of the experimental test rig with a calibration inertia fitted is shown in Figure 1.

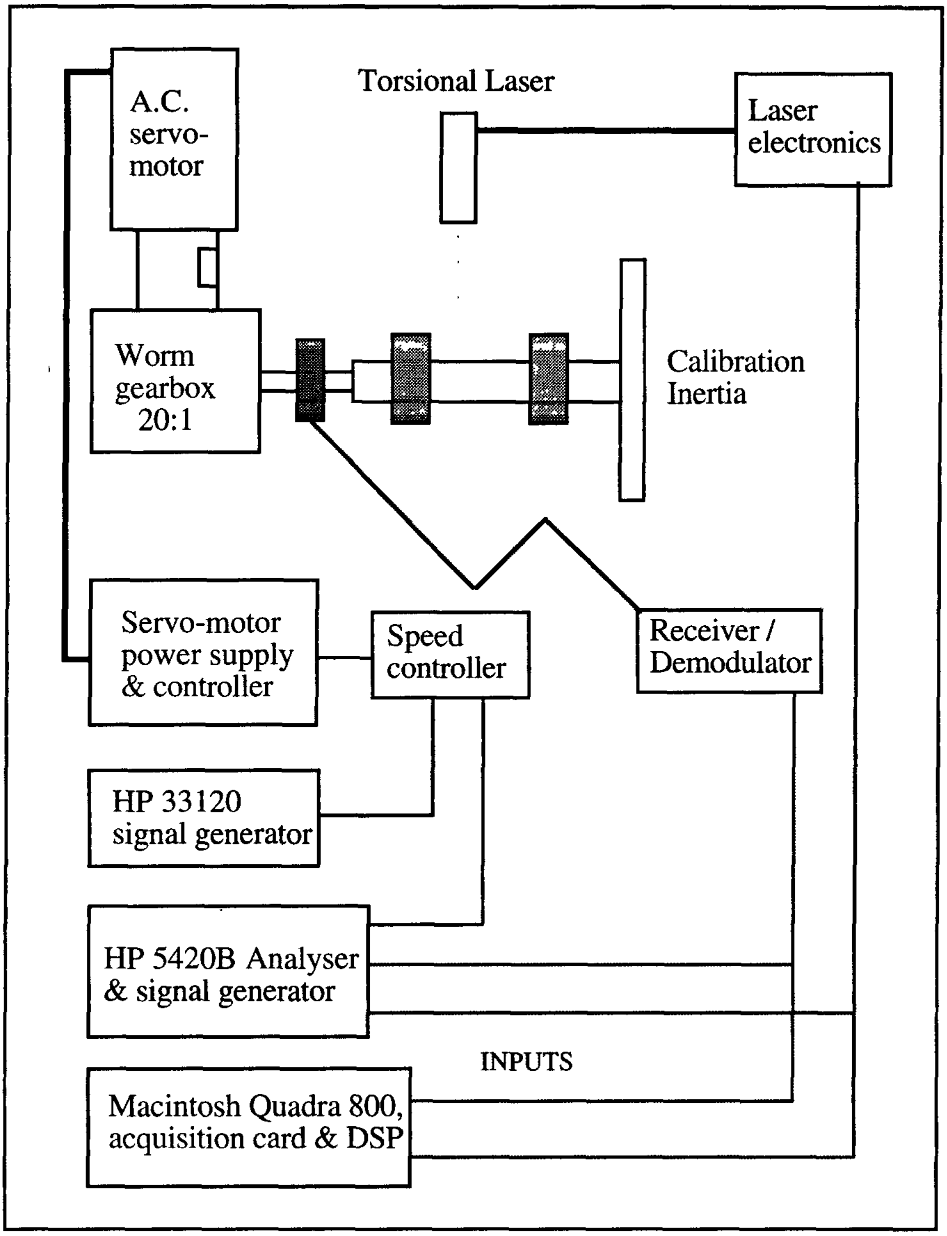

Figure 1. Schematic diagram of the experimental system.

2.2 Experimental Conditions

Decisions were required concerning the particular conditions to be tested. These included:

Wet or dry grinding?
A number of factors influence the decision to grind wet or dry:

1. Whether the subsequent process is wet or dry.
2. The availability of water.
3. Wet grinding requires less power per tonne.
4. Wet grinding does not need dust control.
5. Wet grinding uses more steel grinding material per tonne of product.

When the subsequent processing to be carried out is wet, wet grinding is the most logical choice, and as a consequence most minerals are wet ground. As a result, in the experimental testing the standard case was chosen to be that of wet grinding.

The charge volume and proportion of media in the charge.
The charge volume of a tumbling mill is the volume of the mill interior filled with grinding media. Overflow ball mills are restricted to having a maximum charge volume of approximately 45% to prevent balls discharging. Although this was not a problem for the test rig a charge of 45% was used to simulate the practical situation. The proportion of media that made up the charge was decided upon after consultation with an expert practitioner. Figure 2 shows the proportions of the charge that represent the typical grinding mill.

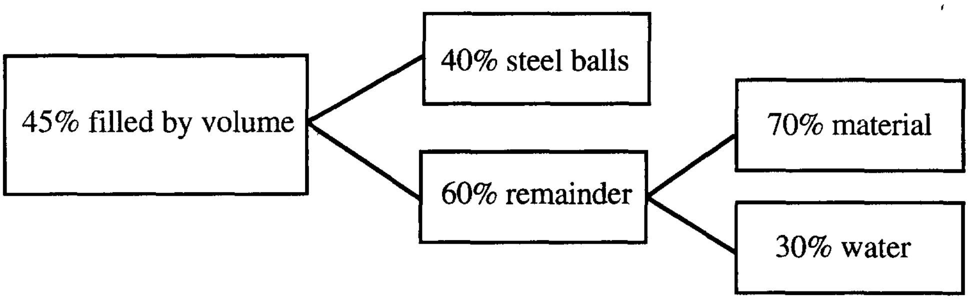

Figure 2. Proportions of media in the experimental mill.

Size and type of grinding media
In consultation with the expert practitioner it was decided that the material should be crushed in a jaw crusher and then sieved. The material was sieved with a 4.8 mm sieve. The ore that was used in this set of experiments is similar to that which enters the mill in a typical gold mining process.

Liners, lifters and the media action
The purpose of liners is twofold; (i) to protect the mill shell from wear, and (ii) to reduce slip between the shell and grinding media. They are made of steel or rubber and as they wear they are replaced. An industrial mill is fitted with liners and it should be noted that the liners add inertia to the mill but add no stiffness. Liners come in various profiles; they may be smooth or have a profile designed to reduce slip between the media and the mill. The experimental rig was not fitted with liners. The media action inside the mill changes when lifters are placed in the mill. The lifter profiles are varied to optimise the milling process. To vary the parameters in the experimental mill, lifters were made that could be inserted and removed from the mill. These were constructed of 20 mm square tube held against the mill wall by continuously threaded rod, as shown in Figure 3.

3. EXPERIMENTAL RESULTS

3.1 Method

A method that could be used to accurately find the inertia and damping of a mill was required. Plots of mobility (torsional velocity/torque) were used because the values of the inertia and damping of the mill could be obtained without considering the rest of the torsional system.

If it is assumed that the mill plus media act as an inertia and viscous damper then the following relationships will hold:

For the magnitude of mobility:

$$\alpha = \frac{1}{\sqrt{(I\omega)^2 + C_v^2}} \tag{1}$$

For the phase of mobility:

$$\mathrm{Tan}\ \phi = \frac{I\,\omega}{C_v} \tag{2}$$

where C_v (the assumed viscous damping coefficient) is assumed to be connected to earth.

This gives two unknown variables, I and C_v, and two equations. The values of inertia and damping can be varied in the above equations and the results compared with the experimental values. This allows the matching of the phase and magnitude plots simultaneously.

The assumption of viscous damping must be tested and the accuracy of measuring inertia and damping examined to substantiate this method of analysis.

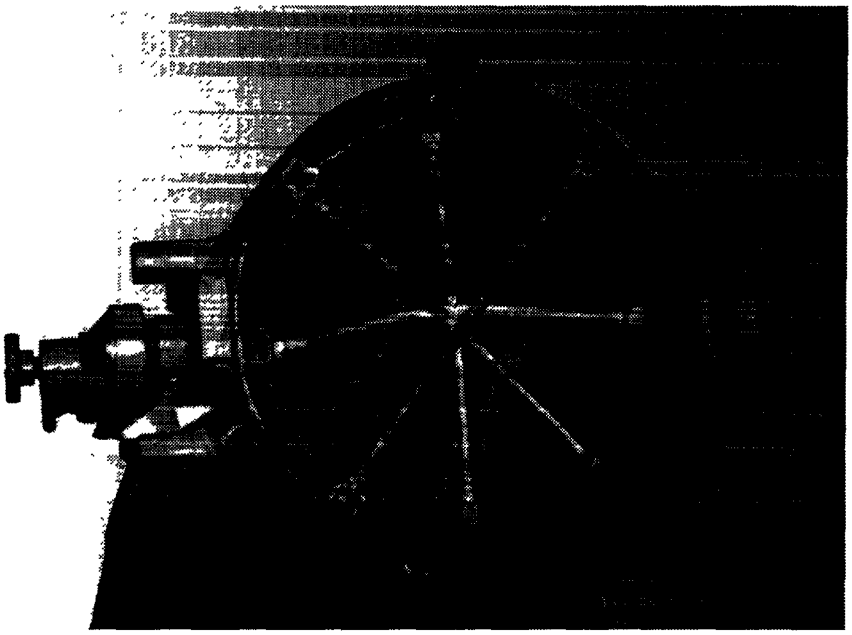

Figure 3. Photograph of the test mill fitted with temporary lifters.

3.2 Experimental Setup

The experimental rig set up that most closely reflects that of a normal milling process was chosen. This case is wet media with four lifters. The values of inertia and damping were varied in the modelling with the objective of obtaining a good fit with the experimental results. Also, in order to test the sensitivity of the results, values of inertia and damping were investigated either side of the best fit. The best fit values were I = 1.7 kg.m^2 and C_v = 5 N.m.s/radian. To test the effect of varying inertia, values of 1.5 and 1.9 kg.m^2 were investigated with C_v = 5 N.m.s/radian, as shown in Figure 4(a); which compares the measured mobility of the ball mill compared to theoretical results. For the damping, values of 2.5 and 7.5 N.m.s/radian were investigated with I = 1.7 kg.m^2, as shown in Figure 4(b). A typical coherence function is shown in Figure 4(c).

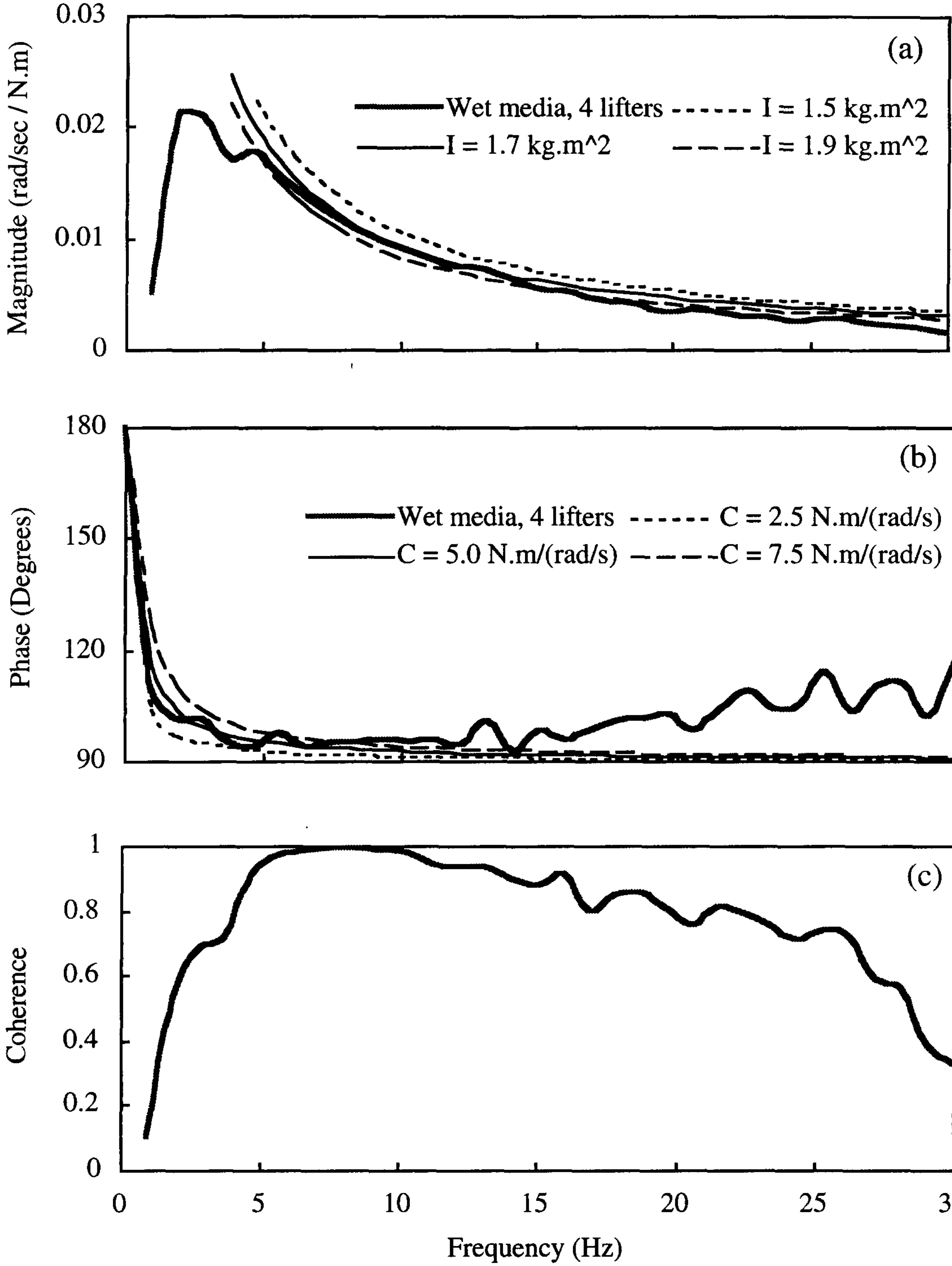

Figure 4. Comparison of measured torsional mobility for the ball mill filled with wet media and fitted with four lifters with theoretical results: (a) magnitude - measured versus theoretical (fixed damping of 5 N.m.s/rad and variable inertia); (b) phase - measured versus theoretical (fixed inertia of 1.7 kg.m^2 and variable damping), and (c) measured coherence.

Figure 4 shows that:

(1) There is not an exact match in shape between the theoretical and experimental curves. The points of maximum deviation coincide with areas of low coherence. The frequency band from 5 Hz to 15 Hz is the crucial area for matching as it has the highest coherence.

(2) The coherence dips at 1 Hz. This is the running speed of the drive shaft.

The theoretical effect of varying the inertia by $\pm$ 0.2 $kg.m^2$ is easily distinguished on the magnitude plot. From this comparison the accuracy of the plot matching is approximately 0.1 $kg.m^2$, corresponding to an error of approximately 6% in the estimated inertia value.

When the damping is varied there is only a small change in the magnitude plot. This is because the magnitude of the damping term, C_v, is small when compared to the magnitude of the inertia term $I\omega$. The phase plot is more sensitive to the value of C_v and was used to determine the estimated damping value. Unfortunately, the change in shape of the phase plot is most pronounced in an area of poor coherence below 5 Hz. This means that the damping estimate has limited accuracy. A change of $\pm$2 N.m.s/radian is detectable in Figure 4(b) given the above considerations. The error given by this change is $\pm$ 40% of the "best" value of 5 N.m.s/radian.

3.3 Inertia And Damping Results From Various Experiments

Values of inertia and damping were obtained in the same manner for each experimental rig set up. The results obtained are shown in Table 1.

Table 1. Inertia And Damping Results.

Experimental Setup	**Inertia** ($kg.m^2$)	**Damping** (N.m.s/radian)
Wet media, 4 lifters	1.7	5
Wet media, no lifters	1.6	2.5
Dry media, 4 lifters	1.6	6
Water (only), 4 lifters	1.5	1.5
Empty drum	1.5	0

4 CONCLUSIONS

A method has been found to simultaneously determine the inertia and damping of the mill when it is loaded. This method does not require the damping of the other components of the system to be known.

The error associated with this method is, inertia $\pm$ 10% and damping $\pm$ 40%. Viscous damping was assumed in these calculations. This assumption seems reasonable for the frequency range where good coherence was obtained, ie. up to 15 Hz. A more comprehensive investigation of the suitability of viscous damping being used is described by Heidecker [4].

From a study of Table 1 a number of points can be ascertained.

Inertia:

1. An increase in inertia occurs when milling media is placed in the drum. Note: when water alone is placed in the drum this does not occur. This suggests the increase in inertia is dependent on the amount and density of the material in the drum.

2. When lifters are placed in the mill there is an increase in the inertia of the mill, as would be expected. There is also an increase in the mean torque when the lifters are placed in the mill. This is due to a reduction in slippage between the media and the mill.

Damping:

1. An increase in damping occurs when media are placed in the mill. Note the difference in damping between drum empty, water in drum and wet or dry media in drum. As the density of media increased so did the damping.

2. The effect of lifters on the damping. When the lifters were removed there was a decrease in damping in the system.

It is clear that the investigation described in this paper needs to be extended. However, it is hoped that the results presented will be of interest to those who encounter torsional vibration problems with ball mills.

5. REFERENCES

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