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Some initial findings on the behaviour and design of mine-shaft steelwork and conveyances

by G.J. KRIGE*

SYNOPSIS

The results of a series of measurements of accelerations and guide-roller loads made on mineshaft conveyances are given, together with the predictions of these accelerations and loads based on a computer programme. The analysis of the measurements is described, and the results of the measurements and computer predictions when compared show reasonable agreement.

Some parameters that increase the guide-roller loads are identified, and their effects are studied, together with some comments on the total loads. Finally, a possible format for a design specification is discussed, and some tentative design values are given.

SAMEVATTING

Die resultate van 'n reeks metings van versnellings en leirollerbelastings van mynskagvervoermiddels word aangegee, tesame met voorspellings van hierdie versnellings en belastings aan die hand van 'n rekenaarprogram. Die ontleding van die metings word beskryf en 'n vergelyking van die resultate van die metings en rekenaarvoorspellings toon 'n redelike mate van ooreenstemming.

Sommige parameters wat die leirollerbelasting verhoog, word geïdentifiseer en hul uitwerking bestudeer, terwyl daar kommentaar oor die totale laste gelewer word.

Ten slotte word 'n moontlike formaat vir 'n ontwerpspesifikasie bespreek en 'n paar tentatitiewe ontwerpwaardes aangegee.

Introduction

A research programme aimed at a better understanding of the dynamic behaviour of mine-shaft steelwork and conveyances has been described in some detail by Krige and Kemp[1]. In this work, a computer model, called DISCS, was set up to predict conveyance behaviour through a step-by-step solution of the equations of motion, and tests were conducted at a total of fourteen different mineshafts. These were all production shafts, so that time was at a premium and the tests had to be limited to measurements of the face roller loads and accelerations in the same direction. Nevertheless, very useful information was obtained, and this has been complemented by similar work carried out by Structural Dynamics Research Corporation (SDRC)[2,3].

The major findings of this work and some design implications are discussed in detail in this paper. (It should be noted that the shafts are identified only by code numbers to preserve anonymity.)

Observations and Calculations of Roller Loads

The analysis, which is performed on the measured *in situ* data and on the DISCS results, includes a probability analysis of peak roller loads, a spectral analysis to establish the frequency characteristics, and fatigue-loading cycle counting. Typical data on acceleration and roller loads are shown in Fig. 1.

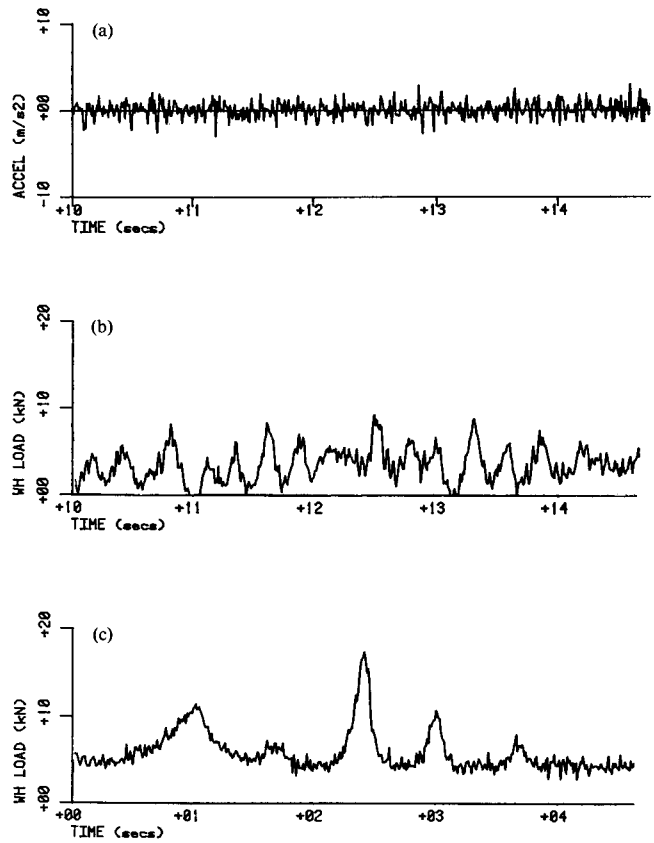

Fig. 1—Typical measured data
(a) Acceleration, Shaft 2
(b) Roller load, Shaft 2
(c) Roller load, Shaft 4

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Analysis of Peak Roller Loads

Although roller loads are limited between zero and a maximum value when the slipper hits the guide, a probability analysis is felt to be valid because of the following three considerations.

(1) At the lower limit, the peak roller load between buntons was so seldom equal to zero (i.e. the roller was off the guide for an entire bunton space) that the analysis would not be affected.
(2) At the upper limit, the peak roller loads did not show several loads at the same value, indicating that, within the values used for the analysis, the upper limit was reached seldom enough for the analysis to be valid.
(3) The analysis may predict long-term maximum loads that are greater than normal maximum loads. However, this is reasonable since the normal maximum loads may be exceeded, for example when a slipper is badly worn or broken, or when spillage restricts roller movement.

In general, the probability distribution of the measured data will be both skewed and mesokurtosic. In order to make statistical predictions of maximum loads, Alport[4] developed computer programmes known as POWER and SMEMAX, which transform these data to an equivalent normal distribution and then, after statistical evaluation, transform the predictions back to the raw values. These programmes were based on the work by Chander *et al.*[5] and Bethlahmy[6] in return-event analyses.

The roller loads were analysed to determine the peak value during each time interval taken to travel between two adjacent buntons. SMEMAX was then used to determine the following predictions of peak roller load between any adjacent pair of buntons:

- Mean of the peak values in each time interval.
- Standard deviation of the peak values.
- Mean + 3,8 standard deviations representing a probability of exceedance of 0,01 per cent for a normal distribution. In each case, this value was of a similar magnitude to the maximum measured roller load.
- Expected maximum roller load in a time period representing the passing of 1 million buntons (representing approximately one week of operation).

The measured and extrapolated roller loads obtained in this way on 9 conveyances in 7 different shafts are given in columns 5, 6, and 7 of Table I. The equivalent peak roller loads predicted theoretically by the use of DISCS are also identified for 5 shafts in columns 8 and 9 of the table. In arriving at these results, the author made some adjustment of the input values to allow for misalignment, damping, and, in one case, slipper/guide clearance, so that the results would be consistent with the observations. This was necessary because of uncertainty regarding actual misalignment and damping, and to prevent high forces due to slipper/guide contact in the DISCS model. These input values were always kept within reasonable practical limits.

Misalignment was generated on a random basis within the limits specified. The results for Shaft 4 are considered to be less reliable because DISCS assumed buntons of equal stiffness on both sides of the conveyances, which was significantly incorrect in that case. Nevertheless, DISCS predicted peak roller loads within 35 per cent of the measured values.

The following observations arise from these results.

(i) An assessment of the variation of roller load with hoisting speed in the various shafts indicates approximately a linear increase of roller loads with hoisting speed, except in two cases (Fig. 2). The first was Shaft 2, where the roller load was due primarily to a high preload, and was probably magnified by shaft misalignment. The second case was Shaft 4, where there was an exponential increase with decreasing speed, which was considered to be caused by resonance between the skip bending frequency and the roller revolution at 10 m/s.
(ii) Measurements at Shaft 1 indicated that, where the bunton stiffness varied significantly on different sides of the shaft, the peak roller loads were very much higher on the side with stiff buntons and were approximately proportional to the difference in bunton stiffness. No absolute relationship was observed between roller load and bunton stiffness in cases where this difference in stiffness did not exist. In view of the apparent importance of this phenomenon, DISCS has subsequently been updated to allow for differing bunton stiffnesses, and now shows higher loads on the side with stiffer buntons.
(iii) Neither measurements nor parametric studies using DISCS have shown that bunton spacing is an important variable, although most of the measurements were in shafts with the same bunton spacing, i.e. 4,57 m. Only two sets of measurements were in shafts with a bunton spacing of 6,10 m, two with bunton spacings of 3,68 m, and two with bunton spacings of 4,0 m, which are probably insufficient to be conclusive.
(iv) High preloads did not reduce the dynamic behaviour in the two shafts where measurements were obtained with higher preloads. In both cases, the mean load increased approximately by the increased preload, and the standard deviation increased as well (as shown by the skip in Shaft 5). The increased standard deviation is due to either increased dynamic behaviour or an increased effect of misalignment with a higher preload. The effect of this increased standard deviation is to increase the predicted long-term roller loads. It thus appears that high preloads should be avoided in order to reduce roller loads and give longer roller life. The analyses using DISCS did not indicate an increase in dynamic behaviour as the roller preload was increased.
(v) In general, there was not a significant difference between the roller loads on conveyances travelling up or down. This to some extent confirms the assumptions made in DISCS that gravity loads can be neglected.
(vi) No significant difference was recorded between roller loads on empty and fully loaded conveyances. A study of the relationship between roller loads with a 0,01 per cent probability of exceedance and

TABLE I

PEAK ROLLER LOADS AT 15 m/s (kN)

Shaft (1)	Roller (2)	Weight condit. (3)	Dirn of travel (4)	Measured peak* Mean (5)	Measured peak* Std devn (6)	Measured peak* 0,01% prob. exceed (7)	DISCS peak Mean m (8)	DISCS peak 0,01% prob. exceed (9)	Fatigue design eqn Load range/ bunton (10)	Fatigue design eqn Shaft cat. (11)	Fatigue design eqn 0,01% prob. exceed (12)	Fatigue design eqn 10% loaded convey. wt (13)
1	Top	Full	Up	2,9	0,47	4,7	5,7	10,4	3,3		6,8	13,4
105-man cage	Top	60%	Up	4,6	0,65	7,1					6,8	10,5
	Top	Empty	Both	4,0	0,58	6,2	4,7	8,4	3,4		6,8	6,1
	Bot. 1	Full	Up	8,2	1,35	13,3	6,0	13,8	4,1	1	11,7	13,4
	Bot. 2	Full	Up	5,3	0,58	7,5	6,1	13,6	3,5	$\kappa = 1,4$	10,1	13,4
	Bot. 1	60%	Up	7,9	1,25	12,7					11,7	10,5
	Bot. 2	60%	Up	5,5	0,57	7,7					10,1	10,5
	Bot. 1	Empty	Both	7,2	1,27	12,0	4,6	8,2	4,4		11,7	6,1
	Bot. 2	Empty	Both	5,4	0,59	7,6	4,7	8,2	3,7		10,1	6,1
2	Top	Full	Up	8,0	1,64	14,2			6,2	3	15,7	13,4
105-man cage	Top	Empty	Down	8,1	1,45	13,6			6,0	m = 1,3	15,7	6,1
	Bot.	Full	Up	8,3	1,88	15,5			9,2		20,4	13,4
	Bot.	Empty	Down	8,7	1,95	16,1			8,5	$\kappa = 1,6$	20,4	6,1
3†	Bot.	Full	Up	9,2	1,16	13,6			8,8		18,4	23,1
14 t skip	Bot.	Full	Down	9,1	1,65	15,4			9,4	2	18,4	23,1
	Bot.	Empty	Up	10,7	1,42	16,1			12,6		18,4	9,1
	Bot.	Empty	Down	10,3	1,64	16,5			12,2		18,4	9,1
4	Top	Full	Up	4,3	1,34	9,4	4,1	12,7	4,7	2	21,0	29,3
21 t skip	Top	Empty	Both	4,7	1,48	10,3	3,0	6,0		$\kappa = 1,5$	21,0	13,6
	Bot.	Full	Up	6,7	1,19	11,2	5,5	19,9	12,6		30,9	29,3
	Bot.	Empty	Both	7,0	2,15	15,2	2,9	6,4			30,9	13,6
5‡	Top	Full	Both	4,0	0,45	5,7	4,3	7,8			7,3	28,0
18 t skip	Top	½ full	Both	4,0	0,41	5,6					7,3	19,0
	Top	Empty	Both	3,7	0,39	5,2	3,6	5,4		1	7,1	10,0
	Top	Empty	Both	4,1	0,49	6,0					7,4	10,0
6	Top	Full	Both	9,7	1,00	13,5	8,1	14,9		3	17,7	31,0
19 t skip	Top	Empty	Both	9,2	0,84	12,4	6,2	8,5		m = 1,2	17,4	12,0
	Bot.	Full	Both	14,3	2,36	23,3	10,3	18,4			24,3	31,0
	Bot.	Empty	Both	13,2	1,85	20,2	6,3	9,3			23,7	12,0
6	Top	Full	Up	8,2	1,7	14,7			9,4	3	13,1	16,7
10 t skip	Top	Full	Down	8,6	1,4	13,9				m = 1,4	13,1	16,7
	Bot.	Full	Up	10,5	1,8	17,3			12,6		17,7	16,7
	Bot.	Full	Down	10,2	1,3	15,2					17,7	16,7
7	Top	Full	Both	2,6	0,80	5,6	1,2	5,5	1,6		8,1	13,6
10 t skip	Top	80%	Both	1,9	0,68	4,5					8,1	11,8
	Top	Empty	Both	1,6	0,56	3,7	1,2	3,9	1,5	2	8,1	3,8
	Bot.	Full	Both	1,5	0,46	3,2	1,4	7,4	2,6		10,4	13,6
	Bot.	80%	Both	0,9	0,38	2,3					10,4	11,8
	Bot.	Empty	Both	1,3	0,85	4,5	1,5	6,3	2,3		10,4	3,8
1	Bot. 1	Full	Up	12,9	1,81	19,8			8,4		20,9	18,0
10 t skip	Bot. 2	Full	Up	6,1	0,61	8,4			5,0	3	14,6	18,0
	Bot. 1	Empty	Both	12,2	1,78	19,0			9,2		20,9	8,0
	Bot. 2	Empty	Both	6,1	0,48	17,9			4,4		14,6	8,0

*Peak roller load per time interval taken to travel between 2 consecutive buntons

†Wheel loads in Shaft 3 are out-of-plane of guides

‡4th set of results for Shaft 5 reflect measurements with increased preload

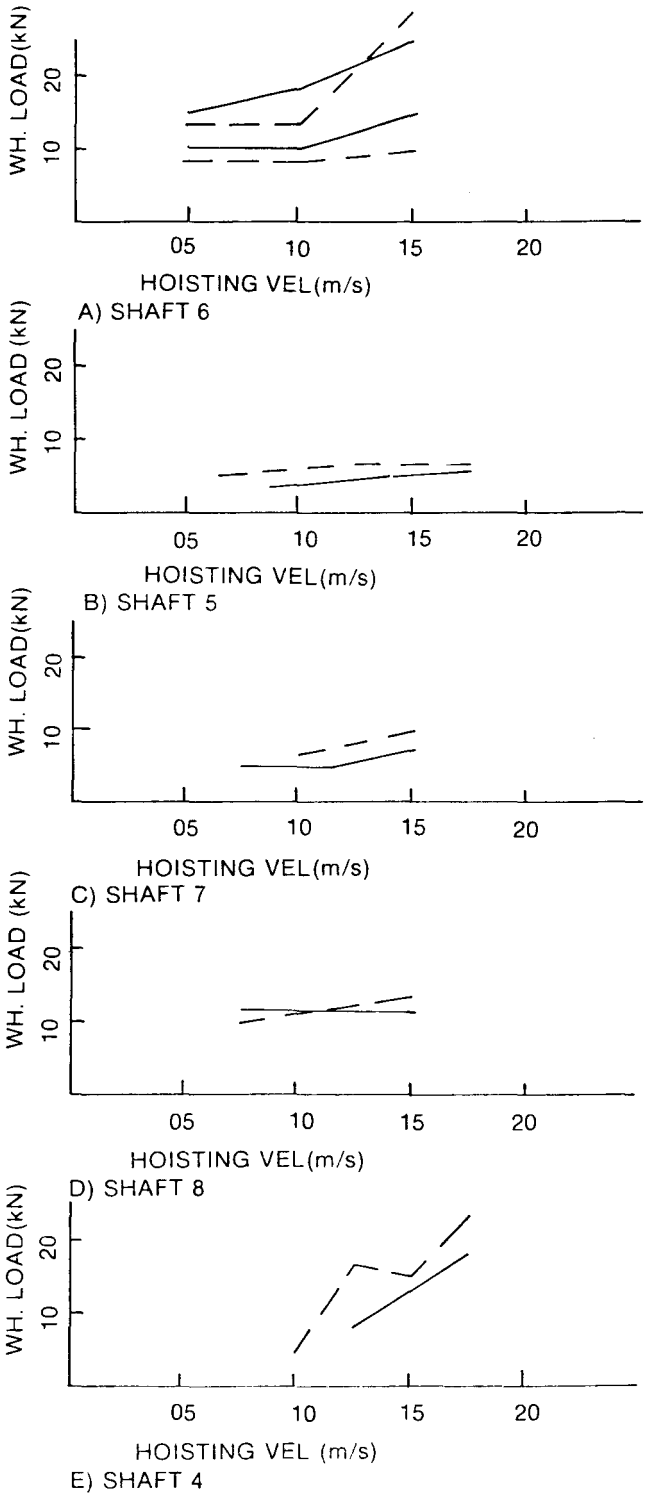

Fig. 2—Comparison of measured forces (—) and forces predicted by DISCS (- - -)

either empty or full weight, or length of conveyance, indicated the least variability with the empty weight (coefficient of variation of 0,33 over all measurements taken to date). As the roller loads on bigger conveyances were generally higher, the empty weight appeared to be the best parameter to describe the size of the conveyance.

(vii) In general on skips, the bottom-roller loads are higher than the top-roller loads. The ratio of the top-roller loads to bottom-roller loads is of the same order as the ratio between the distances from the centre of gravity of the skip to the bottom rollers and to the top rollers, and the difference may relate to this in some cases. In other cases, the relative magnitude of the roller loads is probably more a function of transom mass and of the stiffness of its connection to the body of the skip.

(viii) On the basis of measurements at one shaft, it appears that accelerations normal to the plane of the guides are significant and that the roller loads in this direction should be considered in future.

(ix) The reasonably good agreement between the measured roller loads and the roller loads predicted by DISCS as shown in Table I and Fig. 2, and the fact that DISCS gives qualitatively similar results when defined parameters are altered, indicate that, for roller loads, DISCS can be used with some confidence for parametric studies.

Fatigue Analysis of Roller Loads

The measurements of roller loads on seven conveyances were analysed by Fotopoulos[7] to determine the fatigue spectrum in the following way.

- A computerized form of the Reservoir method[8] described in BS 5400 was developed for cycle counting, and was shown to be sufficiently accurate in view of other uncertainties in the measurements.
- Each fatigue cycle represented by a range of roller load was converted to an equivalent number of cycles of 5 kN load on the assumption of an S–N curve for fatigue strength that is linear on a log scale with a slope of −3 (applicable to the majority of welded details in BS 5400) and Miner's law for fatigue damage.
- The equivalent number of cycles of 5 kN load was converted by use of the same assumptions made for a range of roller loads that will cause the same fatigue damage as the counted fatigue cycles if applied once for every bunton passed. The result is recorded in column 10 of Table I.

Fotopoulos indicated an acceptable degree of correlation in these measurements of fatigue damage between different runs on the same shaft. On the other hand, when he subdivided the observations on one particular run in a shaft into lengths of record of 5 or 10 seconds duration, unacceptably high variations were observed between these records, indicating irregular behaviour over the length of the shaft. This may be a useful approach for an assessment of the relative alignment of different portions of a shaft. In addition, it was observed that a few very high roller loads (Fig. 1c), when converted to an equivalent number of cycles of a lower load such as 5 kN, will significantly alter the assessment of the equivalent number of cycles.

The influence of various factors on roller loads described in the previous section were found to apply in the same way to the equivalent number of cycles of fatigue loading. This is probably not surprising since the same

measurements of roller load were used in both cases, but it does provide greater confidence in this interpretation.

Accelerations

The plots of acceleration measurements were studied so that the peak acceleration could be identified. The results are illustrated in Table II. In the best shafts where measurements were taken, accelerations rarely exceeded $4 m/s^2$. In addition, shafts with accelerations up to $9 m/s^2$ do not appear to present any problems, and this may be an appropriate limit for maintenance purposes.

TABLE II
ACCELERATION VALUES

Shaft	Skip or cage	Peak measured accelerations, m/s^2					
		Speed 5 m/s		Speed 10 m/s		Speed 15 m/s	
		Full	Empty	Full	Empty	Full	Empty
5	18 t skip			4	4	5	5
6	19 t skip		3		5	11	8
	10 t skip	4		6		13	
7	10 t skip	5	2	4		9	4
1	10 t skip		6			9	11
	cage				3	3	4
2	cage					10	12
8	skip			8	8	12	12
4	21 t skip			3	4	7	8
9	cage		5		8		10
10	skip						12
	cage		4		12		
11	skip	5	4	5	4	4	6

In general, DISCS showed a similar level of peak accelerations, and again the effects of varying major parameters, such as velocity, were qualitatively similar to the measured effects.

Frequencies

The frequencies that predominated in the spectral analysis of the accelerations and roller loads, irrespective of hoisting speed, are compared in Table III with the forcing frequency associated with the passing of buntons and revolution of the roller. The frequency of revolution of the rollers was also often evident in the power spectrum, but the frequency of the passing of buntons was evident only in cases of high preload. The frequencies in the passing of higher numbers of buntons were not individually located because the sampling frequency did not give a sufficiently small resolution. The values of spectral density at these low frequencies did not appear to warrant a detailed study of the low-frequency excitation. The 'natural' frequencies, which are not dependent on velocity, are of two types, as follows:

(1) There were typically one or two, and sometimes three, frequencies below 5 Hz that would probably reflect the natural frequencies of 'freebody' rotation and translation of the conveyance.

(2) The frequencies in the second group were generally higher than 9 Hz, and probably represent natural frequencies of flexural deformation of the conveyance.

In general, the power spectra for accelerations and for roller loads gave similar significant frequencies. The only important distinction was that the accelerations tended sometimes to emphasize the natural frequencies, rather than the velocity-dependent forcing frequencies. This is to be expected since the accelerations are mainly the response of the conveyance to the roller loads, unless there is frequent slipper contact with the guides.

A comparison of the measured frequencies produced the following interesting, and to some extent, unexpected results.

(a) The frequency of the passing of buntons as defined in Table III, which initially was expected to be the primary exciting frequency, was significant only at Shafts 1 and 2, where a high preload was applied. Even at these shafts, it was much less significant in the acceleration measurements, indicating that the preload was being transferred from side to side through the transoms without significantly affecting the dynamic behaviour. It is clear from Table III that, of the shafts with high wheel loads (i.e. Shafts 1, 6, and 9), none of the measured natural frequencies coincided with bunton-passing frequencies at the test speeds. Dynamic magnification would, therefore, not be expected at these frequencies, and certainly none was recorded.

(b) The frequency of revolution of the rollers appears to be one of the most important frequencies in most of the measurements. It is felt that this is, in fact, the resonant frequency in two of the bad shafts mentioned above. At Shaft 9, there was a natural frequency at 18,5 Hz, which is the roller revolution frequency at 15m/s. On the skips in Shaft 6, a simple calculation shows that the natural frequency of vibration of the roller itself about its pivot is about the same as the frequency of revolution of the roller. This has been confirmed by measurement, and indicates probable dynamic magnification due to resonance within the roller set itself.

(c) In general, it was difficult to clearly distinguish decreasing natural frequencies with increasing mass, as shown in Table IV. This is probably to be expected with cages carrying men, who would not form an integral mass with the cage but would tend to sway as the cage moves. This is borne out by the reduction in roller loads in loaded cages, which is due to the much higher damping within the load as the men move against one another and relative to the cage. In some, but not all, cases of skips, there was a definite reduction of the two lower natural frequencies. As the rock load in skips would be expected to form an integral solid mass with the body of the skip, this is as expected. In other cases, this frequency reduction was not evident at all, a fact that could not be explained adequately.

At the lower frequencies in Table IV, representing overall translation and rotation of the conveyance in the plane of the guides, there was good correlation between DISCS predictions and the measured natural frequencies.

TABLE III
PROPOSED MISALIGNMENT LIMITS

Misalignment variable	Limiting values		
	1. Good shaft	2. Average shaft	3. Poor shaft
Projected values of max. accelerations, m/s^2	< 4	> 4 < 9	> 9
(a) Overall off-vertical, m	< $D/2000$ < 0,5	> $D/2000$ < $D/1000$; < 0,5	> $D/1000$ > 0,5
(b) Off-vertical increment between buntons/bunton-spacing, Δ/A	< 0,001	> 0,001 < 0,002	> 0,002
(c) Gauge Error, δ Minimum, mm Maximum, mm	> 0 < 2	> 0 > 2; < 4	< 0 > 4
(d) Mismatch without grinding, mm	< 0,3	> 0,3 < 0,6	> 0,6
(e) Equivalent roller irregularity, mm	< 0,5	> 0,5 < 1,0	> 1,0

D = Depth below sheaves; A = Bunton spacing; Δ = Off-vertical misalignment increment

TABLE IV
FREQUENCY MEASUREMENTS AND PREDICTIONS
Note: Frequency of wheel revolution is 9,4 Hz at 7,5 m/s, and 18,8 Hz at 15,0 m/s

		'Natural' frequencies of 'rigid-body' rotation and translation				'Natural' frequencies of tors. and flex. deformation		Bunton passing frequency at hoisting speeds	
		Measurements		DISCS					
Shaft	Skip or cage	Full	Empty	Empty		Measurements	DISCS	7,5 m/s	15,0 m/s
5	18 t skip	1,1 3,1	1,1 3,1	1,8 3,5		10,0 17,5	16,0	1,6	3,3
6	18 t skip	1,5 to 2,1 range	1,5 to 2,1 range	1,2 2,3		13,5 19,0, 20,7	12,0	1,2	2,4
7	10 t skip	1,5 2,5 5,0	1,5 2,5	1,8 3,6		9,5	7,8	2,0	4,1
1	cage	3,1 4,0	3,1 4,0	3,1 3,5		9,0 10,5 14,5	9,7 11,2	1,6	3,3
4	21 t skip	0,9 1,8 2,9	0,7 1,6 2,5	1,1 3,8		5,2 8,7 12,0	6,0	1,9	3,8
11	skip	1,5 4,3	1,3 3,7	–		–	–	1,6	3,3
9	cage	–	1,2 2,3 3,2	–		18,5	–	1,9	3,8

This prediction was not good at the higher-order frequencies (representing deformation of the conveyance) owing to the difficulty of assessing the actual stiffness of bridles and other structural elements.

Some Factors Leading to High Total Forces

The total force between conveyances and guides is the sum of roller loads and forces on slipper plates. Some of the variables that increase roller loads have been identified above from the measurements. The following were the most important of these:

(a) linear increase of roller load with increasing hoisting velocity,
(b) increased roller loads with significantly different bunton stiffnesses on the two sides,
(c) increased roller loads with high preload and stiff buntons, and
(d) increased roller loads with bigger conveyances.

Parametric Studies

SDRC[2,3], in performing detailed measurements in the worst section of a very bad shaft, have highlighted the extremely high slipper contact forces that can occur, and the important variables. The effects of some such variables could not be determined from the measurements described here, but the reasonable agreement between the measurements and DISCS predictions showed that parametric studies using DISCS should be expected to give valid results. The effects of several variables have been studied using DISCS and these are described below.

(1) Slipper/guide clearance is a layout dimension that has traditionally been kept small to prevent the derailment of conveyances. A study of probable forces between conveyance and guides at varying clearances has shown a typical pattern as in Fig. 3. At very small clearances, there is very frequent slipper/guide contact, but dynamic movements are restricted and total contact forces are typically 1,5 to 2 times the 'no contact' roller loads. As the clearance is increased, dynamic movements become greater and total contact forces increase (usually more markedly at higher hoisting speeds) to between 2 to 6 times the 'no contact' roller loads. Further increase in clearance leads to much less frequent slipper/guide contact, thus reducing probable contact forces, until some point is reached at which slipper/guide contact does not occur. Beyond this point the forces remain constant at the 'no contact' roller loads. The magnitude of these forces predicted by DISCS is consistent with measurements of total contact forces made by Structural Dynamics Research Corporation[2] in one particular shaft. The implication of this is that slipper/guide clearances should be increased sufficiently to ensure 'no contact' operation of conveyances, and, in order to maintain current safety levels against derailment, the depth of guide sections should be increased.

(2) Until recently, guide stiffness has not been a parameter that could be altered by the designer, since each mining house used a standard guide section. However, it has for some time been suggested that stiffer guides would improve the performance of shafts. Parametric studies of this variable have shown that there is usually an optimum guide stiffness where the dynamic forces are at a minimum, shown at point A on Fig. 4. As the stiffness decreases below the optimum value, the forces increase rapidly as high slipper contact forces become a problem. As the stiffness increases above the optimum value, there is a slight increase in forces because the whole spring system is stiffer. Further parametric studies will be necessary to identify the guide stiffness, or the ratio of guide stiffness to bunton stiffness, at which this optimum value is likely to occur.

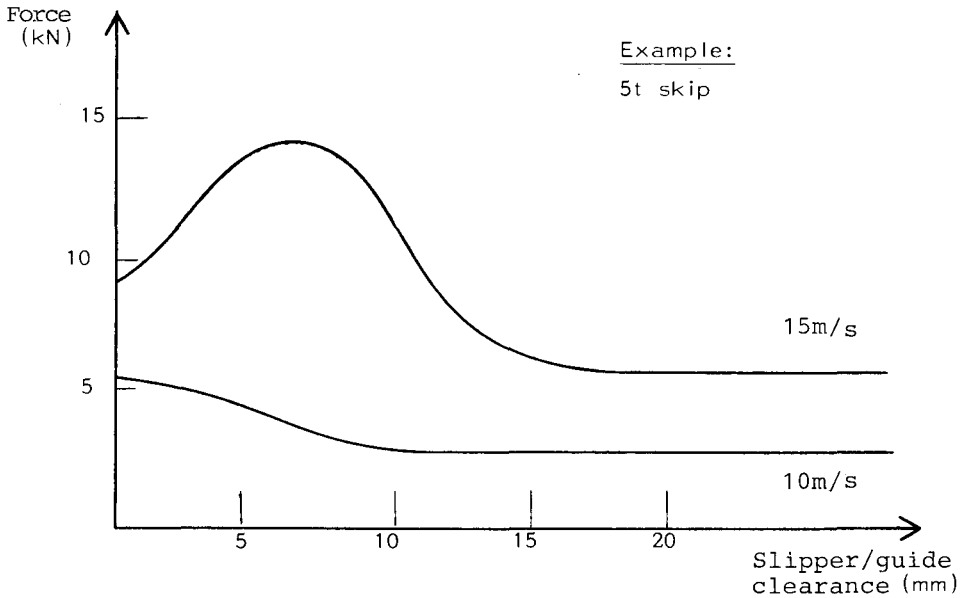

Fig. 3—Relationship between force and clearance

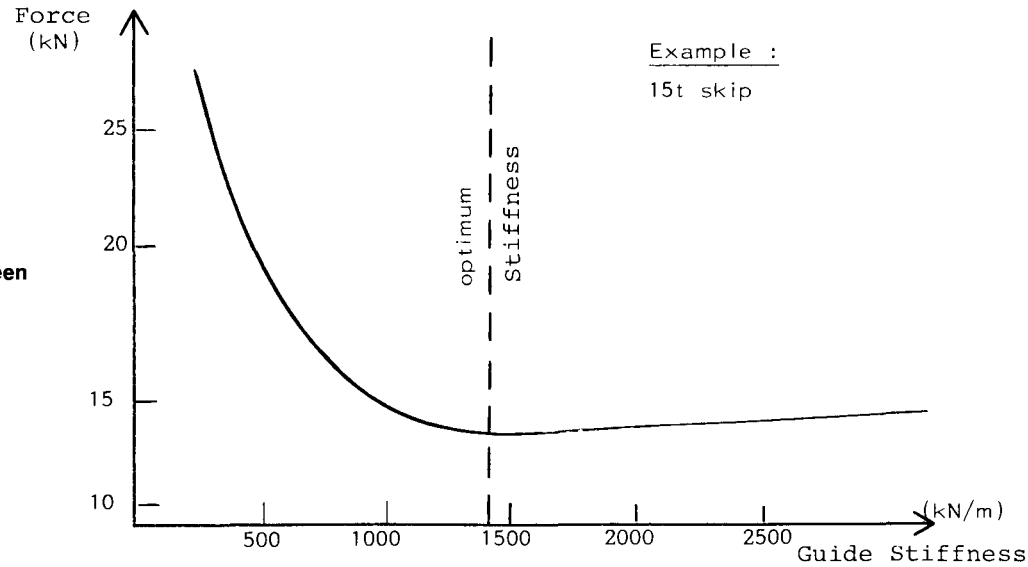

Fig. 4—Relationship between force and guide stiffness

(3) Roller-spring stiffness is a further variable that affects the dynamic forces. Parametric studies have shown that, in general, there is a small increase in forces as the roller-spring stiffness increases, except that, if the stiffness alters to bring about resonance, then there is a high peak as indicated by the dashed line in Fig. 5. The forces also increased markedly if the stiffness is decreased sufficiently for the greater movements to allow slipper/guide contact to occur. This is shown by the dotted line on Fig. 5.

(4) Misalignment could be assessed only from the engineer's judgement in the shafts in which tests were conducted. A parametric study using DISCS showed an approximately linear increase of roller loads with misalignment between consecutive buntons. This is to be expected because accelerations of a mass at a particular frequency are proportional to the amplitude of displacement.

Dynamic Magnification

Total loads may also be increased by a dynamic magnification that is associated with resonant conditions between the exciting frequencies (associated with the passing of buntons or roller rotations) and the natural frequencies of vibration of the conveyance or the roller set. Reports have been made of instances of high dynamic magnification that have resulted in serious consequences for the shaft and conveyances. In a plot relating total loads and velocity of conveyance, an approximately linear increase is expected normally, but, under conditions approaching resonance, a magnification hump is observed above this linear relationship, as shown in Fig. 5. A section of this hump was observed in a few of the shaft measurements, and the complete hump was predicted using DISCS. A simplified analytical approach to this problem has been described by Reinke[9] and the C S I R[10]. In the measurements taken during this project, dynamic magnification was observed in the following three shafts:

(i) Shaft 9 at a frequency of about 18,5 Hz, which is considered to reflect resonance between the frequency of revolution of the rollers at a hoisting speed of approximately 15 m/s and the frequency of the fundamental flexural mode of vibration of the conveyance;

(ii) Shaft 6 at a frequency of about 20 Hz, which may represent resonance between the frequency of revolution of the rollers at a speed of approximately 15 m/s and the frequency of vibration of the rollerset or a flexural-deformation mode of the conveyance;

(iii) Shaft 4 at a frequency of 12 Hz and a hoisting speed of 10 m/s.

Although not measured as part of this investigation, dynamic magnification could also occur at significantly lower frequencies if the frequency of the passing of buntons corresponds to the lower-frequency translational or rotation modes of the conveyance including the spring-loaded rollers. A necessary design criterion for a shaft is therefore that the frequencies of revolution of the rollers or of the passing of buntons should be separated from the natural frequencies of the conveyance and rollerset in accordance with normal dynamic design procedure. The natural frequencies can be assessed theoretically or *in situ* by the use of some form of excitation.

Recommendations for Design

As a result of the observations and theoretical predictions on the various shafts discussed above, certain preliminary and tentative guidelines can be proposed for the design of shafts. It should be emphasized that the qualitative behaviour represented by these proposals is more definitive and relevant than the actual numerical values described in the subsequent expressions. Future measurements may well indicate that these numerical values should be altered or refined.

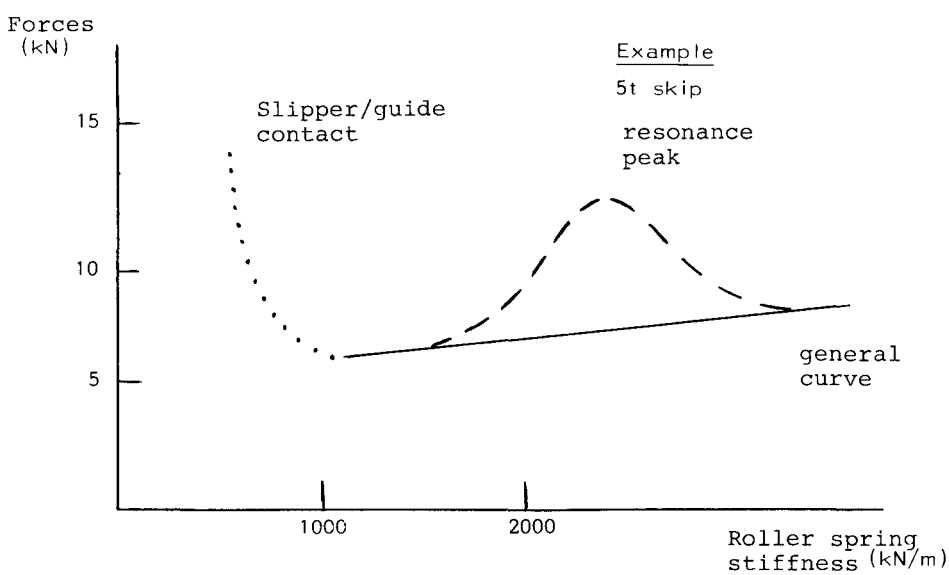

Fig. 5—Relationship between force and roller stiffness

Lateral Accelerations and Misalignment Limits

Five basic components of misalignment or damage should be recognized as follows.

(a) Overall distance of the guide from the true vertical position. This does not have any noticeable effect on the behaviour of the conveyance, but it should be limited to ensure that the conveyance is basically running vertically, and to limit the horizontal component of rope pull. It should therefore be specified as a ratio of misalignment to distance from surface.

(b) The increment of the guide position from true vertical, from one bunton to the next. This is an important factor in determining the dynamic behaviour and should be carefully controlled. This is defined in terms of the off-vertical increment divided by the bunton spacing, i.e. the off-vertical slope of the guide.

(c) The error in the gauge between guides. This can be controlled to acceptable levels reasonably easily by the use of a template, and it is also of significant importance in dynamic behaviour. Reduction in guide gauge could lead to the jamming of conveyances and should therefore not be permitted. An increase in guide gauge could lead to rollers coming away from the guides, which results in increased roller loads and severe tyre wear, and should therefore be limited to a small amount.

(d) Guide mismatch at joints. This leads to high shock loads and should thus be minimized, either by some system ensuring that guides of very nearly equal height are placed consecutively down the shaft, or by the grinding of joints where there is a mismatch to a maximum slope of 1:20.

(e) Damage, wear, or ovalling of the roller. This is represented by a defined equivalent irregularity each time the roller rotates.

These misalignments have been categorized in Table III in terms of the following three groups:

- Good shafts: projected maximum accelerations $< 4\ m/s^2$,
- Average shafts: projected maximum accelerations $> 4\ m/s^2$ but $< 9\ m/s^2$,
- Poor shafts: projected maximum accelerations $> 9\ m/s^2$.

On the basis of these criteria, the shafts at which measurements have been taken are categorized in column 11 of Table I in terms of 1 = Good shaft, 2 = Average shaft, and 3 = Poor shaft. The misalignment values related to these accelerations were obtained from the engineer's assessment of the shaft and the values that produced such accelerations in DISCS.

Design Equation for Roller and Slipper Contact

The expression for the design value of peak force, P, during the time interval of passing between two adjacent buntons is expected to be of the following form (on the assumption that resonance has been checked as described above and is not a problem):

$$P = L_d + W_d + S_d = L_m(1 + g_1.C_L) + W_e\, g_1\, C_w + (S_m + g_2\,.\,S_v), \quad (1)$$

where L_d = design value of total contribution due to roller preload

W_d = design value of total contribution due to conveyance weight

L_m = mean roller preload

C_L = coefficient of variation of roller load due to dynamic variations associated with the level of preload

g_1 = statistical constant for a defined probability of occurrence of roller loads

W_e = empty weight of conveyance

C_w = coefficient of variation of roller load due to dynamic variations associated with conveyance weight

S_d = design value of slamming or slipper to guide contact force

S_m = mean slamming or slipper to guide contact force

g_2 = statistical constant for a defined probability of occurrence of slamming

S_v = standard deviation of slamming or slipper to guide contact force.

The first term in the expression represents the influence of roller preload and is a function of the following variables:

- Preload, L_o, set in the rollers at the bank.
- Increase in preload to mean magnitude, L_m, associated with mean gauge reduction, δ, and mean spring stiffness of roller in shaft compared with conditions at bank.
- Dynamic increase associated with roller preload and related to the ratio of bunton stiffness to which the two guides are attached, κ (>1,10).

$$\text{Thus, } L_m = K_s\left(\frac{L_o}{K_b} + \frac{\delta}{2}\right) \approx \left(L_o + K_s\,.\,\frac{\delta}{2}\right), \quad (2)$$

where K_s = mean roller spring stiffness in shaft

K_b = roller spring stiffness at bank.

$$\text{Thus, } L_d = L_m\,[1 + g_1.a_1.\,(\kappa - 1)] = [L_o + K_s\,\frac{\delta}{2}]\,[1 + g_1.a_1.\,(\kappa - 1)] \quad (3)$$

The constant a_1 was assessed from the measurements and theoretical results to be approximately equal to 0,08.

The second term in the expression for roller load represents the influence of the conveyance weight. It has already been noted that the weight of an empty conveyance, W_e, appears to be the most consistent parameter in the assessment of this dynamic contribution. Other variables that are considered to influence the conveyance-related component of the roller load are the speed of travel, V, and a misalignment factor, m. The misalignment factor is an attempt to obtain a single measurement of the influence of roller irregularity, off-

vertical increment, and mismatch as described in Table III. It is assumed to have a value of 0,5 for misalignments meeting the limits for a 'good shaft', 1,0 for an 'average shaft', and 2,0 for a 'poor shaft'.

$$\text{Thus, } W_d = W_e.g_1.\ C_w$$
$$= W_e.g_1.\ (a_2.\ V.m). \quad \text{.....(4)}$$

The constant a_2 has been assessed from the measurements and theoretical results to be approximately equal to 0,0025 for skips and 0,0035 for cages.

The design value of peak roller load for the bottom rollers is therefore

$$P_b = [L_o + K_s\frac{\delta}{2}]\ [1 + g_1.a_1.(\kappa - 1)] + W_e.g_1.(a_2.V.m).$$

The design value of peak roller loads for the top rollers is assumed on the basis of the measurements to be 75 per cent of the bottom roller loads,

i.e. $P_t = 0{,}75\ P_b$.

The resulting values of design roller loads (i.e. the first two terms in the expression for P) are given in column (12) of Table I, based on the following assumptions.

- The shaft categories are defined in column (11) of this table, where 1 = a good shaft (m = 0,5), 2 = an average shaft (m = 1,0), and 3 = a poor shaft, for which m is assumed to be equal to (maximum measured acceleration/$9m/s^2$) as defined in column (11). The basis of this categorization is the acceleration measurements given in Table II as interpreted in Table III. In a design situation in which the condition at which the shaft is going to be maintained is unknown, it would be necessary to make a conservative assumption such as m = 1,5.
- The mean value of the roller preload, L_m (Equation 2), is assumed in most cases to be equal to the mean measured preload (calculated in the fatigue analysis) or half the mean peak load (column (5) of Table I) in the two shafts where fatigue loading has not been analysed. The values K_s, L_o, and the gauge misalignment δ (Table III) would have to be identified in a design situation.
- A value of g_1 of 3,8 is assumed, representing a 0,01 per cent probability of occurrence for a normal distribution. This result can then be compared in Table I with the equivalent measured value (column (7)) and the DISCS value (column (9)).

The term for slamming and slipper to guide contact is included since it is a potential cause of high forces, as identified by Structural Dynamics Research Corporation[2,3]. At this stage, further analytical and measurement work is required before design equations can be proposed.

The assessment of the fatigue life should allow for the application of roller forces, i.e. the first two terms of Equation (1), at every bunton, and for the application of the slipper forces at a reduced number of buntons, depending on the probability of slipper contact with the guide. Thus, fatigue design is based on the two forces:

P_1 applied N_1 times
P_2 applied N_2 times,

where

$P_1 = L_d + W_d$
$P_2 = S_d$
$N_1 = 2 \times T \times H$ for steelwork
$N_1 = 2 \times B \times T \times H$ for conveyances
$N_2 = P \times N_1$ for steelwork and conveyances
T = number of trips per hour
H = hours of operation during the life of the shaft
B = number of bunton levels in shaft
P = probability of slipper contact on the guide.

As an alternative to this approach to the assessment of the design forces, a computer programme such as DISCS could be used.

Conclusions

On the basis of the measurements and theoretical analyses undertaken in this project, the magnitude of the roller loads appears to be increased by the following parameters:

- Significantly by the hoisting speed, guide misalignments between buntons and across the gauge, empty weight of conveyance, roller preload, large differences in bunton stiffness, slipper guide clearance, and guide stiffness.
- To a smaller degree by roller-spring stiffness and bunton stiffness.

The roller loads were not significantly affected by the direction of travel and whether the conveyance was empty or full. Existing design formulae assuming a maximum roller load equal to 10 per cent of the loaded weight of the conveyance appear to be reasonably safe in most cases, but a formula identifying the observed contribution during this project of the most important variables is defined in the paper. An approach for the identification of the load range that should be applied once for each bunton passed in fatigue design is also described. This was shown to be of similar magnitude to the mean value of the measured peak loads in each time interval corresponding to the passing of one bunton.

Critical dynamic conditions that are not predicted by conventional design formulae may occur if resonance develops between the forcing frequencies, represented by revolution of the roller or possibly passing of buntons, and the natural frequencies of rigid-body rotation or translation, flexural or torsional deformation, and displacement of the rollersets.

Acknowledgements

The author acknowledges with thanks the substantial support received from the C S I R, the Chamber of Mines of South Africa, Dorbyl Limited, and the Department of Civil Engineering, University of the Witwatersrand, Johannesburg, as well as the help and guidance of many individuals in the mining industry.

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Resources of Southern Africa

The Associated Scientific and Technical Societies of South Africa, in collaboration with the Council for Scientific and Industrial Research, will stage a major conference on the resources of Southern Africa from 12th to 14th November, 1986, in the Linder Auditorium at the Johannesburg College of Education in Parktown, Johannesburg.

The theme of the Conference will be a re-evaluation of Southern Africa's resources in the light of developments over the past decade, and the potential contribution of science and technology in the utilization of those resources to the benefit of the peoples of the region.

The objective of the Conference is to review the findings and recommendations of the Conference 'Resources of Southern Africa—Today and Tomorrow', which was held in 1975, and to see the extent to which trends then evident are varying.

The topics, which will be addressed in five sessions, are as follows:

- Development of demographic and human resources, including population growth, education and training, technology, and productivity
- Agriculture, including food and water resources
- Energy, including renewable sources of energy
- Materials, including renewable resources such as timbers and fibres
- Infrastructure, including
  - transport, communications, electric power, housing
  - public health care and water supply
  - environment, recreation, and tourism.

The organizers are hoping to attract speakers and delegates from the other States in Southern Africa that made significant contributions to the previous Conference.

The keynote address will be presented by Dr J.G. Garbers, President of the Human Sciences Research Council. Dr N. Stutterheim, Chairman of the Telephone Manufacturers of S.A. (Pty) Ltd and a past president of the Associated Societies, will present the closing address.

Further information is obtainable from

Miss Nicky Fehrsen
The Associated Scientific and Technical Societies of Southern Africa
Kelvin House
2 Hollard Street
Johannesburg 2001.
Telephone: (011) 832-2177.

Impact loading and material behaviour

An International Conference on Impact Loading and Dynamic Behaviour of Materials is to be held in Bremen from 18th to 21st May, 1987.

The aim of the Conference, to be held in English, is to provide an up-to-date comprehensive assessment of recent progress and unsolved problems in the fields of impact and high-strain-rate loading. The following topics will be discussed.

- Deformation mechanisms and their modelling
- Impact mechanics and materials behaviour of metals, ceramics, and composites (metallic and non-metallic)
- Failure criteria
- Constructive equations including computer simulations, shockwave effects, dynamic fracture phenomena, shearband phenomena such as generation conditions (penetration or other tests), and material aspects
- High-speed forming processes.

For more information contact

Conference Secretariat
Deutsche Gesellschaft für Metallkunde e.V.
Adenauerallee 21
D-6370 Oberursel
West Germany.
Telephone: 06171/4081.

Letter to the Editor

The following letter has been received from Mr J. Levin, c/o Mintek (Private Bag X3015, Randburg, 2125 South Africa).

In the March issue of the Institute's *Journal*, on page 96, there is an interesting account of the development of the Black Mountain operations. Readers are told that Phelps Dodge established the existence of a large deposit of valuable base-metal minerals in 1970 and that, in 1976, Phelps Dodge sought a partner in the exploitation of the deposit. However, nothing is said of anything that might have happened in the years between 1970 and 1976. Perhaps nothing did happen, and the years were allowed to slip by without any effort to recover the costs of the exploration and then to profit from it. Perhaps, in the forlorn expectation of a rise in the value of base-metal concentrates, it might have been decided that delay would, ultimately, be advantageous. Or was there difficulty in finding a partner in the project and, if that was the case, what information additional to that already available from the geological exploration was provided to show that the working of the deposit was technically and economically feasible?

I believe that anyone who was sufficiently interested to read the article as published would also be interested in information that would fill the gap in it, indicated above.

Mr R.A. Snodgrass (Gold Fields of South Africa, P.O. Box 1167, Johannesburg 2000) has replied as follows.

It is more correct to state that 'Phelps Dodge suspected the existence of a large deposit of valuable base metals in 1970'.

Phelps Dodge registered as an exploration company in the second half of 1970, and up to that date reports on the mineral deposits at Aggeneys had not been favourable.

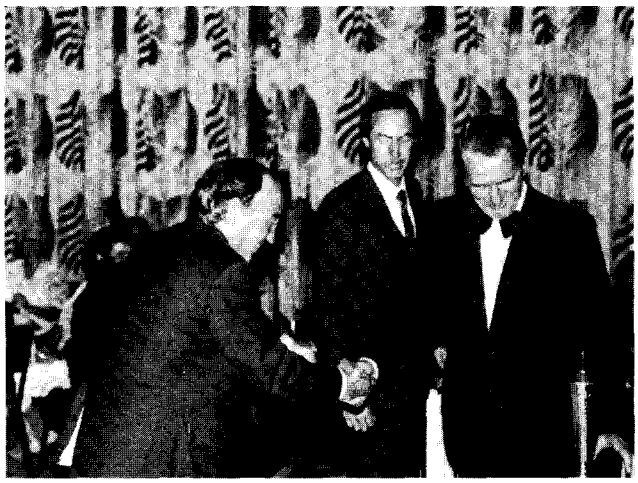
Drilling of the first borehole started on 21st June, 1971. Exploration, which included extensive drilling, the cutting of crosscuts, and the opening of an adit, continued until the end of 1974, when a feasibility study was initiated. The study was completed in 1976, and later that year Phelps Dodge sought a partner to exploit the deposit.

More details of the history of Black Mountain are given in *Papers and Discussions* 1980–1981 of the Association of Mine Managers of South Africa.

Tribology awards

At its recent Annual Dinner, attended by approximately 120 people, the South African Institute of Tribology honoured those who had made significant contributions to the science of tribology in 1985. *Tribology*—derived from the Greek word *tribos*, meaning to rub—is the study of wear caused by interactive faces in relative motion to one another.

In his speech, the patron of the Institute, Dr Louw Alberts, Director General of Mineral and Energy Affairs, praised the Institute for the progress it had made since formalizing its existence barely twelve months ago. During this time, it had not only gained international recognition but had also been well supported by local industry, and research and academic institutions.



Left to right: Professor Tony Ball, Mr Graham Wright, and Dr Louw Alberts

The President's Award, known as the Louw Alberts Award, was made to Professor Tony Ball, Head of the Faculty of Metallurgy and Materials Engineering, University of Cape Town. This award was made in recognition of the outstanding contribution that Professor Ball has made to tribology in the research and development of wear-resistant materials, particularly in the mining industry. The award for the best technical paper presented in 1985 was made to Mr Michael Neale of Michael Neale & Associates, U.K., for a paper entitled 'Understanding Engine Wear'. The third award—for best technical achievement—was not made. Instead, recognition was given to the contributions of the current President of the Institute, Mr Graham Wright, and the Institute's Secretary, Mr Roger Pacey, in formalizing the Wear Society, which they had formed over three years ago. Together with a group of people involved in the area of wear, they had launched the South African Institute of Tribology.

In his speech, the Guest Speaker, Mr John Hall, Executive Director of Barlow Rand, stressed the need to communicate with individuals at all levels. Through communication, better working conditions would be achieved that would eventually lead to a better South Africa.