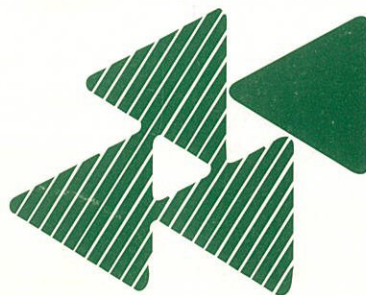


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**SIMRAC PROJECT GEN 420**

**A MANEYLAW, G NORMAN  
and F H von GLEHN**



**Mining  
Technology**

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SIMRAC GEN420

**AN EXAMINATION OF METHODS  
WHEREBY NOISE LEVELS IN  
CURRENT AND NEW MINING  
EQUIPMENT MAY BE REDUCED**

**SIMRAC PROJECT GEN 420**

**A MANEYLAW, G NORMAN  
and F H von GLEHN**

**DECEMBER 1997**

**BBE**  
Mining and Industrial Consultants

 **IMC**  
IMC Health and Safety Consultants

## **Executive summary**

An extensive literature review of international work [mainly completed in the 1970s and 1980s] on mining equipment noise control has been carried out. The sources of noise on percussion rock drills, continuous miners, dust scrubbers and fans, longwall machinery and trackless vehicles are identified. Noise control techniques, for retrofit and for incorporation in new equipment, are proposed and noise reductions and order-of-magnitude costs quantified. A database structure and data capture forms are proposed which will enable equipment noise levels to be compared.

Equipment noise control is considered in terms of four key areas: mineral winning, tunnel drivage, ventilation and transport systems. Noise control treatments are discussed in terms of the acoustic principles of sound insulation and absorption and vibration isolation and damping. For some equipment, the application of simple acoustic treatments can give significant reductions in noise levels. For other equipment there are no "quick fix" solutions while some acoustic treatments, which give small but worthwhile reductions in noise level, have drawbacks in terms of reduced performance.

### **• Rock drills**

Measurements on unsilenced pneumatic and water-hydraulic percussion drills have indicated noise exposure levels in the range 110 dB(A) to 114 dB(A) and 100 dB(A) to 106 dB(A) respectively.

Drilling noise is the sum of three major noise sources - exhaust noise (absent in hydraulic drills), drill steel noise and drill body noise. Drill steel noise results from transverse and longitudinal vibrations of the rod. Drill body noise is produced by moving and impacting parts inside the drill exciting the drill body and becomes apparent only when exhaust noise and drill steel noise have been reduced significantly.

Noise reduction techniques include exhaust mufflers, drill body enclosures, constrained layer damping collars and sheaths and concentric drills. Noise reductions of up to 15 dB(A) have been reported.

- **Continuous miners**

Measurements of noise from continuous miners and roadheaders give operator noise levels between 98 dB(A) and 104 dB(A) when cutting and loading. The major noise sources are the chain conveyor, cutting noise and the machine mounted dust scrubber. Secondary noise sources are the drive train and the hydraulic system.

Noise reduction techniques include the application of constrained layer damping to chain conveyor deck plates, damped cutting heads and resiliently mounted picks. Noise reductions of up to 8 dB(A) have been reported.

- **Longwall systems**

On longwall coalfaces, the armoured face conveyor and the shearer are the main noise sources. Noise levels at the shearer operator's position have been measured at 100 – 105 dB(A).

Techniques for noise reduction include the elimination of discontinuities from, and the application of damping to, armoured face conveyor line pans, close shielding/enclosures for crushers and stage loaders and stiffening/damping of the shearer cutting drum. Reductions of 4-6 dB(A) have been reported.

- **Trackless vehicles**

Extensive measurements of noise from trackless vehicles gave mean noise exposure levels of 99 dB(A). The engine generally constitutes the major noise source and noise may come from the exhaust, the intake and the casing. The cooling fan, transmission, drive train and hydraulic systems can also be significant sources of noise.

Standard noise control techniques include enclosure, absorption and vibration isolation. Noise reductions up to 12 dB(A) have been reported.



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# 1. Introduction

Mining is noisy by virtue of its inherent operations involving drilling, rock breaking and transporting rock and material. The layout and dimensions of underground workings are dictated by the mineral deposit, mining methods etc. and the result is that conditions are often extremely cramped, resulting in noise levels being enhanced by reflection. In addition, the headings and working faces are constantly moving, making remedial measures difficult. These conditions, which are peculiar to underground mining, complicate conventional noise abatement arrangements and at the same time increase exposure levels.

## 1.1 Legislation, codes of practice and guidelines

The Mine Health and Safety Act<sup>(1)</sup> came into force in 1997 and among its objectives are:-

- To protect the health and safety of persons at mines
- To require employers and employees to identify hazards and eliminate, control and minimise the risks to health and safety at mines

Section 5 (1) of the Act states "To the extent that it is reasonably practicable, every manager must provide and maintain a working environment that is safe and without risk to the health of employees".

Section 21 (1) of the Act states "Any person who -

(a) designs, manufactures, repairs, imports or supplies any article for use at a mine

must ensure, as far as reasonably practicable -

(i) that the article is safe and without risk to health and safety when used properly ....."

Thus there are responsibilities on mine managers and equipment suppliers in terms of health and safety at mines.

The Act has given new impetus to the issues of noise control and hearing conservation within the South African mining industry, where there are already requirements under Regulation 4.17 the Minerals Act of 1991<sup>(2)</sup> for a manager to take steps to reduce the equivalent personal noise exposure ( $N_{eq}$ ) to below 85 dB(A), and where this is not

possible, to implement a hearing conservation programme which complies with the recommendations given in SABS083:1983 "Code of Practice for the Measurement and Assessment of Occupational Noise for Hearing Conservation Purposes" (as amended)<sup>(3)</sup>.

The document "Guidelines for the Implementation and Control of a Hearing Conservation Programme in the South African Mining Industry"<sup>(4)</sup>, produced by the Special Sub-Committee on Hearing Conservation of the Chamber of Mines of South Africa, outlines a policy broadly similar to that adopted in the UK coal mining industry, and its recommendations were designed to assist in achieving compliance with the requirements of SABS083. It consists of the following components: -

- Noise zones and noise measurement
- Training and Education
- Hearing Protection
- Medical and Audiometric Services
- Noise Control and Planning
- Machinery Suppliers' Scheme
- Research

Within the South African mining industry, as elsewhere, the emphasis has been on the provision of hearing protection, with the backing of appropriate education and audiometric services. Also, a zoning policy has been adopted, similar to that operating in the UK mining industry, but with differences in terms of zone action levels and measurement procedures. This has resulted in extensive noise data being available at some mines which could be used for establishing an industry wide database on equipment noise levels, exposure levels and numbers of workers affected.

## **1.2 Noise measurements in the South African mining industry**

Extensive work has been completed under previous projects to measure the octave band noise levels of underground equipment and to provide equivalent noise exposure levels for workers in a range of jobs, with and without hearing protection<sup>(5),(6),(7)</sup>. For underground gold mining, Kielblock gives unprotected noise exposure levels for a range of job categories, as shown in Table 1.1. It is estimated that the level of 111 dB(A) for pneumatic drill operators affects approximately 90,000 workers.

TABLE 2.1: WORKERS EMPLOYED IN THE SOUTH AFRICAN MINING INDUSTRY (1996)

MINING TYPE	NUMBER OF WORKERS			% OF OVERALL TOTAL
	UNDERGROUND	SURFACE	TOTAL	
GOLD	239543	62574	302117	61.02
PLATINUM	58719	15616	74335	15.01
COAL	28792	30914	59706	12.06
DIAMOND	6122	9204	15326	3.10
CHROME	3968	2663	6631	1.34
IRON ORE	1772	4214	5986	1.21
COPPER	732	4403	5135	1.04
GRANITE	589	2644	3233	0.65
LIMESTONE	585	2292	2877	0.58
CLAY	130	883	1013	0.20
OTHER MINES	4263	14512	18775	3.79
OVERALL TOTAL	345215	149919	495132	100

TABLE 2.2: MINING EQUIPMENT NUMBERS

DESCRIPTION	A	B	C	D	E	F	G	H	J	TOTAL
<b>LOCOMOTIVES</b>										
DIESEL	2	28	547	0	26	349	5	1338	476	2771
BATTERY	1	18	1149	0	2	128	56	685	1816	3855
<b>FLAMEPROOF EQUIPMENT</b>										
GATHERING ARM LOADERS	0	0	0	0	27	179	1	4	0	211
SHUTTLE CARS	0	0	0	0	50	720	7	95	0	872
COAL CUTTERS	0	0	0	0	79	176	3	9	0	267
BATTERY SCOOPS	0	0	0	0	46	70	0	9	0	125
LOAD HAUL DUMPERS	0	0	0	0	31	285	7	17	0	340
CONTINUOUS MINERS	0	0	0	0	17	203	2	18	0	240
SHEARERS	0	0	0	0	4	14	0	1	0	19
DRILL RIGS	0	25	12	0	4	111	2	46	57	257
ROOFBOLTERS	0	0	0	0	80	364	0	33	0	477
TRACTORS	0	0	0	0	71	434	3	17	0	525
COMPRESSORS	0	0	0	0	50	200	8	25	0	289
PERSONNEL VEHICLES	0	0	0	0	25	357	2	24	0	408
FEEDER BREAKERS	0	0	0	0	22	248	2	22	256	559
<b>WINCHES</b>										
SCRAPERS	0	91	9037	0	21	1383	86	7434	12874	30926
MONO ROPE CONVEYORS	0	1	2845	0	45	92	12	1420	1987	6402
<b>TRACKLESS MINING</b>										
DUMPERS	215	212	27	22	195	290	153	214	340	1668
LOADERS (INC. AIR)	116	88	991	0	4	242	103	319	931	2794
SHOVELS	128	47	4	30	18	33	40	54	140	494
BULLDOZERS	236	60	13	13	31	171	47	61	223	855
LOAD HAUL DUMPERS	95	84	75	95	151	41	76	370	179	1166
PERSONNEL VEHICLES	45	187	35	0	95	784	370	575	322	2413
SERVICE VEHICLES	0	478	17	0	12	310	347	203	121	1488

A WESTERN CAPE    B NORTHERN CAPE    C FREE STATE    D EASTERN CAPE    E KWAZULU NATAL  
 F MPUMALANGA    G NORTHERN REGION    H GAUTENG    J NORTH WEST REGION



TABLE 2.3: NOISE LEVELS FOR A RANGE OF JOB CATEGORIES IN GOLD AND PLATINUM MINING

JOB CATEGORY	OCTAVE BAND SOUND PRESSURE LEVEL (dB(A))										SUM (dB(A))	N <sub>eq</sub> (DOSIMETRY)			
	12.5	250	500	1k	2k	4k	8k								
DEVELOPMENT DRILL OPERATOR;-															
UNSILENCED PNEUMATIC	94	102	108	109	111	108	106	106	116	111					111
SILENCED PNEUMATIC	89	94	97	100	103	104	100	100	108	104	100	100	108	104	104
WATER-HYDRAULIC	77	79	85	86	100	102	98	106	106	102	98	106	101	101	101
STOPE DRILL OPERATOR;-															
UNSILENCED PNEUMATIC	95	101	108	110	112	111	109	109	117	111	109	117	112	112	112
SILENCED PNEUMATIC	88	90	95	100	104	105	103	103	110	105	103	110	105	105	105
WATER-HYDRAULIC	75	77	87	88	105	107	103	103	110	107	103	110	105	105	105
DEVELOPMENT TEAM (EXCL. DRILLERS)	90	99	101	104	105	104	102	102	111	104	102	111	103	103	103
STOPE TEAM (EXCL. DRILLERS)	92	102	106	108	109	107	104	104	114	107	104	114	102	102	102
PNEUMATIC LOADER OPERATOR	87	94	96	98	100	101	100	100	107	101	100	107	106	106	106
DIESEL LOCO CREW	78	82	86	90	90	89	80	80	96	90	80	96	94	94	94
ELECTRIC LOCO CREW	82	85	86	86	86	82	73	73	92	82	73	92	93	93	93

TABLE 2.4: NOISE LEVELS FOR A RANGE OF JOB CATEGORIES IN COAL MINING

JOB CATEGORY	OCTAVE BAND SOUND PRESSURE LEVEL (dB(A))										SUM (dB(A))	N <sub>eq</sub> (DOSIMETRY)
	125	250	500	1k	2k	4k	8k					
CONTINUOUS MINER OPERATOR	89	93	97	96	93	90	81				102	96
CONTINUOUS MINER ASSISTANT	92	95	99	96	95	91	83				103	95
SHUTTLE CAR OPERATOR	78	85	88	86	84	78	70				92	94
CONTINUOUS MINING TEAM LEADER	85	96	100	100	104	99	89				108	96
LONGWALL SHEARER OPERATOR	85	93	98	100	96	94	85				104	101
LONGWALL SHEARER ASSISTANT	79	86	93	96	93	90	85				100	100
LONGWALL CRUSHER ATTENDANT	73	82	84	87	86	84	82				92	96
LONGWALL TEAM LEADER	82	90	97	98	96	92	82				103	97
LHD DRIVER	80	84	90	94	95	96	83				100	99
UTILITY VEHICLE DRIVER	86	87	91	97	96	93	85				101	93

### 3. Equipment noise control

Throughout the 1970's and 1980's, a significant amount of work was completed in mining equipment noise control. Much of this work was done in the U.S.A. by the U.S. Bureau of Mines (U.S.B.M.), and in the U.K. and other EU countries with part funding from the European Coal and Steel Community (E.C.S.C.).

Investigations were carried out into the sources of noise on a range of mining equipment including continuous miners and roadheaders, longwall shearers, underground transport, trackless vehicles and rock drills. Numerous reports and papers were published, characterising the noise from these items of equipment and detailing the development of noise control treatments.

Development work was completed into the redesign of some equipment to produce "low noise" versions for possible manufacture. For retrofit treatment, significant reductions in noise levels for some equipment were demonstrated by the application of standard acoustic principles.

An introduction to the sources of machinery noise and the application of these acoustic principles, including insulation, absorption, enclosure, isolation and damping, is described here.

Noise reduction measures fall into two general categories:-

- noise control at source
- noise reduction by treatment remote from the source

Often, the first of these can only be carried out in the design of the machine and requires detailed knowledge of the working of the machine and the noise producing processes.

The second category can be applied without detailed knowledge of the machine and the mechanics of the noise production, but requires knowledge of the spectrum of the noise if it is to be effective. In many cases this is the only practical option because of time and cost considerations.

### 3.1 Noise control at source

As an example consider a diesel engine which has pistons moving up and down in the cylinders. The motion of the pistons produces harmonically varying forces within the engine and the engine structure responds to these forces, is set into vibration and radiates noise. Counterbalancing of the engine reduces the force to a minimum, but it is impractical to eliminate it entirely.

If the main noise generating structure has a natural frequency within the range of the engine out of balance frequency, it can be expected that high noise levels would be produced over part of the engine speed range due to resonance effects. Although this situation should not occur in a well designed engine, there are many machines that resonate. Secondly, nearly all rotating machines are out of balance to some extent and the rotational frequency of the machine is nearly always a prominent peak in the noise spectrum.

The response of the noise radiating structure can be reduced by reducing the excitation, that is, the out-of-balance forces, or by altering the structure itself. This can be done by damping the structure, altering its natural frequencies or isolating the structure from the forcing mechanism.

In addition to out of balance forces, other mechanisms for noise production in machinery are:-

- excessive clearance in bearings
- aerodynamic noise
- electrical noise
- pressure variations in hydraulic systems
- stick-slip between rubbing surfaces

and these can be considered in relation to the noise production of machine components.

#### • **Bearing Noise**

Plain journal bearings for rotating shafts can rumble or knock when periodically forced because the radial clearances are too large. Reduction of this clearance is not always



easy to achieve. Additionally, the knock can sometimes be the result of the shaft running at its natural bending frequency.

"Screech" is a common bearing noise in light electrical machines and occurs soon after the machine has started up from cold. It is thought that this is caused by poor lubrication, allowing stick-slip motion between the shaft and bearing, which produces a high frequency excitation that is amplified by the machine casing.

Oil film whirl can be another cause of noise in well-lubricated plain bearings, in which the shaft performs a circular motion within the clearance of the bearing. This can be dealt with by changing the bearing clearance or the viscosity of the lubricant.

In rolling bearings the noise is generated by irregularities in the bearing tracks, balls or rollers, and cages. This can give a complex noise signature, at a number of different frequencies and all their harmonics. A major cause of noise is damage to the tracks during poor assembly.

## • Gear Noise

Gears produce noise because each pair of teeth produces an impulse as they come into contact and produce a sliding contact as they move against each other. Accuracy in manufacture is of primary importance for low noise, poor quality control causes variations in the loading between teeth as the gears rotate. Generally straight tooth spur gears are noisier than helical, angled tooth gears. The impulsive contact is virtually eliminated in the latter as the teeth slide into engagement gradually.

Noise produced by the gears in a machine is often at frequencies corresponding to the modes of the structure of the machine, rather than at gear tooth frequencies. This is to be expected as tooth errors in gears produce forcing over a wide range of frequencies. Sometimes, the noise is at the natural frequencies of the gears, and this can be reduced by applying damping to the gears or by reducing the clearances. Another option, where feasible, is to replace one or both gears of a pair with a non-metallic gear. Because of the relatively low stiffness of the non-metallic gears, they are more able to deal with manufacturing inaccuracies.

## • Hydraulic Pump Noise

High pressure hydraulic systems can produce high levels of noise, emanating from pressure pulsations, usually produced in the pump. Noise control can be achieved by either reducing the rate at which the pressure rises in the fluid passing through the pump, by allowing a small pressure leakage from the pump outlet back to the inlet, or by reducing the stiffness of the system into which the fluid is pumped. The design of these systems is highly complex.

Cavitation is another source of noise in hydraulic systems. If the pressure in the system at some point falls below the saturated vapour pressure, small bubbles of vapour form. When the pressure increases, these bubbles collapse very quickly, producing noise.

In hydraulic systems, noise is radiated from the pipework, the pump casing and the reservoir, the latter usually being the dominant source. Additionally, valves produce noise due to hydromechanical instability. This can be lessened by reducing the speed of valve opening, adding damping or using mechanical techniques to cancel out the oscillating forces.

## • Electric Motor Noise

On electric motors, the cooling fan is often the main source of noise, producing broadband noise and discrete frequencies. Straight bladed, bi-directional fans are the noisiest types. Fairly simple redesign, or use of curved blade unidirectional impellers, can effect reductions of 5 - 10 dB(A). Further noise reductions are possible with silencers on the inlet and outlet.

As the rotor rotates, magnetic noise produces many discrete frequencies. It is a function of the number of rotor and stator slots, the flux density of the magnetic field, the coil winding and the size of the air gap. Magnetic noise should be minimised at the design stage, but can sometimes be problematic in cases where insufficient care has been taken in the design.

## 3.2 Noise reduction by treatment remote from the source

Enclosures are of the partial or full type. Obviously, full enclosure of a troublesome noise source is the most effective, but problems with machinery overheating and maintenance access must be addressed.

Silencers can be used to reduce noise from a moving gas stream. They are either of the reactive type, which are designed for each specific installation, or the absorptive type, which are effective over a broad frequency range above 250 Hz.

### • Enclosures

For large enclosures (when the distance between the machinery surfaces and the enclosure walls is greater than one wavelength at the frequencies of interest), the performance of the enclosure can be straightforwardly estimated. There are some basic rules that must be followed to design an effective enclosure:-

- 1) the enclosure skin must be of sufficient weight to provide the necessary sound reduction index (as a first approximation, the sound reduction is linearly related to the surface density of the enclosure skin).
- 2) maximum absorption at all important frequencies must be provided inside the enclosure. If this is not done, the reverberant field inside the enclosure builds up significantly.
- 3) All unnecessary openings in the enclosure should be sealed, as their effect on the performance of the enclosure can be quite severe.
- 4) The enclosure should be mechanically isolated from the noisy machine so that it is not driven to vibrate. This isolation also includes all service pipework/ducting etc. to the machine.

The most effective way to provide a high sound reduction index for an enclosure, without resorting to extremely thick and heavy walls, is to use a double skin construction. Two single skin walls, isolated from each other and the surrounding supporting structure, give a sound reduction equal to or greater than the arithmetic sum of their individual sound

reductions. If the two skins are fully coupled together, their combined sound reduction is 5 - 6 dB(A) greater than the single skin. The usual construction, with an air gap between the two skins and common supporting structure, gives a sound reduction somewhere between these two extremes.

Partial enclosures, or hoods, can be used successfully for shielding machine operators from nearby noise sources. It is difficult to predict their performance due to diffraction of noise around the edges of the enclosure.

For close fitting enclosures, for which the air gap between the machine surfaces and the enclosure walls is less than one half wavelength, there is considerable coupling between the vibrating machine and the enclosure. The sound reduction is significantly reduced (allowances up to 10 dB are made).

Lagging with acoustic material can reduce sound radiated from pipes. An absorbent, resilient layer is wrapped around the pipe and covered with a heavy, outer skin. The absorbent layer isolates the outer skin from the pipe vibrations and damps the resonances introduced by fitting the outer skin. Suitable materials for the outer skin are asphalted paper, neoprene sheeting, lead loaded PVC and lightweight metal sheet.

### • Duct Linings and Silencers

The design of absorptive linings to attenuate the noise travelling along ducts is complex. Fortunately, good results can be achieved with designs that are far from perfect. A reasonable approximation for the attenuation due to an absorptive lining is given by:-

$$A = \alpha^{1.4} P / S \quad (3.1)$$

where;

A = attenuation in dB/m

P = perimeter of duct in m

S = cross sectional area of duct in m<sup>2</sup>

$\alpha$  = absorption coefficient of lining (frequency dependent) – see Section 3.3:

Absorptive silencers for fan/duct systems are a standard, off the shelf product supplied by a number of manufacturers. Lined ducts are also used in enclosures where openings are

required for ventilation or for product movement (e.g. a conveyor belt entering and leaving an enclosure, which is built around some noisy process such as crushing).

Reactive silencers, for example those used in vehicle exhaust systems, consist of expanded sections in a pipe or duct, along which noise propagates. The noise reduction is due to reflection and cancellation of sound waves. Their performance is frequency dependent and attenuation is a function of the ratio of the cross sectional areas of the expanded section and the pipe. Multi-chamber silencers are more effective than single chamber silencers. The design of reactive silencers is complex and a large body of practical knowledge has been built up over the years, principally by specialist manufacturers.

### 3.3 Materials for noise control

Materials are available from a host of manufacturers for the control of noise and vibration. Generally, each material performs the separate function of sound insulation, sound absorption or vibration damping. Care needs to be exercised when deploying these types of material in hazardous atmospheres such as coal mines, where restrictions on the use of non-metallic materials apply.

#### • Sound Insulation Materials

Sound insulation materials are used in the construction of walls, barriers and enclosures, where it is required to maintain a noise level difference across the material. The performance of a sound insulation material is described in terms of its **Sound Reduction Index (SRI)**, also known as **Transmission Loss**, which is a measure of the ratio of the incident sound power on the material to the transmitted sound power.

Analysis of an ideal panel of sound insulation material results in the **Mass Law**, which states that the sound reduction index is a function of the superficial mass of the panel,  $M$  ( $\text{kg}/\text{m}^2$ ), and the frequency of the incident sound,  $f$  (Hz):-

$$\text{SRI} = 20 \log_{10}(M.f) - 43 \quad \text{dB} \quad (3.2)$$

This is supported by common sense observation. The denser a panel, the better sound insulation it is likely to provide. Also, materials are much better at reducing the transmission of high frequency sound than low frequency sound.

Real walls and enclosures behave in rather more complex ways than predicted by the mass law, although this relationship provides a good approximation to the behaviour and is used in design calculations. The effects of stiffness at low frequencies and an effect known as "coincidence" at mid to high frequencies alter the sound reduction index and produce deviations from the mass law.

High density polymer curtains, sometimes faced with absorbent foam, can be used as sound insulation material in many applications where movability is important.

Sound reduction index data for extensive ranges of materials are available in many acoustics textbooks and are available from suppliers and manufacturers.

## • **Sound Absorption Materials**

The performance of a porous or fibrous material in absorbing sound is defined in terms of absorption coefficient,  $\alpha$ , defined by the ratio of absorbed sound power to incident sound power. As with sound reduction index, absorption varies with frequency and also the thickness of the material, density and fibre or pore size, the fundamental parameter being flow resistance.

Open cell polymer foam is a versatile acoustic absorbent material. The foam is non-fibrous and therefore no shedding or fraying occurs, even under the effect of air flows. If unprotected, the surface is open to the ingress of oil and other contaminants. Also, some untreated foams are flammable and generate smoke while burning and care should be taken in choosing suitable materials.

There is a wide range of commercially available glass fibre material available for noise control purposes. Materials for sealing the surface (to reduce shedding) are available.

Mineral wool, or rock fibre, is manufactured from various types of stone. Panels of such material display similar characteristics to glass fibre. However, mineral wool is classified as non-combustible.

In installations, the fibrous materials such as glass fibre and mineral wool are faced with a polymer material (<50 microns thick) to prevent shedding and to protect the absorbent against the ingress of contaminants. This membrane is then overlain with a perforated panel, such as perforated metal plate, for protection against physical damage, and to provide an effective earth should any static electrical charge build up on the membrane. Perforated plate open areas of 30% and above show no detrimental effects on the sound absorption performance of the material.

As with sound insulation materials, absorption data for extensive ranges of materials are available in many acoustics textbooks and are available from suppliers and manufacturers.

## • Damping Materials

Modern machine structures employ manufacturing techniques in which welding has replaced riveted or bolted joints. This has resulted in structures with much lower damping since the joints provided most of the damping in older structures.

Significant increases in damping of machine structures and casings can be obtained by adding proprietary damping materials. There are two ways to do this. With non-constrained damping, the damping material is applied to the structure as a single layer and it absorbs energy by contracting and extending longitudinally as the structure vibrates. With constrained (or "sandwich") layer damping, the damping material is covered by a layer of extensionally stiff material (usually metal sheet). As the structure vibrates, large shear stresses are set up in the constrained damping layer and energy is dissipated.

The two methods are equally effective for damping layer weights between 10% and 20% of the weight of the panel to be damped. Below 10%, the constrained layer method is likely to be more effective. The degree of damping is limited by thickness and weight restrictions and the application of damping layers is most effective when applied to large thin covers and plates which have relatively low stiffness.

## 4. Hand held rock drills

Percussion drills are the most serious noise problem in the mining industry because of their extremely high noise levels and widespread use. Measurements in South Africa on unsilenced pneumatic percussion drills have shown levels of the order of 117 dB(A) at the operator's ear, with corresponding measurements of noise exposure levels in the range 110 - 114 dB(A). Based on an allowable noise exposure level of 85 dB(A) for an 8 hour shift, the exposure time for a level of 117 dB(A) is 18 seconds.

Measurements on pneumatic percussion drills, fitted with commercially available exhaust silencers, have shown levels of the order of 109 dB(A) at the operator's ear, with corresponding measurements of noise exposure in the range 102 - 108 dB(A).

Measurements on water-hydraulic percussion drills have shown levels of the order of 108 dB(A) at the operator's ear, with corresponding measurements of noise exposure levels in the range 100 - 106 dB(A). Based on an allowable noise exposure level of 85 dB(A) for an 8 hour shift, the exposure time for a level of 106 dB(A) is 3,8 minutes.

Studies have shown that drilling noise is the sum of three major noise sources - exhaust noise, drill steel noise and drill body noise. Exhaust noise is the most severe and is produced by the high velocity cyclic release of air to the atmosphere. Mixing of the high velocity air with the relatively still atmosphere gives rise to broadband random noise and the cyclic nature gives it the characteristic staccato sound. Exhaust noise has been measured at 110 - 120 dB(A) at the operator's ear. Drill steel noise results from the transverse and longitudinal vibrations excited by the piston impacts on the shank and is of the order of 105 - 110 dB(A) at the operator's ear. Drill body noise is produced by the moving and impacting parts inside the drill exciting the drill body which radiates noise. This is the least important of the three sources and only becomes significant when the exhaust noise and the drill steel noise have been reduced significantly.

A large amount of work has been done, particularly in the U.S. and the U.K., to develop noise reduction treatments for coal and hard rock pneumatic drills and numerous reports have been published<sup>(1) - (10)</sup>. Initially, work concentrated on reducing exhaust noise (and, as a consequence, drill body noise). Once the exhaust noise had been reduced significantly, the drill steel noise became dominant and efforts were concentrated on reducing this



source. Recently new "low noise" designs for coal stoper drills and hard rock drills have been developed, with variable success.

## 4.1 Retrofit treatments

Numerous muffler designs have been developed by equipment manufacturers, mining companies and government researchers, and mufflers are available from manufacturers for fitment to their equipment. The most effective muffler is the wraparound jacket type, which can reduce drill body noise as well as exhaust noise.

Summers and Murphy<sup>(1)</sup> investigated the effectiveness of several designs of muffler and also the application of a damping "collar" on the drill steel. The collar consisted of a 15 cm long steel tube bonded onto the drill steel, near the drill end, with a viscoelastic, cured urethane product, which dissipated vibrational energy in the drill steel by acting as a constrained layer damping system. The damping collar is shown in Figure 4.1 (taken from reference (1)). The application of the wraparound jacket muffler and the drill steel damping collar reduced the noise level of the test drill from 115 dB(A) to 102 dB(A).

This work was extended by Visnapuu and Jensen<sup>(2)</sup> who developed a new case/muffler and applied the drill steel constrained layer principle to the full length of the drill steel. The noise level of the test drill was reduced from 115 dB(A) to 97 dB(A). However, performance tests showed a measurable loss in penetration rate due to these two noise control treatments.

This work was considered by Hawkes and Wright<sup>(3),(4)</sup> who confirmed that significant reductions in noise level were possible by the application of constrained layer damping to the drill steel. However, they also showed that reductions in drilling rate were evident, both with the full length steel sheathing and, to a lesser extent, with the collaring. They also noted that the constrained layer systems were subject to abrasion and possible consequent failure of the viscoelastic bond. They recommended an independent "shroud tube" which covered the drill steel, and entered the hole with it, but was not attached to it.

Related retrofit work was carried out in the U.K. by British Coal<sup>(5),(6)</sup>. Trials were conducted with various exhaust silencing arrangements, including ducted exhaust, on a range of pneumatic rock drills. Ducting the exhaust away was effective in reducing exhaust noise, but the inconveniences of this method outweighed the benefits for hand-

held equipment. Additional noise reductions were achieved with the application of a telescopic cover to the drill steel, which collapsed as the drill steel entered the hole. As with the shrouded drill steel trials in the U.S.A., doubts were expressed regarding the handlability of this drill steel covering method.

They confirmed that the bulk of drill steel noise was due to the bending waves in the drill steel, not the longitudinal waves which provide the useful work in breaking the rock. They stated that the energy in the bending waves was much lower than in the longitudinal waves, as would be expected from axial impacts, but that, in worn or poorly designed machines, the bending wave energy could approach 20% of the original piston impact energy. They put forward several proposals for reducing the bending waves in the drill steel, and hence reducing the radiated noise:-

- **chuck / drill steel shank tolerances**

Misalignment of the drill steel with the axis of impact is one of the major sources of bending waves. Chuck/shank clearances of 1.5 mm and greater can lead to misalignments of up to 1° with an 83 mm chuck. Tightening the tolerances will increase manufacturing costs but is a necessary part of reducing drill noise.

- **increase drill shank length**

The length of the drill steel shank is not critical and limitations are ultimately imposed by machine length restrictions and manufacturing considerations. Together with tightened tolerances, increased shank length will significantly decrease misalignment.

- **ensure chuck/bore alignment**

In many drills, the chuck is a loose fit in the chuck housing and there is considerable play in the chuck drive assembly, both of which contribute to misalignment. Redesign will eliminate this unwanted radial and axial movement and maintain axial alignment of the drill steel with the axis of impact.

- **optimising the thrust**

When the drill steel is out of contact with the rock the constraints on drill steel vibration are reduced. Thus, optimising the thrust is necessary not only for good drilling rates but also for reduced noise levels. For example, it is more difficult to operate rifle bar drills at optimum thrust than drills with independent rotation. With the latter, the thrust is not critical for smooth operation and the thrust can be optimised and drill steel bounce minimised.

## **4.2 Coal stoper drill redesign**

The coal stoper drill retrofit treatments developed in the U.S. were only partially successful and the U.S.B.M. sponsored a redesign project to obtain greater noise reductions and improved drilling performance<sup>(7)</sup>. Work was concentrated in four main areas:-

- (1) redesign of the drill steel rotation mechanism and other drill parts
- (2) development of a compact, effective muffler/enclosure
- (3) development of a drill steel shroud tube
- (4) redesign of drilling controls

The redesigned coal stoper was tested in several operating coal mines.

- **Redesign of the drill steel rotation mechanism and other drill parts**

Standard stoper drills achieve drill steel rotation through a rifle bar arrangement, which means that, for any particular drill, the rotation torque is constant. The rotational speed is therefore dependent on the thrust provided by the drill feed leg. The redesigned stoper drill used an independent drill steel rotation system ( a separate air motor and gear arrangement) which improved drill performance because the rotational speed was not dependent on thrust. The piston was always able to travel through its full stroke, thus increasing drilling power, and rotational speed could be changed to suit different rock conditions without affecting the piston blow frequency. Also, the many internal impact points of the rifle bar system were eliminated, thus reducing the high frequency rattling noise of standard stoper drills. Because the fluted hole in the centre of the piston

(required for the rifle bar arrangement) was no longer necessary, the piston diameter was reduced, resulting in a smaller overall drill body diameter and facilitating the addition of the muffler/enclosure.

Because the piston was no longer responsible for rotation, it was redesigned to serve as the valve controlling the flow of compressed air within the cylinder; this "valveless" method was more efficient and problem free than a valve system and the elimination of the "flapper" and "kicker-port" valves negated another source of high frequency noise.

The annular clearance between the chuck and shank was reduced and the upset shoulder on the standard drill steel was eliminated. These design changes reduced the misalignment and rattling impacts occurring at the top of the drill body and reduced the severity of transverse waves in the drill steel produced by off-centre impacts.

- **Development of a compact, effective muffler/enclosure**

The new design necessitated the development of a special muffler/enclosure. The inner part of the muffler/enclosure consisted of a series of ring-shaped, perforated metal baffle plates around the drill body. The outer shell consisted of two layers - an inner of aluminium and an outer of EAR Isodamp 1002 polymer. During drilling the exhaust air from the cylinder and rotation motor moved up through the perforated baffles and left the muffler/enclosure near the top of the drill.

The muffler/enclosure attenuated both drill body noise and air exhaust noise. The baffle plates vibrated to prevent ice build-up on the inner surfaces of the muffler/enclosure and a flexible deflector plate near the exhaust port of the piston chamber directed the air towards the top of the drill which also helped to reduce icing.

- **Development of a drill steel shroud tube**

As recommended by Hawkes and Wright<sup>(3),(4)</sup>, an independent shroud tube was used to reduce drill steel noise. The shroud tube was a simple steel tube designed to fit a 25 mm hexagonal drill steel and 35 mm to 44.5 mm drill bits. Its outer diameter was small enough to allow it to follow the drill bit into the hole, and its inner diameter was large enough to keep it from touching the drill steel. The tube was connected to the drill body through a rubber sleeve to provide isolation from drill body vibration.

- **Redesign of drilling controls**

All controls were mounted on the feedleg, enabling the operator to stand further away from the drill than on a conventional machine. Hammer, thrust and rotation controls were located together for easy operation. The throttle was equipped with a special "collaring" position, a "full on" position and a special "drill retract" position.

- **Underground tests**

Six prototypes were manufactured and four of these were tested in operating underground coal mines. The average noise level of these drills was 102 dB(A), an improvement of approximately 15 dB(A) on standard drills. Also, the redesigned drills were lighter and their penetration rates were greater than the drills they replaced. There were no freezing problems, and minimal wear was noted when the drill parts were examined at the conclusion of the field tests.

The one disadvantage with the redesigned stoper drill was that the shroud tube required removal and replacement during drill steel changing. Since this was time consuming, operators often drilled without the shroud tube. However, noise levels without the shroud tube were approximately 107 dB(A), substantially lower than standard stoper drills or stoper drills with retrofit mufflers.

### **4.3 Hard rock drill redesign**

Following the success of the redesigned stoper drill, U.S.B.M. sponsored a programme to redesign a hand-held drill suitable for use in hard rock mines<sup>(8),(9)</sup>. The basic design features were the same as for the coal stoper drill - independent rotation, valveless operation, muffler/enclosure, drill steel shroud tube and redesigned controls. However, the size, shape and stroke length of the piston had to be changed substantially to achieve the higher blow energy required in the hard rock drill.

There were other differences between the hard rock drill and the previous design. Firstly, the outer cover of the hard rock drill was made of cast aluminium rather than the aluminium-EAR composite. This reduced its weight and made it easier to fabricate.

Secondly, the ring shaped baffle plates were dispensed with because the flexible exhaust deflector alone was found to be sufficient to inhibit ice build-up. Thirdly, the drill cylinder was mounted within the outer cover through rubber pads, which isolated the cover from the drill cylinder vibration.

A production ready prototype of the quiet hard rock (QHR) drill produced noise levels of 104 dB(A) with the drill steel shroud tube and 107 dB(A) without the drill steel shroud tube. A production model of this drill was eventually developed, shown in Figure 4.2 (taken from reference (9)).

#### **4.4 Development of concentric drill steels**

As mentioned earlier, although the use of constrained layer damping on drill steels gave significant reductions in noise level, there were problems with this technique. It reduced the drilling rate, the outer sheath was subject to abrasion and the viscoelastic bond could fail, allowing the outer sheath to move axially on the drill steel. The alternative technique of using an independent shroud tube, as in the redesigned drills discussed above, also could be problematic. The drill operator could not observe drill steel rotation, the shroud tube interfered with chip extraction (thereby increasing the minimum hole size required) and drill steel changing was more difficult due to the presence of the shroud tube and retainer.

The U.S.B.M. sponsored a program to develop concentric drill steel designs for both machine mounted drifters and for hand held drills, which would give significant reductions in drill steel noise, maintain drilling rates, be durable and cause no difficulties for the drill operator<sup>(10)</sup>.

Concentric design drill steels use an inner rod to transfer the percussion pulse to the bit and an outer tube for transmission of rotation torque. The outer tube also acts as a cover over the inner rod, containing noise generated by bending waves in the inner rod. The reduction of drill steel noise is determined by the degree of enclosure of the inner rod and transmitted vibrations exciting the outer tube. The flushing medium is transported to the bit in the annular space between the inner rod and the outer tube.

- **Drifter concentric drill steel design**

The final design for the drifter concentric drill steel is shown in Figure 4.3 (taken from reference (10)). The complete system consisted of six basic elements:-

1. inner tappet which receives hammer blow and transmits the pulse to the drill steel inner rod
2. shank drive member which fits into existing drifter chuck and transmits torque
3. concentric drill steel inner rod which receives the pulse from the tappet and transmits the pulse to the bit
4. concentric drill steel outer tube which is threaded at both ends for transmission of torque and containment of inner rod
5. bit adapter which is threaded to the end of the outer member of the concentric drill steel with splines to transmit torque to bit
6. bit which fits into bit adapter with spline coupling for rotation and receives pulse from inner rod and transmits the pulse to the rock surface for chipping

The system was compatible with existing drifters although the hammer required modification since no water or air needle was required and the impact face did not have a hole in it. The flushing medium was fed into the concentric drill steel with a swivel built into the chuck end-cap. The entire system, ready for assembly, is shown in Figure 4.4 (from reference (10)).

- **Hand held concentric drill steel design**

The design for the hand held concentric drill steel was rather different from the drifter version. The drill steel was an integral design with a retaining collar. To drill a deeper hole, a longer complete drill steel must be inserted into the drill chuck.

The flushing water flowed from the drill water needle into the concentric drill steel in a similar manner to conventional integral drill steels. This required a hole in the end of the inner rod which terminated at a depth coinciding with the start of the outer tube. Three holes were drilled radially into the end hole to allow water to pass into the annular space between the inner rod and outer tube. This is shown in Figure 4.5 (taken from reference (10)).

At the bit end, a taper bit could be attached directly to the inner rod end. Provision was made to drive the bit with the outer tube by using an interlocking finger arrangement attached to the outer tube and machined into the bit skirt. As at the shank end, the flushing water passed through the annular space to the bit end where three radial holes connected to an axial hole drilled from the taper end, allowing water to flow through the bit. This is shown in Figure 4.6 (reference (10)).

• **Field testing of drifter concentric drill steels**

The drifter concentric drill steel was tested at four sites. At the first site there were a number of problems with equipment failure and the noise level measurements were inconclusive although they did show some reduction by using the concentric drill steel. Following these tests, a new hammer was fabricated using solid bar stock and the bit adapter was modified to use a solid construction rather than the initial welded assembly.

After these modifications to the drifter concentric drill steel, tests were carried out at a second site, where significant reductions in drill steel noise were measured. The results were;-

	At Operator	15cm From Drill Steel
Concentric Drill Steel	107 - 109 dB(A)	107 dB(A)
Conventional Drill Steel	110 - 112 dB(A)	120 dB(A)

The drilled holes were approximately 9 m deep, requiring the use of three drill steel sections. The drilling performance was improved by using the concentric drill steel. Drilling with the first section was at an average rate of 60 cm/minute with both types of steel. The second section drilled at an average of 49 cm/minute for the conventional drill steel against an average of 60 cm/minute for the concentric drill steel. The third section drilling rate was 43 cm/minute for the conventional drill steel against 60 cm/minute for the concentric drill steel. Endurance tests at this site highlighted problems with the uncoupling forces required and the service life of the threads.

At the third site, employing a different type of drill, the concentric drill steel failed before the noise measurements were completed. Due to the presence of high exhaust noise, minor noise level differences were noted. With both types of drill steel, the penetration rate was approximately 152 cm/minute in this softer strata.



At the final site, the drilling rate was approximately 60 cm/minute for the concentric drill steel and approximately 55 cm/minute for the conventional drill steel. As before, the concentric drill steel gave the same 10 - 12 dB(A) reduction in noise level at a position close to the drill steel.

- **Field testing of hand held concentric drill steels**

The concentric drill steel for hand held drills was tested using the QHR (quiet hard rock) drill discussed above. Measured noise levels at the operator showed approximately 107 dB(A) with the conventional drill steel and 104 dB(A) with the concentric drill steel. The concentric drill steel reduced the mid frequency noise significantly but increased the noise at frequencies of 8000 Hz and above, a phenomenon which was also observed with the drifter concentric drill steel. The drilling rates with the conventional and concentric drill steel were comparable; this was as expected since the hammer impacts directly on the drill steel in both cases.

- **Conclusions from concentric drill steel work**

After the completion of the testing of the concentric drill steels, it was concluded that the concept showed promise as a viable alternative to conventional drill steels, although it was recognised that commercial application of concentric drill steels would require substantial development. A commercial company is currently developing this work, although it is understood that the latest developments are the subject of a patent dispute.

## **4.5 Latest developments**

A novel concept for a quiet pneumatic percussion drill is being developed by Harper and Radzilani at CSIR Mining Technology Division<sup>(11)</sup>. They recognised the fact that, whilst effective exhaust silencing and constrained layer damping of the drill steel provided significant reductions in noise level, it resulted in reductions in drill performance. Additionally, the frequent contact between the drill steel and wall of the drilled hole, when using standard thrustleg drills, resulted in the destruction of the constrained layer and failure of the drill steel treatment. They inferred that, if the reduced drill performance could be recovered by an alternative method and drilling could be carried out without contact

between the drill steel and the walls of the hole, both these treatments could be incorporated in a new design. Both of these objectives could be achieved if the thrust was directed along the drill axis.

Their design is shown in Figure 4.7 (taken from reference (11)). A modified rockdrill was used as the piston in what was effectively an enlarged thrust leg. The thrust was axial to the drill steel, requiring approximately 80 kPa to move the drill along the tube. Also, the drill body and drill steel were enclosed at all times, resulting in significant reductions in noise levels.

Initial testing with the pre-prototype were very encouraging, with noise levels below 95 dB(A), and penetration rates slightly above that of a conventional drilling machine. A number of design problems were identified during these initial tests which were resolved before going on to prototype testing. At present, a number of machines are being manufactured for underground evaluation trials.

## **4.6 New equipment**

Apart from the design of Harper and Radzilani, which has promise, but which is in the early stages of development, no totally effective noise control treatment has been developed for hand held rock drills. At present the most effective available method for reducing noise is the fitment of mufflers which reduce exhaust and drill body noise. It is recognised however, that slight reductions in drill performance occur when mufflers are fitted. Most manufacturers produce effective mufflers for their equipment.

Where possible, it is always better to specify hydraulic drills, which produce significantly less noise than unsilenced pneumatic drills due to the absence of exhaust noise. The main source of noise is the drill steel noise.

The QHR drill developed in the U.S.A. was not as successful in the field as had been hoped. Drilling performance was down on conventional drills. The company which produces the drill are putting their effort into the development of a water powered drill. However, manufacturers should be aware of the underlying concepts behind the development of "quiet" rock drills.

Efforts should be made to tighten chuck/drill steel tolerances, increase shank length where possible, reduce unwanted chuck movement and employ independent rotation of the drill steel. These will all contribute to the reduction of bending waves in the drill steel, and hence the reduction of radiated noise.

As mentioned earlier, the concentric drill steel concept showed promise, particularly for the larger boom mounted drills. It is possible that further developments will result in production drill steels which employ this design, although it is understood that nothing is available from manufacturers at present.

## 4.7 Retrofit summary

The following are simple treatments which are either well proven and readily available (mufflers) or which have been shown to be reasonably effective but have not been widely employed in the field (damped drill steels). It is recommended that a field trial be carried out where their performance and durability are assessed. As part of the field trial, the importance of good drill maintenance, in order to maintain tight chuck/drill steel tolerances, would be quantified.

- Fitment of wraparound muffler 7 - 10 dB(A) reduction  $\pm$ R1000
- Constrained layer damped drill steel (collar) with wraparound muffler 10 - 12 dB(A) reduction  $\pm$ R1100
- Constrained layer damped drill steel (full length) with wraparound muffler 10 - 15 dB(A) reduction  $\pm$ R1200
- Shrouded drill steel with wraparound muffler 10 - 15 dB(A)reduction  $\pm$ R1200

## 4.8 References

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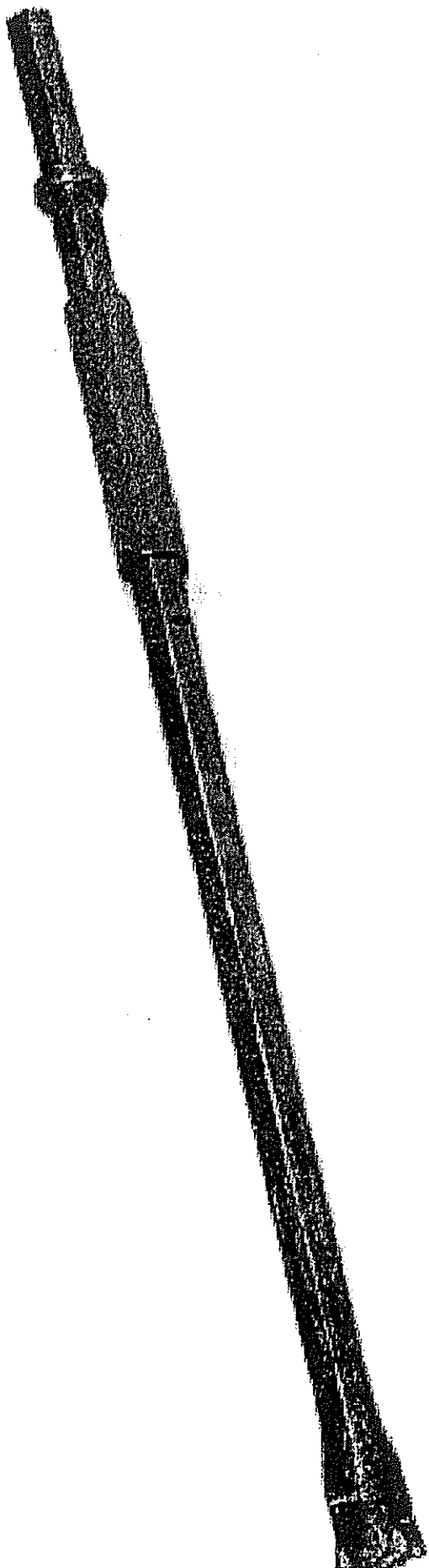


FIGURE 4.1: DRILL STEEL WITH DAMPING COLLAR

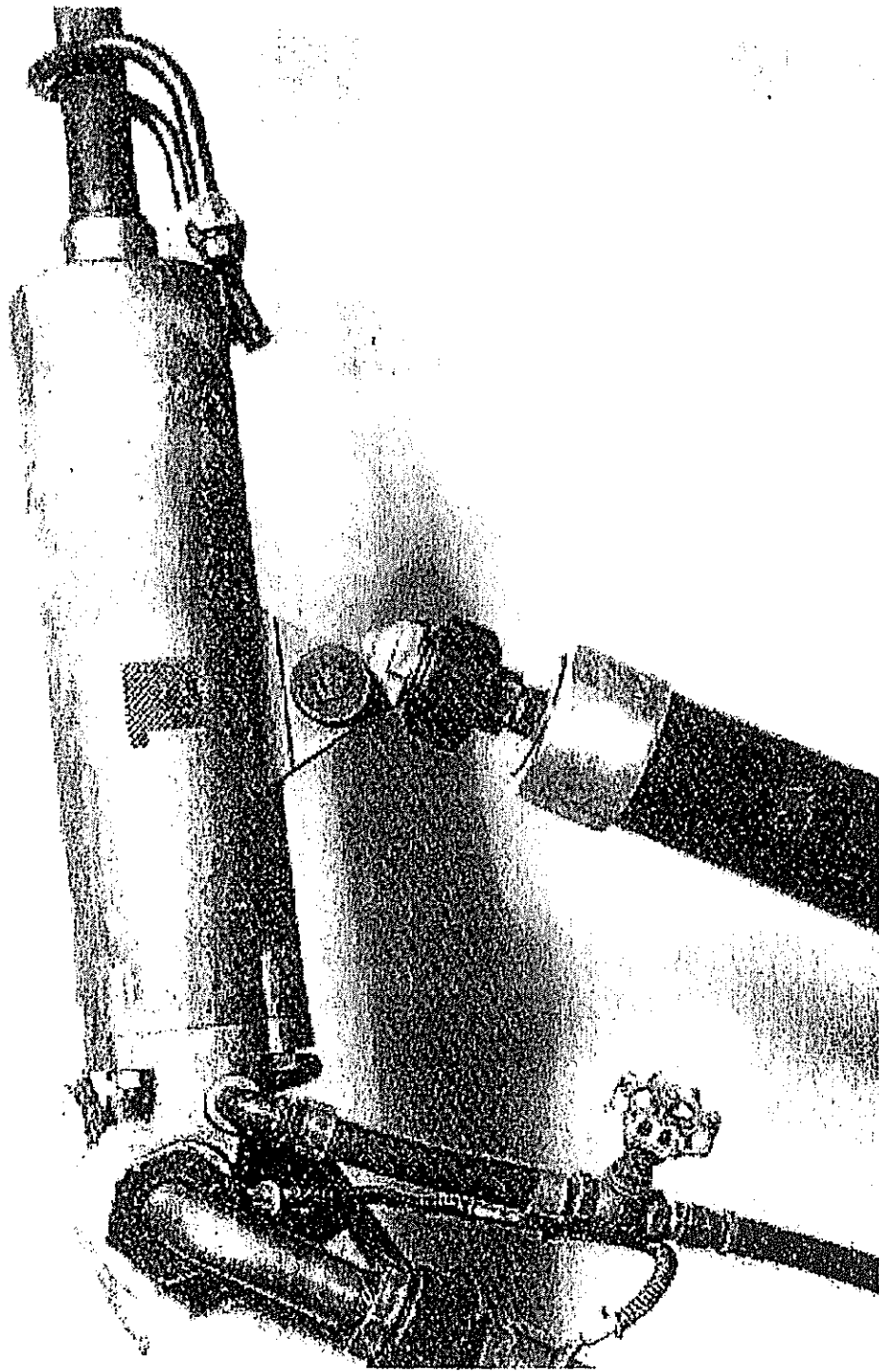


FIGURE 4.2: QUIET HARD ROCK DRILL

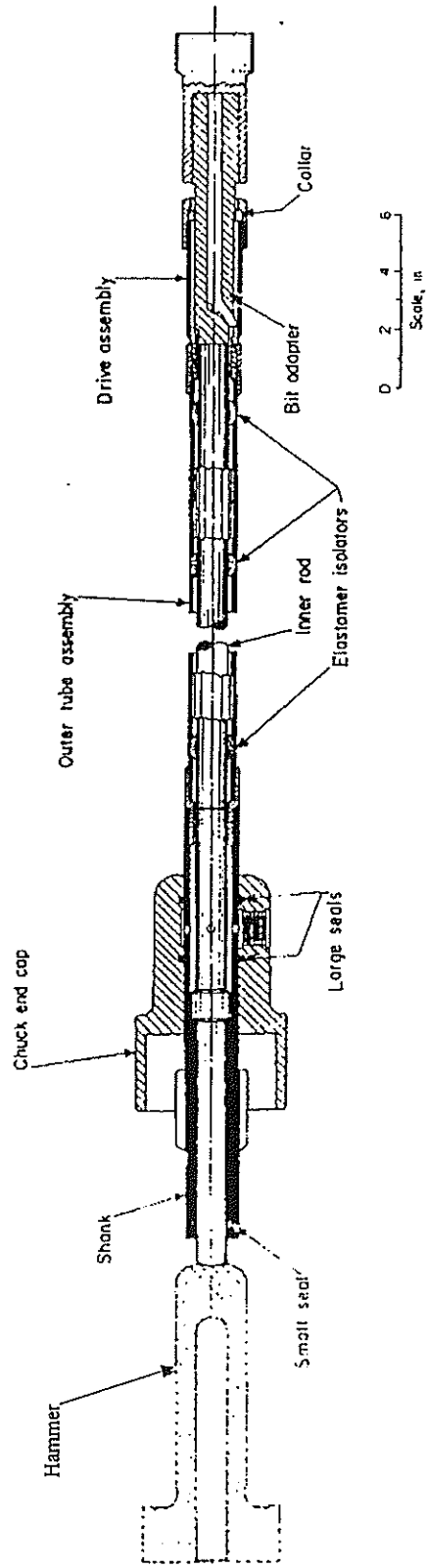


FIGURE 4.3 : CONCENTRIC DRILL STEEL DESIGN

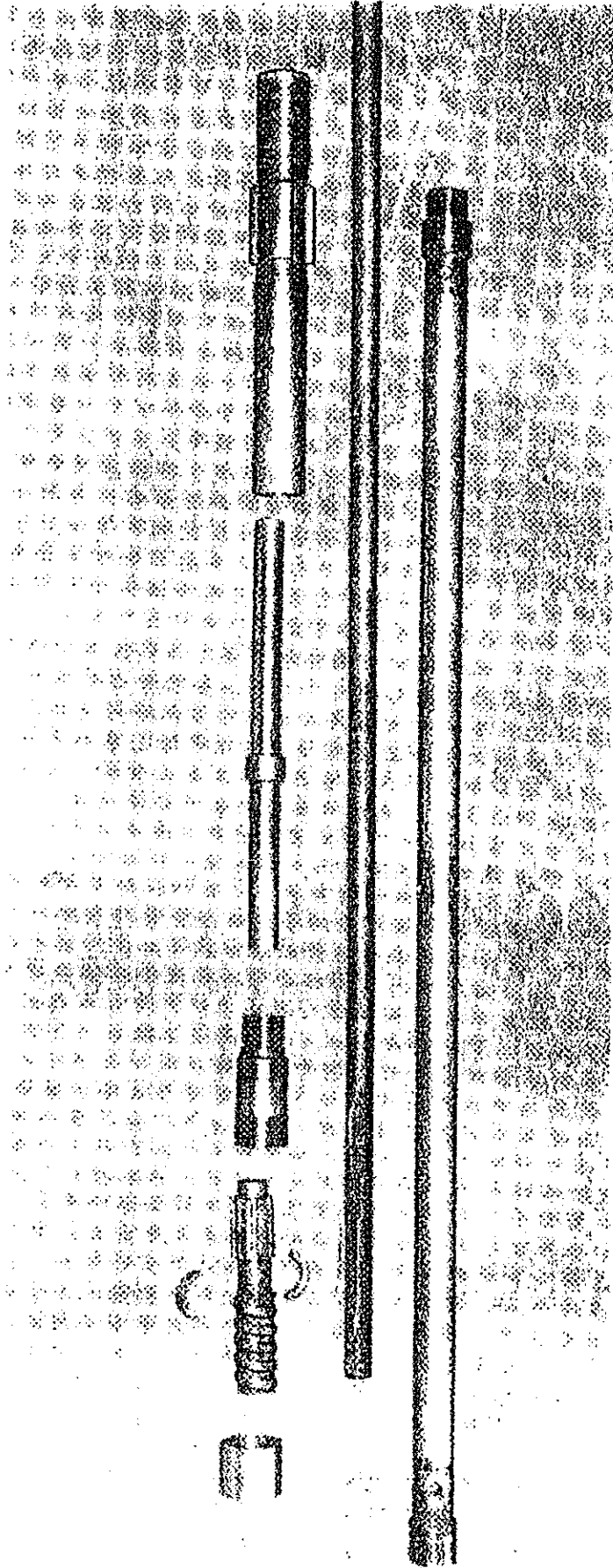


FIGURE 4.4: DRIFTER CONCENTRIC DRILL STEEL



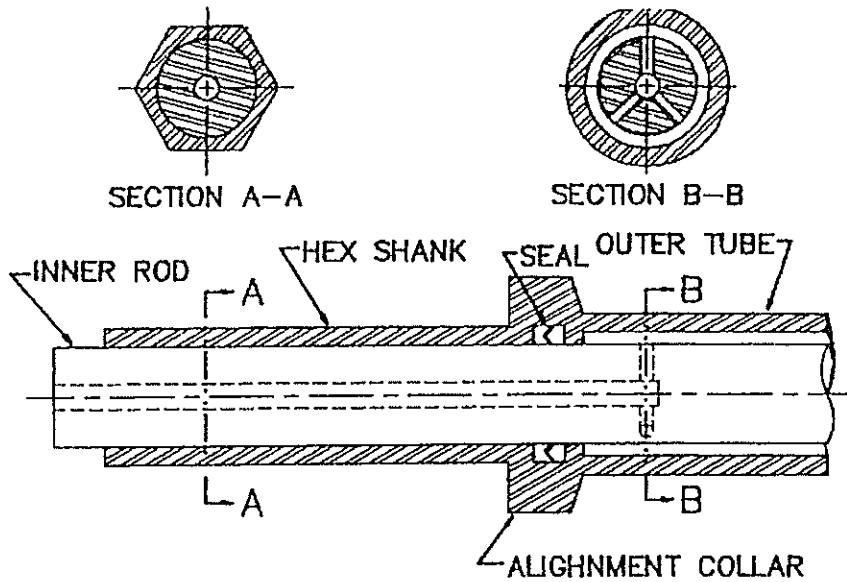


FIGURE 4.5: HAND HELD CONCENTRIC DRILL STEEL/SHANK END

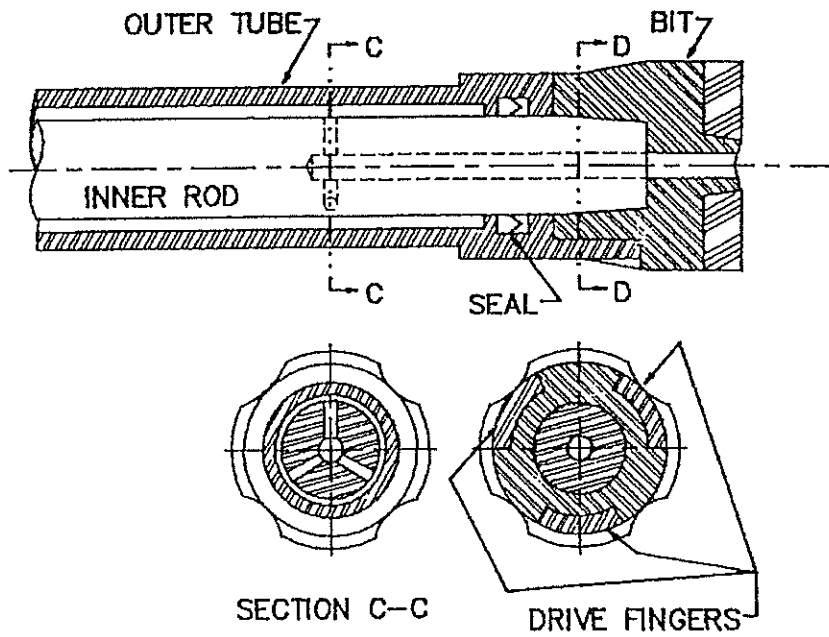


FIGURE 4.6: HAND HELD CONCENTRIC DRILL STEEL/BIT CONNECTION

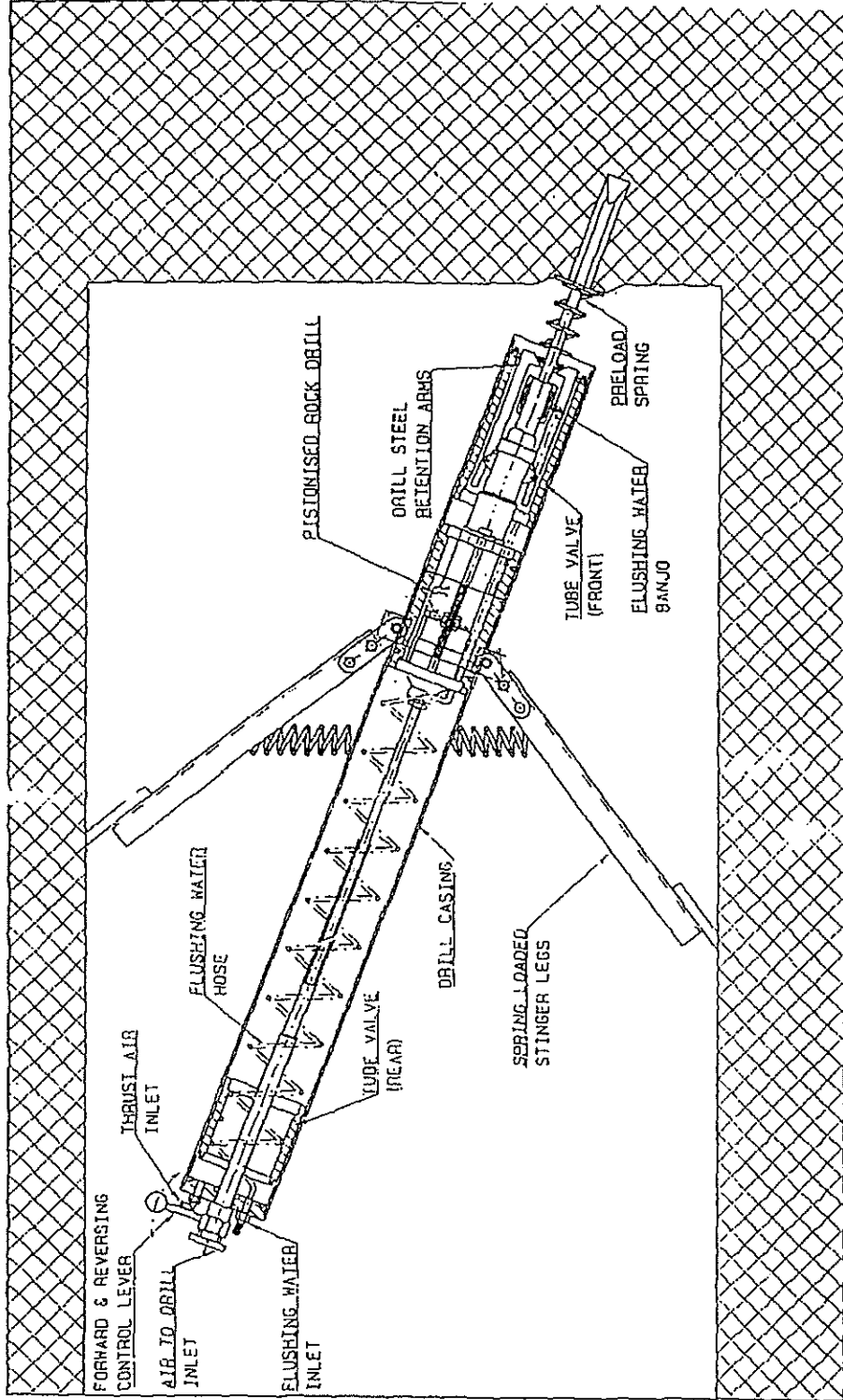


FIGURE 4.7: NOVEL DRILL DEVELOPMENT

## 5. Boom mounted rock drills

As with hand held drills, the three major sources of noise on pneumatic, boom mounted drill rigs are air exhaust, drill steel and drill body. Exhaust noise reduction techniques have included piping the exhaust away from the operator, attaching a muffler to the exhaust port and placing the entire drifter inside an acoustic enclosure. The last technique has been the most successful since it reduces drill body noise as well as exhaust noise, and is the least susceptible to freezing. Drill steel damping collars and shroud tubes have been used on boom mounted drills to suppress drill steel noise. This technique is effective since collars are not physically coupled to the drill steel. Typical noise levels for unmuffled pneumatic boom mounted drills are above 115 dB(A) at the operator's position.

In terms of noise control, hydraulic drills have an advantage over pneumatic drills because exhaust noise is non-existent. However, the noise from the drill steel and the drill body are still present, and these can combine to produce noise levels above 110 dB(A) at the operator's position.

In parallel with the work on hand held pneumatic drills, retrofit and redesign work has been carried out for pneumatic boom mounted drills, and several reports have been published<sup>(1),(2),(3)</sup>.

### 5.1 Retrofit treatments

A workable retrofit package was developed by Dixon et al under a U.S.B.M. contract<sup>(1)</sup>. They developed an acoustic enclosure for the drifter which is shown in Figure 5.1 (taken from reference (1)). The top half of the enclosure hinged open from the bottom half, allowing easy access to the drill.

A schematic of the enclosure is shown in Figure 5.2 (from reference (1)). The exhaust air exits the drill radially, strikes a silicone rubber deflector at the top of the enclosure, and moves forward to escape through the front opening. The deflector is very flexible and shakes off any ice that begins to form on it. After passing the deflector, the exhaust air enters the fibreglass lined muffler section at the front of the enclosure. The fibreglass is held in place by perforated metal plate and a thin surface layer protected it from oil and

water contamination. The exhaust air then leaves the enclosure through the front opening.

A drill steel shroud tube was also developed for the retrofit treatment and is shown in Figure 5.3 (from reference (1)). The outer diameter of the shroud tube was slightly smaller than the bit diameter, allowing the tube to enter the hole. The inner polymer layer rode loosely on the drill steel. The foam interlayer absorbed some of the vibration imparted to the polymer. Exhaust air from the muffler enclosure travelled forward through the annulus between the steel and the shroud tube, escaping just behind the bit.

Performance of the drill, with and without the retrofit noise control treatments, was evaluated above ground and underground in an operating mine. Results of the underground tests are shown in Table 5.1 (from reference (1)). As can be seen, noise levels at the operator were 12.5 to 15 dB(A) lower than with the untreated drill. The durability of the noise control treatments was evaluated by drilling over 3000 metres of hole in the underground tests, approximately 150 metres of this with the shroud tube. Very little damage to the enclosure was noted. Acceptance of the muffler enclosure by mine personnel was very good, with no significant interference with routine drill maintenance and reported penetration rate at least as good as the unmodified drills in use at the mine.

Over the same period of time, The National Coal Board in the U.K. carried out extensive tests on a range of boom mounted rotary and rotary-percussive drills<sup>(2)</sup>. Hydraulic rotary, hydraulic rotary-hydraulic percussion and hydraulic rotary-pneumatic percussion types were tested in a limestone mine.

The results for the hydraulic rotary drills are given in Table 5.2. For the hydraulic rotary drills, the drills operating at low feed thrust tended to generate the lowest noise levels. The exception to this was drill D, which had a mid range drill thrust value of 1290 kgf, but the lowest operator's noise level of 86 dB(A). No correlation on the basis of drill penetration rate seemed apparent.

The results for the hydraulic rotary-hydraulic percussion drills are given in Table 5.3. Drills H, I and P were tested with both "crossbit" and "3 winged" bit types. In each case, the "3 winged" bit type showed higher penetration rates with lower noise levels at the operator's position. It was concluded that, if the noise radiated from rotary percussive drills was to be minimised, care should be taken in selecting bit type and machine operating pressures to

suit the strata conditions. Effective use of the drills can show a 3 - 5 dB(A) reduction for the same machine, coupled with a reduced drilling time because of the increased penetration rate.

The results for the hydraulic rotary-pneumatic percussion drills are given in Table 5.4. Both of the drills were tested with and without exhaust mufflers. The pneumatic drills were noisier than their hydraulic equivalents, having mean levels at the operator's position of 101 and 104 dB(A). When the drills were fitted with mufflers, the levels fell to 97 and 98 dB(A), in line with the levels for the hydraulic drills. The results confirmed that the noise level at the operator's position was dependent on the pneumatic operating pressure, a change from 0,55 MPa to 0,82 MPa resulting in a change in noise level from 104 dB(A) to 106 dB(A).

## 5.2 Drill redesign

Under a further U.S.B.M. contract, Dutta and Runstadler developed a redesign of a boom mounted drill with independent rotation<sup>(3)</sup>. A view of the drill is shown in Figure 5.4 (taken from reference (3)). The rotation motor was removed from the drifter and located at the front of the feed channel. This design change required the use of a specialised drill steel called a "kelly bar". The drifter supplied percussion to the rear end of the kelly bar and the new rotation mechanism supplied torque to the front.

The modified drifter was placed inside a two piece muffler/enclosure, which could be opened for easy access to the drill. A drill steel shroud tube was used which was of a different design from that used on the retrofitted drill. As shown in Figure 5.4, the shroud tube was a collapsible steel coil of approximately 20 cm diameter which did not enter the hole during drilling. It was suspended between the front of the drifter enclosure and the rear face of the kelly bar rotation mechanism. The shroud was completely extended at the start of drilling and collapsed as the drifter moved towards the rock face. The exhaust air from the drifter moved forward through the shroud tube and a rubber stinger that was pressed against the rock face. The stinger is shown in Figure 5.5 (from reference (3)).

The measured noise level underground was in the region of 100 dB(A), significantly down on the unsilenced levels of 110 to 115 dB(A). Some problems with the inability to see the rotation of the drill steel were noted.

### 5.3 New equipment and recommendations

As with hand held rock drills, muffler/enclosures are available for boom mounted drills from most manufacturers and should be specified when ordering new equipment. Again, hydraulic equipment is intrinsically quieter than pneumatic equipment, because of the absence of exhaust noise, although the potential noise reduction is limited because of the continuing presence of drill steel noise.

Acoustic cabs are available from some manufacturers and should be considered as an option where they are acceptable in an underground situation.

The development of drill shroud tubes (including collapsible shroud tubes) should be pursued.

The concentric drill steel concept, discussed in section 4.4, showed real promise as an effective and robust means of reducing drill steel noise without a detrimental effect on drilling performance, particularly for boom mounted drills. It is possible that production models may be available in the future.

### 5.4 Retrofit Summary

- |   |                         |         |
|---|-------------------------|---------|
| • Drifter muffler                               | 4 - 8 dB(A) reduction   | ±R3000  |
| • Drifter muffler with drill steel shroud       | 8 - 12 dB(A) reduction  | ±R3500  |
| • Drifter muffler with constrained layer collar | 8 - 10 dB(A) reduction  | ±R3500  |
| • Addition of acoustic cab                      | 30 - 40 dB(A) reduction | ±R30000 |

## 5.5 References

- (1) Dixon, N.R. et al, 1982. Development of a Prototype Retrofit Noise Control Treatment for Jumbo Drills, BuMines Contract H0387006, Final Report, July
- (2) National Coal Board. 1984. Noise Exposure and Control in Mine Headings and Roadways, and in Passenger Transport Systems, ECSC Final Report 7255-17/040/08, MRDE, U.K.
- (3) Dutta, P.K. and Runstadler, P.R. 1985. Development of Prototype Quiet Jumbo Drills, BuMines Contract H0395025, Final Report

TABLE 5.1: UNDERGROUND TESTS OF RETROFIT BOOM MOUNTED DRILL

DRILL POSITION	UNTREATED DRILL dB(A)	TREATED DRILL dB(A)	NOISE REDUCTION dB(A)
COLLARING HOLE (3M OF STEEL)	117.5	105	12.5
MIDDLE OF HOLE (1.8M OF STEEL)	116	101	15
END OF HOLE (0.6M OF STEEL)	115	101.5	14



TABLE 5.2: TEST RESULTS FOR ROTARY HYDRAULIC DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
					OPERATOR	1 m FROM DRILL
A	380	1270	2.10	2-WING RADIUS	90	99
B	650	1230	2.05		91	101
C	350	680	2.00		90	99
D*	400	1290	1.35		86	90
E	400	1480	1.85		92	100
F	620	760	1.60		90	100
G	450	2545	4.05		92	103
H*	450	2680	2.77		93	-
I*	350	204	1.20		88	94

\* ROTARY PERCUSSIVE DRILLS OPERATED IN ROTARY MODE

TABLE 5.3: TEST RESULTS FOR HYDRAULIC ROTARY-HYDRAULIC PERCUSSION DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	BLOWS (/minute)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
						OPERATOR	1 m FROM DRILL
J	350	1430	2800	2.00	cross bit	97	104
K	-	-	-	2.05	cross bit	95	105
L	-	-	-	1.46	cross bit	101	108
M	200	1360	3400	1.70	cross bit	97	105
N	-	-	-	-	cross bit	99	104
O	170	1360	3600	1.65	cross bit	99	108
D	400	1290	5300	3.65	2 wing radius	92	99
I	350	820	3100	1.30	cross bit	100	106
I	350	820	3100	1.50	3 wing radius	99	106
P	450	1200	3600	1.52	cross bit	97	106
P	450	1200	3600	3.10	3 wing radius	93	105
H	460	1725	3000	0.96	cross bit	98	107
H	460	1725	3000	2.77	3 wing radius	95	106

TABLE 5.4: TEST RESULTS FOR HYDRAULIC ROTARY-PNEUMATIC PERCUSSION DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	BLOWS (/minute)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
						OPERATOR	1 m FROM DRILL
Q STANDARD	400	1150	3100	3.05	RADIUS	104	110
Q MUFFLED	400	1150	3100	3.05	RADIUS	98	105
R STANDARD	-	-	-	-	-	101	107
R MUFFLED	-	-	-	-	-	97	105

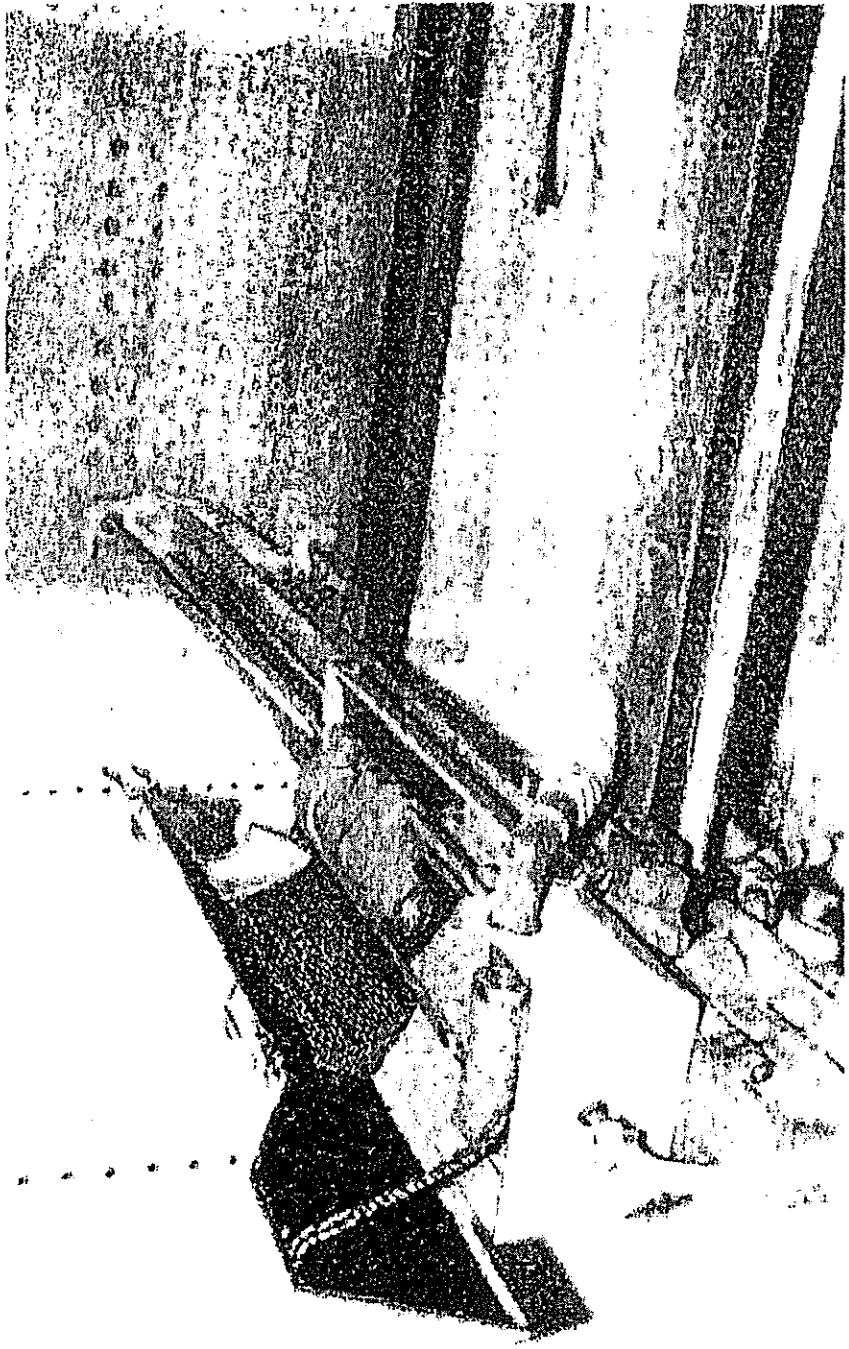


FIGURE 5.1: RETROFIT ACOUSTIC ENCLOSURE ON DRILL

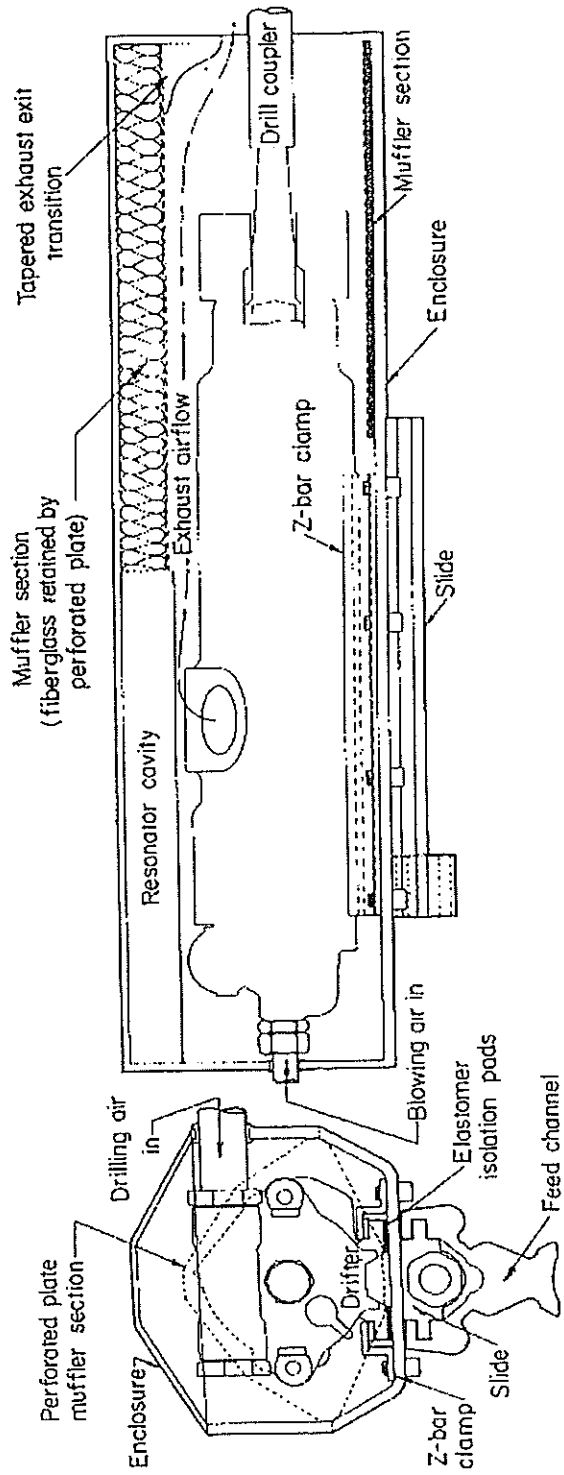


FIGURE 5.2: RETROFIT DRILL BODY ENCLOSURE

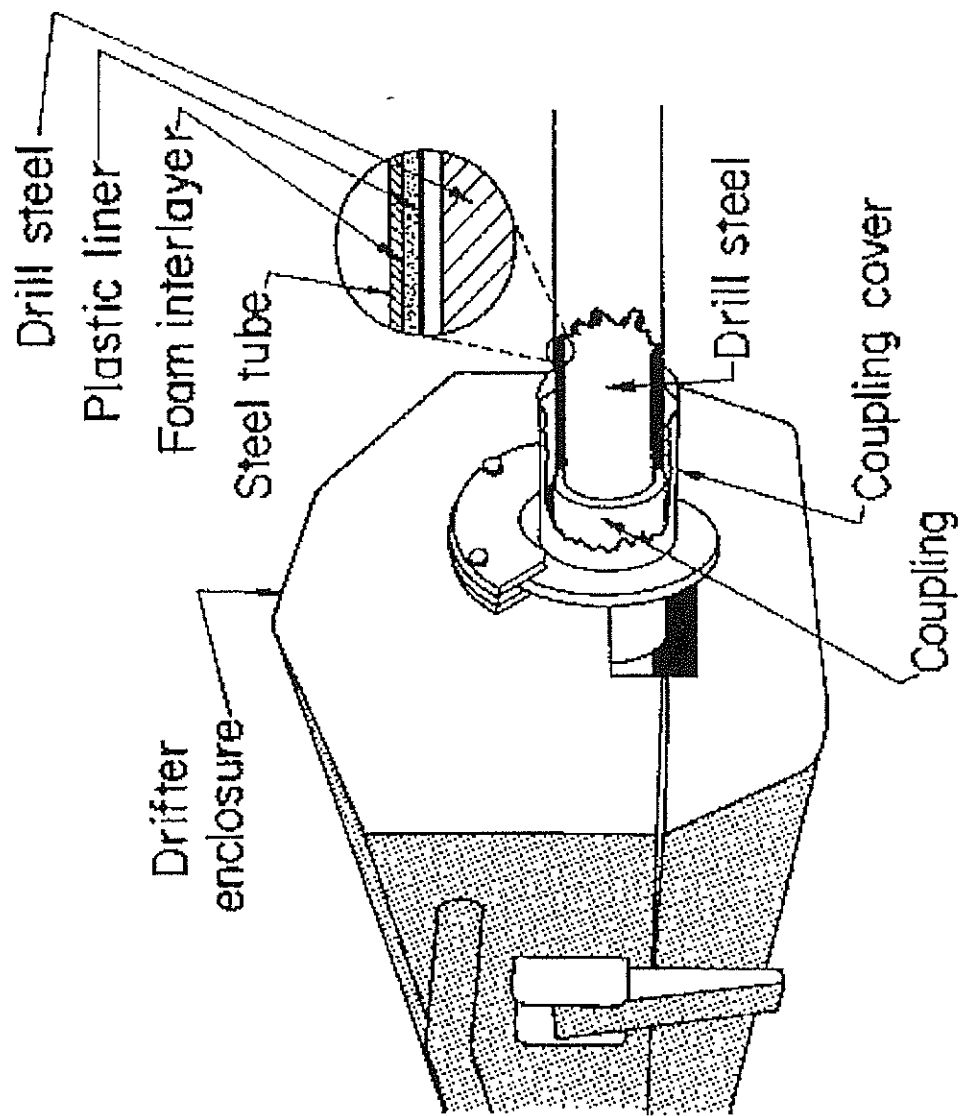


FIGURE 5.3: RETROFIT DRILL STEEL SHROUD TUBE

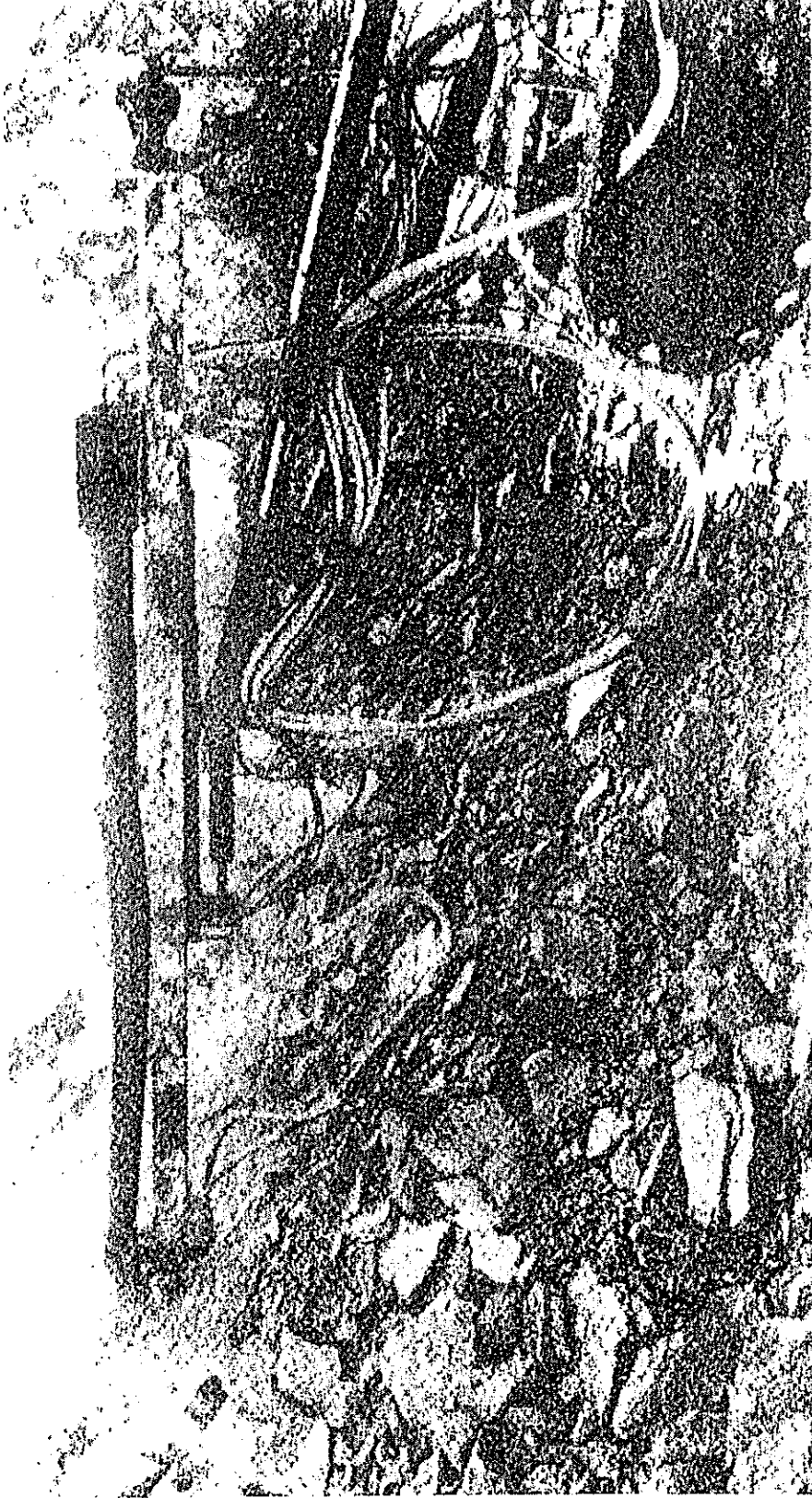


FIGURE 5.4: REDESIGNED DRILL WITH COLLAPSIBLE SHROUD TUBE

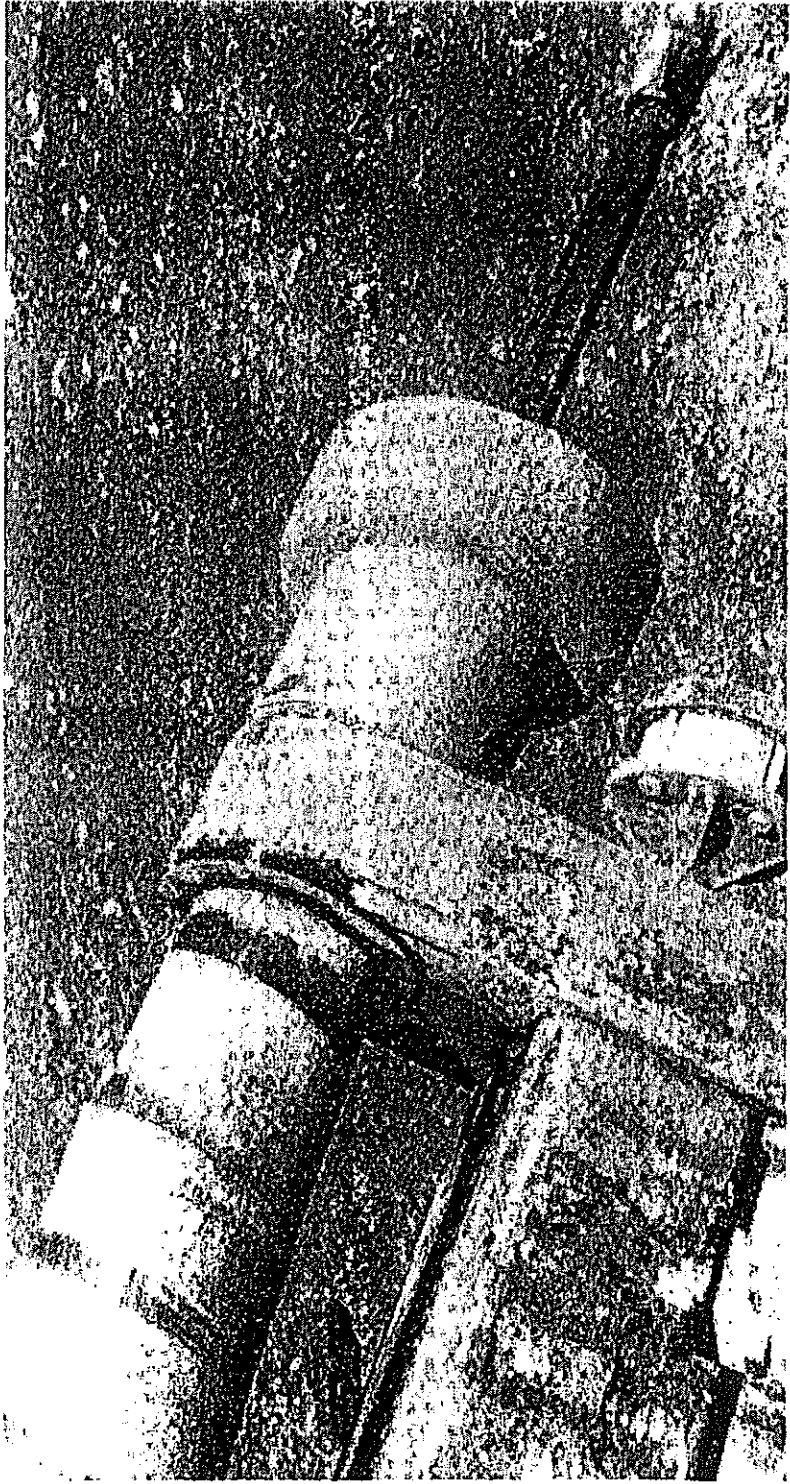


FIGURE 5.5: STINGER ARRANGEMENT



## 6. Continuous miners and roadheaders

Continuous miners and roadheaders are two other types of machines in the coal mining industry whose operation is responsible for the generation of high noise levels. The continuous miner and the roadheader perform the basic mining operations of cutting mineral from the face and transporting it to the rear of the machine. Other machines or conveyors then remove the mineral outbye and eventually to the surface.

The major noise sources on continuous miners are the chain conveyor and the cutting noise. Where auxiliary ventilation at the cutting face is provided by a machine mounted dust scrubber then this is also of major significance. Scrubber noise is dealt with in Chapter 7 of this report. Secondary noise sources on a continuous miner are the drive train and the hydraulic system.

Noise data on continuous miners and roadheaders has been collected from within the South African coal mining industry and from reports from the USA and the United Kingdom. No other papers on noise from these machines were found in the literature search. Kielblock<sup>(1)</sup> and Marx and Franz<sup>(2)</sup> report personal noise exposures,  $N_{eq}$ , for continuous miner operators in South Africa of 94/95 dB(A). Underground measurements of noise levels from some continuous miners and roadheaders in the British coal mining industry<sup>(3)</sup> gave operator noise levels between 98 dB(A) and 104 dB(A) when cutting and loading. With only the central chain conveyor operating noise levels were between 87 and 106 dB(A). These levels are summarised in Table 6.1. In the USA an extensive survey<sup>(4)</sup> by the U.S.B.M. of some twenty six continuous miners from four manufacturers gave a range of operator exposure levels for various operations. For full operation, levels of 93 to 103 dB(A) were measured, while cutting produced 89 to 104 dB(A), and chain conveyor noise levels were between 86 and 99 dB(A). Other U.S.B.M. reports<sup>(5), (6)</sup> gave conveyor only noise levels between 101 and 108 dB(A) on one particular type of continuous miner (Jeffrey 120).

U.S.B.M. major research programs investigated the generation and reduction of conveyor noise<sup>(5),(6),(7),(8)</sup> and coal cutting noise<sup>(9),(10),(11),(12),(13),(14),(15),(16)</sup>. The investigations applied noise control principles and techniques to various mining machines which were tested in the laboratory and in underground working operations.

## 6.1 Chain conveyor noise

References (5) to (8) present the results of U.S.B.M. research programmes to reduce the noise of conveyors used in coal mining. Reference (8) quantifies the noise sources on a continuous miner and describes the treatments used to bring about a noise reduction of 11 dB(A) on a Jeffrey Model 202 loader in an above ground test facility. The loader type of machine was used because of the similarity of the chain and flight type conveyor to that of a continuous miner. The noise reduction techniques implemented were vibration isolated tail roller and top and bottom deck isolation strips. The isolated tail roller comprised an elastomer sleeve isolating the chain from the tail roller. A steel outer shell provided protection for the elastomer. The isolation strips on the decks were protected by aluminium strips covering the top of the elastomer. It is reported that, although these treatments were not tested underground, the above ground operational durability for both treatments was acceptable. The report also states that the techniques are retro-fittable to existing conveyors.

References (5) and (6) describe a U.S.B.M. program whereby noise reduction techniques were tested above ground and underground on a continuous miner conveyor. The above ground tests were performed on a Jeffrey 120M conveyor and the noise level at the operator's position was reduced from 102 to 91 dB(A) using techniques with satisfactory above ground durability. Figure 6.1 (from Reference (6)) shows the noise reduction techniques. The techniques evaluated in this particular above ground program were: constrained layer damping of the upper and lower conveyor decks and side walls, polymer wear strips on the upper and lower decks, a modified tail roller, smoother discontinuities and a modified take-up plate. Figures 6.2 and 6.3 (taken from Reference (6)) show sketches of the treatments. As can be seen in the figures, the top surface of the upper conveyor deck, the bottom surface of the lower deck and the outer faces of the sidewalls were treated with the constrained layer damping. In addition the deck plates were isolated from the conveyor chain and flights by resilient wear strips. These were also backed with rubber to further isolate the decks. A larger diameter tail roller was installed which was mounted on resilient support slides. The takeup plate between the tail roller and the rear portion of the upper deck was mounted on resilient support pads. The most effective treatments were the constrained layer damping, the resilient wear strips and the larger diameter tail roller. Estimated noise level reductions were 5, 3 and 2 dB(A) respectively.

Not all of these treatments were tested underground; only the upper and lower deck constrained layer damping technique was tested, although treatment was to six machines. Operator noise levels from these treated machines, compared with eleven untreated machines, showed average reductions of 2 dB(A) when loading and 5 dB(A) with no coal on the conveyor. Reference (6) states that the constrained layer damping treatment was subsequently integrated into the design of all new Jeffrey chain conveyors.

An ECSC study<sup>(17)</sup> of noise from machines in face-ends and headings reported noise reductions of 5 dB(A) from a continuous miner after damping treatment to the chain conveyor. Initially the chain conveyor noise of a standard Joy 12CM11 continuous miner, shown in Figure 6.4, was reduced experimentally by simple covering of the top of the conveyor with noise barrier/absorption material (foam/lead/foam sandwich). These initial tests showed potential noise reduction at the operator position of at least 5 dB(A). Subsequent noise measurements of two continuous miners, the first with a conventional chain conveyor and the second with a modified conveyor, produced 5 dB(A) lower noise levels (101 vs. 106 dB(A)) compared with the modified deck. The modifications to the second chain conveyor comprised welding of extra steel plates to the underside of both the upper and lower decks to provide damping of the conveyor deck. These design modifications were undertaken during the manufacture of the machine.

It is reported<sup>(18)</sup> that since 1989 all Joy continuous miners have incorporated a damped chain conveyor as standard. Each deck is actually two steel plates with a "filling" material between the plates. Reported noise reduction was of the order of 3 dB(A). Until 1996 the "filling" used between the steel plates was lead shot but Joy now use sand fill rather than lead. Sand replaced lead because of problems with disposal of the lead when a conveyor was worn out or came to the end of its life. Prior to the use of this type of conveyor damping, Joy tested various noise reduction techniques in the early 1970's on a 14BU10 Loader. Significant noise reductions were obtained with the following: full deck liners of urethane, nylon liners under the chain and modification of the deck geometry to smooth exit and entry at foot shaft and tail roller. No significant change occurred with shock absorbers in the flight tips, isolation of the tail roller and modification of the chain geometry. Reducing chain speed and reducing the number of flights "yielded no improvement without destroying the machine's ability to do its job." The full deck urethane liners reduced the level from 105 dB(A) to 99 dB(A) and further 2 dB(A) reduction with the deck geometry modifications. Such a machine was then subjected to underground trials. However, "Within three months about half the improvement was lost and within six months it had virtually all been lost. This was mainly due to the inability to keep the

urethane liners in place in the conveyor." Joy did make several design changes to chain and deck geometry and tail roller size but these modifications did not contribute more than 1 dB(A) long-term reduction.

The conveyor damping method referred to above, (Reference (17)), had previously been used with varying degrees of success on other machines within British Coal in the UK. Another ECSC project on reduction of noise from mining machines<sup>(19)</sup> reports a reduction of 10 dB(A) for a similarly treated chain conveyor on an Anderson Strathclyde RH22 Roadheader.

A manufacturer's information leaflet<sup>(20)</sup>, from Long-Airtox, on two models of continuous miner describes "suppressed-noise discharge chain conveyors". The model CM-525 has a 30 inch "sound-dampened conveyor" and the model CM-728 has a 36 inch "sound-dampened conveyor".

Further chain conveyor noise research by the U.S.B.M. was reported in 1986 by Burks and Peterson<sup>(21)</sup>. An approach they considered to be promising was the use of either a urethane coating or a urethane sleeve on the chain flights as vibration isolation to soften the impact between the chain and the conveyor structure. An above ground test produced a reduction of 5 - 10 dB(A). The previously discussed experience of the failure underground within six months of the Joy continuous miner urethane conveyor liners may be of note here. This raises the question of the durability of such methods. However, the Joy tests were some twenty seven years ago and the efficiency of metal/plastic bonding systems should have improved with technological progress. Further investigation may be worthwhile.

Effects of chain speed, type of chain and chain tension were also investigated by the U.S.B.M. Noise levels from conveyors are known to depend on chain speed and tests examining the effect of chain speed (Reference (7)) suggested the following expression for the overall A-weighted sound pressure level ( $L_p$ ):

$$L_p = 10 \log_{10} \left( \frac{V}{100} \right)^3 + 86.5 \quad \text{dB(A)}$$

where  $V$  is the chain velocity in feet per minute.

A variable speed conveyor drive system was investigated by the U.S.B.M. In addition to the primary benefit of noise reduction through conveyor speed reduction, there is also the secondary benefit of more coal on the conveyor, in itself producing lower noise, when running at slower speeds for certain portions of the cutting cycle.

Reference (3) reports lower noise levels from an Anderson KBII Miner-Bolter in the UK running at slower chain speeds. When tested underground, the KBII produced levels at the operator of 90 dB(A) for a chain speed of 200ft/minute compared with operator levels of 100 dB(A) and 105 dB(A) (Joy 12CM18 and Joy 12CM11 respectively) measured underground during the same investigation with chain speeds of greater than 400 ft/minute. However, it should be noted that these noise levels are not directly comparable because the KBII had another considerable difference, that of the absence of significant discontinuities on the scraper conveyor, producing a much smoother running surface than on the other machines.

## 6.2 Coal cutting noise

Reference (14) reported that "The largest noise reductions can be achieved through very slow and deep coal cutting and/or cutting force isolation techniques ...implementation of these noise and vibration control concepts by the industry as a whole will require a significant design and re-tooling effort". In practice, slow and deep coal cutting has already been incorporated within coal face machinery designs where major gains have been obtained with respect to dust control. Significant further speed reductions are unlikely due to production restraints.

The U.S.B.M. investigations in references (9) to (16) found that noise generating mechanisms associated with coal cutting are:

- (i) cutting head noise, caused by vibration of the structure holding the cutting tools;
- (ii) fracture noise, caused by air rushing into the cracks produced when the coal breaks;
- (iii) face radiation noise, caused by vibration of the solid coal;
- (iv) machine structural noise, produced when coal cutting forces caused vibration of other machine components.

It was stated that, of the above, the cutting head noise was dominant.

In the U.S.B.M. studies three basic coal cutting noise control concepts were considered - reduced bit velocity, cutting head structural response alteration and noise control through cutting force isolation.

- **Reduced bit velocity**

Theoretical models and controlled laboratory tests predicted noise reductions by reducing cutting speed. Reference (11) discusses a theoretical 3 dB(A) noise reduction by halving the cutting speed. However, it was not feasible to implement a change in cutting speed during the experimental program on a continuous miner and therefore the actual reduction attainable in the field was not determined. In relation to concerns of reduced operational performance, reference (11) states that "the bit velocity can be reduced while maintaining coal production by increasing the depth of cut and/or the number of bits per cutting line." Clearly, these factors have implications on manufacturing design parameters of the continuous miner.

In 1992 Murphy<sup>(16)</sup> reported that the U.S.B.M. had developed an entirely new coal cutting concept to reduce dust generation at the source. The problems inherent with rotary cutting could be controlled by changing to linear cutting. A linear drum cuts easily and smoothly, with a minimum of dust and fines, and *at a slow bit velocity*. The slow bit velocity is the important factor as far as noise reduction is concerned.

- **Cutting head structural response alteration**

An **auger miner** cutting head was noise treated by stiffening the cutting head helix by addition of steel plate welded to the helix and drum core, and filling the void thus created with sand. Additionally the damping of the cutting drum core was increased. These techniques are illustrated in Figures 6.5, 6.6A and 6.6B (from Reference (11)), photographs of an untreated auger and the noise reduced auger.

The stiffening achieved a four-fold increase in the helix first natural frequency and damping achieved a ten-fold increase in helix damping at the resonances below 2000 Hz. Operational tests occurred over a six month long period at a colliery. Noise reductions were 6 dB(A) at the machine operator position (96 vs. 102 dB(A)) and 9 dB(A) at the jacksetter position (further inbye than the machine operator) (97 vs. 106 dB(A)). The

reduction at the operator would have been greater if other noise sources had not been present.

Similar noise control techniques were incorporated in a redesigned longwall shearer cutting drum. Field tests at two mines gave noise reductions of 5 dB(A). Further details are discussed in Chapter 8 of this report "Longwall Systems".

**Continuous miner** cutting noise was reduced experimentally above ground by treatment of the cutting head. References (13) and (14) give details of U.S.B.M. noise control work on the cutting head of a Lee Norse HH105 continuous miner. The treatment incorporated wrapping the surface of the drums with a layer of foam overlaid with a heavy vinyl barrier material. Cutting tests with a treated and an untreated drum produced reductions of approximately 5 dB(A), demonstrating the importance of noise radiated from the cutting head. A photograph of the wrapped cutting drum is shown in Figure 6.7 (from Reference (13)).

Reference (18), from a continuous miner manufacturer, briefly describes testing of a damped cutting drum in 1981 which gave a measured noise reduction of 3 dB(A). This was achieved by the manufacturer gluing a layer of vibration damping material on the outside of the drum and covering this with steel plate for wear protection. This treatment reduced cutting head vibration and hence radiated noise. Joy claim the method is workable and appears to have a reasonable service life.

### • **Noise control through cutting force isolation**

An alternative noise control treatment for a **continuous miner** cutting head involved attempts by the U.S.B.M. to isolate the input cutting force from the cutting head.

The technique involved attempts to isolate the dynamic forces of the cutting bits from the major noise radiating structures of the continuous miner. Difficulties in producing an effective isolated cutting bit were overcome by production of an isolated cutting *ring*. Isolation of the cutting forces was achieved by mounting the cutting bits on large rings, with each ring isolated from the central drive shaft of the head by four elastomeric bushings. This involved re-design of the cutting head as shown in the design plans of Figure 6.8 (taken from Reference (11)) and Figure 6.9 (taken from Reference (15)). The report of the U.S.B.M. contract, Reference (14), details underground tests carried out

after isolated cutting head treatment to a Jeffrey 1028 continuous miner. The first isolated cutting ring tested used an experimental bit lacing pattern called staged lacing. This method used sets of bits, rather than single bits, arranged in a helical pattern around the drum, as shown in Figure 6.8. If successful this concept would have assisted the development of further noise control by using isolated cutting tools of much larger mass and lower natural frequency than the isolated cutting bit discussed above. However, underground testing showed that the staged bit lacing was operationally less effective than the standard lacing, with a decreased rate of advance. Noise reduction was 3 dB(A). The staged lacing was removed and replaced by standard scroll lacing which produced better operational performance. In hard cutting conditions the cutting noise levels were reduced by 5 dB(A) (91 vs. 96 dB(A)) and when cutting coal alone the cutting noise was reduced to less than 90 dB(A). Frequency spectra comparing the noise from cutting with the standard cutting head and the isolated cutting head are shown in Figure 6.10 (from Reference (14)). It is reported that the isolated cutting head performed well over a five month underground test and that the elastomeric isolator bushings showed no signs of wear. Although some failures occurred with the capture bolts the report suggests remedial measures to overcome this minor problem.

Apparently the isolated cutting head concept is applicable to all continuous miners with chain drive.

### **6.3 Other noise sources on continuous miners**

The previously discussed noise control treatments have targeted the dominant noise sources of the cutting head and chain conveyor. Other noise sources such as the drive train and hydraulic system represent secondary noise sources because of their smaller contribution to the overall noise levels. Reference to Table 6.1 shows that noise from these sources is usually below 90 dB(A).

After noise reduction of the dominant sources has been achieved the use of standard noise control techniques would be more applicable to these secondary noise sources. Effective methods are discussed elsewhere in this report, for example, in the next section on roadheader noise and in Chapter 9 "Trackless Vehicles/Load Haul Dump Machines".



## 6.4 Other noise sources on roadheaders

In addition to detailing the conveyor noise reduction on the RH22 roadheader, reference (19) describes extensive studies and attenuation of other principal noise sources on the machine. Unlike the continuous miners already discussed, the principal roadheader sources included the electric drive motor, located just behind the driver, and the machine hydraulics. Both these sources were successfully treated on the Anderson RH22. Measured attenuations were up to 10 dB(A) and 13 dB(A) respectively. Originally the electric motor drive to the hydraulic circuit generated levels of 96 dB(A), caused by the motor cooling fan. The hydraulics noise was also a major contribution to operator noise exposure. In particular the tonal noise of one of the flow dividers was dominant. Figure 6.11 shows the frequency spectrum of noise at the driver's position on the RH22 and illustrates the dominant characteristic of the discrete tonal levels. High tonal peaks from the drive motor cooling fan, hydraulic flow divider and to a lesser extent the hydraulic pump, can clearly be seen. The drive motor tones occurred at the fan blade passing frequency and its harmonics. A similar harmonic series was caused by one of the hydraulics flow dividers.

Re-design of the motor cooling fan by reducing the diameter and improving the aerodynamics reduced the levels of turbulence and resulted in noise reductions of 10 dB(A) at 1m on the axis of the fan inlet and 8 dB(A) at 1m from the outlet at the side of the motor. Design sketches illustrating the differences between the original, standard, fan and the final version of the new design are shown in Figure 6.12 and photographs are shown in Figure 6.13A and 6.13B. As a result of the fan noise reduction, a directive was issued requiring that all new British Coal Specification 625 motors were to be fitted with low noise impellers. The RH22 hydraulics noise was reduced by 13 dB(A) by re-design of the hydraulics circuit, part of which replaced a flow divider with one of a more appropriate duty, and acoustically enclosing the hydraulics power pack. The enclosure comprised a noise transmission barrier shell of 5mm steel plate and internal lining of sound absorbing mineral wool panels covered with steel mesh for protection. The absorbent panels reduced the reverberation noise inside the hydraulic compartment. The photograph in Figure 6.14 shows the hydraulic power pack enclosure with acoustic lining.

## **6.5 Retrofit of existing equipment**

Significant portions of Joy mining machines are fabricated in South Africa. To allow for increased seam height standard continuous miners are modified locally: height is increased, pivot point raised, etc. This lends itself to retro-fit noise control treatments, but many of the aforementioned techniques would be more applicable for incorporating into new designs rather than retro-fit options.

A retro-fittable technique, tested by the U.S.B.M. and discussed earlier, is that of fitting isolating strips on the upper and lower decks of chain conveyors. Another technique, that of constrained layer damping of conveyor deck plates, may be retro-fittable, but with some difficulty and would be much more suited to incorporation into new machines. Some manufacturers already incorporate this technique into their designs of continuous miners.

## **6.6 New equipment**

Many new continuous miners are manufactured with damped chain conveyor decks. In fact at least one manufacturer is producing a sandwich type deck as standard on all continuous miners. Reported noise reductions of this type of treatment are of the order of 3 to 10 dB(A), although the 10 dB(A) claim may be a little optimistic.

Isolation of the chain and flights from the upper and lower decks of chain conveyors should also be considered, either for further research, or for implementation by manufacturers. Care must be taken with the method of fixing the treatment to the conveyor so as not to negate any potential noise reduction achieved by the "isolation".

A damped continuous miner cutting drum giving a noise reduction of 3 dB(A) is available from at least one manufacturer.

An alternative to new design or re-designed machinery which is finding favour with manufacturers is the operation of continuous miners and roadheaders by radio remote control. This can benefit the machine operator by moving him further away from the high noise levels and thereby reduce his noise exposure. Remote control can be fitted to all current production Joy continuous miners.

## 6.7 Retrofit summary

• Damped chain conveyor	typically - 5 dB(A) reduction	±R20 000
• Isolating strips/resilient wear strips on conveyor	3 dB(A) reduction	±R10 000
• * Resilient/polymer conveyor liners or chain/flight coatings	5 dB(A) reduction	±R10 000
• Radio remote control	to be determined	±R500 000
• Isolated cutting head	5 dB(A) reduction	±R50 000
• Damped cutting drum	3 dB(A) reduction	±R30 000
• Re-designed motor fan impeller	8 dB(A) reduction	±R5 000
• Improved hydraulic power pack enclosure	8 dB(A) reduction	±R10 000

\* Would require further research.

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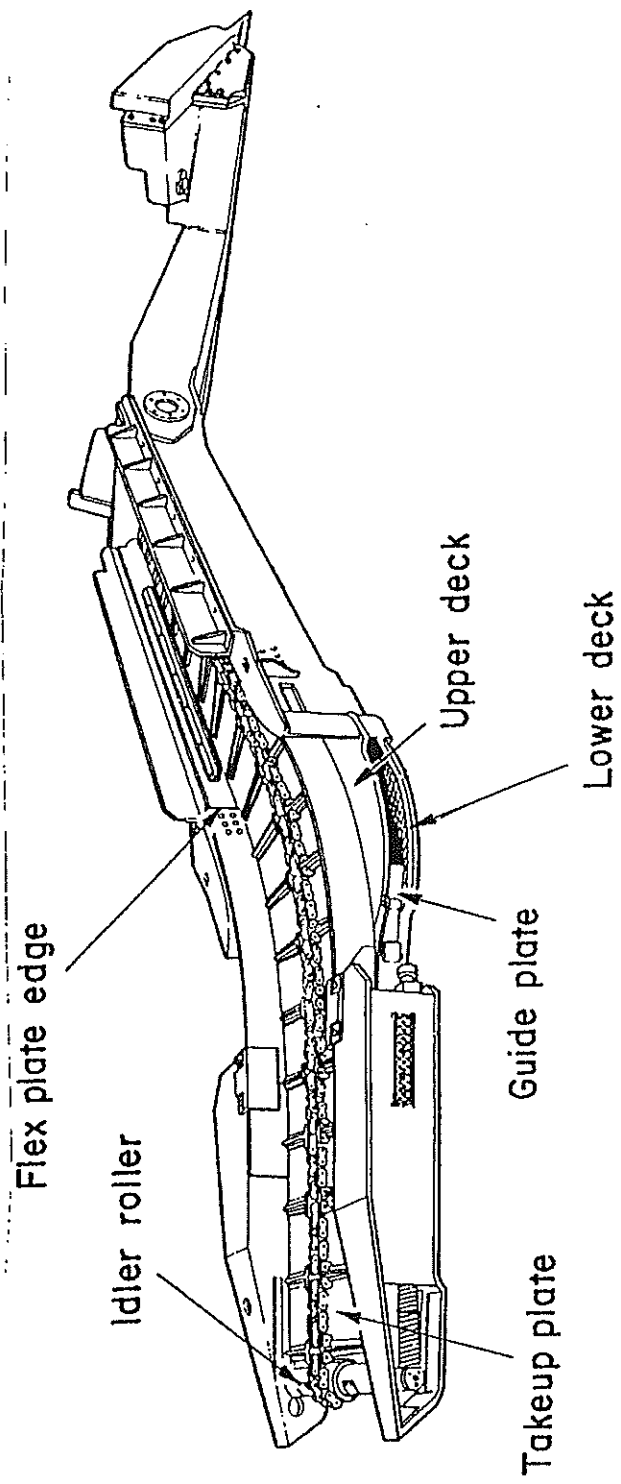


FIGURE 6.1: NOISE PRODUCING COMPONENTS OF A CONTINUOUS MINER CHAIN CONVEYOR

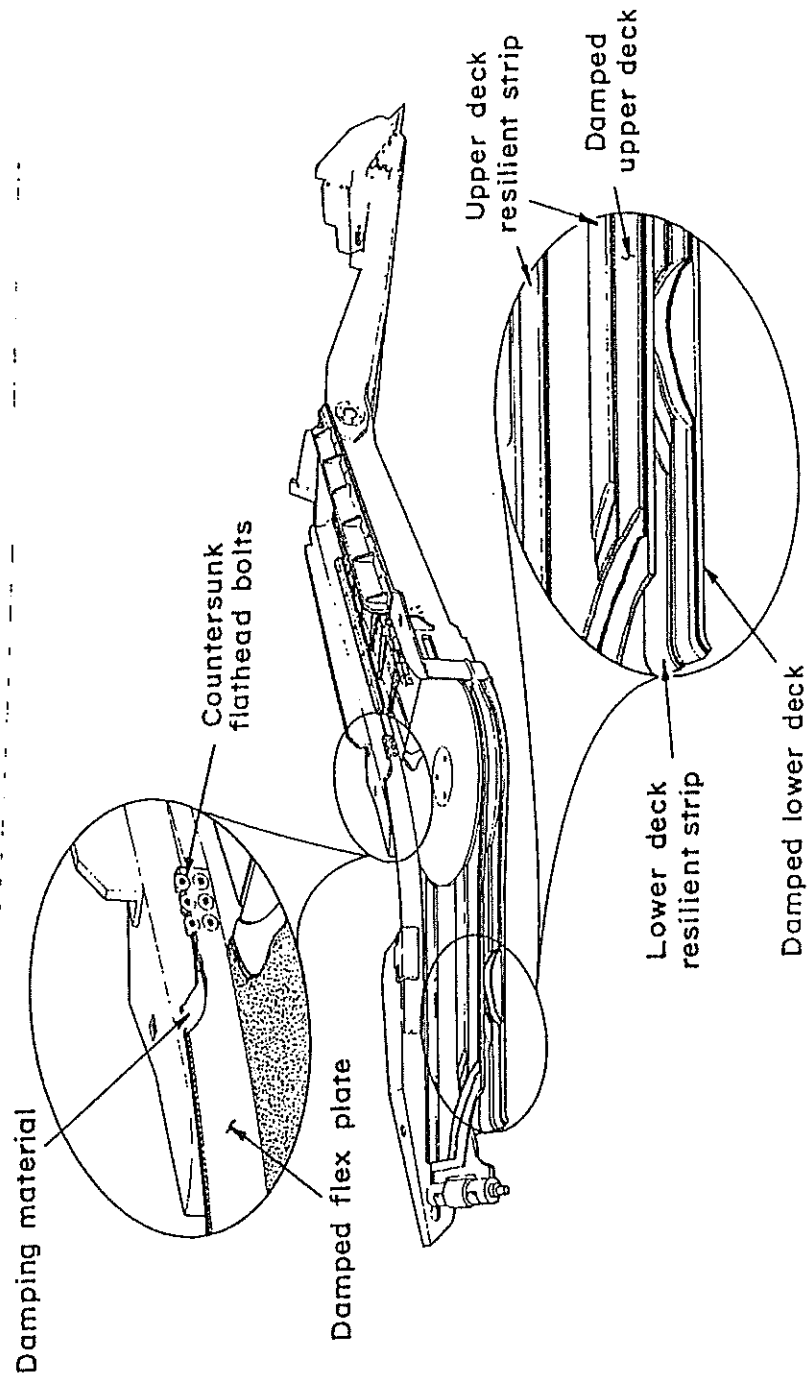


FIGURE 6.2: NOISE CONTROL TREATMENTS ON CONTINUOUS MINER CONVEYOR DECKS AND SIDEWALLS

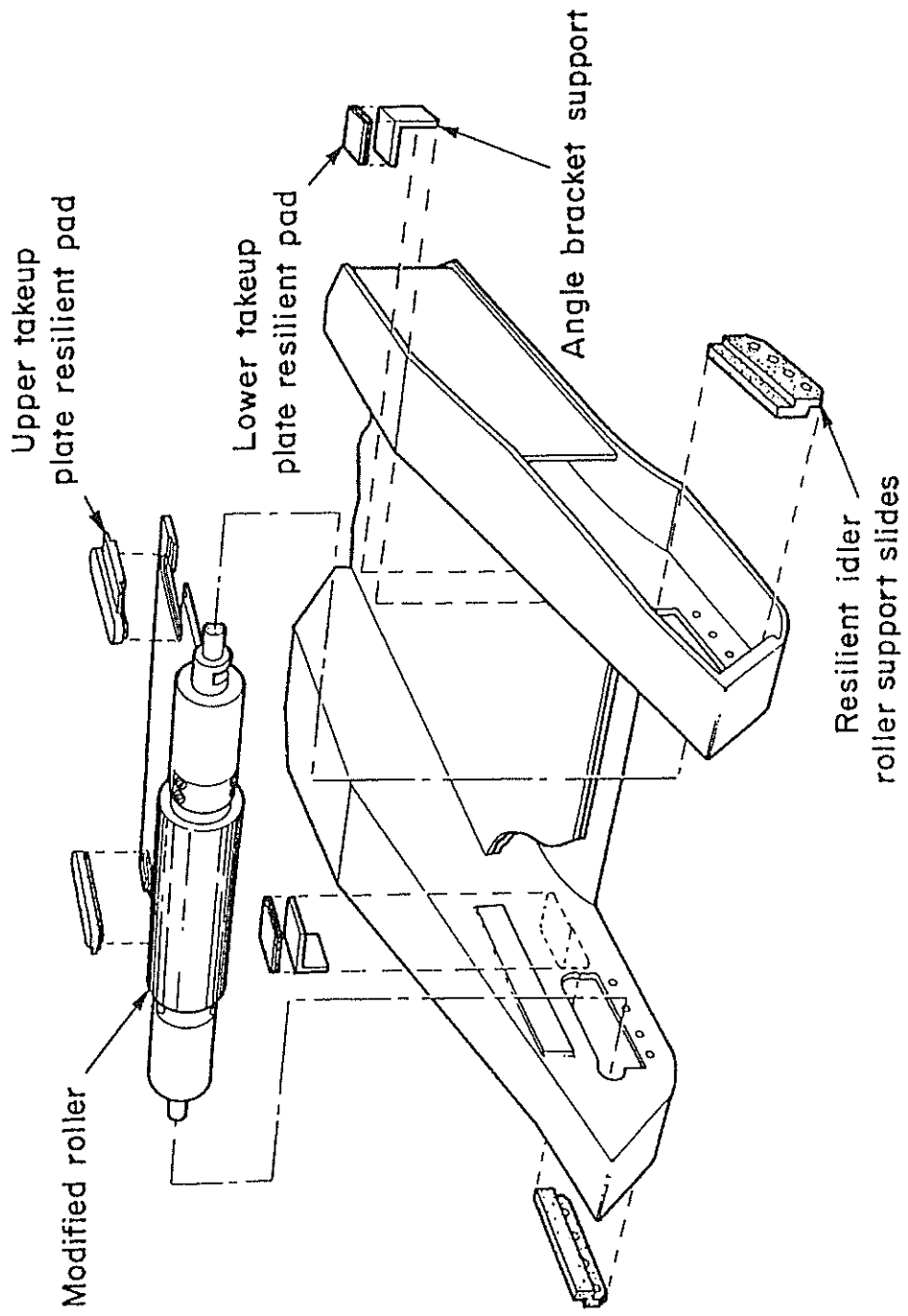


FIGURE 6.3: NOISE CONTROL TREATMENTS ON CONTINUOUS MINER CONVEYOR TAIL ROLLER AND TAKE-UP PLATE

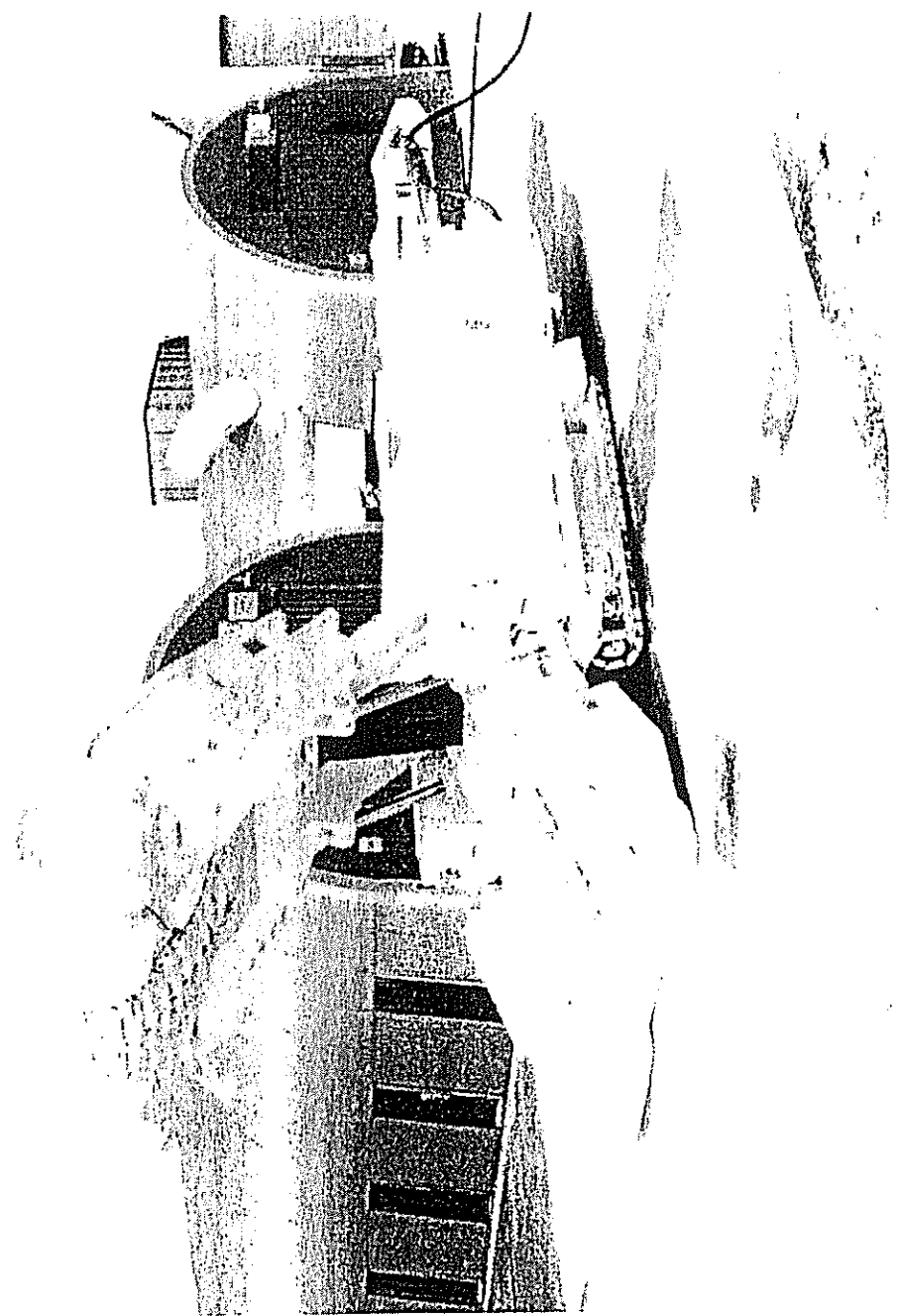


FIGURE 6.4: JOY 12CM11 CONTINUOUS MINER



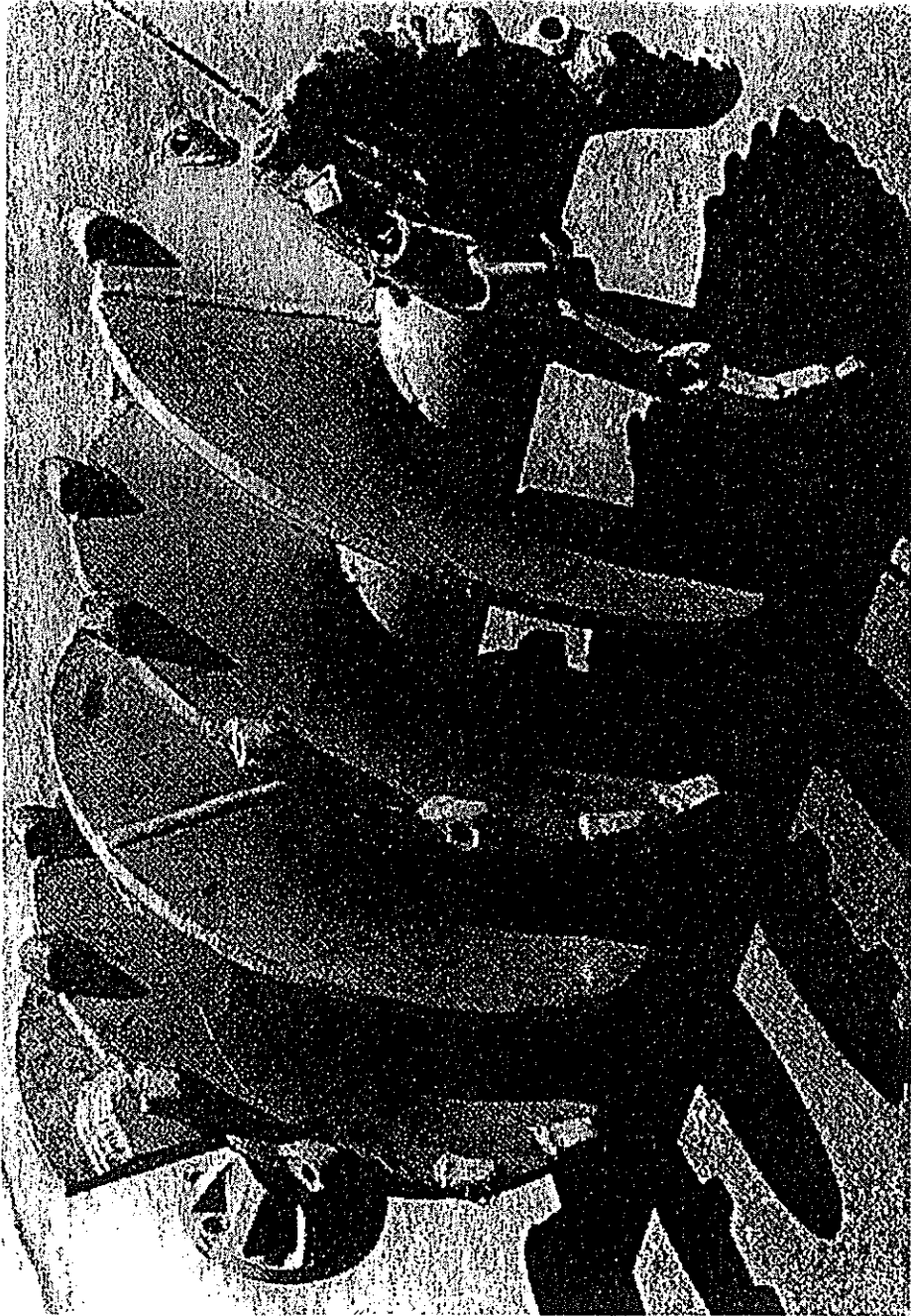


FIGURE 6.5: STANDARD AUGER MINER CUTTING HEAD

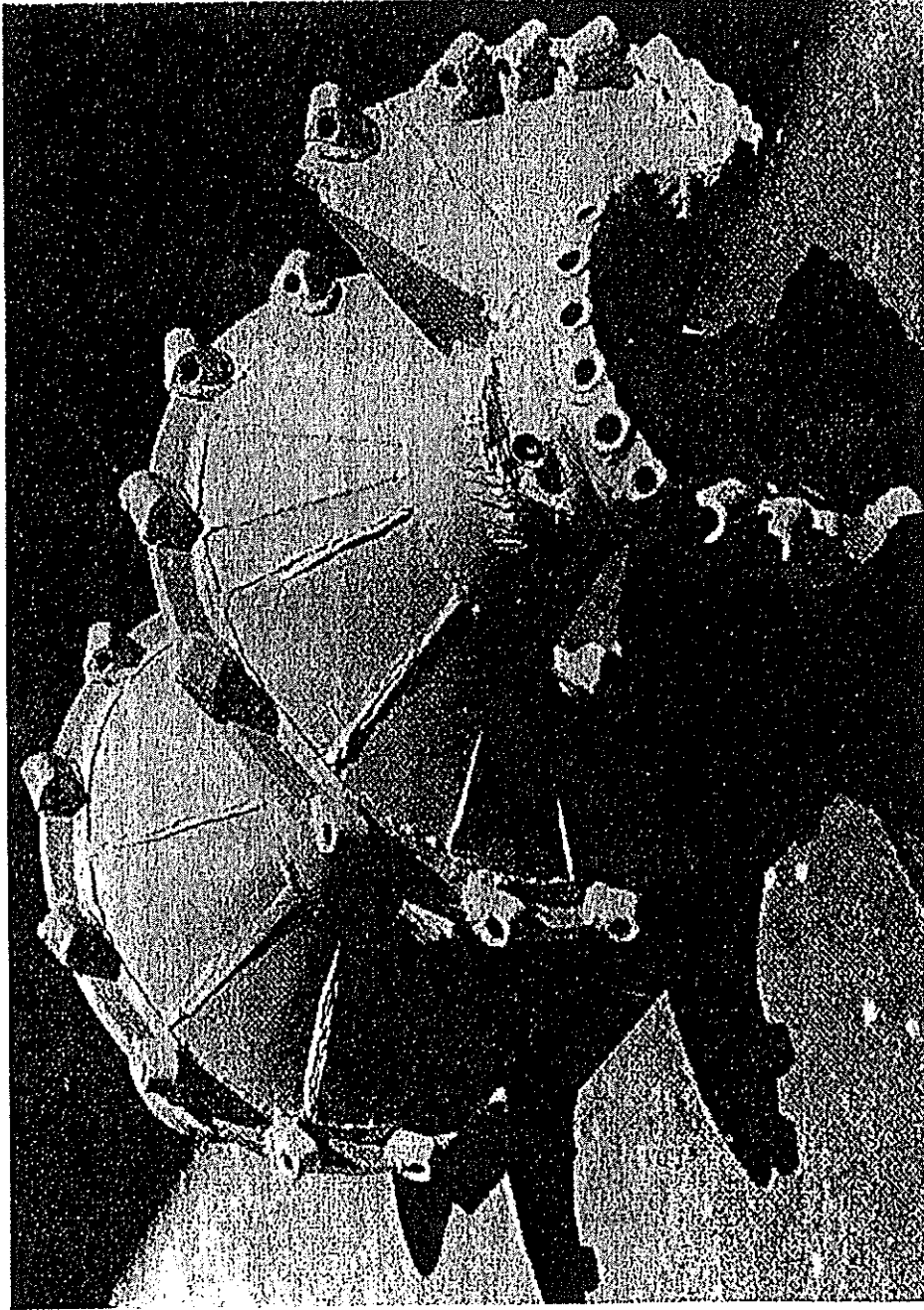


FIGURE 6.6A: REDUCED-NOISE AUGER MINER CUTTING HEAD

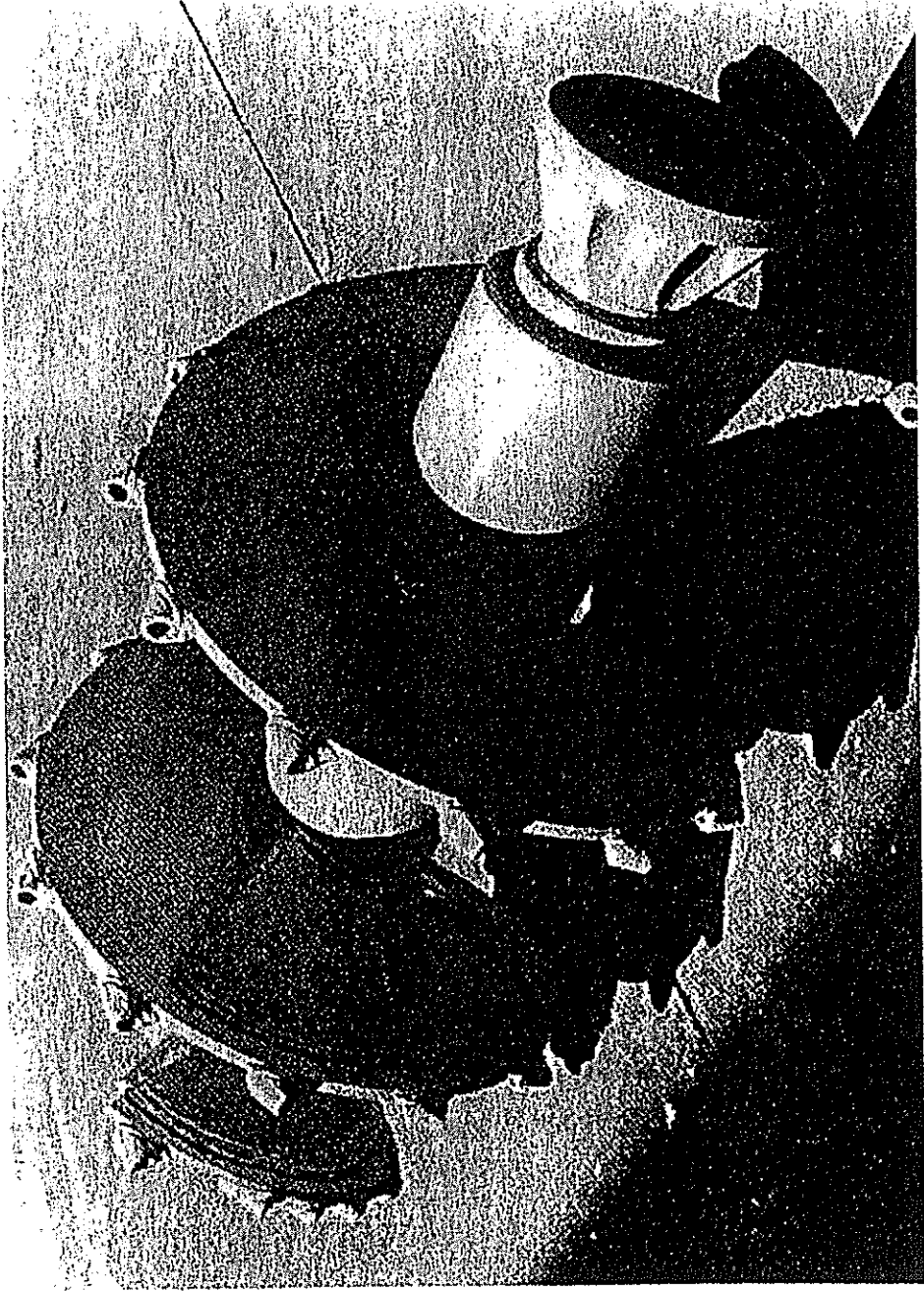


FIGURE 6.6B: REDUCED-NOISE AUGER MINER CUTTING HEAD



FIGURE 6.7: WRAPPED CUTTER HEAD OF CONTINUOUS MINER

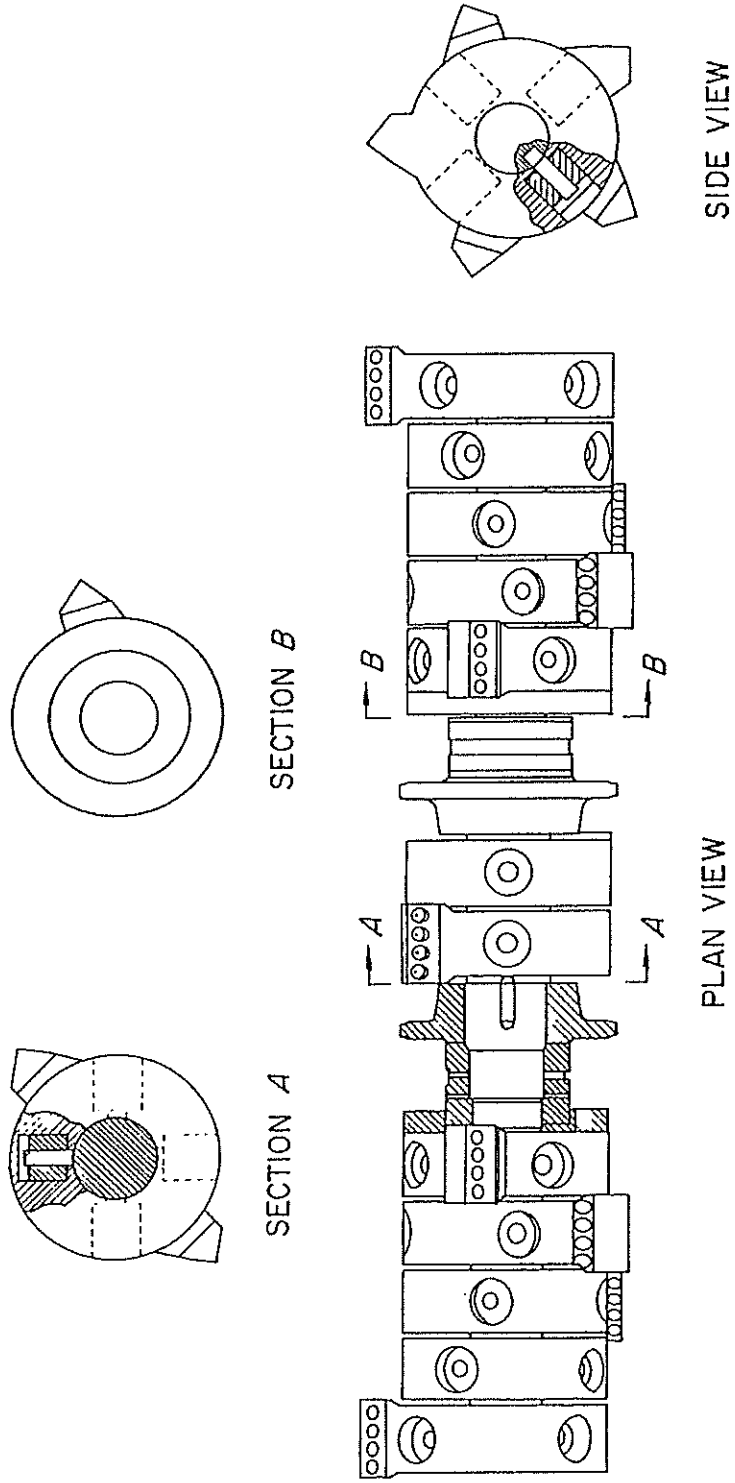


FIGURE 6.8: CONTINUOUS MINER ISOLATED CUTTING HEAD WITH STAGED BIT LACING

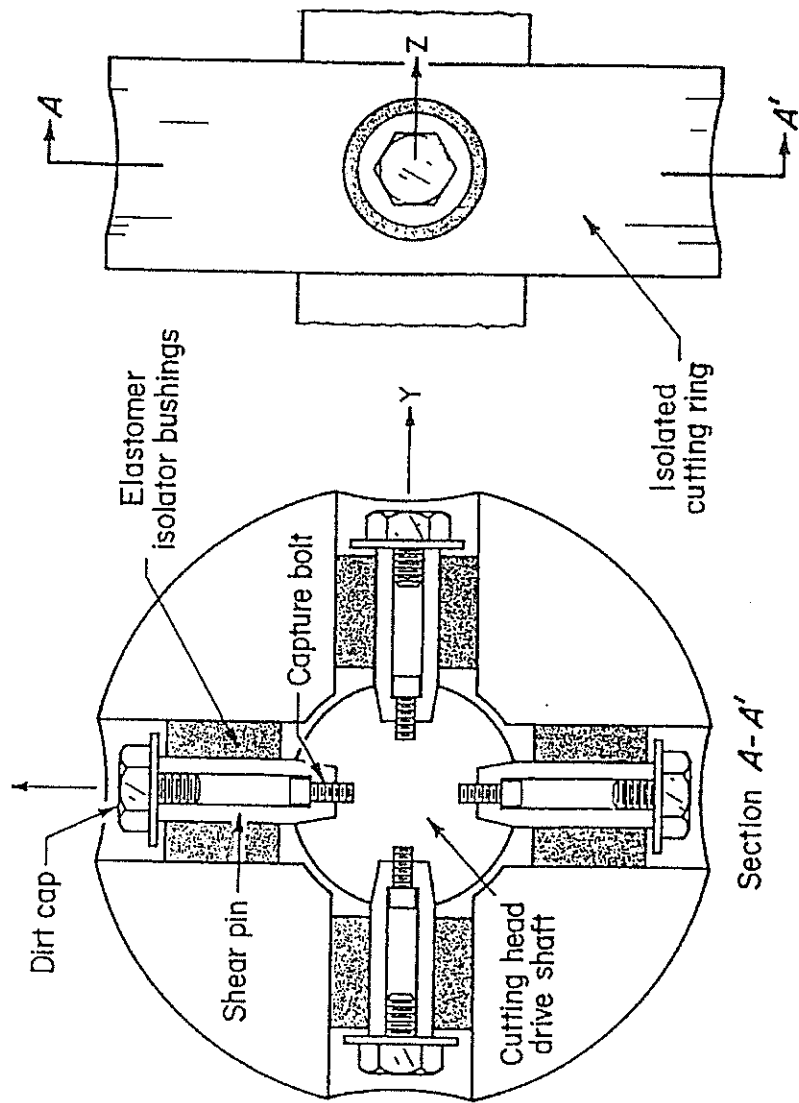


FIGURE 6.9: ISOLATED CONTINUOUS MINER CUTTING HEAD - ISOLATION SCHEME

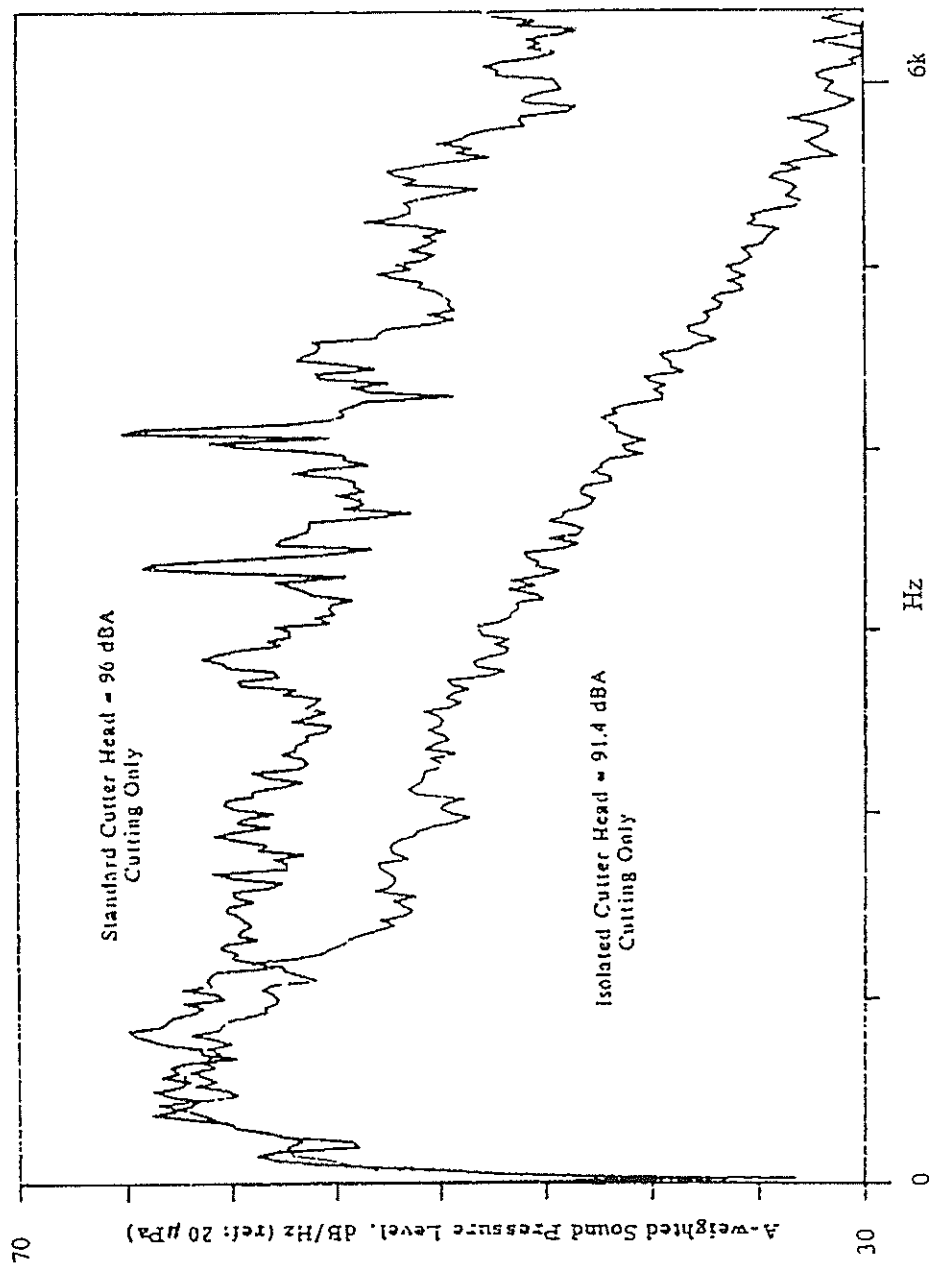


FIGURE 6.10: NOISE CONTROL PERFORMANCE OF ISOLATED CUTTING HEAD - REPRESENTATIVE SPECTRA

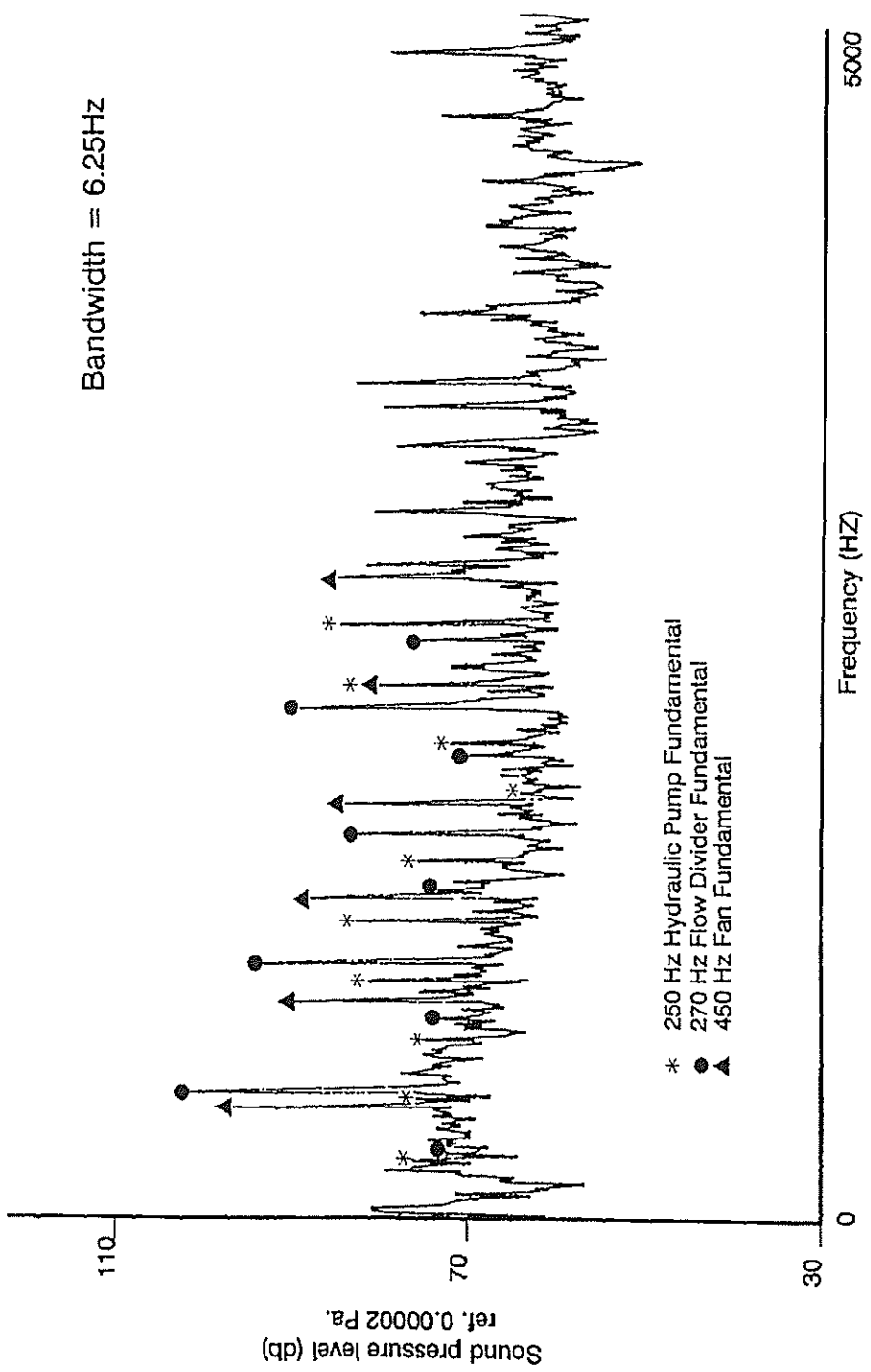
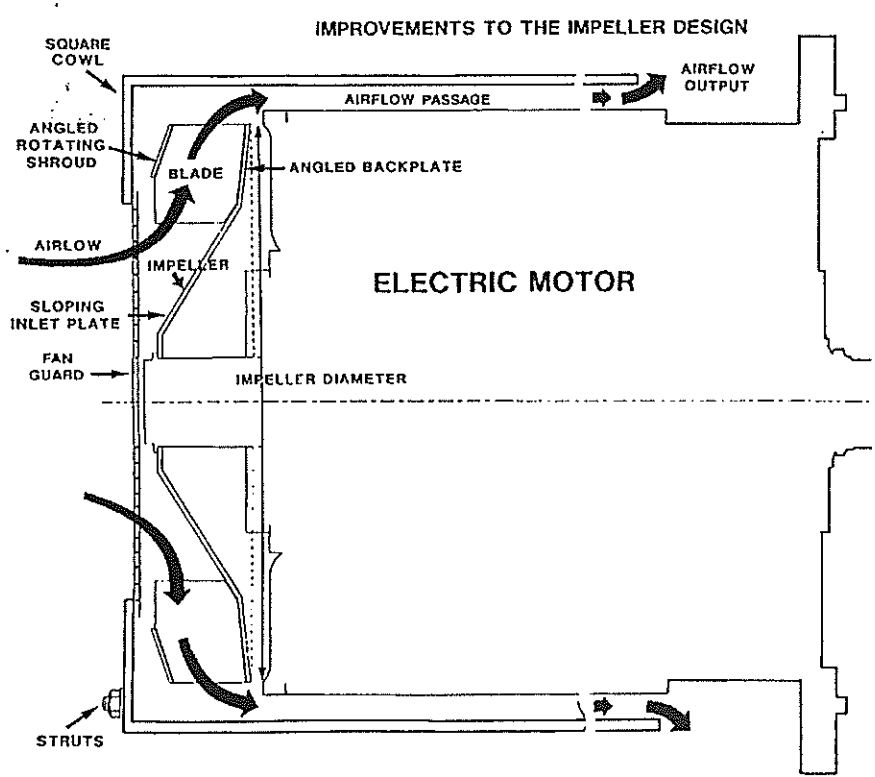
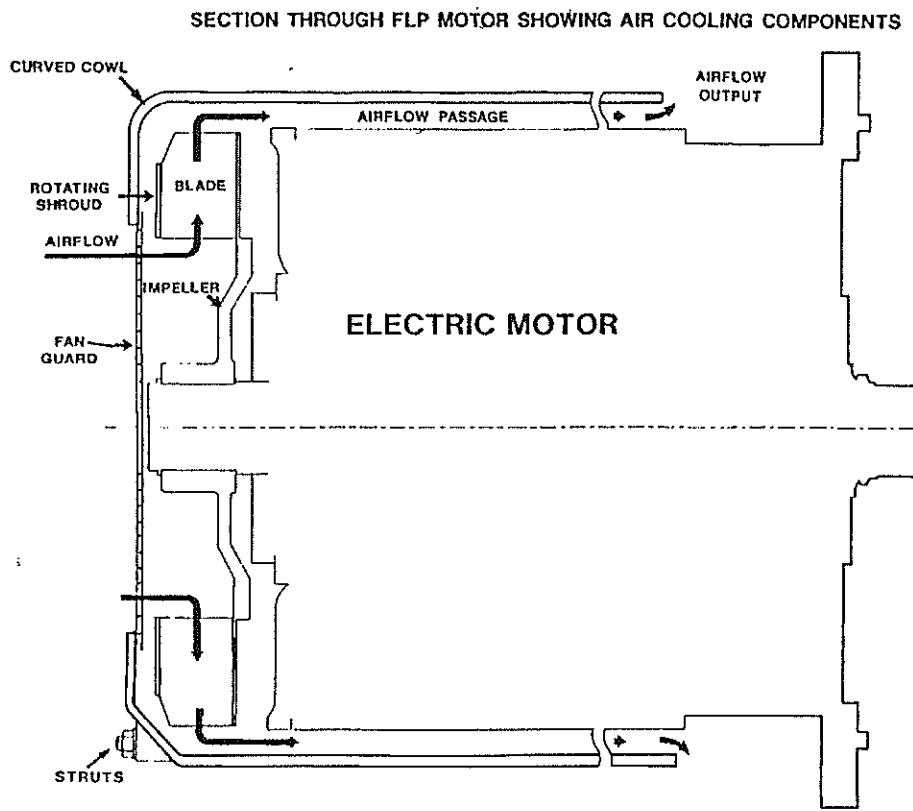
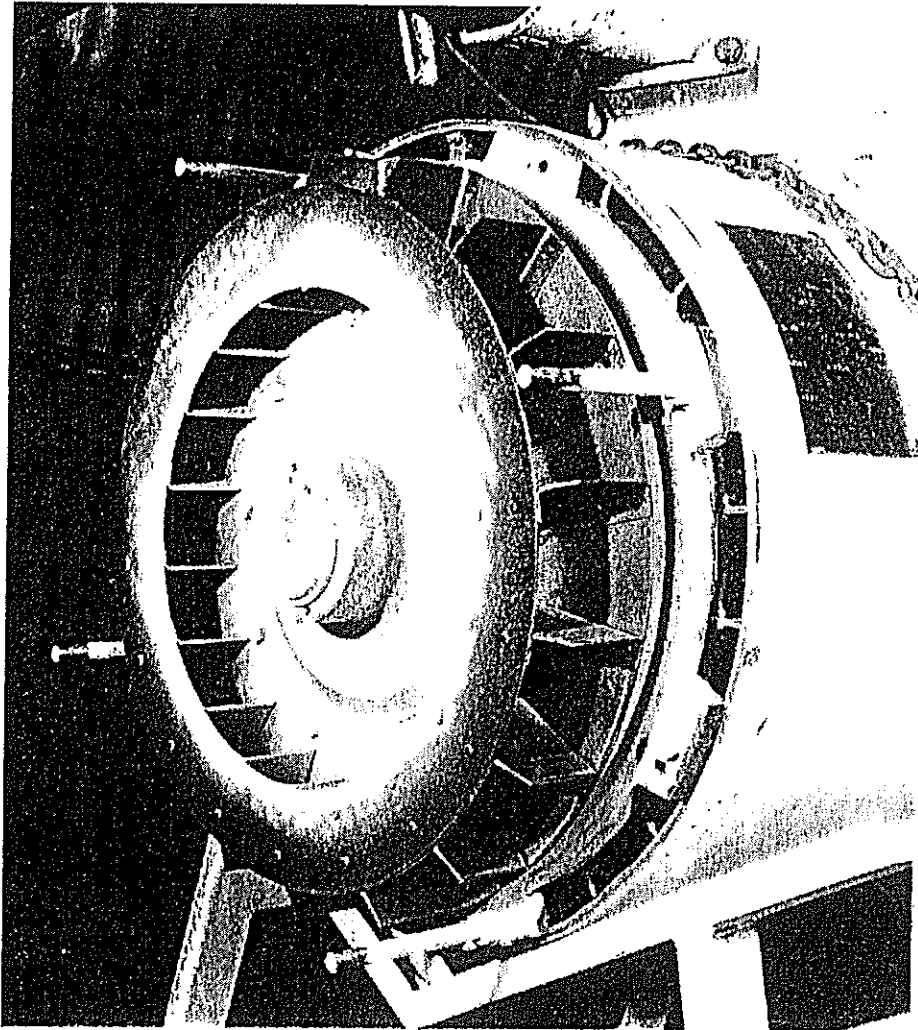


FIGURE 6.11: RH22 NOISE SPECTRUM AT OPERATOR'S POSITION

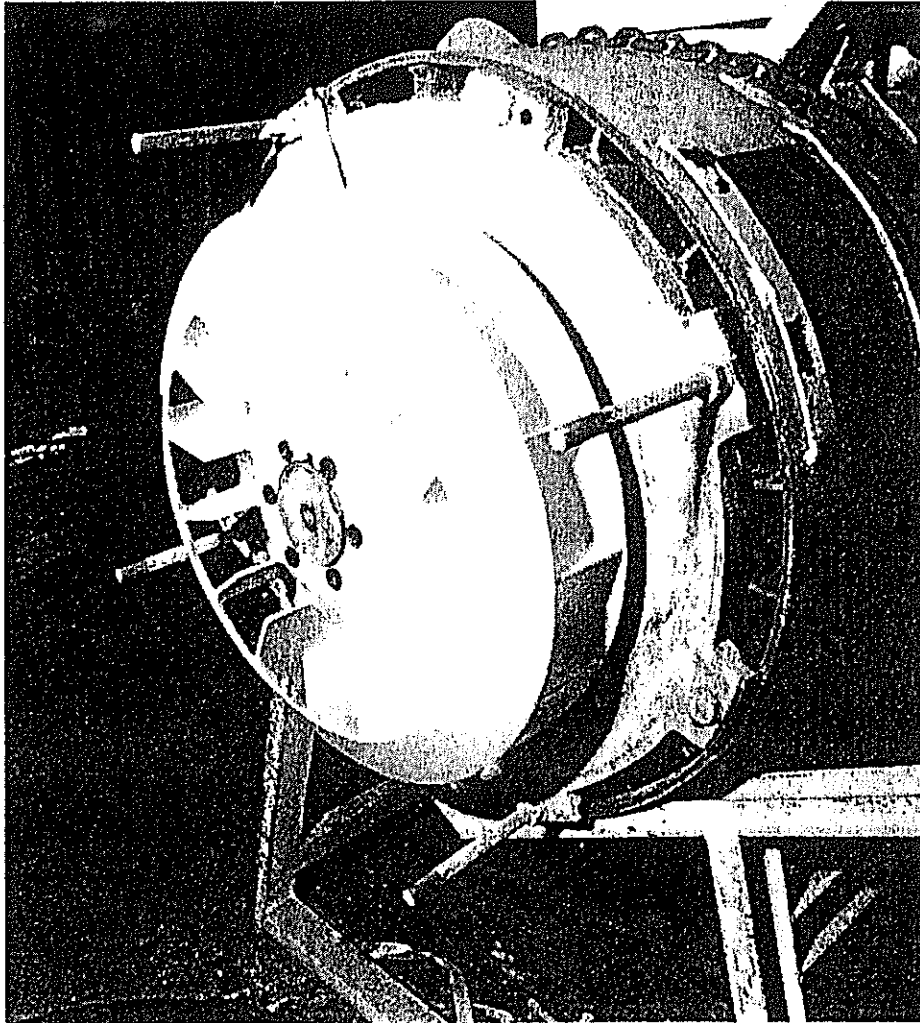




**FIGURE 6.12: ELECTRIC MOTOR BEFORE AND AFTER NOISE CONTROL RE-DESIGN**



**FIGURE 6.13A: STANDARD FAN IMPELLER ON SPEC 625 MOTOR**



**FIGURE 6.13B: LOW NOISE FAN IMPELLER ON SPEC 625 MOTOR**

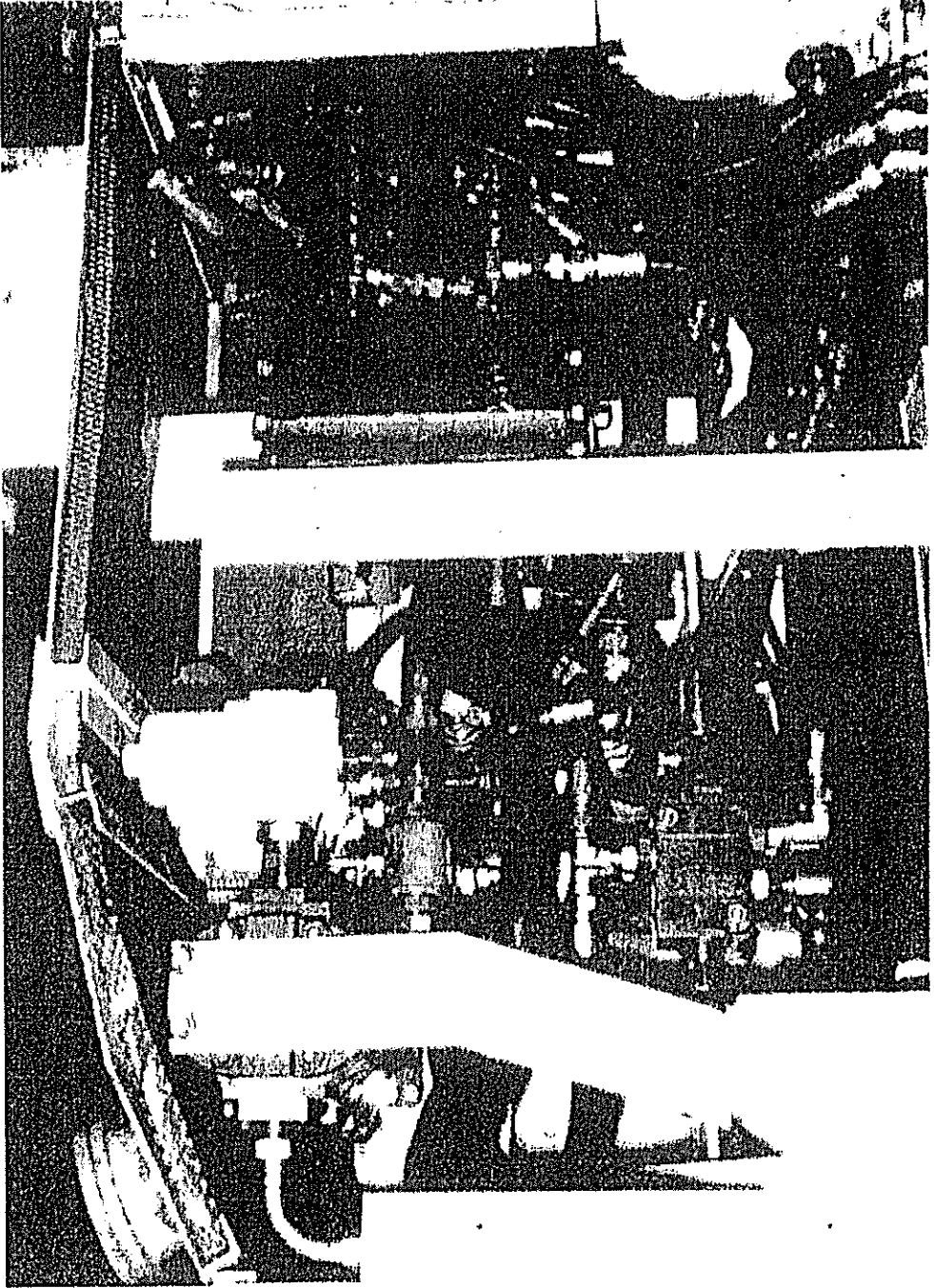


FIGURE 6.14: RH22 HYDRAULICS COMPARTMENT ENCLOSURE

## 7. Dust scrubbers and fans

For obvious reasons it is a requirement that all coal faces be ventilated adequately. With a face being worked by a mechanical miner (continuous miner or roadheader) the ventilation is normally provided by a dust scrubber mounted on the mechanical miner and additional ventilation is provided by an auxiliary fan. The dust scrubber must be operating whenever the mechanical miner is cutting.

Kielblock,<sup>(1)</sup> in collecting noise data from mining operations within the South African coal mining industry, reports personal noise exposures for continuous miner operators in South Africa of 94 dB(A). The noise levels from the scrubber are considered to be responsible for a considerable proportion of operator noise exposure.

Presently there are three types of scrubber in use in South African coal mines. The first of these is a scrubber that was originally developed by British Coal and accounts for a large number of the scrubbers in use. In this system the fan is inbye, i.e. nearest the coal face, after which the air splits over the bifurcated motor section and then into the dust scrubber box section where the dust screens and demisters are located, the cleaned air finally being discharged to the rear of the machine. In the second system, the scrubber box is upstream, with the fan located at the outbye end of the unit. Although a number of these units are still in use but this system is being replaced by a new design from the manufacturers, where the fan is located inbye of the scrubber box and an in-line motor is used.

### 7.1 Fan selection

For a specific ventilation flow-rate and pressure rise a variety of different fan types can be selected. Apart from different impeller types, the fans can be small diameter, but running at high speed, or large diameter running at a lower speed. Fan selection is generally determined by a number of factors including available space, aerodynamic performance and efficiency, and cost.

Fan characterisation by their impellers and flow gives the following types of fans:

- centrifugal - suitable for a low flow-rate to pressure ratio
- axial - suitable for a high flow-rate to pressure ratio
- mixed flow - suitable for an intermediate flow-rate to pressure ratio
- tangential - suitable for a very high flow-rate against minimal resistance.

In comparing noise from different types of fans, i.e., axial, centrifugal and mixed-flow fans Neise<sup>(2)</sup> stated that, of the fans for which he had sound power data, the centrifugal fan with backward curved blades was the quietest.

Over a fairly broad range of their performance curves axial flow fans have, on average, higher specific sound power levels than centrifugal fans with backward curved blades. This is particularly true at the respective points of optimum operation, and if the quietest ones in each of the two classes of fan are compared, the difference is between 6 to 8 dB. When the A-weighted levels are compared the difference is even greater, the axial being some 13 dB(A) higher than the centrifugal. Figure 7.1 shows in graphical form the typical approximate fan sound power level frequency spectra for axial flow fans and centrifugal fans, with the axial fans generating higher noise levels, and generally at higher frequencies. This latter feature is a disadvantage in terms of source dB(A) levels, but can be advantageous when absorptive attenuators are used for noise control, since the performance curves of the attenuators and the fans are more closely matched than in the case of centrifugal fans. A noise absorption coefficient curve for a typical mineral wool absorptive silencer is illustrated in Figure 7.2.

Generally the minimum sound power of most fans coincides with operational optimum fan efficiency and thus proper fan selection, to cover the range of duty required, is advantageous in terms of the noise level and operational costs.

Other general features associated with fan noise are (i) for a given fan, noise increases with increase in the impeller tip speed and (ii) axial fans are more sensitive to distorted inlet flow conditions than centrifugal fans. If a fan has to be situated downstream of a bend or obstruction then increasing the distance between the bend and the fan can reduce noise.

Another factor to be considered in dust scrubber fan selection is in the operational effectiveness of the fan for the job. If replacement of axial fan type with the quieter centrifugal fan type seems worthy of further consideration then a reported problem of dust build up on the fan blades of centrifugal fans should be further investigated.

Impinging slightly upon the re-design aspects later in this section, but still pertinent to fan selection, is the location of the fan within the scrubber unit, either upstream or downstream of the scrubber box. If replacement of the axial fan type by quieter centrifugal or mixed flow fan type is considered to be feasible then the fan downstream location, i.e., outbye the scrubber box, would enable centrifugal to be used since the fan is now on the *clean* side of the unit and hence dust build up on the fan blades would not be a problem. However, with the fan located downstream the scrubber screens and demisters create obstructions to the airflow upstream of the fan and are therefore likely to increase tonal noise generation by interactions between the obstructions and the fan blades. Dust capture efficiency of the scrubber is likely to be reduced with the fan downstream type of design.

## 7.2 Dust scrubber noise

A recent SIMRAC project<sup>(3)</sup> studying the dust capture effectiveness of scrubber systems also reported on noise control testing of a scrubber when in use with continuous miner and roadheader mock-ups in a surface gallery at Swadlincote, UK.

No cutting took place and measured noise levels were obtained with only the scrubber operating. Noise was measured at a number of positions, as shown in Figure 7.3 (taken from Reference (3)), the most important being a typical driver's position and a position at 7.5 metres outbye of the scrubber unit, typical of a shuttle car driver's position. For the continuous miner the driver's position was 2 metres to the right of the scrubber, near the rear of the machine. For the roadheader, the driver's position was chosen to represent an operator using remote control and was located to the left and outbye the rear of the scrubber.

Table 7.1 shows the overall noise levels of over 100 dB(A) from the scrubber noise tests. The effect of fitting an inlet absorptive silencer to the scrubber was to reduce the noise level at the continuous miner driver's position and at the shuttle car driver's position by 2 dB(A). For the roadheader tests similar reductions occurred with the inlet silencer. More

significant reductions were achieved with an outlet absorptive silencer also fitted. An absorbently lined square section outlet silencer of dimension 1.2 metres and axial length 0.9 metres was constructed and fitted to the scrubber outlet. The combined effect of inlet silencer and outlet silencer was to reduce the level at the remote machine operator's position by 8 dB(A), (104 vs. 96 dB(A)). The reduction at the typical position for a shuttle car driver was 5 dB(A), (105 vs. 100 dB(A)).

The test gallery in which the noise tests were conducted was quite reverberant, resulting in measured noise levels that were probably higher than those encountered under normal mining conditions.

The sound from the scrubber at Swadlincote contained a strong tonal characteristic of the blade passing frequency of the axial fan. With six fan blades rotating at 50 Hz a major harmonic series based on the fundamental frequency of 300 Hz was produced. This is seen in Figure 7.4 which shows the narrow band frequency spectra of the scrubber noise at the remote machine operator's position, both with and without silencers on the inlet and outlet. Attenuation of both tonal noise and broadband noise occurred but attenuation of the major tone of the blade passing frequency was only 8 dB. Nevertheless, additional reduction of the overall level from the scrubber unit needs to include further reduction of both the blade passing frequency tone and the underlying broadband noise from 500 Hz to 1500 Hz. A purpose made silencer designed to better match the frequency spectrum would produce a better attenuation. The calculations in Table 7.3 illustrate the likely noise attenuation effect of fitting manufacturers' standard absorptive cylindrical silencers to the inlet and outlet of the scrubber. Estimations of octave band and overall attenuations of silencer performance are shown for the configurations of inlet silencer with both a 1 diameter outlet silencer and a 1.5 diameter outlet silencer. The calculations are based on the noise results from the Swadlincote scrubber and cylindrical passive silencer insertion loss data obtained from manufacturer's information data sheets<sup>(4)</sup>. It can be seen that noise levels at the operator's position, although substantially reduced, remain significantly high at approximately 92/93 dB(A), even with an inlet silencer and a 1.5 diameter outlet silencer fitted.



### **7.3 Jet fan and auxiliary fan noise**

Jet fans are usually free-standing/suspended, typically of the order of 11 kW, and are positioned some distance from the working face. Likewise with auxiliary fans which range in power from approximately 4 kW up to 75 kW. They are generally less significant in terms of personal noise exposures since personnel working at the coal face are some distance from the fans. Ducted auxiliary fans can generally be located at positions where there are no personnel permanently stationed. Silencing may not be required on the ducted side if rigid ducting of more than 30 to 40 m is fitted, since noise decay occurs along the duct.

### **7.4 Noise controls achieved worldwide**

- **Fan absorptive silencers**

Fan manufacturers supply both off the shelf and custom designed fan inlet and outlet absorptive silencers.

It is calculated in Table 7.3 that effective absorptive silencers fitted to both inlet and outlet ends of a scrubber unit would reduce fan noise levels by up to 11 dB(A). The length and bulk of such silencers may, however, be an operational problem, particularly with the outlet silencer being prone to damage if protruding from the rear of the machine. Standard silencers are not efficient in reducing low frequency noise, for example an optimum lining thickness of 0.28 m is required for the scrubber fan blade passing frequency of 300 Hz discussed earlier. Attenuation can be improved by the addition of centre pods in the absorptive silencers but not without increasing pressure losses.

- **Fan reactive silencers**

The development of prototype reactive silencer elements for underground fans formed part of a European Coal and Steel Community research project<sup>(5)</sup> conducted by British Coal. Reactive silencers were studied in an effort to obviate the potential contamination by dust and water of absorptive silencers. Laboratory testing of the reactive elements

gave reasonable attenuations of 5 to 10 dB(A) at lower frequencies but no significant attenuation above 1 kHz. Further investigation of scrubber noise control by reactive silencers might yield improved benefits. The concept of reactive silencers located either within the scrubber unit casing or as a "bolt-on" silencer is worthy of further consideration.

- **Fan active noise control**

The basic concept of active noise control is the cancellation of an existing noise by the creation of an "anti-noise" of the same amplitude and frequency as the offending noise but exactly 180° out of phase.

Active noise control is now a well proven solution to many low frequency noise problems associated with fans and compressors in general industrial applications. Advantages of active attenuation are in the reduction of low frequency noise, particularly tonal noise, and in the reduced pressure drop of the system compared with absorptive silencers with splitters or centre pods. The fan blade passing frequency tonal components of the noise produced by scrubbers would seem to be the ideal target of cancellation by active noise. However, current disadvantages of active attenuation are in the length of duct required, the intrinsic safety aspect of the electronics system and the initial capital costs.

Noise attenuation of an auxiliary ventilation fan by active noise cancellation was reported by Stein & Bartholomae<sup>(6)</sup> of the U.S.B.M. Experimental tests conducted on a 10,000 cfm (4,7 m<sup>3</sup>/s) vaneaxial auxiliary fan with 16 inch outlet ducting gave some noise attenuation, particularly at low frequencies below approximately 500 Hz, and led to the conclusion that "active noise control applied to auxiliary ventilation fans should provide reductions greater than those presently obtainable through conventional noise suppression techniques, but a total noise control system will have to utilise a passive silencer to reduce noise at frequencies above 500 Hz."

Leventhall and Wise<sup>(7)</sup> describe a hybrid active/passive silencer designed to work in HVAC systems. They also quote a tonal noise reduction of 32 dB by active attenuation of a dust control fan in an industrial plant ventilation system.

- **Fan re-design/modifications**

Methods of noise control of fans are numerous and well documented in many papers, articles and books. The paper by Neise, Reference (2), referred to earlier, reviews the generation mechanisms and control methods of fan noise and lists many references addressing various aspects of fan noise control. Manufacturers should be encouraged to optimise their fan designs to acoustic principles taking advantage of the wealth of information available on this heavily researched and documented topic.

The relationship between fan speed and sound power generated by a fan can be approximately represented by the following equation quoted by Watson and Addison<sup>(8)</sup>:

$$L_{W2} - L_{W1} = 10 \log_{10} (N_2 / N_1)^n + 10 \log_{10} (D_2 / D_1)^7 \quad \text{dB}$$

where  $L_W$  = fan sound power level

$N$  = fan speed

$D$  = fan impeller diameter

$n$  = 5 to 8

If blade tip clearance can be reduced the fan efficiency is increased, hence allowing the fan to be run at a lower speed and thereby reducing the fan noise level. Thus a reduction in blade tip speed of, say, 25% would result in a reduction in fan noise of between 6 and 10 dB.

In terms of minimising blade tip clearance fan casings can be manufactured with abradable blade track linings.

A European Coal and Steel Community research project<sup>(9)</sup>, re-designed the impeller of a fan used on cooling systems of underground locomotives and trackless vehicles within British Coal. A new non-metallic fan was designed, with aerofoil section blading, to improve the aerodynamic efficiency and hence enable noise to be reduced by lowering the fan speed. Tests on a prototype fan, running at 1600 rpm compared to the conventional 3000 rpm gave noise level reductions of 12 dB(A). However, due to the rapid deployment of trackless vehicles in the industry it became necessary for any new fan to be retrofittable within the existing cooling circuit. Another prototype fan was designed and

manufactured, which gave noise results which were no better than a standard metal bladed fan<sup>(6)</sup>. A basic problem of tonal blade passing noise occurs when fans have guide vanes and/or straighteners. The closer the vanes are to the rotor the higher the noise level that is produced, due to interaction of the downstream blades with turbulence from upstream ones. This becomes severe at spacings of approximately 2x blade chord length and generally a spacing of 5x blade chord is desired. Few axial or mixed-flow fans are now designed with inlet guide vanes.

With regard to axial fans, there are a number of design considerations which can lead to reduced noise exposure. Guide vanes downstream of the impeller improve the pressure development and efficiency but, as discussed earlier, can also have a detrimental effect on noise levels. The design and positioning of downstream guide vanes should be examined for reduced noise potential. Bifurcated fans are very inefficient. Possible development of an external motor, out of the airstream, and incorporating a right-angle gearbox drive would be worthy of consideration. The gearbox could also be used to vary the fan speed, which at present is defined by either the two-pole or four-pole motor speed.

Burrell and Stout<sup>(11)</sup>, in a review of the historical development of fans for auxiliary ventilation in British coal mines discuss a proposed specification for a new fan to replace the current, relatively inefficient, fans. It is stated that for the majority of driveages, now using forcing systems, bifurcated fans are unnecessary because of the absence of high concentrations of dust and water. The use of straight through fans with aerofoil blades and impeller clearances of 1.5 mm (compared with previous clearance specification of 2.5 mm) significantly increases the efficiency of fans. An indication of the improvements which can be gained by using such fans is shown in Figure 7.5 (taken from Reference (11)). This shows the characteristics of a commercially available 80 kW mixed flow fan together with the standard NCB 90 kW fan, both fitted with inlet and outlet guards. Burrell and Stout calculated that the mixed flow fan could deliver a required airflow to a greater distance and at a much greater efficiency (80% vs. 40%) compared to the current 90 kW fan. To maximise efficiency the fan capacity control should be by variable speed or variable vane angle. As discussed earlier, greater fan efficiency would allow lower operational speed and hence produce lower noise levels.

- **Fan/scrubber casing treatment**

A noise control option used in South Africa is a *silenced casing* type of jet fan described in a paper by Langmore<sup>(12)</sup>. This reports a measured attenuation on a prototype 572/18,5 kW unit of 12 dB(A) (92 vs. 104 dB(A)) at "one metre in the air system with a nozzle velocity of 46,5 m/s." The design incorporated an absorbent lining and perforated plate, along the lines of an "integral silencer", which formed the fan casing. Dimensionally the silenced casing fan was 20% longer than the standard fan.

The same manufacturer also offers a silenced casing as an option for auxiliary ventilation fans.

Assessment of the contribution of the dust scrubber box casing and internal regeneration noise from the screens and demisters for potential noise reduction should be conducted to evaluate this for retrofit treatment and incorporation into any re-design of new scrubbers.

## **7.5 Recommendations for improving South African position**

- **Retrofit of existing equipment**

Well designed acoustic silencers on the dust scrubber inlet and outlet would reduce the scrubber noise. Absorptive silencers for the full range of fans/scrubbers are available from most manufacturers. These can provide reductions of, typically, 11 dB(A), depending on the size of fan. However, the operational practicality of incorporating an outlet silencer may not be straightforward due to the additional length and bulk at the rear of the machine.

According to manufacturer's data a silenced casing fan can provide up to 12 dB(A) reduction compared with a conventional fan.

Assessment of the contribution of the dust scrubber casing and internal regeneration noise from the screens and demisters for potential noise reduction should be conducted

to evaluate this for retrofit treatment and incorporation into any re-design of new scrubbers.

### • Re-design options/new equipment

Attention should be paid to reducing the noise at source, possibly by re-design of the impeller and consideration of fan tip speed and tip clearance. Selection of fan type should be reviewed to maximise efficiency and minimise fan noise radiation.

Manufacturers should employ acoustic re-design principles for low noise fans and dust scrubbers. As with retro-fit of existing equipment, new fan units should be examined on noise performance principles in addition to airflow performance.

The development of active noise control is continuing rapidly and is diversifying to smaller applications. Currently some small systems are available for as low as R3000 (Digisonix<sup>(13)</sup>) but costs depend on numbers of units and complexity of system. It would appear that this method of noise control will need further research and development before being applicable to such arduous conditions as underground mining.

As discussed earlier, a fan employing a "silenced casing" type of design is available, which claims a 12 dB(A) improvement on the conventional design.

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POSITION	OVERALL NOISE LEVEL IN dB(A)					BEST REDUCTION ACHIEVED ie
	UNTREATED	INLET SILENCER	INLET SILENCER & OUTLET SILENCER	OUTLET DEFLECTOR	INLET DUCTING & OUTLET DEFLECTOR	
1	104	103	96	106	105	8
2	104.5	103.1	99.5	105.2	105	5
3	107.2	101.1	97.4	107.1	106.2	10
4	103.3	101	96.8	104.2	105.7	6
5	108.7	108	107.2	111.3	109.9	2
6	107.8	106.9	104.7	108.4	108.3	3
7	103.4	99.4	96.3	102.9	102.6	7
8	108	98.9	97	108.9	103.1	11
9	107.7	98.3	96.1	108.3	101.9	12

POSITION 1 IS TYPICAL (REMOTE CONTROL) MACHINE OPERATOR POSITION  
POSITION 2 IS TYPICAL SHUTTLE CAR DRIVER POSITION

TABLE 7.1: MEASURED SCRUBBER NOISE LEVELS FROM TESTS ON A MOCK ROADHEADER AT SWADLINCOTE TEST SITETABLE



SWADLINCOTE SCRUBBER - MEASURED LEVELS IN SIMRAC GALLERY

POSITION 1 - TYPICAL MACHINE OPERATOR (REMOTE CONTROL) POSITION

SCRUBBER WITH INLET SILENCER

	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
Resultant Noise Level dB(A)	50	64	98	92	96	94	87	78	103

SCRUBBER WITH INLET SILENCER  
+ SQUARE SILENCER ON OUTLET

	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	75	78	98	89	91	88	83	74	
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
Resultant Noise Level dB(A)	49	62	89	86	91	89	84	73	96

TABLE 7.2: MEASURED SCRUBBER NOISE LEVELS  
AT TYPICAL MACHINE OPERATOR POSITION

ENGART SILENCERS NOISE DATA FROM MANUFACTURER'S INFORMATION SHEET

SCRUBBER WITH INLET SILENCER (MEASURED LEVELS AT SWADLINCOTE)  
 POSITION 1 - TYPICAL MACHINE OPERATOR (REMOTE CONTROL) POSITION

CYLINDRICAL SILENCER - SIZE 762 ON OUTLET

1D SILENCER ON OUTLET	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	103
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
A-weighted Level	50	64	98	92	96	94	87	78	
Dynamic Insertion Loss 1D *	-4	-3	-9	-17	-14	-12	-13	-14	
Noise Contribution From Outlet	46	61	89	75	82	82	74	64	
Estimated Noise Contribution From Inlet									89
Resultant Noise Level at Position 1									93

1.5D SILENCER ON OUTLET	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	103
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
A-weighted Level	50	64	98	92	96	94	87	78	
Dynamic Insertion Loss 1.5D *	-5	-5	-11	-20	-17	-15	-15	-17	
Noise Contribution From Outlet	44	56	86	73	81	83	77	69	
Estimated Noise Contribution From Inlet									89
Resultant Noise Level at Position 1									92

\* Silencer length 1D = 1 diameter, 1.5D = 1.5 diameters

TABLE 7.3: ESTIMATIONS OF SCRUBBER NOISE LEVELS USING  
 ENGART OUTLET SILENCER

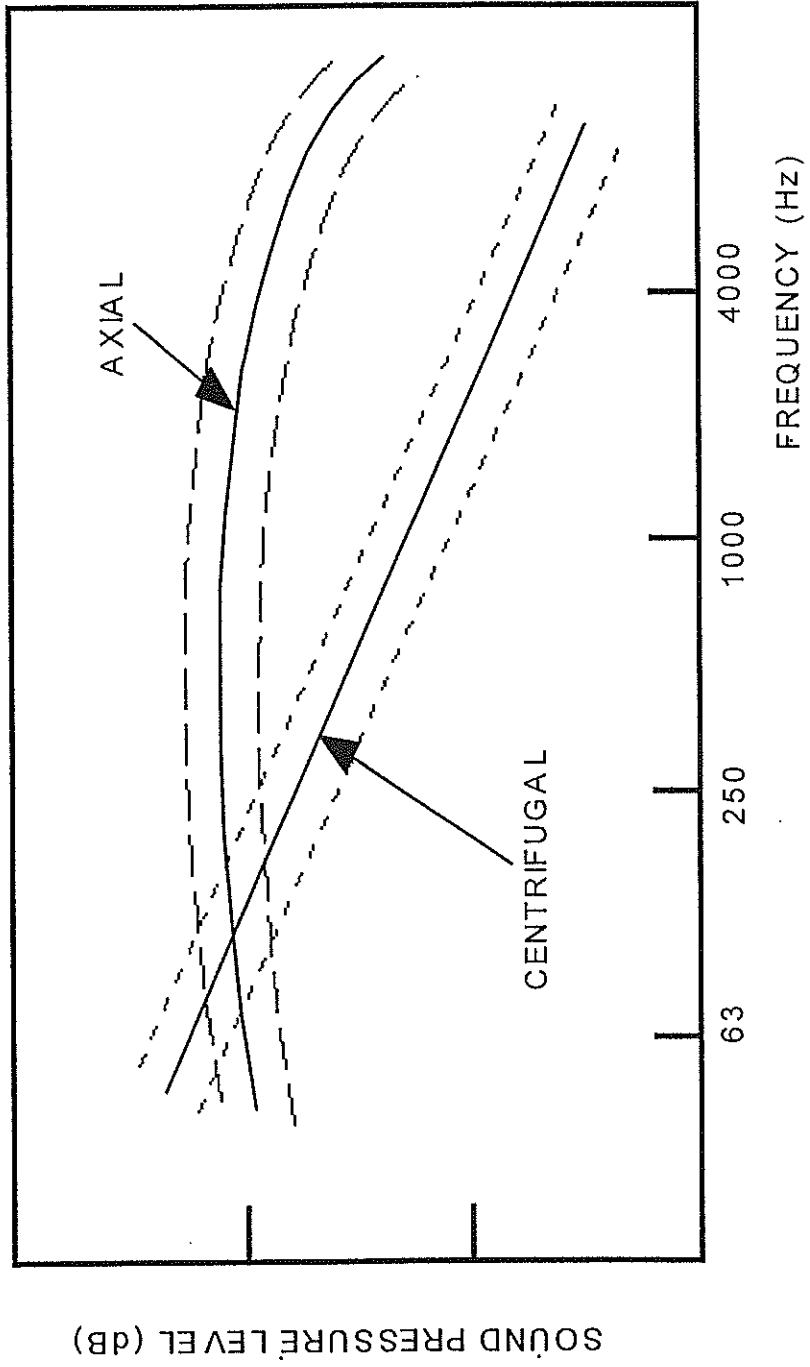


FIGURE 7.1: SOUND POWER SPECTRA OF FANS

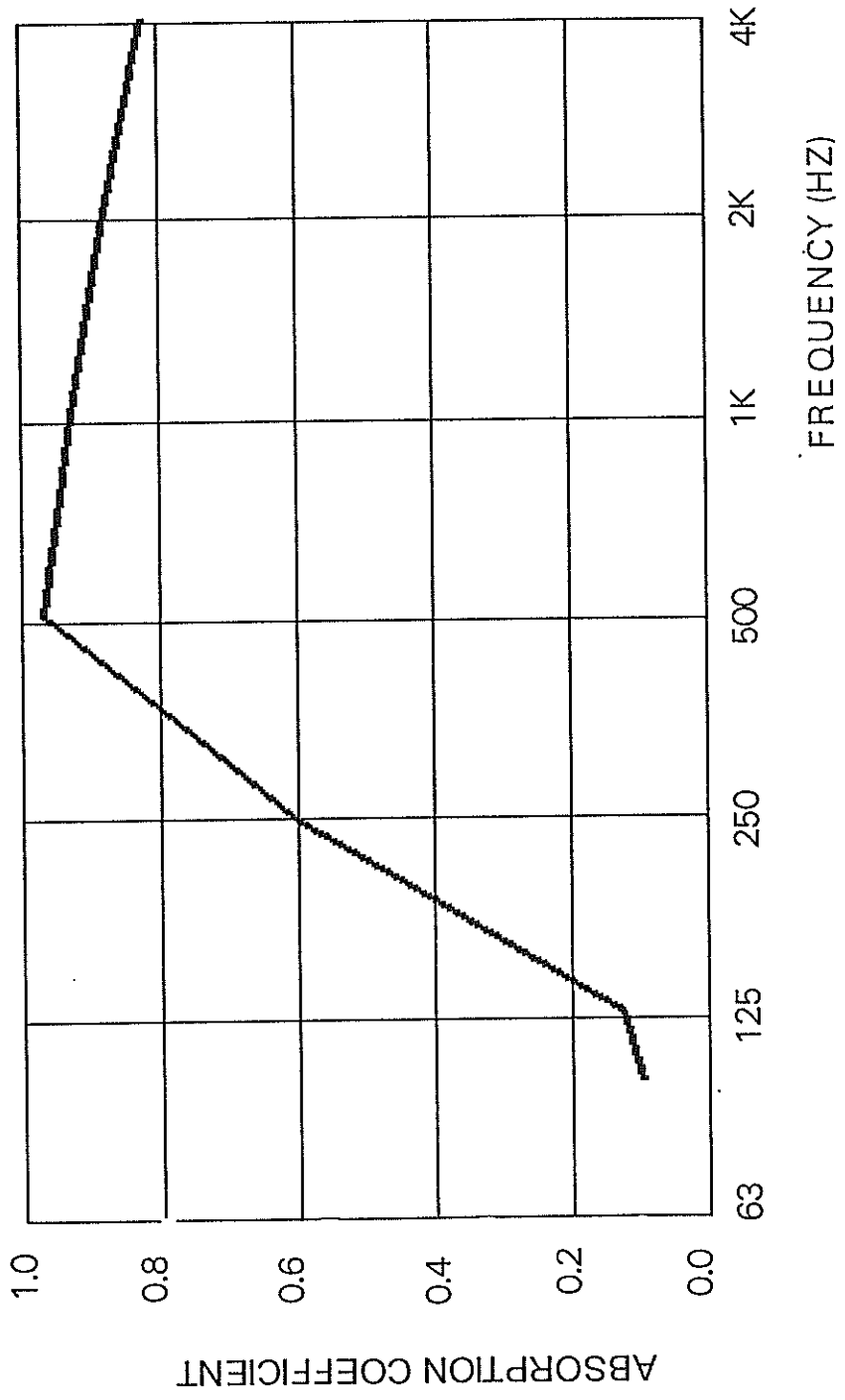
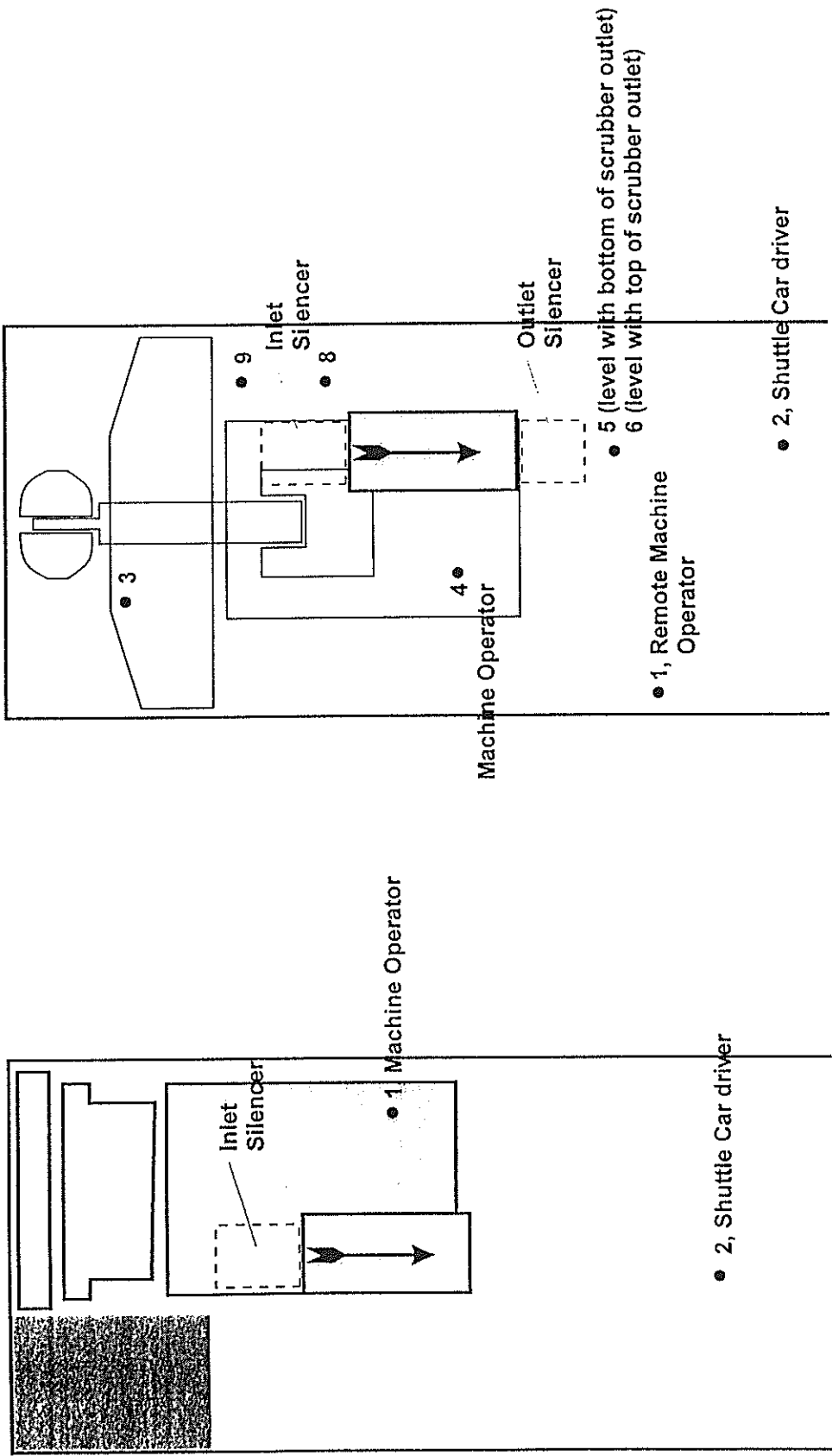


FIGURE 7.2: TYPICAL NOISE ABSORPTION - MINERAL WOOL



a. Continuous Miner

b. Roadheader

FIGURE 7.3: LAYOUT OF THE SCRUBBER TESTS AT SWADLINCOTE SHOWING NOISE MEASUREMENT POSITIONS

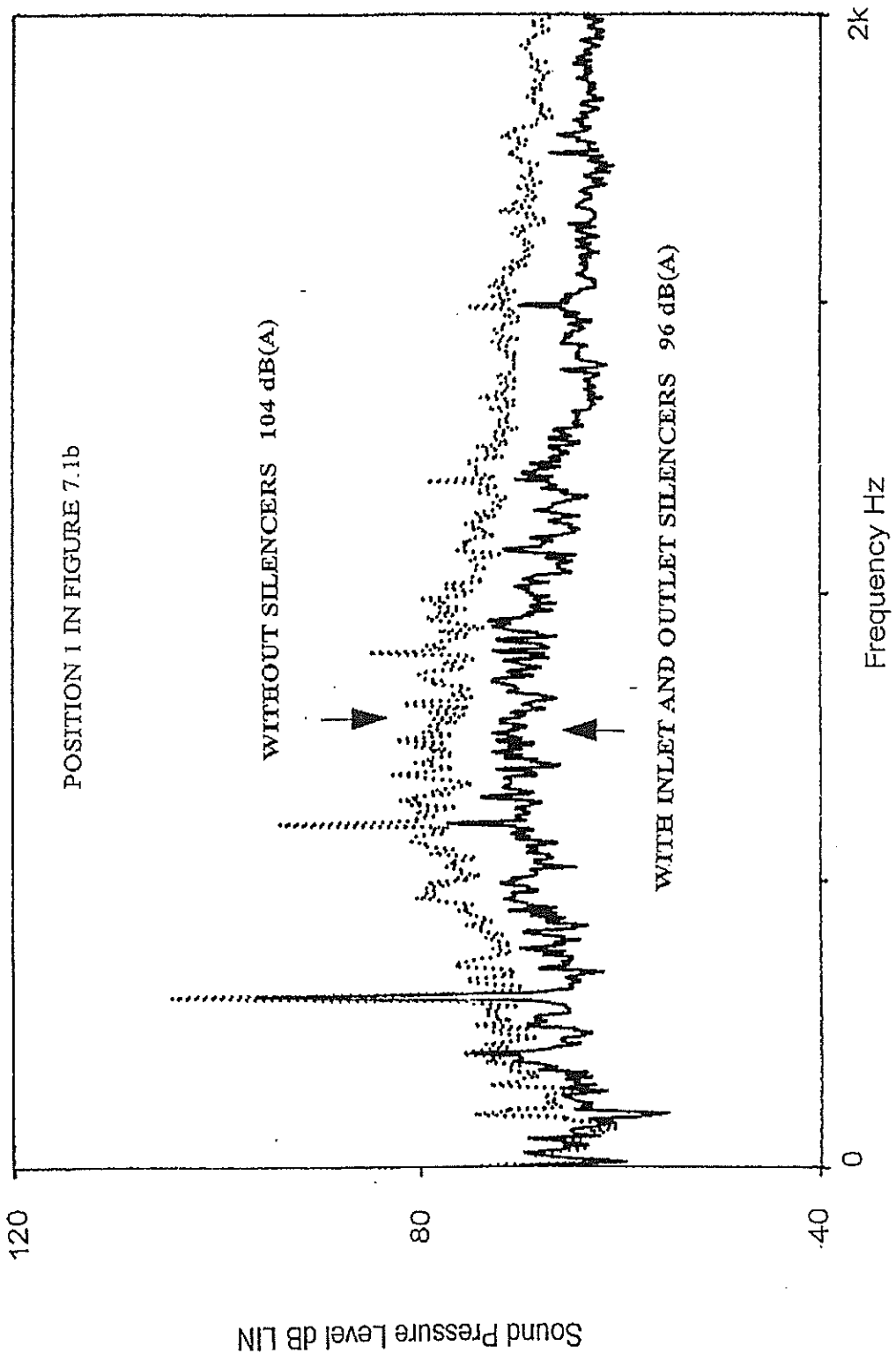


FIGURE 7.4: SCRUBBER NOISE FREQUENCY SPECTRA - WITH AND WITHOUT SILENCERS

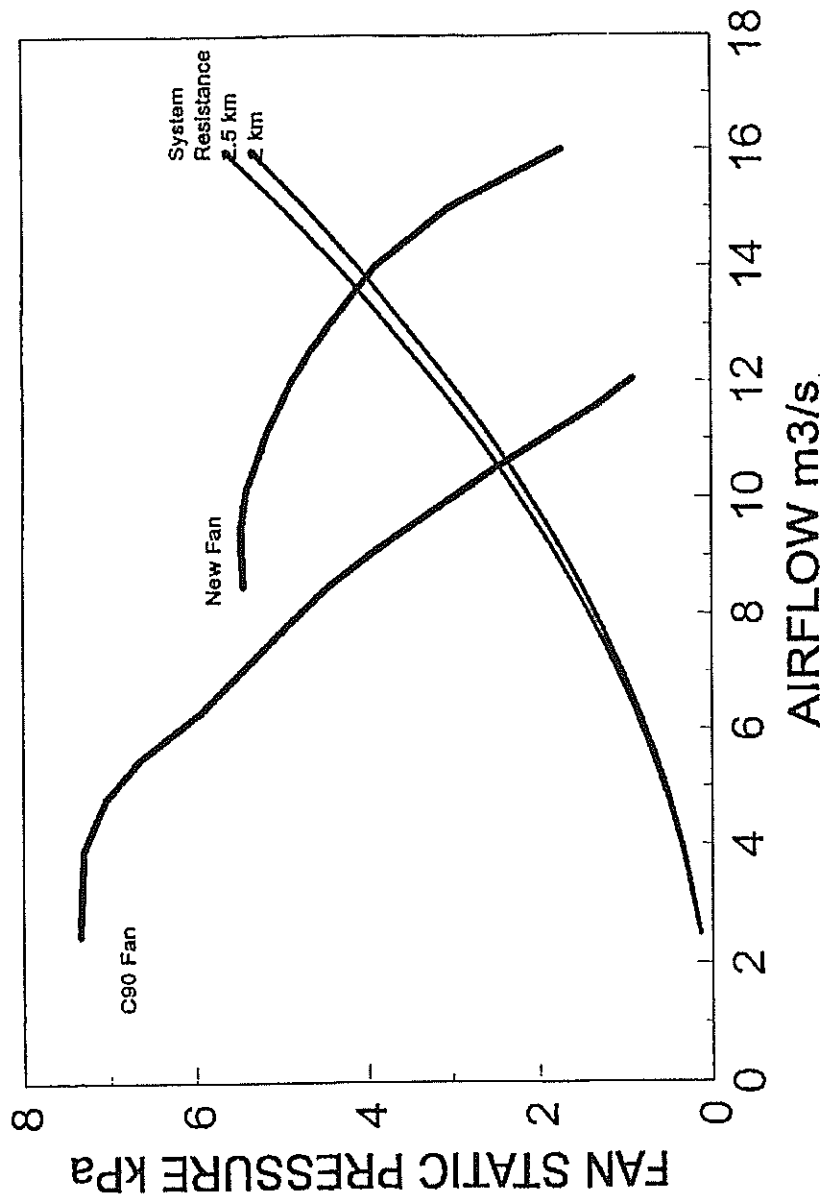


FIGURE 7.5: AUXILIARY FAN CHARACTERISTICS C90 AND NEW FAN

## 8. Longwall systems

There are several major sources of noise in a longwall district. On the face itself, the armoured face conveyor (acf) and the shearer (or shearers) are the main noise sources. Noise levels at the shearer operator's position have been measured at 100 - 105 dB(A). The noise levels on the face due to the operation of the armoured face conveyor vary according to the type and size of conveyor, and whether it is running empty or full. Noise levels at the mid face position, 1 m from the conveyor, have been measured at 90 - 105 dB(A) without coal, and 80 - 90 dB(A) with coal.

The situation at face-ends is complex because of the congested space and concentration of machinery. The resultant noise level is made up of noise from armoured face conveyor drives, the stage loader, which is similar in character to the armoured face conveyor, crushing equipment and other ancillary equipment. Noise levels at a range of positions in the face end have been measured at 90 - 102 dB(A), depending on the amount of activity.

Extensive work has been carried out, both in the U.S. by U.S.B.M. and in the U.K. by British Coal, into characterising the sources of noise on longwall faces and developing methods of reducing noise levels<sup>(1)-(5)</sup>. Some of these methods are novel and show promise for effective reductions in noise levels on longwall faces, although, up to now, application in the field has been limited.

### 8.1 Armoured face conveyors

The main sources of noise on armoured face conveyors are;

- drive head gear noise
- chain squeal
- impact of flights and chain on line pan joints

Drive head noise is mainly from the meshing of gears and varies little with loading. The noise is affected strongly by pitch errors in gears and precision quality gears. Ground or shaved profile gears are 2 - 3 dB(A) quieter than hobbed or milled gears.

Away from the drive head, the major source of noise is impact of flight bars on the line pan joints. On an empty conveyor, this increases with flight and chain mass, conveyor



speed, degree of misalignment (snaking or undulation) and reduces slightly with pan stiffness and weight. A summary of British Coal data for a range of chain and flight arrangements is given in Table 8.1., which shows the results of indoor surface tests with and without coal. Levels underground would be 3 - 4 dB(A) lower. The following conclusions can be drawn;-

- the lightest chain and flight for the duty should be used
- outboard chain is quieter than inboard chain
- the heavier 222 mm pan section is quieter than the earlier 190 mm type
- chain speed should be the slowest compatible with effective coal clearance

British Coal also carried out work to introduce damping to line pans. Tredomen 222 line pans were manufactured with a laminated base construction, having a 20 mm thick plate on top of a 5 mm thick plate welded along the edges. In laboratory impact response measurements, the resonant frequency peaks above 500 Hz were all effectively reduced and a total overall noise reduction of 8 dB(A) was predicted. Subsequent surface tests with a line of fifteen pans showed actual reductions of 6 dB(A) without spill-plates and 4 - 5 dB(A) with spill-plates when running empty.

## **8.2 Longwall shearers**

In contrast to the armoured face conveyor, shearers are noisier when under load. The primary noise sources are cutting noise, motor noise, gearing and hydraulics and haulage noise on chain hauled machines. Cutting noise tends to be the dominant source, although the other sources can contribute to the exposure of the operator.

The process of breaking coal from the face with rotating picks is inherently noisy. As detailed in section 6, a large amount of work has been carried out to define the parameters which influence cutting noise and to assess the possibilities for reduction of cutting noise levels. It was found that three separate noise generating mechanisms were associated with the coal cutting process;-

- machine structure radiated noise, caused by the vibrational response of the shearer to the cutting forces and to a lesser extent by
- fracture noise, caused by air rushing into the cracks produced when the coal fractures
- face radiated noise, caused by the vibrational response of the coal face to the cutting forces

Laboratory investigations and field studies, carried out by U.S.B.M. have confirmed that machine radiated noise was the major contributor to the noise at the operator's position<sup>(2),(3),(4)</sup>. The cutting head was the machine structure that radiated most noise and it was concluded that a significant reduction in noise level could be achieved by altering the vibrational response of this component.

These investigations were confirmed by work done in the U.K. by British Coal<sup>(5)</sup>. A surface rig was constructed which consisted of a shearer moving on a length of armoured face conveyor and cutting blocks of aggregate material. An array of microphones around the machine picked up the cutting noise and recorded it on multi-channel tape recorders. The rig was used to quantify the contributions of the various sources to the overall noise level. In addition, extensive lagging work was carried out with the cutting head. Separately lagging the individual components of the cutting head enabled the contributions to the noise levels of the vanes, barrel and face ring to be quantified. Fully lagging the cutting head gave reductions of 7 - 8 dB(A) at the operator's position over the unlagged condition.

The U.S.B.M. work culminated in the development of a new design for the cutting head. A typical shearer cutting head is shown in Figure 8.1. The vanes, face ring and barrel have natural frequencies in the range below 200 Hz, and are constructed of poorly damped steel plate, making them very responsive to the low frequency forces of coal cutting, and with large areas to radiate noise. In the new design the stiffness of the vanes and face ring was increased to raise their lower natural frequencies above the region where coal cutting forces are strongest. Also, the damping of the barrel was increased as much as possible since it was not amenable to stiffness changes.

The standard barrel was covered in a 6.4 mm sheet of elastomer, on top of which was a 25.4 mm steel plate, providing a constrained layer damping treatment. This had a negligible effect on the conveying volume (the open volume between the vanes and the barrel).

The vanes and face ring were removed altogether and each bit block was mounted on its own massive, rigid stand, thereby eliminating the most significant noise radiating structure. A view of the new design is shown in Figure 8.2 (taken from reference (4)).

Underground noise measurements showed that the redesigned cutting head gave a 5 - 6 dB(A) reduction over a conventional head at the operator's position. Noise levels were approximately 95 dB(A) with the redesigned head against 100 - 101 dB(A) with the conventional head.

Initially, because of doubts about the conveying performance of the new cutting head, a 25 mm polymer sheet was bolted to the bit stands to replace the absent vanes. However, underground tests showed this to be unnecessary. The large size and tapered shape of the bit stands provided adequate conveying surface. The overall performance of the redesigned cutting head was good. It operated underground for seven months, during which time it extracted over 425,000 tonnes of coal without failure.

The control of mechanical noise from haulages and hydraulic drives is a matter of good design. Precision quality helical gears with ground profiles should be used rather than straight spurs. Bearing housings should be isolated from outer casings and hydraulic machinery and pipes should be isolated from the large radiating surfaces of casings. Close fitting shields, which reduce noise radiation from machine casing surfaces, can give up to 5 dB(A) reduction for existing equipment, without interfering unduly with robustness, heat dissipation or compactness. The use of tracked haulage is also beneficial since it eliminates the clatter of haulage chain on conveyor flight bars on undulating faces.

### **8.3 Face ends**

The noise control principles that apply to armoured face conveyors and continuous miner conveyors apply equally to stage loaders. Damping and isolation techniques can be applied to the stage loader to reduce the vibrational energy being put into the structure. In addition, damped steel cover plates can be used over the length of the stage loader, provided that large lumps of coal are excluded, which would be the case if a crusher/breaker was being used. Surface tests with temporary covers on a stage loader running empty gave noise reductions of 3 - 4 dB(A)<sup>(5)</sup>.

Underground vibration and noise measurements on crusher/breakers identified the main radiating panels. For one breaker, the vibration data was processed to calculate the contributions of the individual panels to the total noise at positions close to the breaker. A schematic of the breaker is shown in Figure 8.3, with the individual panels numbered B1 through B8. The results of the processed vibration data are shown in Figure 8.4, which

shows calculated octave band noise levels and the measured octave band noise levels. Correlation between the calculated and measured levels was good. The large side panel B2 was the major contributor.

Temporary lagging of the breaker gave reductions of 3 - 4 dB(A) at positions close to the breaker<sup>(5)</sup>. The lagging material was a mineral wool/thin lead sheet sandwich encased in brattice cloth. This "acoustic quilt" product was extremely versatile and had approval for a number of noise control applications underground in coal mines, where there are restrictions on the use of non-metallic materials.

For some static and semi-static machinery, such as haulages, winches and transfer points, the use of an enclosure can provide cost effective noise control of the order of 10 - 20 dB attenuation. The enclosure should preferably be of heavy construction (e.g. metal sheet) and lined with acoustic absorption material to prevent the build-up of internal noise. When openings in the enclosure are necessary for cables, haulage ropes etc., these should enter through acoustically lined ducts, as shown in Figure 8.5. In addition to their use in coal mining, small enclosures can be used in the Gold and Platinum mining industries to reduce the noise from the large numbers of winches employed underground.

Where enclosure is not possible, for reasons of safety or limited space, partial barriers can be very effective in providing up to 10 dB protection for an operator who has to work in close proximity to a noisy machine. Ideally, the barrier should be of heavy construction, and lined with absorbent material on the side facing the noise source. As mentioned previously, the acoustic absorbent should be protected from contamination and damage with a PVC or polyester membrane and a perforated metal cover, earthed to dissipate static electricity. Figure 8.6 shows an example of a barrier in use underground, installed to provide noise reduction for the operator of a rope haulage engine.

Strategic siting of equipment, within the constraints of operational efficiency, can have beneficial effects in terms of reduced noise levels. There is sometimes sufficient flexibility to move some equipment, such as hydraulic power packs for roof supports, further up the gate away from the main work area, and so reduce this contribution to the overall noise. Also, the siting of work stations/control panels can have a significant effect on the noise exposure of some face-end workers. This is illustrated in Figures 8.7 and 8.8, which show face end layouts for two mines within British Coal. SITE 1 employed a crusher, the stage loader was equipped with steel covers and the stage loader drive was of the in-line type. Measured noise levels along the gate ranged from 89 to 95 dB(A), the maximum level

being in the vicinity of the crusher. The stage loader motor/drive were relatively quiet with noise levels of 89 - 90 dB(A) at 1 m distance.

SITE 2 was also equipped with steel covers on the stage loader, with gate end boxes mounted over these. The stage loader motor and drive were of the standard type. Measured noise levels along the gate ranged from 82 to 97 dB(A), the maximum level being in the vicinity of the stage loader drive and the minimum level being in the vicinity of the control panel.

## **8.4 Conclusions and recommendations**

Mechanical noise on longwall shearers can be reduced by the specification of precision helical gears and isolated bearing housings. Where possible, the hydraulics should be isolated from the main radiating surfaces of the machine. Enclosure of the hydraulics should be implemented.

The potential of low noise shearer cutting heads should be investigated. The cutting head developed by U.S.B.M. showed real promise and operated underground successfully for over seven months. Operator noise levels were reduced by 5 - 6 dB(A) over a standard cutting head.

The more detailed work in this area being carried out by British Coal had reached the stage where a prototype design was being drawn up for field testing when British Coal Funding ceased due to the impending privatisation of the industry. At present IMCL are proposing to continue this work under a EU collaborative project on underground machinery noise.

The use of remote control on face machines should be investigated. This gives the possibility of moving the operator further away from the noise sources on the machine.

Good design practice for armoured face conveyors should reduce, as much as possible, discontinuities which cause impact noise. Damped line pans can give significant noise reductions up to 5 dB(A) and their fabrication is simple and straightforward. For the drivehead, use of precision quality gears and close shielding of machine casings can give noise reductions from this source of 2 - 3 dB(A).

As with the armoured face conveyor, damping and isolation treatments can be implemented on stage loaders at the design and manufacture stage, more details of which are given in Section 6 on Continuous Miners, where scraper conveyor design is considered. Covers for stage loaders are available from suppliers, or can be manufactured in-house.

## 8.5 Retrofit summary

• Enclosure of hydraulics on face machine	4 - 5 dB(A) reduction	±R10000
• Minimise impact points and damp line pans on face conveyor	4 - 5 dB(A) reduction	±R50000
• Covers for stage loader	0 - 4 dB(A) reduction	±R10000
• Reduce drop height at transfer points	not established	----
• Close shielding for crusher/breaker	3 - 4 dB(A) reduction	±R10000
• Where possible, remove noisy equipment from the main work area	not established	----
• Enclosures for haulages/winches	10 - 20 dB(A)	±R10000
• Barriers to shield noisy equipment	5 - 10 dB(A)	±R10000

## 8.6 References

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FIGURE 8.1: STANDARD SHEARER CUTTING HEAD

TABLE 8.1: NOISE FROM AFC LINE PANS

CHAIN SIZE (MM)	PAN TYPE	CONFIGURATION	NOISE LEVEL (MID FACE) dB(A) AT 1 METRE	
			NO COAL	COAL
19	190MM IIF	TWIN OUTBOARD	91-95	84-94
22	222MM	TWIN OUTBOARD	92	84
22	222MM	TWIN OUTBOARD (CONT)	95	84
22	222MM TREDOMEN	TWIN INBOARD	94	-
22	222MM (OTHER)	TWIN INBOARD	103	93
26	222MM TYPE A	SINGLE CENTRE STRAND	98	89
26	222MM TYPE B	SINGLE CENTRE STRAND	101-103	90-93
26	222MM TYPE C	SINGLE CENTRE STRAND	109	-





FIGURE 8.2: REDESIGNED SHEARER CUTTING HEAD

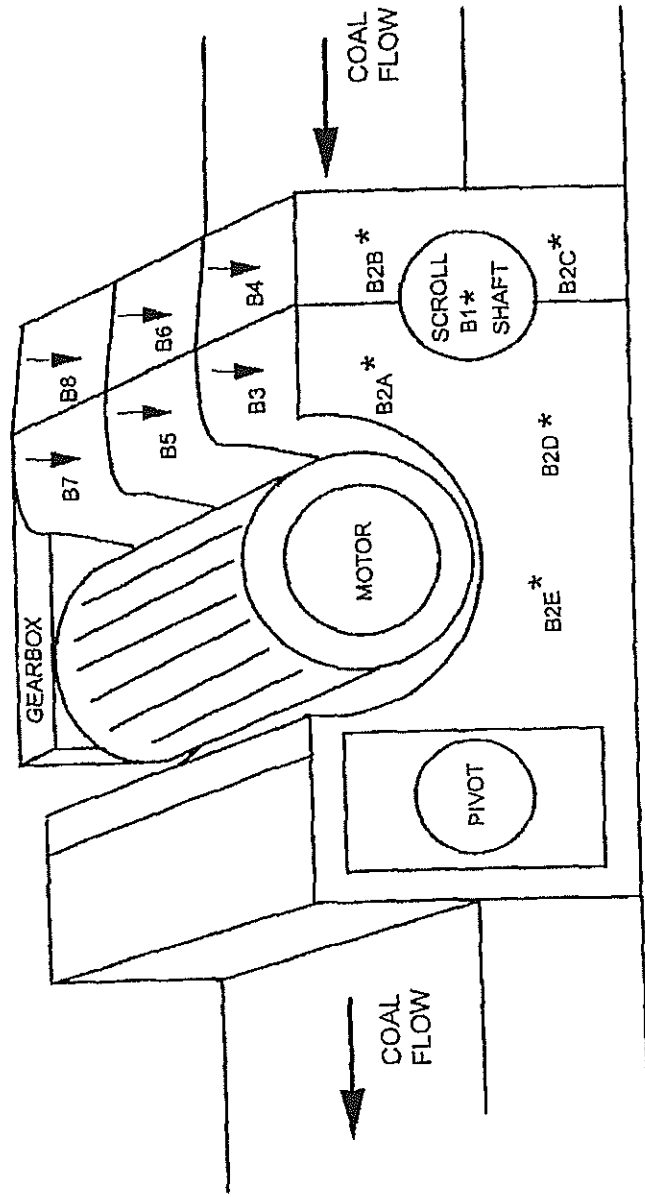


FIGURE 8.3: SCHEMATIC OF BREAKER SHOWING VIBRATION MEASUREMENT POSITIONS

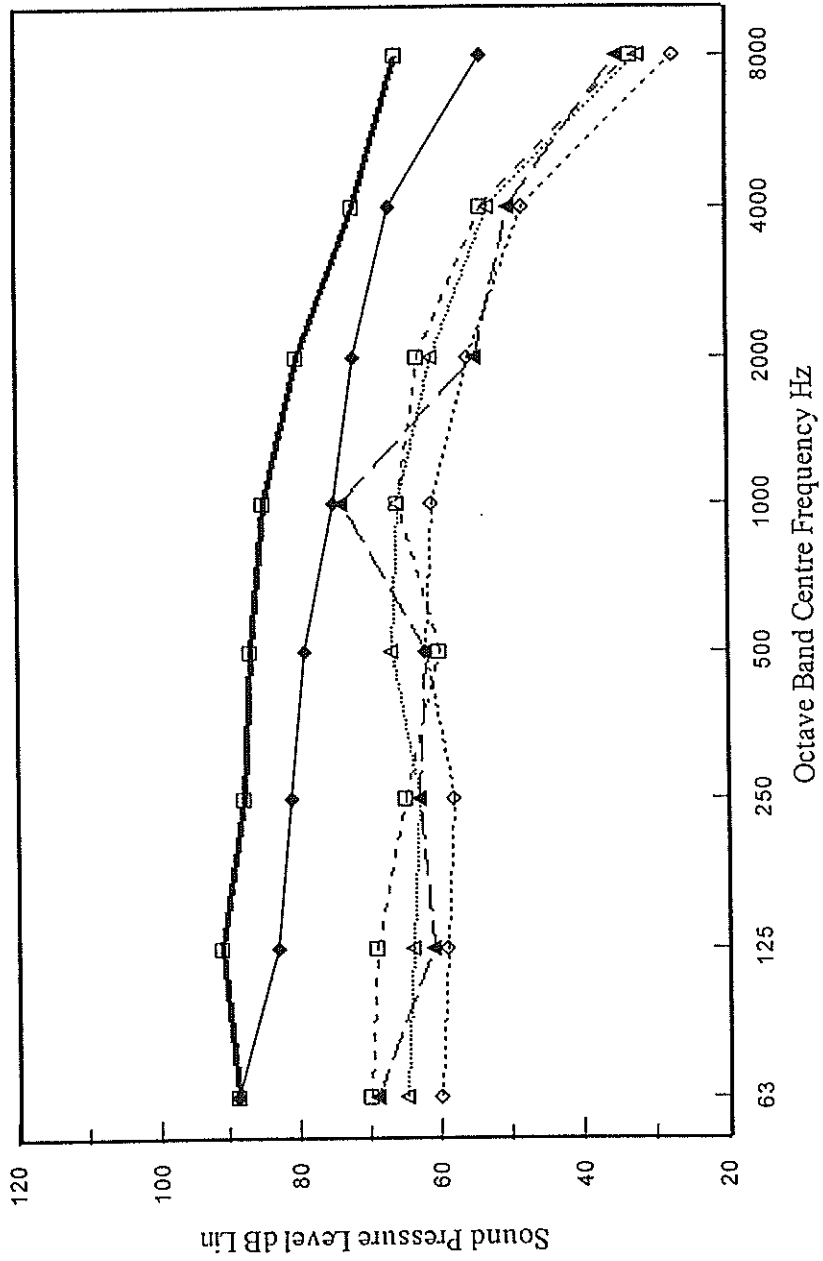


FIGURE 8.4: BREAKER MEASURED AND CALCULATED NOISE LEVELS

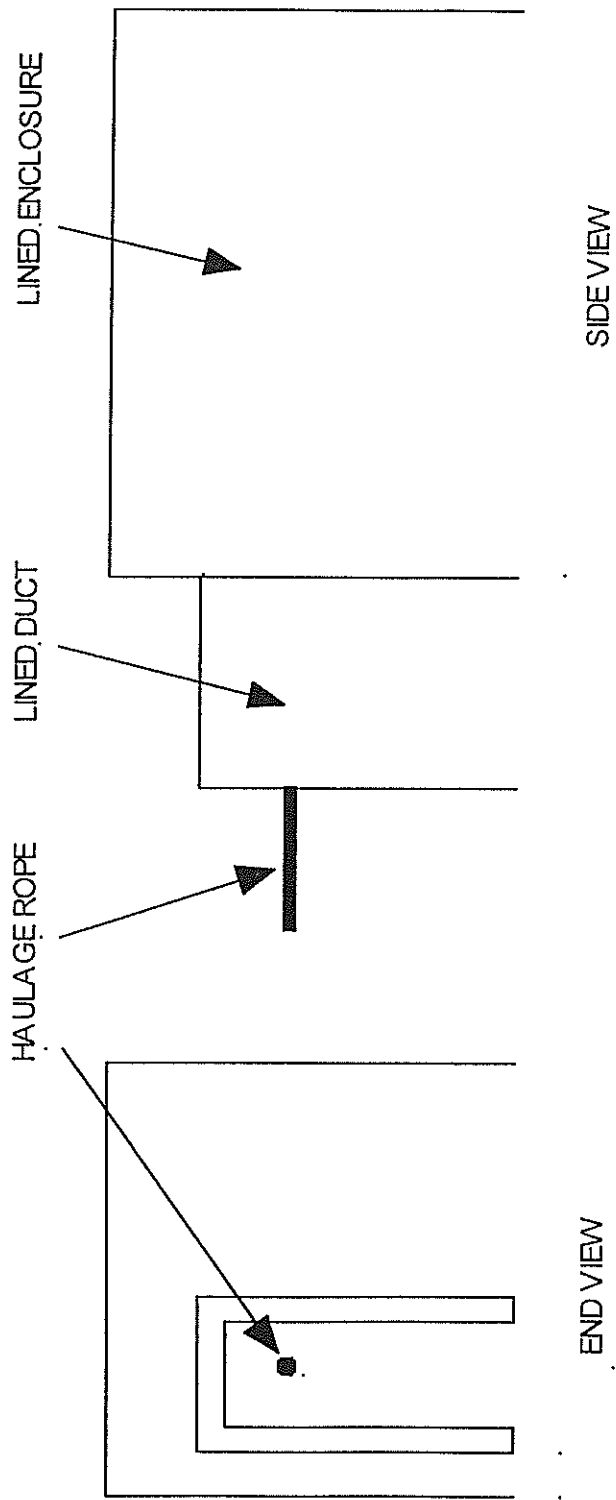


FIGURE 8.5: SIMPLE ENCLOSURE FOR HAULAGE OR WINCH



FIGURE 8.6: NOISE BARRIER IN UNDERGROUND INSTALLATION

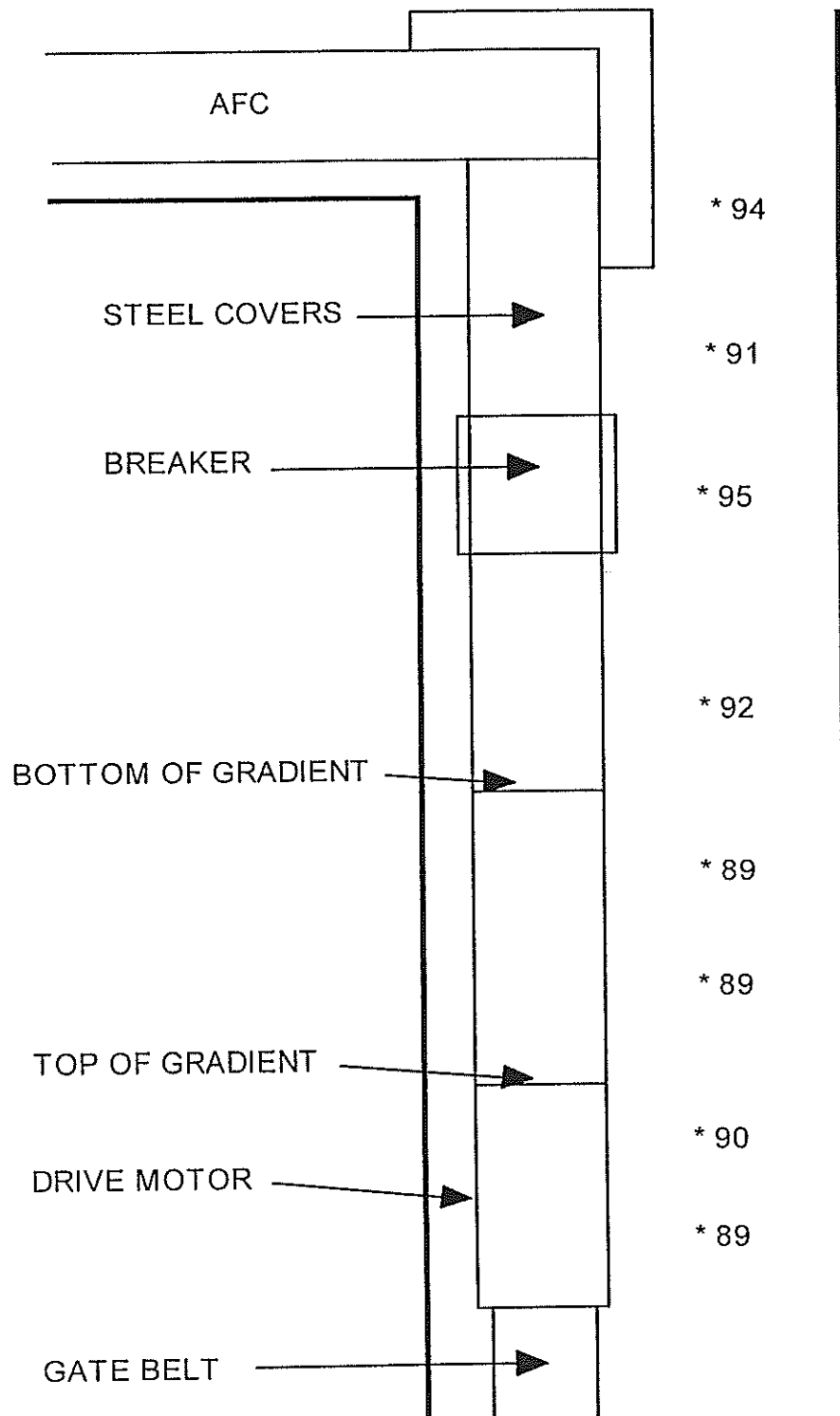


FIGURE 8.7: EXAMPLE FACE END LAYOUT/SITE 1

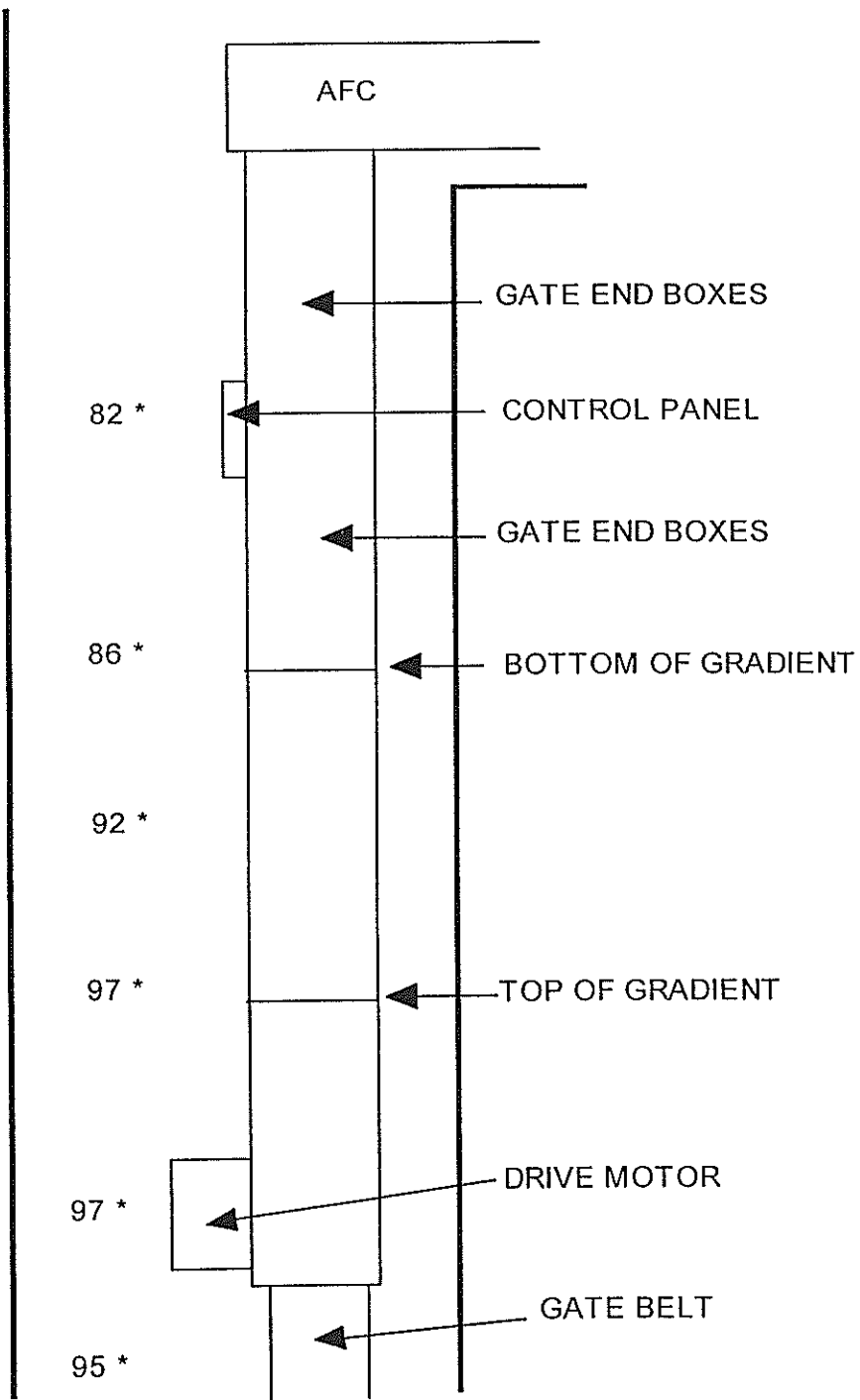


FIGURE 8.8: EXAMPLE FACE END LAYOUT/SITE 2

## 9. Trackless vehicles

Extensive measurements of noise from diesel powered mining equipment have shown levels at the operator's position of over 100 dB(A), with corresponding mean noise exposure levels of 99 dB(A). In this type of equipment, the engine generally constitutes the major source of noise. The engine noise comes from the exhaust, the intake and the casing. Also, the cooling fan can be a significant source of noise, as can be the transmission, drive train and hydraulic system. Noise radiated from the various sources reaches the operator by propagation through the air (directly and with reflections) and, also, vibrations produced by the engine and other mechanical components travel through the vehicle frame to lighter structures, which radiate noise.

The application of standard noise control techniques is very effective in quietening this type of equipment, as has been demonstrated in a number of studies in recent years.<sup>(1)-(5)</sup> Practical and economic considerations generally do not permit modification to the primary noise sources, or to replace them with quieter ones (except in the development of new machines). Consequently, practical noise reduction requires shielding of the operator from the propagation paths of the noise.

Full enclosures are the best means of blocking the radiation of noise from engines and transmissions. The effectiveness of an enclosure increases with the surface density of its walls and can be enhanced if the interior surfaces of the enclosure are faced with acoustically absorptive material. Partial enclosures or barriers can be used where cooling and/or access requirements do not allow full enclosure. These are considerably less effective because of sound propagating out of the openings, although their performance can be enhanced by lining with acoustically absorptive material.

Reactive silencers, or mufflers, are used to reduce engine exhaust noise. A muffler must be matched to a particular engine so that it is acoustically effective but does not produce excessive back pressure. Vehicles approved for use in coal mines often have wet exhaust conditioners which effectively silence the exhaust noise. Absorptive silencers are used to reduce noise from air inlets and outlets. They consist of acoustically lined ducts or louvres faced with acoustically absorptive material and should be chosen to provide the desired noise attenuation without excessive airflow obstruction.



Inserting vibration isolation elements into the propagation path can attenuate vibration propagation in structures. These elements need to be much softer than the structures they connect; for example rubber mounts placed between a vibrating engine and its supports, or a flexible hose inserted in a run of rigid hydraulic tubing.

The noise control treatments can be conveniently considered under the three headings of engine noise control, drive train noise control and operator's compartment noise control. The following sections detail these treatments which were applied to six vehicles - four Wagner vehicles and two Eimco vehicles<sup>(2)</sup>.

## **9.1 Engine noise control**

The engine compartment was modified with an enclosure that was lined with absorptive material. The enclosure consisted of a treated hood, side panels and belly pan with ducts for cooling air exhaust. Rubber gasketing material was applied to the edges of all the panels to isolate them from the frame of the vehicle. The panels were lined with 2.5 or 5 cm thick acoustic absorption material which was protected by perforated steel or expanded metal.

Exhaust noise is predominantly at low frequencies and is best controlled by using a commercial muffler. Four of the machines were fitted with mufflers and two machines used the filled water scrubber technique. Additionally, five of the six machines had the exhaust manifolds wrapped with heat resistant material to reduce the temperatures inside the engine enclosure.

Acoustical modifications were installed at the cooling fan area on three of the six machines. An acoustically treated baffle was attached to the fan grille and the areas adjacent to the fan were faced with acoustical absorption material.

## **9.2 Drive train noise control**

The cover to the transmission and torque converter compartments on each vehicle was treated with absorptive material. Additionally, the open side of the transmission on one machine was closed off with 10 gauge steel that was treated with acoustically absorptive material.

On three machines, the interior of the water and fuel tank compartment was treated with acoustical absorption material, as was the interior of the torque converter compartment. On four of the six machines, the interior surfaces of the transmission compartment were similarly treated.

Vibration isolation treatment was applied to four of the machines. The mounting of the transmission was changed from being rigidly attached to the frame to being supported by resilient elastomeric mounts.

### **9.3 Operator's compartment noise control**

Modifications were made to the operator's compartment on all of the machines. For the three smaller machines, the transmission was located immediately to the right of the operator and for the larger machines, the transmission was located directly in front of the operator. In each case, all openings that permitted noise to escape from the transmission compartment were sealed. On the larger machines, acoustically absorptive material covered in expanded metal was applied to the surfaces around the foot pedals. On one of the larger machines, the operator's compartment was isolated from the machine frame by resilient, elastomeric mounts instead of the normal rigid mounting.

Three of the six machines were equipped with an operator's canopy for falling object protection. These were treated with the acoustic absorption material covered in expanded metal. This treatment, and others for the engine compartment and transmission, are illustrated in Figures 9.1, 9.2 and 9.3 (all taken from reference (4)).

### **9.4 Performance of noise control treatments**

The thermal performance of the noise control treatments was closely monitored. Data for one machine indicated an increase in oil temperatures of less than 9 °C and typically less than 20 °C in the air temperatures in the enclosures.

A summary of the acoustic performance of the noise control treatments is given in Table 9.1, which is taken from reference (4). Three of the six machines were new units and, hence, no in-mine measurements were obtained for the untreated condition.

The resulting noise levels of the fully treated machines ranged from 88 to 95 dB(A). Where treated and untreated data were available, the noise reductions ranged from 4 to 10 dB(A). Further noise reductions were achieved with the installation of an acoustic cab, which is commercially available for some models. The resulting noise levels for the machines were then in the range 80 to 85 dB(A), although acceptability in underground conditions could be a factor.

Similar work was carried out in the UK<sup>(5)</sup> on trackless vehicles. Initially, the treatment consisted of enclosing the engine compartment and transmission with sound deadened steel panels lined with acoustic absorbent. Particular attention was paid to the cooling fan, with acoustic splitters installed in the fan intake, and work was initiated to reduce cooling fan noise at source by designing a new aerofoil bladed fan.

The cooling fans on these machines have curved plate steel blades and a low aerodynamic efficiency in the region of 20%. The fan operates in a pushing mode through the radiator with the incident air being drawn over the engine block. This results in the fan operating in a highly turbulent airflow. The need to avoid sparking due to impacts between the fan blades and the fan cowl requires a relatively large blade tip clearance, further reducing the efficiency of the system. The requirement was to design a fan which would;

- operate efficiently in a highly disturbed incident airflow
- not produce sparking if in contact with a close fitting cowl
- meet the industry's other safety requirements

The fan was designed to give a flow of 5 m<sup>3</sup>/s at a pressure head of 500 Pa and a rotational speed of 1600 rpm. The material used for the blades was carbon fibre reinforced Nylon 66. It was predicted that the new fan would be up to 15 dB(A) quieter than the standard fan.

Rig tests indicated noise reductions of 6 -12 dB(A) in the pulling mode and 8 dB(A) in the pushing mode. This was below the predicted reduction, but still significant. However, because of the contraction of the British coal industry and the reduced numbers of trackless vehicles being supplied, this work was not developed into a production model. The effectiveness of non-metallic aerofoil bladed fans in reducing noise levels was demonstrated, however, and could be exploited in new vehicle developments.

## 9.5 Conclusions and recommendations

Manufacturers can supply vehicles with the engine compartment and transmission compartment treatments described in the previous sections. Care should be taken to ensure that all unnecessary openings in these compartments and in the operator's compartment are closed up. Where it is possible to line the operator's compartment with acoustic absorbent, this should be done.

In the longer term, quieter diesel engines and redesigned cooling systems are the most effective ways to reduce noise levels. The development of quieter, aerofoil bladed fans, as described above, should be pursued. Additionally, the use of alternative fan types should be investigated. Centrifugal and mixed flow fans can be 10 dB(A) quieter than axial flow fans and provide the advantage of having axial intake with radial discharge, allowing an arrangement with a front end air intake discharging to either side of the vehicle, away from the operator.

## 9.6 Retrofit summary

Engine and transmission compartment lining kits can be fitted to existing equipment in mine workshops. Care must be taken to seal unnecessary gaps in the compartments and to reduce the excitation of body panels by the use of neoprene gaskets on covers, as described earlier.

- Engine/transmission enclosure and operator's compartment treatment with exhaust muffler 5 - 12 dB(A) reduction  $\pm$ R10000
- Addition of acoustic cab 12 - 20 dB(A) reduction  $\pm$ R40000

## 9.7 References

- (1) Patterson, W.N., Huggins, G.G. and Galaitisis, A.G. 1975. Noise of Diesel Powered Underground Mining Equipment: Impact Prediction and Control, BuMines OFR 58-75

- (2) **Huggins, G.G., Madden, R. and Murray, B.S. 1997.** Noise Control of an Underground Load Haul Dump Machine, BuMines OFR 125-78
- (3) **Daniel, J.H., Burks, J.A., Bartholomae, R.C., Madden, R. and Unger, E.E. 1981.** Noise Control of Diesel Powered Underground Mining Machines, BuMines IC 8837
- (4) **Bobick, T.G. and Madden, R. 1984.** Current Status of Load Haul Dump Machine Noise Control, Proceedings, Bureau of Mines Technology Transfer Seminars, Pittsburgh, PA and Denver, CO, USA
- (5) **National Coal Board, 1984.** Noise Exposure and Control in Mine Headings and Roadways, and in Passenger Transport Systems, ECSC Final Report 7255-17/040/08

TABLE 9.1: PERFORMANCE OF NOISE CONTROL TREATMENTS / A-WEIGHTED LEVELS

TEST CONDITION	WAGNER ST-2B	WAGNER ST-2D	EIMCO 912D	WAGNER ST-5A	WAGNER ST-8A	EIMCO 918
SURFACE HIGH IDLE: <sup>(1)</sup> BEFORE AFTER	97 - 98 90 - 91	NA NA	101 91	100 - 101 NA	NA 94	99 90
MINE HIGH IDLE: BEFORE AFTER	98 93 - 95	99 88	NM 91 - 92	NM 93	95 <sup>(2)</sup> 90 - 91	NM <sup>(3)</sup> 94
OPERATING: BEFORE AFTER	102 <sup>(4)</sup> 97 - 99 <sup>(4)</sup>	101 - 103 <sup>(4)</sup> 96 <sup>(4)</sup>	NM NA	NM 89 - 92 <sup>(4)</sup>	NA NA	NM NA

Notes:

- NA Not Available
- NM New Machine Treated. Underground Data Not Obtained Prior To Treatment
- (1) Machine Stationary, In Neutral, Maximum Revolutions
- (2) Data Obtained With Some, But Not All, Treatments Removed
- (3) After 1 Year All Treatments Removed. Resulting Noise Level Was 102 dB(A)
- (4) Corresponding Average Noise Levels Based on Dosimeter Data, LOSHA

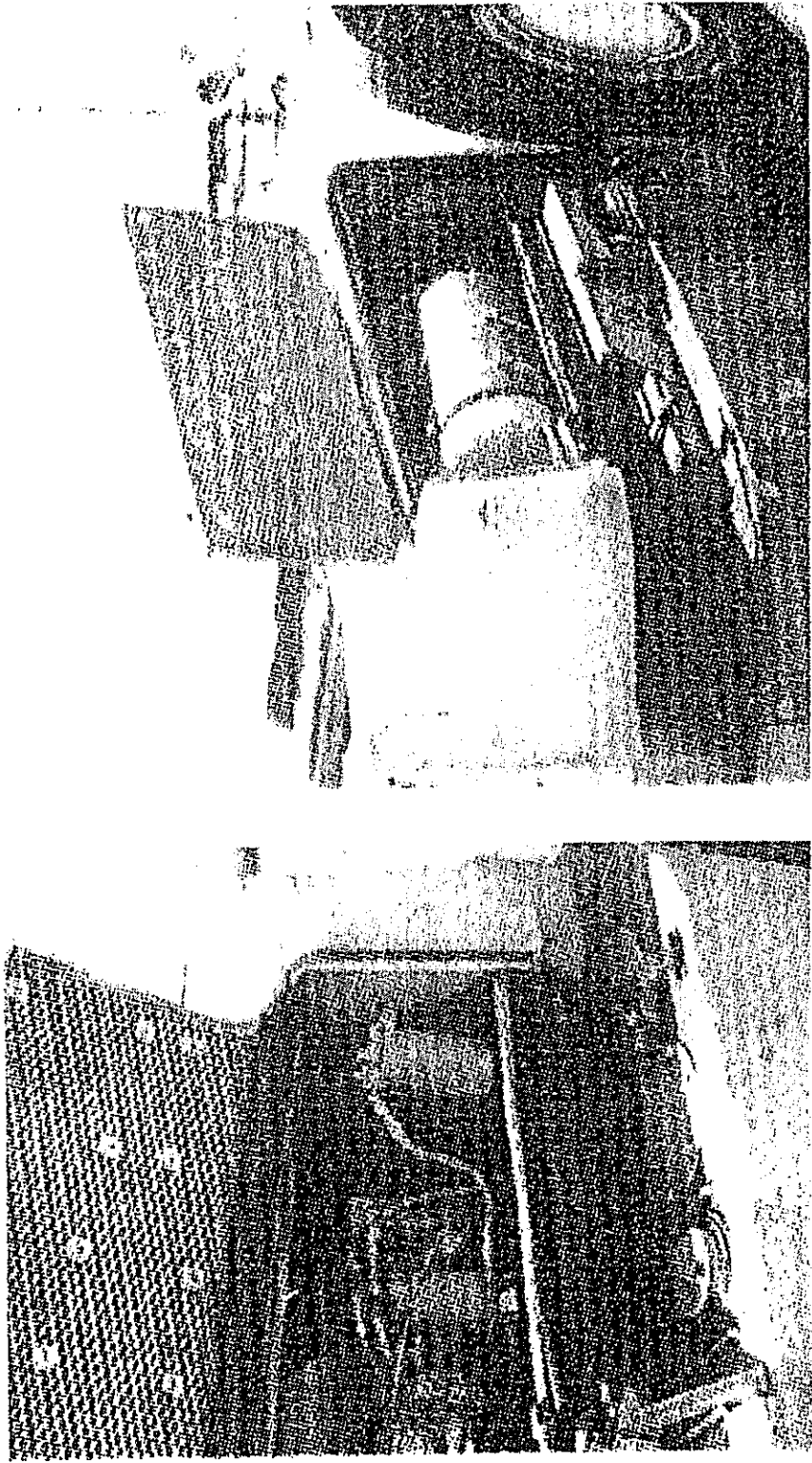


FIGURE 9.1: TREATED ENGINE COMPARTMENT COVERS

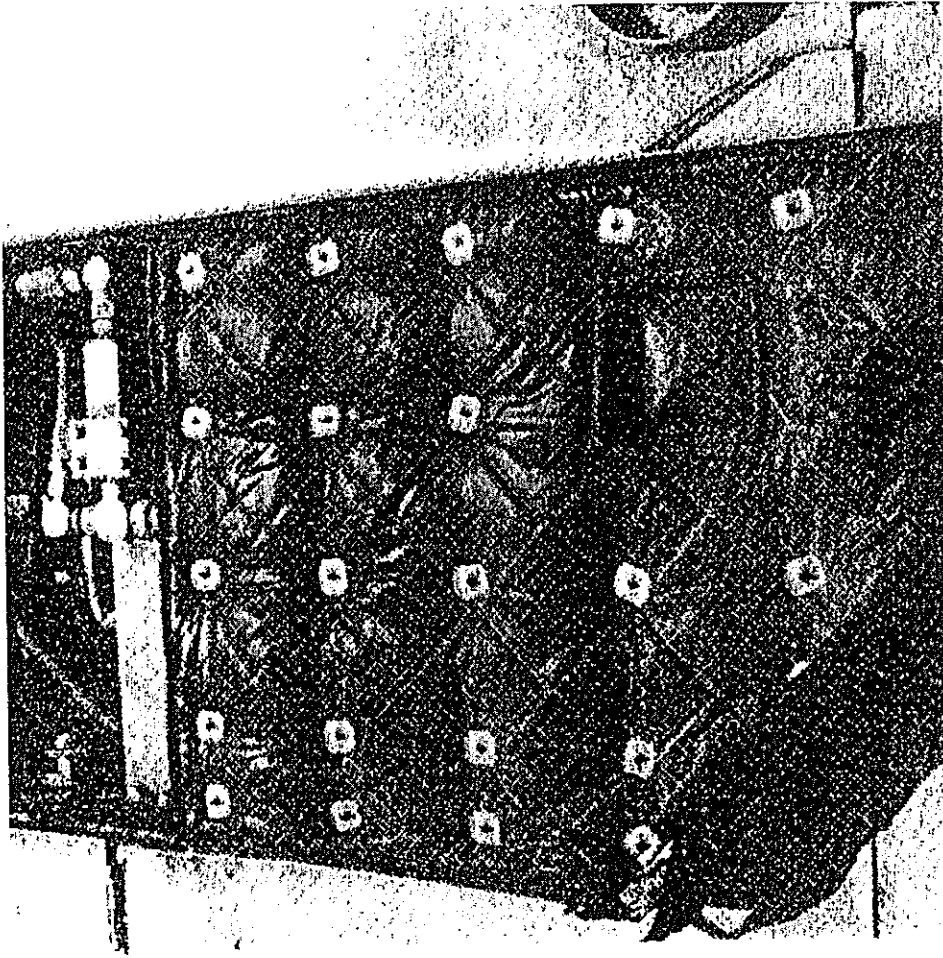


FIGURE 9.2: TREATED TRANSMISSION HOOD



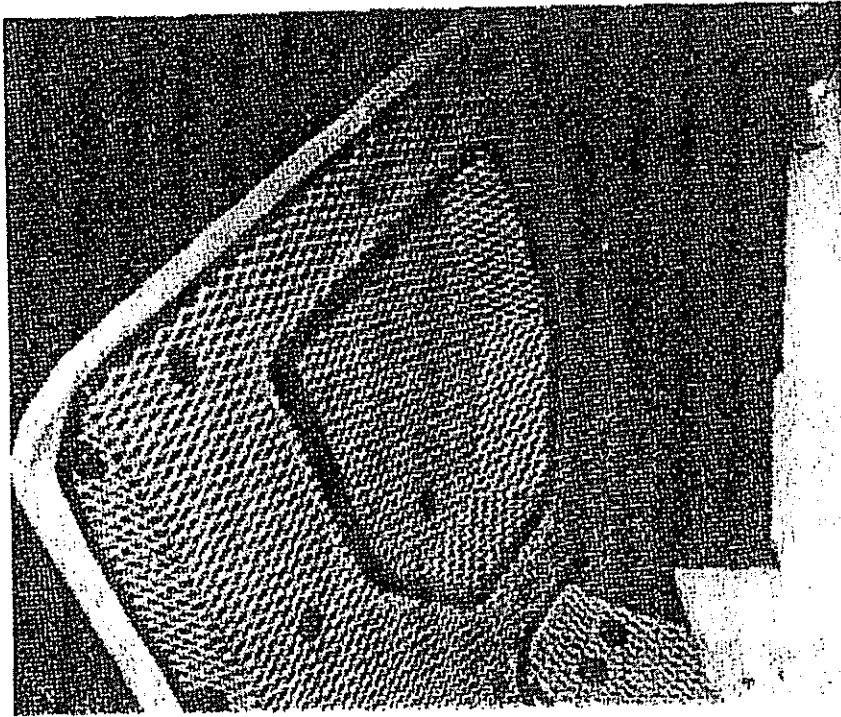
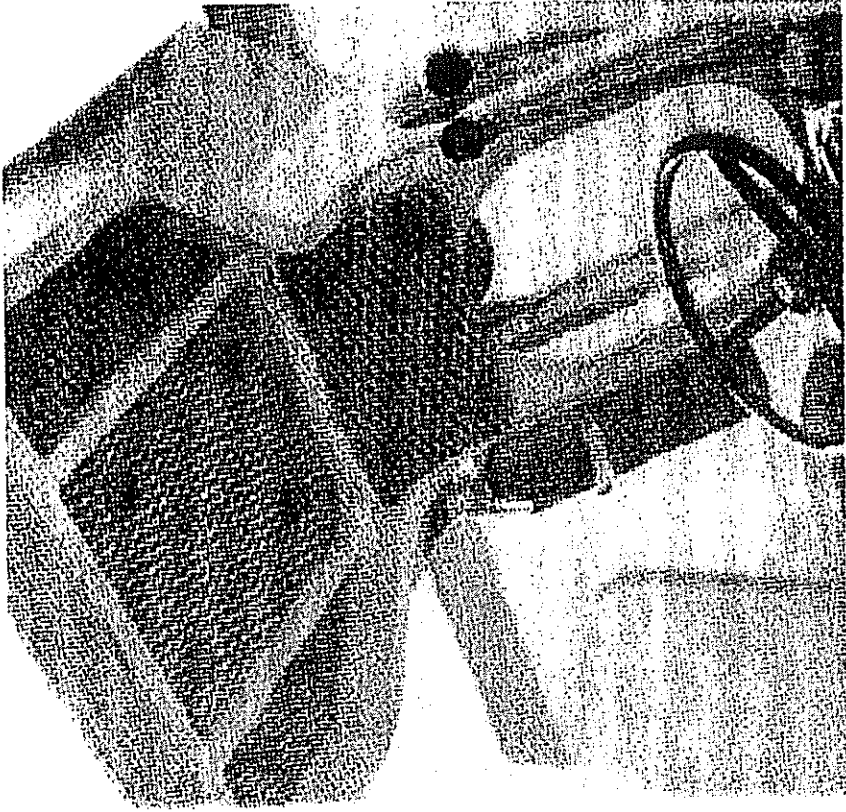


FIGURE 9.3: TREATED OPERATOR CANOPY

## **10. Equipment noise control summary**

In the previous sections noise reduction treatments have been reviewed for a range of underground mining equipment.

Some of these retrofit treatments, and related principles for the design of new equipment, are simple, robust and effective, such as the enclosure and lining of the engine and transmission compartments of Load Haul Dump vehicles.

Others, for equipment which is inherently noisy by virtue of its function, such as rock drills and continuous miners, are more complex, can have drawbacks in terms of equipment performance (although compromises will have to be made to significantly reduce noise levels) and in some cases have not been fully proven in the field.

The retrofit treatments and guidelines for the specification of new equipment are summarised here for the items of equipment considered in the previous sections.

### **10.1 Hand held rock drills**

#### **New equipment**

- All new pneumatic drilling equipment should be fitted with exhaust silencing
- Hydraulic drills should be used in preference to pneumatic drills (hydraulic drills are less noisy because there is no exhaust noise).
- In limited trials there has been significant noise reduction with the use of concentric drill steels (particularly the larger boom-mounted drills) and this approach should be further investigated and developed.
- Tight chuck/drill steel tolerances should be ensured

## Retrofit summary

- |  |                          |        |
|--|--------------------------|--------|
| • Fitment of wraparound muffler  | 7 - 10 dB(A) reduction   | ±R1000 |
| • Constrained layer damped drill steel (collar) with wraparound muffler      | 10 - 12 dB(A) reduction  | ±R1100 |
| • Constrained layer damped drill steel (full length) with wraparound muffler | 10 - 15 dB(A) reduction  | ±R1200 |
| • Shrouded drill steel with wraparound muffler                               | 10 - 150 dB(A) reduction | ±R1200 |
| • Maintain tight chuck/drill steel tolerances                                | -----                    | ----   |

## 10.2 Boom mounted rock drills

### New equipment

- All new pneumatic drilling equipment should be fitted with exhaust silencing
- Where possible, hydraulic drills should be specified. These produce significantly less noise than pneumatic drills due to the absence of exhaust noise
- Acoustic cabs are available from some manufacturers and should be considered as an option where they are acceptable in an underground situation
- The concentric drill steel concept showed promise, particularly for the larger boom mounted drills. Commercial developments should be encouraged

### Retrofit summary

- |   |                        |         |
|---|------------------------|---------|
| • Drifter muffler                               | 4 - 8 dB(A) reduction  | ±R3000  |
| • Drifter muffler with drill steel shroud       | 8 - 12 dB(A) reduction | ±R3500  |
| • Drifter muffler with constrained layer collar | 8 - 10 dB(A) reduction | ±R3500  |
| • Addition of acoustic cab                      | 30-40 dB(A) reduction  | ±R30000 |

## 10.3 Continuous miners and roadheaders

### New equipment

- Where appropriate, remote control should be specified on new continuous miners and roadheaders
- Damped chain conveyor decks should be specified on all new continuous miners and roadheaders
- Acoustically lined enclosure of hydraulics compartment
- The specification of a damped cutting drum on a continuous miner, which is available from one manufacturer, should be considered for evaluation

### Retrofit summary

• Damped chain conveyor	typically - 5 dB(A) reduction	±R20 000
• Isolating strips/resilient wear strips on conveyor	3 dB(A) reduction	±R10 000
• * Resilient/polymer conveyor liners or chain/flight coatings	5 dB(A) reduction	±R10 000
• Radio remote control	to be determined	±R500 000
• Isolated cutting head	5 dB(A) reduction	±R50 000
• Damped cutting drum	3 dB(A) reduction	±R30 000
• Re-designed motor fan impeller	8 dB(A) reduction	±R5 000
• Improved hydraulic power pack enclosure	8 dB(A) reduction	±R10 000

\* Would require further research.

## 10.4 Scrubbers and fans

### New equipment

- Silenced casing jet fans, available from manufacturers, should be specified
- Inlet and outlet silencers should be specified for machine mounted dust scrubbers
- Commercial developments of active noise control systems should be investigated

### Retrofit summary

- |  |                 |       |
|--|-----------------|-------|
| • Fitment of inlet and outlet silencers to fans and dust scrubbers       | not established | *     |
| • Consider optimum siting of fans in relation to work areas of employees | -----           | ----- |

\* Depends on size of fan

## 10.5 Longwall systems

### New equipment

- For longwall shearers, the control of mechanical noise can be reduced with the use of precision quality gears and isolated bearing housings and hydraulic components
- The potential of low noise shearer cutting heads should be investigated
- For armoured face conveyors, reduce, as much as possible, discontinuities which cause impact noise. Additionally, damped line pans can be supplied
- For the drivehead, precision quality gears and close shielding of machine casings can be specified

- As with the armoured face conveyor, damping and isolation treatments can be implemented on stage loaders at the design and manufacture stage

## Retrofit summary

• Enclosure of hydraulics on face machine	4 - 5 dB(A) reduction	±R10000
• Minimise impact points and damp line pans on face conveyor	4 - 5 dB(A) reduction	±R50000
• Covers for stage loader	0 - 4 dB(A) reduction	±R10000
• Reduce drop height at transfer points	not established	-----
• Close shielding for crusher/breaker	3 - 4 dB(A) reduction	±R10000
• Where possible, remove noisy equipment from the main work area	not established	-----
• Enclosures for haulages/winches	10 - 20 dB(A)	±R10000
• Barriers to shield noisy equipment	5 - 10 dB(A)	±R10000

## 10.6 Trackless vehicles

### New equipment

- Specify vehicles with engine compartment and transmission compartment acoustic treatments
- Where possible, line the operator's compartment with acoustic absorbent
- Redesigned cooling systems and the use of quieter, aerofoil bladed fans should be investigated
- The use of alternative fan types should be investigated. Centrifugal and mixed flow fans can be 10 dB(A) quieter than axial flow fans and provide the advantage of having axial intake with radial discharge, allowing an arrangement with a front end air intake discharging to either side of the vehicle, away from the operator

## Retrofit summary

- Engine/transmission enclosure and operator's compartment treatment with exhaust muffler 5 - 12 dB(A) reduction ±R10000
- Addition of acoustic cab 12 - 20 dB(A) reduction ±R40000

## 11. Summary and conclusions

Underground equipment used in the South African mining industry, both hard rock and coal, has been reviewed in terms of numbers employed and resultant noise exposure levels. This review has drawn on extensive underground measurement work carried out in South Africa to characterise the noise emissions of mining equipment and to relate these to noise exposure levels and appropriate hearing protection devices.

A number of mining equipment types has been studied in detail to examine their noise generating components and mechanisms and to assess methods for noise reduction. The range of equipment studied has consisted of hand held rock drills, boom mounted rock drills, continuous miners and roadheaders, dust scrubbers and fans, longwall systems and trackless vehicles. For some equipment, it has been shown that the application of simple acoustic treatments can give significant reductions in noise levels. For other equipment, there are no "quick fix" solutions and some acoustic treatments, which give small but worthwhile reductions in noise level, have drawbacks in terms of reduced performance.

Hand held percussive rock drills are one of the noisiest item of equipment in the mining industry and large numbers are employed in the mines of South Africa. The research and development work, carried out over the last twenty five to thirty years, has been reviewed and proposals for retrofit treatments and development work have been presented. The large body of work available for review indicates the difficulties inherent in reducing the noise levels from this type of equipment.

The necessity of using exhaust mufflers, which reduce drill body noise as well as exhaust noise, has been stressed. Once exhaust noise has been significantly reduced, the dominant noise source is drill rod noise. The potential for noise reduction by using drill rod damping collars and damping sheaths has been shown. The drawbacks of this treatment, in terms of reduced drilling performance and possible loosening of the collar on the drill steel, have also been highlighted. An alternative treatment, consisting of a drill steel shroud tube which does not contact the drill steel, has been described. The drawbacks of this alternative, in terms of handlability and steel changing have been highlighted. However, it has to be recognised that some sacrifices in terms of performance have to be made in order to reduce noise levels, and these have to be weighed against the benefits of reduced worker exposure.



In terms of new developments, the redesign work carried out by U.S.B.M. has been reviewed. This work resulted in a new design for a quiet hard rock drill, but this met with limited success in the marketplace because of reduced performance. However, the techniques employed in this drill may have an influence on future developments by manufacturers. The ongoing work in South Africa, employing exhaust muffling and drill steel damping along with enclosure and in-line thrust, has also been highlighted. Although it is possible to see a number of practical difficulties with this design, it is a novel concept and should be supported through its development and initial proving.

The development of concentric drill steels has been reviewed and it is considered that further work in this area may be justified, although it is not known if any manufacturers are currently pursuing this line. It is a technique that is possibly more suited to the larger drill steels of boom mounted systems rather than hand held drills. Apart from total enclosure of the drill body and drill steel, as in the South African work, it is the only technique likely to give significant reductions in drill steel noise without the attendant problems of reduced drilling performance and/or reduced handability in the mine.

The noise control problems associated with hand held percussive rock drills are also those of boom mounted rock drills. Because of their larger size and reduced requirement for handability, there is more scope for noise control treatments on boom mounted drills. The development work has been reviewed and, as with hand held drills, the main recommendations are for a high performance enclosure of the drill body and the application of a damping collar or a shroud to the drill steel. Noise from the hydraulics and cooling fan on the main body of the drilling rig can be effectively reduced with an acoustically lined enclosure and the fitment of acoustic louvres for cooling inlets and outlets. The fitment of an acoustic cab is the most effective option for reduced operator noise exposure and, where possible, this should be the first choice for noise control.

Remote control is available as an option (for new equipment and for retrofit) for many types of continuous miner and roadheader. Where feasible, remote control should be specified for this type of equipment, as moving the operator further away from the main noise sources is an obvious way to reduce noise exposure levels.

For continuous miners and roadheaders, the major noise sources are machine mounted dust scrubbers, scraper conveyor noise and cutting noise. On roadheaders, and to a lesser extent on continuous miners, electric motor noise and hydraulic system noise also

contribute to the noise exposure of the machine operator and ancillary workers. The development work has been reviewed and noise reduction treatments have been proposed.

Redesigned electric motor impellers and hydraulic systems, together with hydraulic enclosures, have been shown to give significant reductions in noise levels. Scraper conveyor noise can be reduced by the application of damping to the deck. This can be achieved simply by welding steel plates on to the underside of the deck or by including a constrained layer between the plates. Some equipment manufacturers provide damped scraper conveyors of this type on their machines as standard.

The reduction of cutting noise is less certain for continuous miners and roadheaders. Some success has been had with resiliently mounted picks on a continuous miner in a pre-prototype trial, but further development work is needed before the technique can be considered for implementation in the field. Manufacturers should be encouraged to familiarise themselves with the research work undertaken in this area and to recognise the opportunities for development work of their own. It is understood that one manufacturer produces a continuous miner cutting head with structural damping for reduced noise operation.

The available literature for noise reduction on longwall systems has been reviewed and retrofit treatments have been presented. In common with scraper conveyors on continuous miners and roadheaders, the importance of removing discontinuities on afcs and stage loaders has been highlighted, so as to reduce impact noise as much as possible. The possibilities for damping of the line pans by welding steel plates on the underside of the deck plates has been demonstrated, again a technique employed on continuous miners and roadheaders.

On the face machine itself, the importance of cutting noise has been highlighted and the successful testing of a prototype "low noise" shearer cutting head has been reported. It is considered that this work should be followed up, as potentially significant reductions in noise level are achievable with no deterioration in cutting and loading performance. In fact, IMCL are proposing to develop an enhanced "low noise" shearer cutting head, under a forthcoming ECSC funded project, for field testing in the British coal mining industry.

In longwall face-ends, where there is a multiplicity of noise sources, the importance of good planning has been stressed. Possibilities for the enclosure of crushers and the fitment of covers to stageloaders have been highlighted. The potential for the use of simple enclosures and barriers has been indicated; simple installations which have applications in all types of mining.

Acoustically lined enclosures around the engine and transmission compartments of underground diesel vehicles have been shown to give effective reductions in operator noise level. Additional noise control work, including resilient mounting of the transmission (rather than being bolted directly to the vehicle frame) and acoustically lined operator's position and canopy, has been shown to further reduce noise levels. In some cases, operator's noise levels have been reduced below 90 dB(A) and, with the fitment of a full acoustic cab, noise levels below 85 dB(A) are possible. The importance of maintaining the acoustic treatments in good condition whilst the vehicles are in use underground has been stressed.

In reviewing the work which has been completed in mining equipment noise control, it has been shown that there is scope for significant noise reduction for a range of equipment by retrofit treatment and by redesign. However, it has also been shown that, in some cases, although worthwhile noise reductions are possible, noise levels will still be significantly above 85 dB(A), and hearing protection will be required.

A procurement procedure for new equipment has been proposed. This has been based on the procedure implemented in the British coal industry. Within the procedure, the onus is placed on manufacturers and suppliers to take all necessary steps to reduce equipment noise levels. They are also required to provide details of equipment noise levels for assessment by end-users. A noise test procedure, based on the British Coal test procedure and BS7025, has been outlined which enables the measurement and presentation of equipment noise levels in a standard format. This will enable end-users to make informed choices in the selection of equipment, where noise levels should be taken into consideration.

## 12. Acknowledgements

During the course of this project underground visits were made to a coal mine, a gold mine and a platinum mine. Routine noise measurement data was obtained from a number of coal, gold and platinum mines.

Discussions were held with the following South African manufacturers and organisations:

Atlas Copco

Barlows/CAT

Boart Longyear

CSIR Miningtek

Eickhoff

Howden Safanco

Joy Manufacturing

Locked Torque Africa

MAN GHH

## APPENDIX I Procurement procedure and noise test data for new equipment

There are responsibilities on mine managers and equipment suppliers in terms of health and safety at mines to reduce noise exposure to employees. A manager must take the necessary steps to reduce the equivalent personal noise exposure of his employees to below 85 dB(A), where possible. One aspect of this is to purchase equipment designed and manufactured to operate at the lowest noise levels reasonably practicable. To enable this, manufacturers and suppliers need to provide information on equipment noise levels which can be considered by end-users when deciding on the purchase of new equipment. A standard format for the presentation of noise level data is useful when comparing equipment types.

Standard noise test procedures exist for many types of equipment, for example ISO Standards and CAGI-PNEUROP procedures for pneumatic equipment, and they should be used where possible. However, for many types of equipment, test codes do not exist and there is a need to specify how machinery noise testing is to be carried out, including details of:-

- instrumentation
- test environment
- measurement positions
- installation and operating procedures of machine during test
- required measurements
- corrections for background noise
- noise test report

British Standard 7025 (which is based upon, but not identical to, ISO 6081) provides a framework for preparing noise test codes that describe methods for measuring the sound pressure levels at the operator's position(s). The sound pressure levels determined according to such test codes are useful for comparing the noise emissions of different machines.

## AI.1 Procurement policy

Long term noise reduction should include a positive procurement policy which makes sure noise is taken into account when selecting new machinery. The manager of a mine should ensure that:-

- invitations for tender documentation include a requirement for manufacturers and suppliers to provide details of noise levels
- details of noise levels supplied by manufacturers are used in the technical appraisal of the tenders

Manufacturers should ensure that all measures have been taken during the design of machinery to ensure that the noise levels during operation of that equipment in the workplace are kept to the lowest levels reasonably practicable. The provision of noise control treatments should not affect the suitability for purpose of machinery or create safety hazards. It should not unduly impair access to machinery for the purposes of examination and maintenance. Manufacturers intending to supply power operated equipment to mines should arrange for an example of the equipment to be tested in accordance with the procedures of section 11.2.

In terms of acceptable noise levels, a requirement can be stated that:-

- all equipment for use underground shall be designed so as to ensure a maximum noise level of 85 dB(A) at the operator's position or at 1 metre from the equipment surface, when measured in an unrestricted surface or workshop environment.
- where it is apparent that further noise reduction is practicable, a lower maximum noise level may be stipulated
- manufacturers shall ensure that all reasonably practicable measures have been taken during design to reduce equipment noise levels

It has to be recognised that some types of equipment are inherently noisy by virtue of their function. Shearers, roadheaders and continuous miners produce noise levels of the

order of 95 - 100 dB(A) at the operator's position, due mainly to cutting noise, and it is not possible to reduce these levels below 85 dB(A) at present.

Similarly, pneumatic rock drills produce operator noise levels of well over 100 dB(A), even when fitted with mufflers, and a limit of 85 dB(A) is not possible to meet. Obviously, exemptions need to be given in these cases and mines must ensure that operators of exempted equipment are issued with hearing protection devices.

## **AI.2 Noise test and measurement procedure**

A test code is presented for underground mining equipment, which satisfies the requirements of BS 7025, and also specifies further information requirements regarding the noise emission of that equipment. In large part it is based on the test code in the document "A Code of Practice for the Procurement of Underground Machinery and Procedure for Noise Testing" developed within the British coal mining industry.

The reasoning behind the development of the test code was to provide a straightforward and relatively simple procedure for the measurement and presentation of equipment noise data which could be carried out by manufacturers and suppliers without too much difficulty and could be effectively employed by end-users to compare equipment of similar type.

### **• Measurement instrumentation**

Measurements shall be made using an integrating averaging sound level meter conforming to the requirements of IEC804 for Type 1 instrumentation. The meter shall be capable of measuring the equivalent continuous A-weighted sound pressure level ( $L_{Aeq}$ ) and also the unweighted octave band levels in the frequency bands from 31.5 Hz to 8 kHz. The sound level meter shall be calibrated with an acoustic calibrator before and after the measurements are taken. The calibrator shall be checked annually to verify that its output has not changed. In addition, an acoustical and electrical calibration of the instrumentation system shall be carried out at least every two years.

- **Test environment**

The preferred test environment is a free field over a reflecting plane. This can be achieved by testing on a hard surface (asphalt or concrete) in the open air, away from the influence of any buildings or walls, or, secondly, inside a large open plan workshop, away from the influence of any reflecting walls or structures.

Tests conducted in confined areas will increase the measured noise levels, unless nearby walls and surfaces are acoustically absorbent in nature.

- **Measurement positions**

Measurements should be made at four positions around the machine. These positions should normally be central to both sides and ends of the machine and at a distance of 1 metre from the machine surfaces and at a height of 1.5 m.

For machines that are operator attended, a measurement should be made at the operator's head position.

For large machines (any dimension greater than 3 m), measurements should be made at four additional positions around the machine, if possible. These positions should normally be central to both sides and ends of the machine and at a distance of 10 metres from the machine surfaces and at a height of 1.5 metres.

- **Installation and operating procedures of machine during test**

Where practicable, equipment should be tested on full load or in an operating condition giving the maximum noise levels. (Certain types of equipment, such as armoured face conveyors, produce higher noise levels when running empty)

Equipment such as shearers, roadheaders, continuous miners, dinstheaders and crushers produce maximum noise levels whilst processing mineral. Where this equipment cannot be tested on the surface under this loaded condition, tests should be carried out under no load conditions.



Mobile plant can be divided into various types - diesel, electric, hydraulic and pneumatic. Diesel powered equipment should be tested at maximum engine revs. Other equipment should be tested using a drive-by method of measurement: the speed should be close to maximum. The noise level at the operator's position should be measured at this speed also.

### • Required measurements

The equivalent continuous A-weighted sound pressure level shall be measured at each measurement position. The measurement time interval,  $T$ , shall be chosen in such a way that the equivalent continuous sound pressure level,  $L_{Aeq,T}$  can be determined for the specified operating conditions.

If the noise at the measurement position is steady (i.e. with negligibly small fluctuations of level within the period of observation), the measurement time interval shall be at least 15 seconds.

The measurement time interval and operating conditions of the machine shall be reported with the test results.

In addition, the Linear octave band sound pressure levels, in the bands 31.5 Hz to 8 kHz, shall be measured at one position; preferably at the operator's head position, or at the measurement position 1 metre from the machine where the maximum overall level occurs.

Measurements carried out in the open should not be undertaken in rain or when the wind speed exceeds 5 metres/second.

### • Corrections for Background Noise

Background noise measurements shall be taken at the measurement positions. Background noise levels should ideally be at least 10 dB below the level due to the machine being tested. If this is not possible, the background noise levels shall be at least 6 dB below the level due to the machine being tested. Failing this, the test measurement shall be invalid.

Where necessary, the noise measurement at each measurement position shall be corrected for the influence of background noise according to Table 11.1.

**TABLE A1.1: CORRECTION FOR BACKGROUND NOISE**

DIFFERENCE BETWEEN MACHINE NOISE LEVEL AND BACKGROUND NOISE LEVEL (dB)	CORRECTION TO BE SUBTRACTED FROM MACHINE NOISE LEVEL (dB)
<6	MEASUREMENT INVALID
6	1.0
7	1.0
8	1.0
9	0.5
10	0.5
>10	0.0

- **Noise Test Report**

A noise test report shall be prepared detailing the measurements carried out on the equipment. The format of this noise test report is given in Figure A1.1.

**FIGURE AI.1: FORMAT FOR NOISE TEST REPORT**

NOISE TEST REPORT
<p>EQUIPMENT DESCRIPTION</p> <ul style="list-style-type: none"><li>• Supplier</li> <li>• Equipment Name</li> <li>• Type and Serial Number</li> <li>• Specification of Equipment</li> <li>• Description of Normal Equipment Operation</li> <li>• A General Arrangement Drawing to Show Configuration of Equipment and Measurement Positions</li></ul>

## NOISE TEST REPORT

### NOISE MEASUREMENTS

- Equivalent Continuous A-Weighted Sound Pressure Levels,  $L_{Aeq}$ , Around the Machine and at Operator's Position

MEASUREMENT POSITION	MEASURED NOISE LEVEL dB(A)	
	@ 1 METRE	@ 10 METRES
1	---	---
2	---	---
3	---	---
4	---	---
OPERATOR'S POSITION	---	

- Background Noise Levels

MEASUREMENT POSITION	BACKGROUND NOISE LEVEL dB(A)	
	@ 1 METRE	@ 10 METRES
1	---	---
2	---	---
3	---	---
4	---	---
OPERATOR'S POSITION	---	

- Octave Band Noise Levels at Operator's Position or where the Maximum Overall Level at 1 Metre From Equipment Occurs

MEASUREMENT POSITION	OVERALL A-WT	OVERALL LIN	OCTAVE BAND LEVELS									
			31.5	63	125	250	500	1k	2k	4k	8k	
---	---	---	---	---	---	---	---	---	---	---	---	---

## NOISE TEST REPORT

- Character of Noise From Equipment

Does the Noise From the Equipment Contain Any Notable Pure Tones, When Judged Subjectively (e.g. whine or hum)

YES / NO (if YES, describe)

Does the Noise From the Equipment Contain Impulsive Components (e.g. clangs or bangs)

YES / NO (if YES, describe)

- Equipment Operating Conditions During Test

Does This Operating Condition Produce the Highest Noise Level

YES / NO (if NO, what condition produces the highest noise level)

- Details of Any Noise Reduction Features Included In Equipment As Tested (Give Details)

- Test Conditions

- Outdoors (provide sketch of adjacent buildings)
- Indoors (provide sketch and dimensions)

NOISE TEST REPORT

- Any Other Relevant Information

MAKE, MODEL AND TYPE OF SOUND LEVEL METER USED FOR MEASUREMENTS:

.....

DATE OF LAST CALIBRATION OF SOUND LEVEL METER USED FOR MEASUREMENTS  
(ATTACH COPY OF CERTIFICATE):

.....

TEST CONDUCTED BY:

.....

.....

TEST DATE:

.....

## APPENDIX II Noise database

Mines undertake routine noise measurements at specific workplaces, for example, pump stations, fan stations, face drilling, continuous miner sections, etc as part of their hearing conservation programme. However, the format in which the data is reported is not consistent across mines and can thus not be readily incorporated into a common database. Information on specific machines is also not included with the data. As part of this project a database on equipment noise levels was to have been set up. However, as discussed in Section 2, due to the lack of consistent existing noise data, it was not possible to collate data for the development of such a database.

### Development of noise database in British Coal

Fundamental to British Coal policy was the method by which noise assessments were made. This system was not based on the measurement of individual levels of noise exposure, but on the identification of noise zones. It was analogous to British Coal's very successful policy on dust control, whereby dust measurements were taken at particular locations within a mine, not at each person. By using such a system, emphasis was placed on the sources of dust and methods for control, rather than on its effect. British Coal noise policy therefore also placed emphasis on control of the generation of noise by machines, but there were other substantive reasons for adopting a zoning policy.

During 1983, to assess the validity of using noise dosimeter measurements for assessment of personal noise exposure and to compare dosimeter measurements of exposure with calculated exposures (derived from knowledge of sound pressure levels and exposure levels), extensive noise surveys were undertaken at three collieries over a six month period.

Detailed results from these surveys were published in three papers<sup>(1),(2),(3)</sup>. Twenty three underground workers were monitored in fifteen types of job. Where exposure was particularly variable (two job types), it took, on average, 55 consecutive full shift measurements to obtain a  $\pm 1$  dB(A) determination for an individual. For the less variable types of job, this average dropped to 40 and the typical 95% confidence limit for a five shift set of measurements was  $\pm 3$  dB(A).

Clearly, this represented an immense expense in determining reliable noise exposures. When comparing dosimeter measurements of personal exposure with calculations, agreement to within  $\pm 2$  dB(A) was obtained in only about half the cases studied; for the remaining half, the calculated figure underestimated the daily noise exposure by up to 13 dB(A). The results demonstrated the practical difficulties in reliably assessing the noise exposure of individual underground workers in the coal mining industry, caused by the mobility of workers, the wide variation in noise levels experienced at different locations and the variability of machine operating times. It was considered likely that other industries which encountered these problems would have equal difficulty in obtaining reliable assessments of individual workers' noise exposure, particularly where a large workforce is involved.

Noise exposure can be based either on area/equipment measurements or personal measurement. The approach adopted by British Coal was to divide the mine into a number of noise zones. It was believed that this was the most appropriate method. For each zone, noise exposures could then be assessed over a full 8 hour shift by considering the maximum noise level within these zones and the time for which each machine operates. This system had a built-in safety factor in that it generally overestimated noise exposure as, in practice, most personnel are not exposed to the maximum noise levels within the zones.

Under this policy, each colliery manager was responsible for implementing a scheme for noise control, which stipulated that all workplaces and other areas where men travel were to be surveyed, the survey to comprise the following;-

- $L_{Aeq}$  levels at a reference point 1 metre from each noise source
- $L_{Aeq}$  level at the operator's position (or person nearest to the source)
- Impulsive peak noise levels
- 90 dB(A) boundaries
- 85 dB(A) boundaries

The information was recorded on standard workplace survey record forms and computer coded for input to a central database.

## **Proposed database for South African mines**

An outline structure for a database [in Microsoft Access format, see attached disk] is now proposed. The database records are input from workplace survey record forms, an



example of which is given in Figure All.1. The proposed database is based on the system implemented within British Coal as outlined above. It is proposed that individual mines capture information using the format for a trial period after which the format can be assessed and reviewed. For the system to be successful, mine databases will have to be submitted to a central database managed by SIMRAC on a regular basis. In this manner information on noise levels of different equipment types and workplaces will be directly comparable across mines.

Initially, in an attempt to minimise the number of measurements that need to be taken in addition to the normal noise measurements, not all of the measurements listed above are included (for example  $L_{Aeq}$  at 1m). The additional information relates to

- noise sources
- equipment type, manufacturer
- measurement duration – this should be long enough to allow a representative measurement of the  $L_{Aeq}$  for the typical operating cycle of the equipment to be obtained.
- maximum distance to 85 dB(A) SPL
- peak noise level above 120 dB(A)
- number of workers in vicinity
- meter type and serial number

Some of the routine measurements obtained during the present study have been included on the disk containing the database, even though the data does not include all of the required information. Where all of the required information is available on individual mines for existing measurements, then this data can also be input.

## References

- (1) Jones, K. 1986. Personal Noise Exposure, Part 1, Colliery Guardian, Vol. 224, No. 3, March
- (2) Lancaster, G. K. 1986. Personal Noise Exposure, Part 2, Colliery Guardian, Vol. 224, No. 5, May
- (3) McCormick, G. 1987. Personal Noise Exposure, Part 3, Colliery Guardian, Vol. 235, No. 12, December

DATE	REFERENCE NUMBER	SURFACE UNDERGROUND
------	------------------	------------------------

MINE	LOCATION
------	----------

NOISE SOURCE	MANUFACTURER/MODEL No.	NOISE SOURCE	MANUFACTURER/MODEL No.
1.		4.	
2.		5.	
3.		6.	

MEASUREMENTS	
SAMPLE $L_{Aeq}$ AT OPERATOR	dB(A)
MEASUREMENT DURATION	MINUTES
MAX. DISTANCE TO 85 dB(A) SPL	METRES
PEAK ABOVE 120 dB (FLAT RESPONSE)	dB

ASSESSMENT OF ZONE EXPOSURE		
SAMPLE $L_{Aeq}$	TOTAL DURATION	CALCULATED $L_{EP,d}$
dB(A)	HOURS	dB(A)
NUMBER OF MEN WORKING IN VICINITY		
COMMENTS		

METER TYPE	SERIAL NO.
SURVEYED BY	

SKETCH PLAN

FIGURE AII.1 PROPOSAL FOR WORKPLACE SURVEY RECORD FORM

TABLE 2.1: WORKERS EMPLOYED IN THE SOUTH AFRICAN MINING INDUSTRY (1996)

MINING TYPE	NUMBER OF WORKERS			% OF OVERALL TOTAL
	UNDERGROUND	SURFACE	TOTAL	
GOLD	239543	62574	302117	61.02
PLATINUM	58719	15616	74335	15.01
COAL	28792	30914	59706	12.06
DIAMOND	6122	9204	15326	3.10
CHROME	3968	2663	6631	1.34
IRON ORE	1772	4214	5986	1.21
COPPER	732	4403	5135	1.04
GRANITE	589	2644	3233	0.65
LIMESTONE	585	2292	2877	0.58
CLAY	130	883	1013	0.20
OTHER MINES	4263	14512	18775	3.79
OVERALL TOTAL	345215	149919	495132	100

TABLE 2.2: MINING EQUIPMENT NUMBERS

DESCRIPTION	A	B	C	D	E	F	G	H	J	TOTAL
LOCOMOTIVES	2	28	547	0	26	349	5	1338	476	2771
DIESEL BATTERY	1	18	1149	0	2	128	56	685	1816	3855
FLAMEPROOF EQUIPMENT	0	0	0	0	27	179	1	4	0	211
GATHERING ARM LOADERS	0	0	0	0	50	720	7	95	0	872
SHUTTLE CARS	0	0	0	0	79	176	3	9	0	267
COAL CUTTERS	0	0	0	0	46	70	0	9	0	125
BATTERY SCOOPS	0	0	0	0	31	285	7	17	0	340
LOAD HAUL DUMPERS	0	0	0	0	17	203	2	18	0	240
CONTINUOUS MINERS	0	0	0	0	4	14	0	1	0	19
SHEARERS	0	0	0	0	4	111	2	46	57	257
DRILL RIGS	0	25	12	0	4	364	0	33	0	477
ROOFBOLTERS	0	0	0	0	80	434	3	17	0	525
TRACTORS	0	0	0	0	71	200	8	25	0	289
COMPRESSORS	0	0	0	0	25	357	2	24	0	408
PERSONNEL VEHICLES	0	0	0	0	22	248	2	22	256	559
FEEDER BREAKERS	0	0	0	0	0	0	0	0	0	0

A WESTERN CAPE B NORTHERN CAPE C FREE STATE D EASTERN CAPE E KWAZULU NATAL

F MPUMALANGA G NORTHERN REGION H GAUTENG J NORTH WEST REGION

TABLE 2.2: (CONTINUED)

DESCRIPTION	A	B	C	D	E	F	G	H	J	TOTAL
WINCHES	0	91	9037	0	21	1383	86	7434	12874	30926
SCRAPERS	0	1	2845	0	45	92	12	1420	1987	6402
MONO ROPE CONVEYORS										
TRACKLESS MINING										
DUMPERS	215	212	27	22	195	290	153	214	340	1668
LOADERS (INC. AIR)	116	88	991	0	4	242	103	319	931	2794
SHOVELS	128	47	4	30	18	33	40	54	140	494
BULLDOZERS	236	60	13	13	31	171	47	61	223	855
LOAD HAUL DUMPERS	95	84	75	95	151	41	76	370	179	1166
PERSONNEL VEHICLES	45	187	35	0	95	784	370	575	322	2413
SERVICE VEHICLES	0	478	17	0	12	310	347	203	121	1488

A WESTERN CAPE B NORTHERN CAPE C FREE STATE D EASTERN CAPE E KWAZULU NATAL

F MPUMALANGA G NORTHERN REGION H GAUTENG J NORTH WEST REGION

TABLE 2.3: NOISE LEVELS FOR A RANGE OF JOB CATEGORIES IN GOLD AND PLATINUM MINING

JOB CATEGORY	OCTAVE BAND SOUND PRESSURE LEVEL (dB(A))								SUM (dB(A))	N <sub>eq</sub> (DOSIMETRY)
	125	250	500	1k	2k	4k	8k			
	DEVELOPMENT DRILL OPERATOR;- UNSILENCED PNEUMATIC	94	102	108	109	111	108	106		
SILENCED PNEUMATIC	89	94	97	100	103	104	100	108	104	
WATER-HYDRAULIC	77	79	85	86	100	102	98	106	101	
STOPE DRILL OPERATOR;- UNSILENCED PNEUMATIC	95	101	108	110	112	111	109	117	112	
SILENCED PNEUMATIC	88	90	95	100	104	105	103	110	105	
WATER-HYDRAULIC	75	77	87	88	105	107	103	110	105	
DEVELOPMENT TEAM (EXCL. DRILLERS)	90	99	101	104	105	104	102	111	103	
STOPE TEAM (EXCL. DRILLERS)	92	102	106	108	109	107	104	114	102	
PNEUMATIC LOADER OPERATOR	87	94	96	98	100	101	100	107	106	
DIESEL LOCO CREW	78	82	86	92	90	89	80	96	94	
ELECTRIC LOCO CREW	82	85	86	86	86	82	73	92	93	

TABLE 2.4: NOISE LEVELS FOR A RANGE OF JOB CATEGORIES IN COAL MINING

JOB CATEGORY	OCTAVE BAND SOUND PRESSURE LEVEL (dB(A))										SUM (dB(A))	N <sub>eq</sub> (DOSIMETRY)		
	125	250	500	1k	2k	4k	8k							
	CONTINUOUS MINER OPERATOR	89	93	97	96	93	90	81						102
CONTINUOUS MINER ASSISTANT	92	95	99	96	95	91	83						103	95
SHUTTLE CAR OPERATOR	78	85	88	86	84	78	70						92	94
CONTINUOUS MINING TEAM LEADER	85	96	100	100	104	99	89						108	96
LONGWALL SHEARER OPERATOR	85	93	98	100	96	94	85						104	101
LONGWALL SHEARER ASSISTANT	79	86	93	96	93	90	85						100	100
LONGWALL CRUSHER ATTENDANT	73	82	84	87	86	84	82						92	96
LONGWALL TEAM LEADER	82	90	97	98	96	92	82						103	97
LHD DRIVER	80	84	90	94	95	96	83						100	99
UTILITY VEHICLE DRIVER	86	87	91	97	96	93	85						101	93

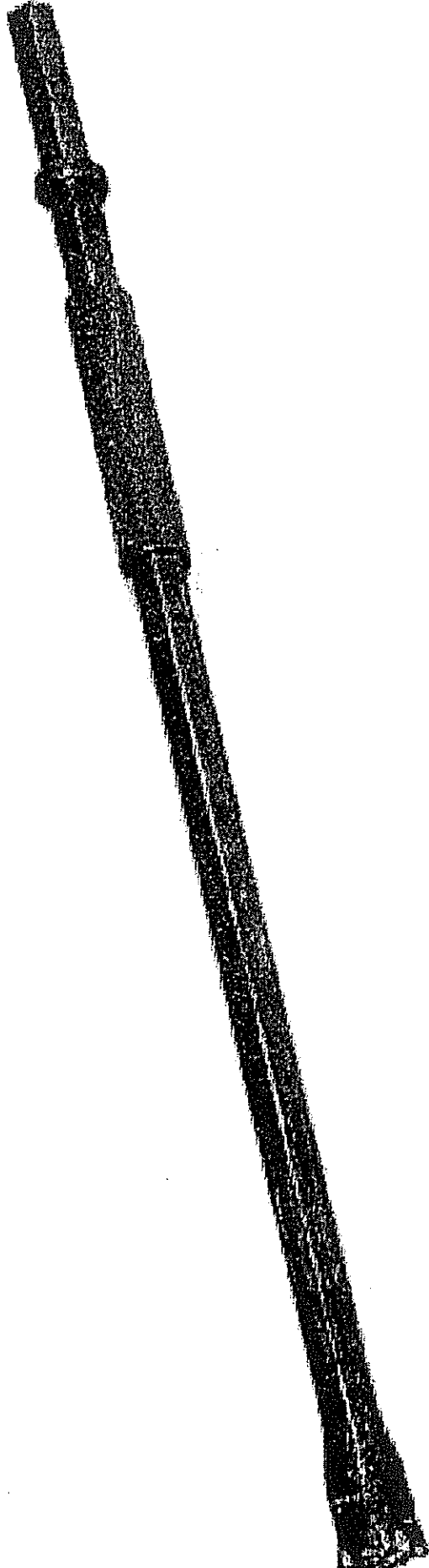


FIGURE 4.1: DRILL STEEL WITH DAMPING COLLAR



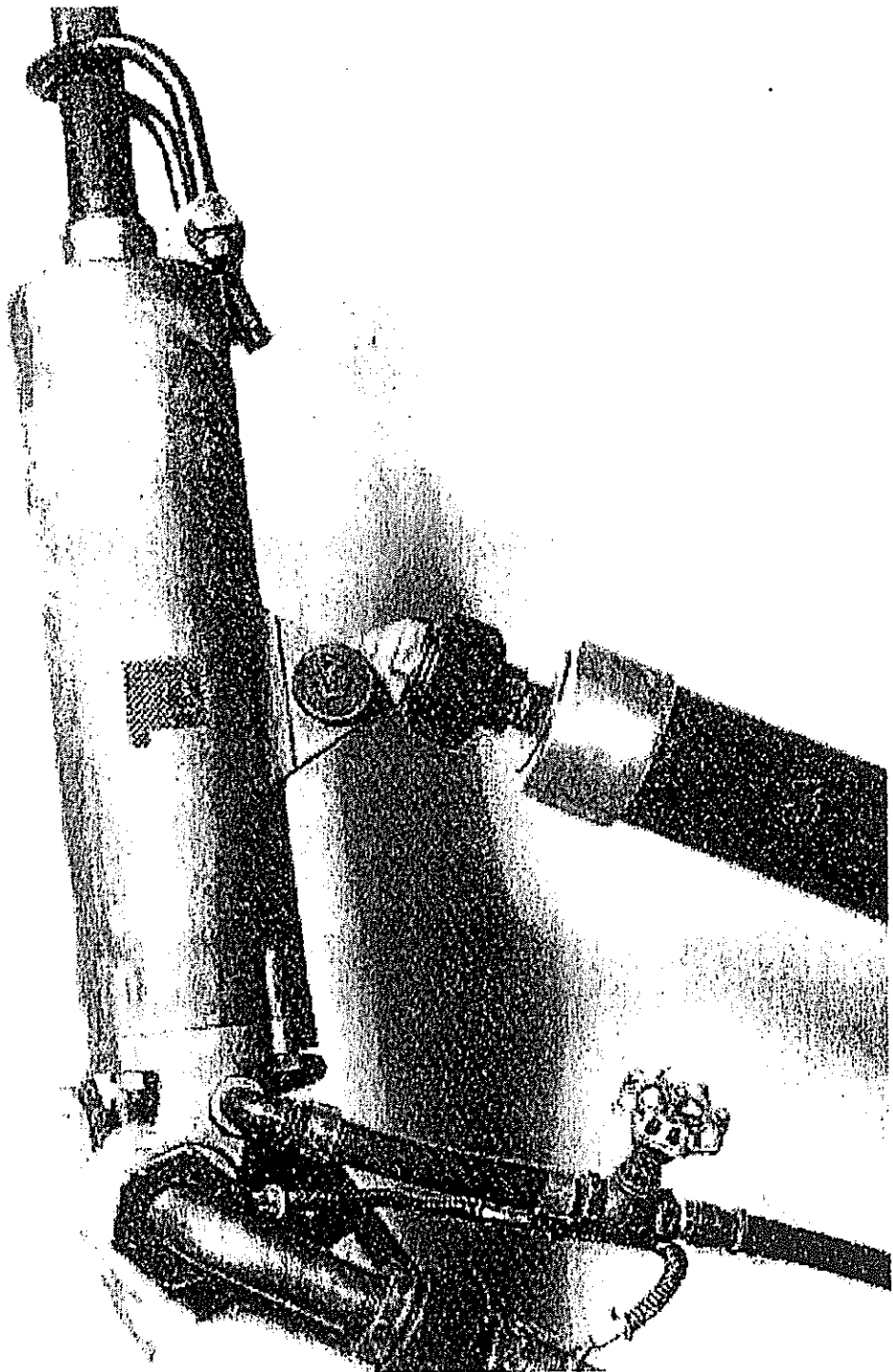


FIGURE 4.2: QUIET HARD ROCK DRILL

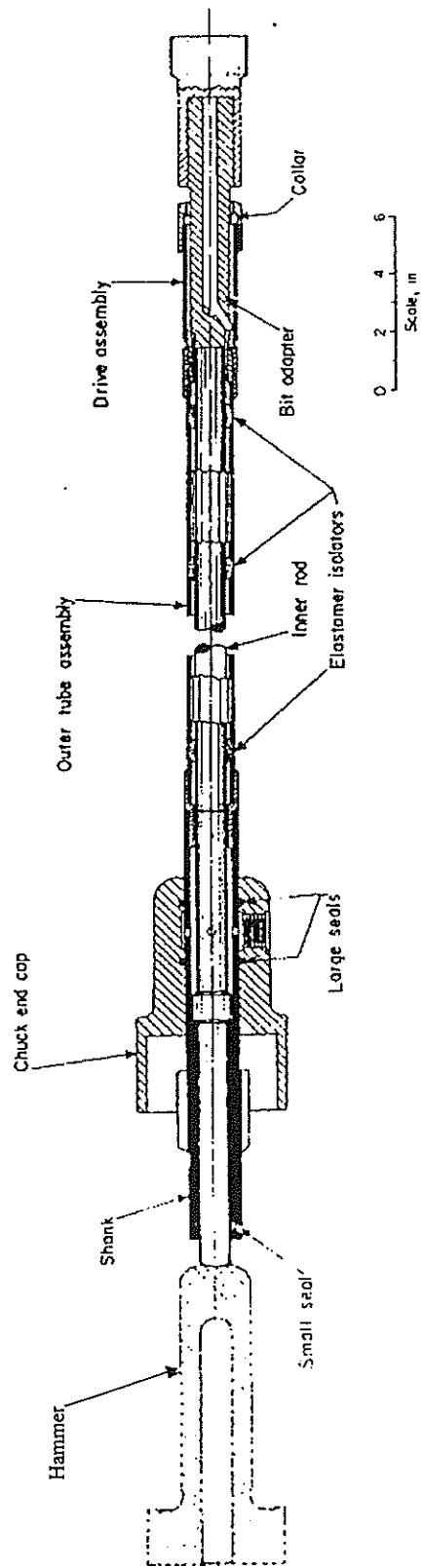


FIGURE 4.3 : CONCENTRIC DRILL STEEL DESIGN

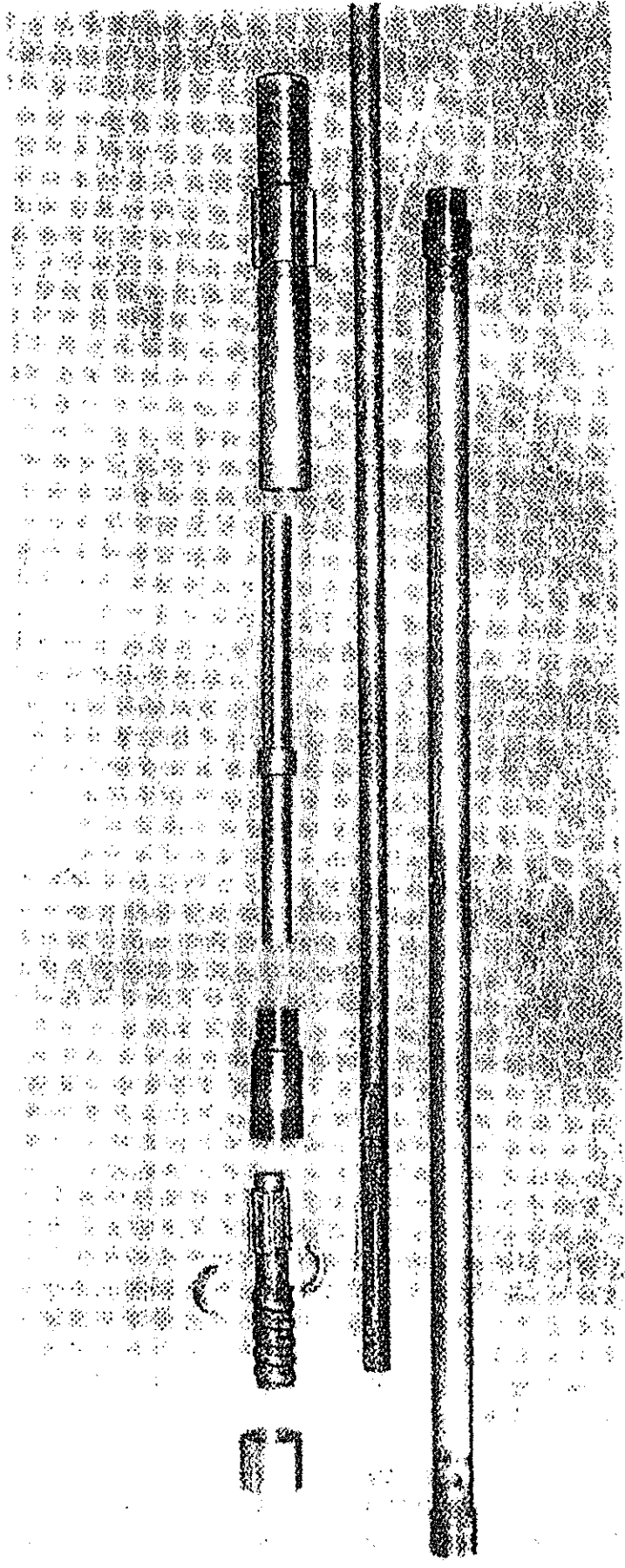


FIGURE 4.4: DRIFTER CONCENTRIC DRILL STEEL

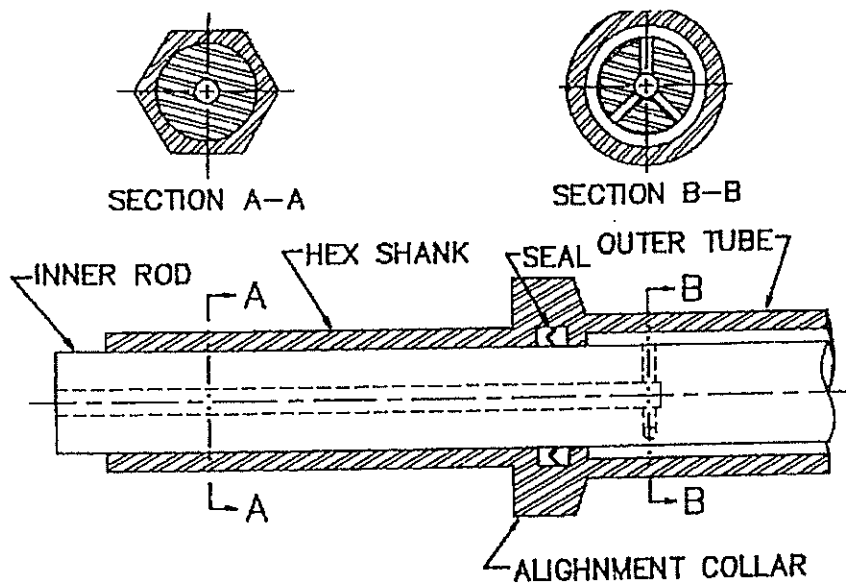


FIGURE 4.5: HAND HELD CONCENTRIC DRILL STEEL/SHANK END

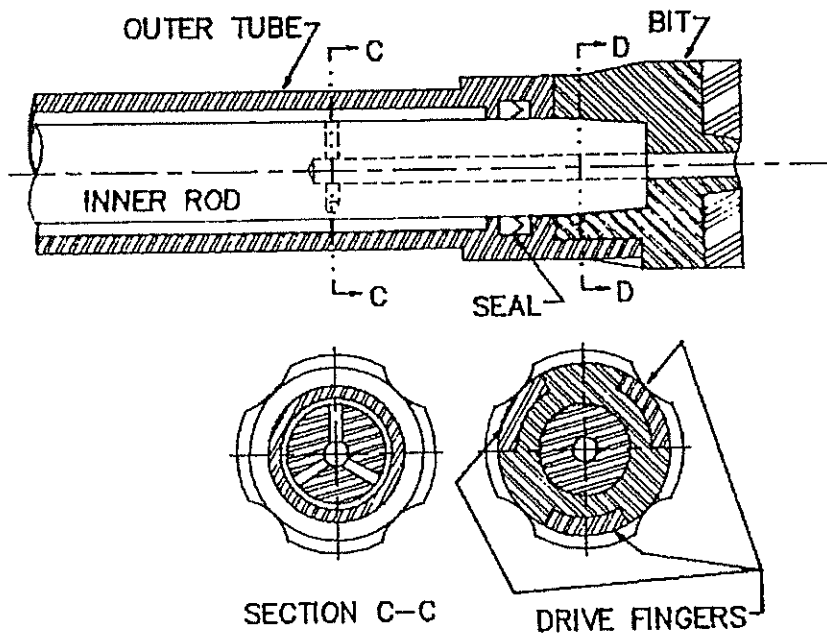


FIGURE 4.6: HAND HELD CONCENTRIC DRILL STEEL/BIT CONNECTION

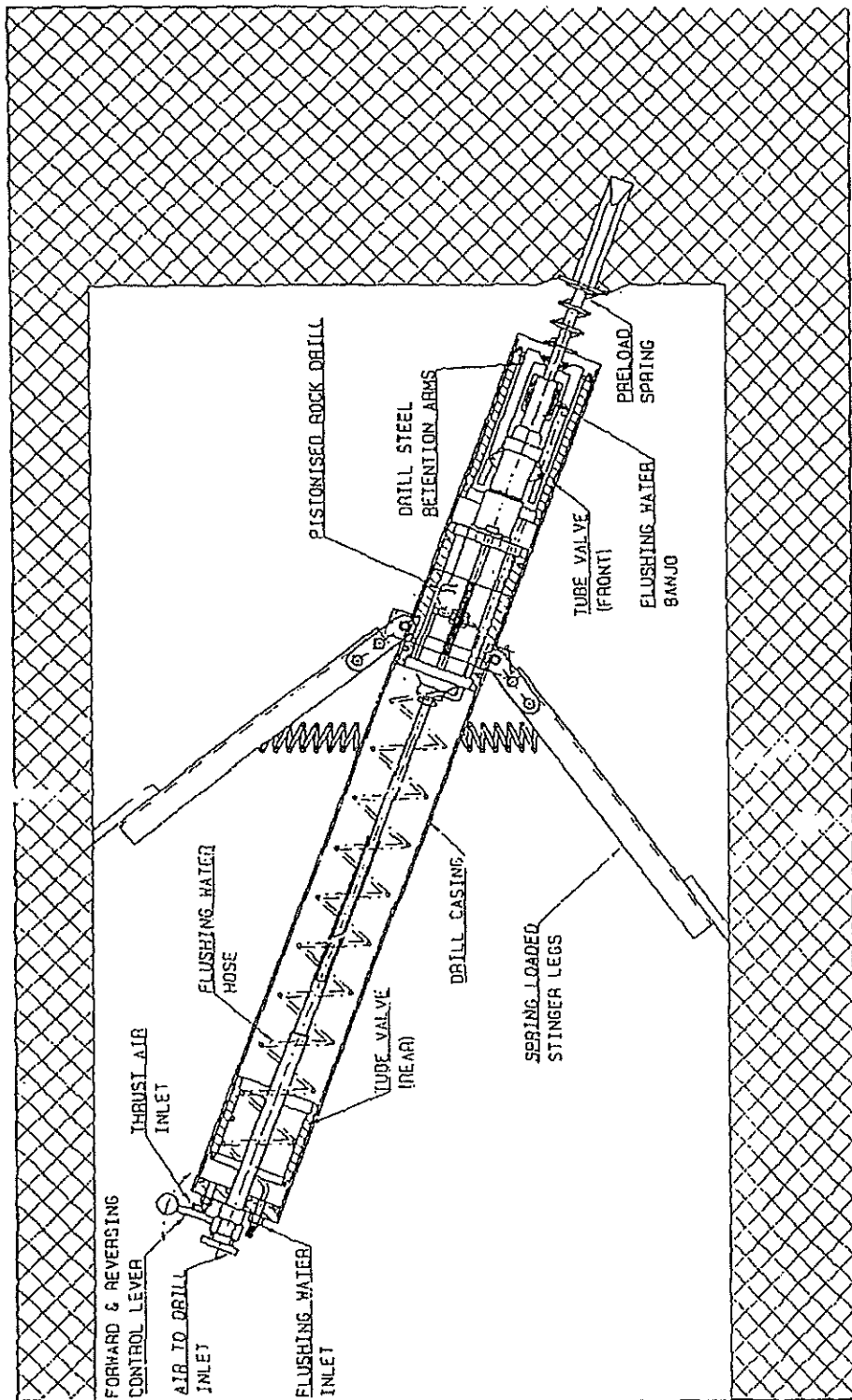


FIGURE 4.7: NOVEL DRILL DEVELOPMENT

TABLE 5.1: UNDERGROUND TESTS OF RETROFIT BOOM MOUNTED DRILL

DRILL POSITION	UNTREATED DRILL dB(A)	TREATED DRILL dB(A)	NOISE REDUCTION dB(A)
COLLARING HOLE (3M OF STEEL)	117.5	105	12.5
MIDDLE OF HOLE (1.8M OF STEEL)	116	101	15
END OF HOLE (0.6M OF STEEL)	115	101.5	14

TABLE 5.2: TEST RESULTS FOR ROTARY HYDRAULIC DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
					OPERATOR	1 m FROM DRILL
A	380	1270	2.10	2-WING RADIUS	90	99
B	650	1230	2.05		91	101
C	350	680	2.00		90	99
D*	400	1290	1.35		86	90
E	400	1480	1.85		92	100
F	620	760	1.60		90	100
G	450	2545	4.05		92	103
H*	450	2680	2.77		93	-
I*	350	204	1.20		88	94

\* ROTARY PERCUSSIVE DRILLS OPERATED IN ROTARY MODE

TABLE 5.3: TEST RESULTS FOR HYDRAULIC ROTARY-HYDRAULIC PERCUSSION DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	BLOWS (/minute)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
						OPERATOR	1 m FROM DRILL
J	350	1430	2800	2.00	cross bit	97	104
K	-	-	-	2.05	cross bit	95	105
L	-	-	-	1.46	cross bit	101	108
M	200	1360	3400	1.70	cross bit	97	105
N	-	-	-	-	cross bit	99	104
O	170	1360	3600	1.65	cross bit	99	108
D	400	1290	5300	3.65	2 wing radius	92	99
I	350	820	3100	1.30	cross bit	100	106
I	350	820	3100	1.50	3 wing radius	99	106
P	450	1200	3600	1.52	cross bit	97	106
P	450	1200	3600	3.10	3 wing radius	93	105
H	460	1725	3000	0.96	cross bit	98	107
H	460	1725	3000	2.77	3 wing radius	95	106



TABLE 5.4: TEST RESULTS FOR HYDRAULIC ROTARY-PNEUMATIC PERCUSSION DRILLS

DRILL TYPE	DRILL SPEED (rev/min)	DRILL THRUST (kgf)	BLOWS (/minute)	PENETRATION (metres/min)	BIT TYPE	MEASURED NOISE LEVEL dB(A)	
						OPERATOR	1 m FROM DRILL
Q STANDARD	400	1150	3100	3.05	RADIUS	104	110
Q MUFFLED	400	1150	3100	3.05	RADIUS	98	105
R STANDARD	-	-	-	-	-	101	107
R MUFFLED	-	-	-	-	-	97	105

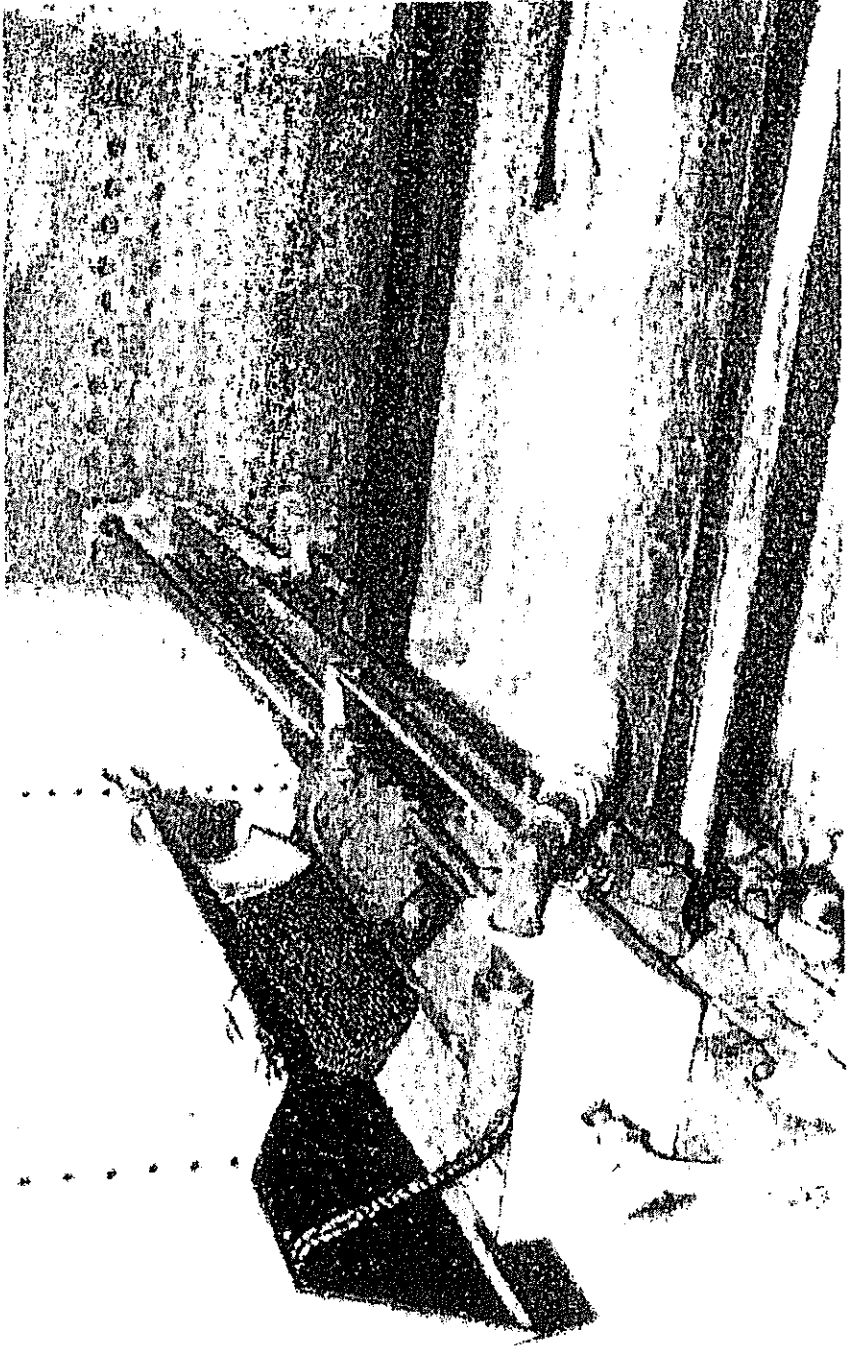


FIGURE 5.1: RETROFIT ACOUSTIC ENCLOSURE ON DRILL

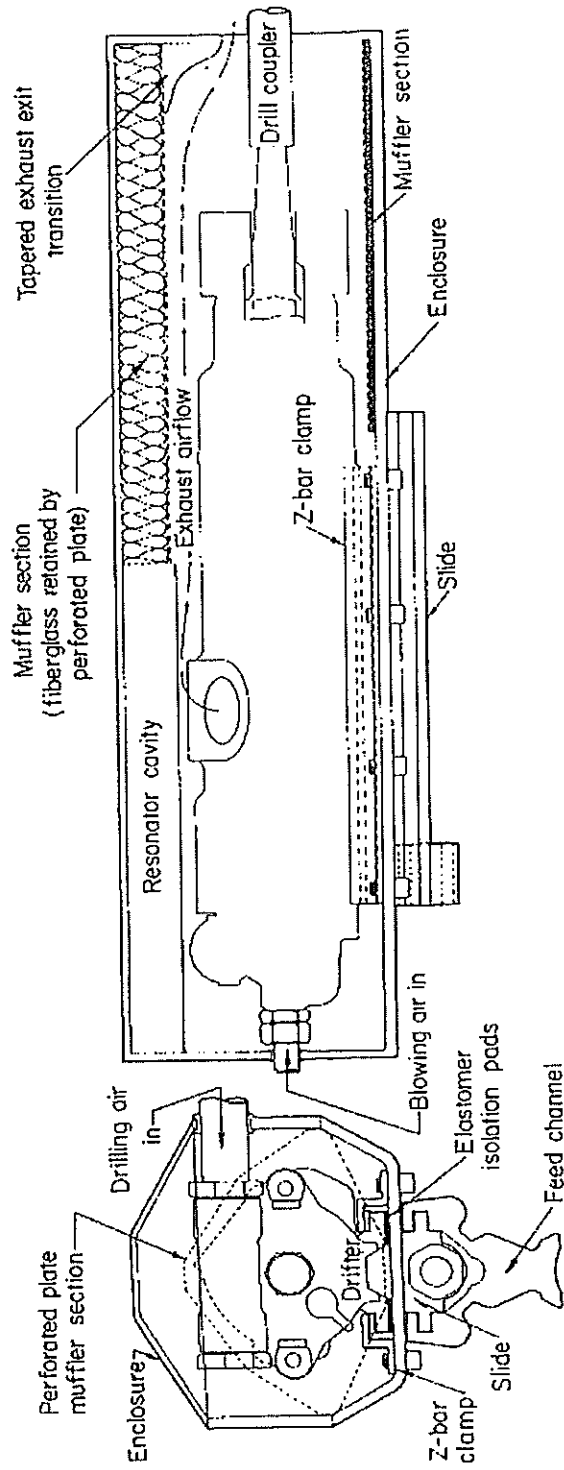


FIGURE 5.2: RETROFIT DRILL BODY ENCLOSURE

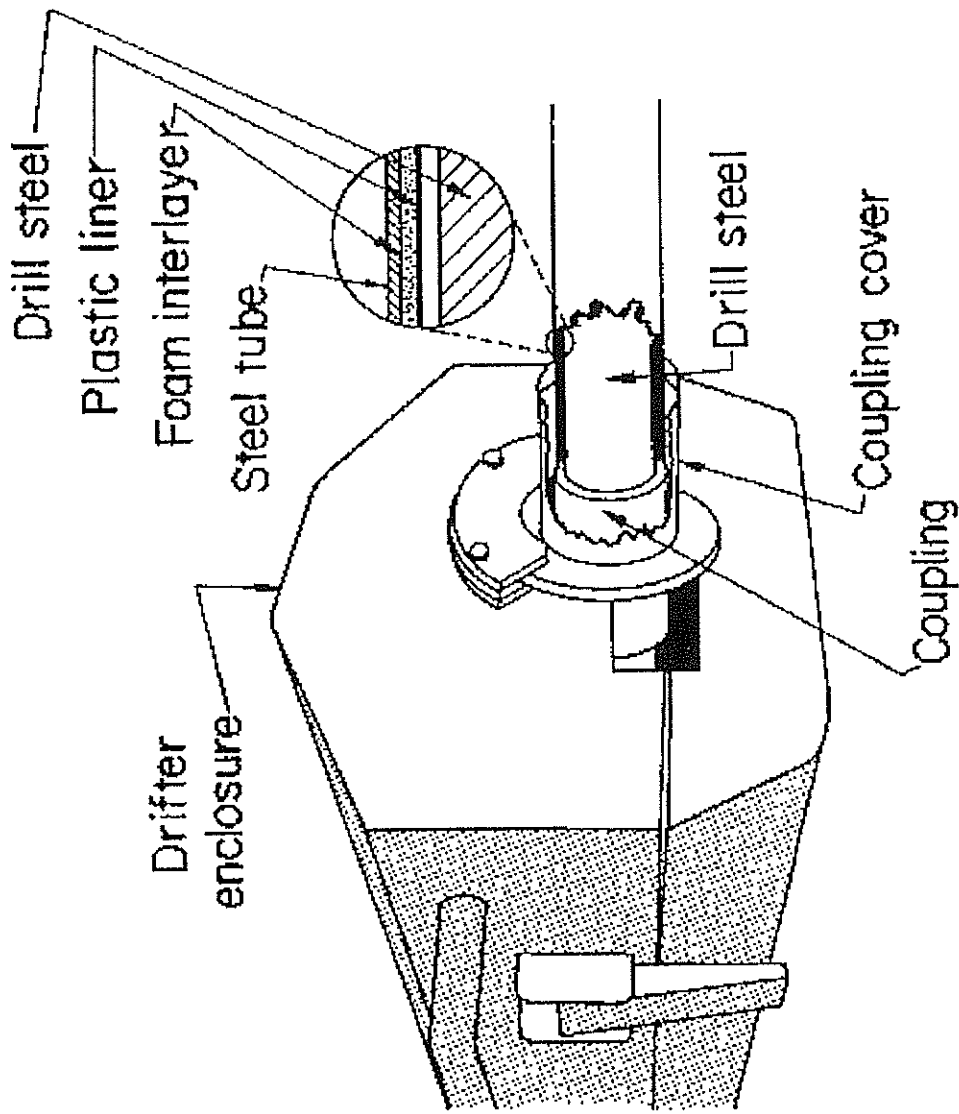


FIGURE 5.3: RETROFIT DRILL STEEL SHROUD TUBE

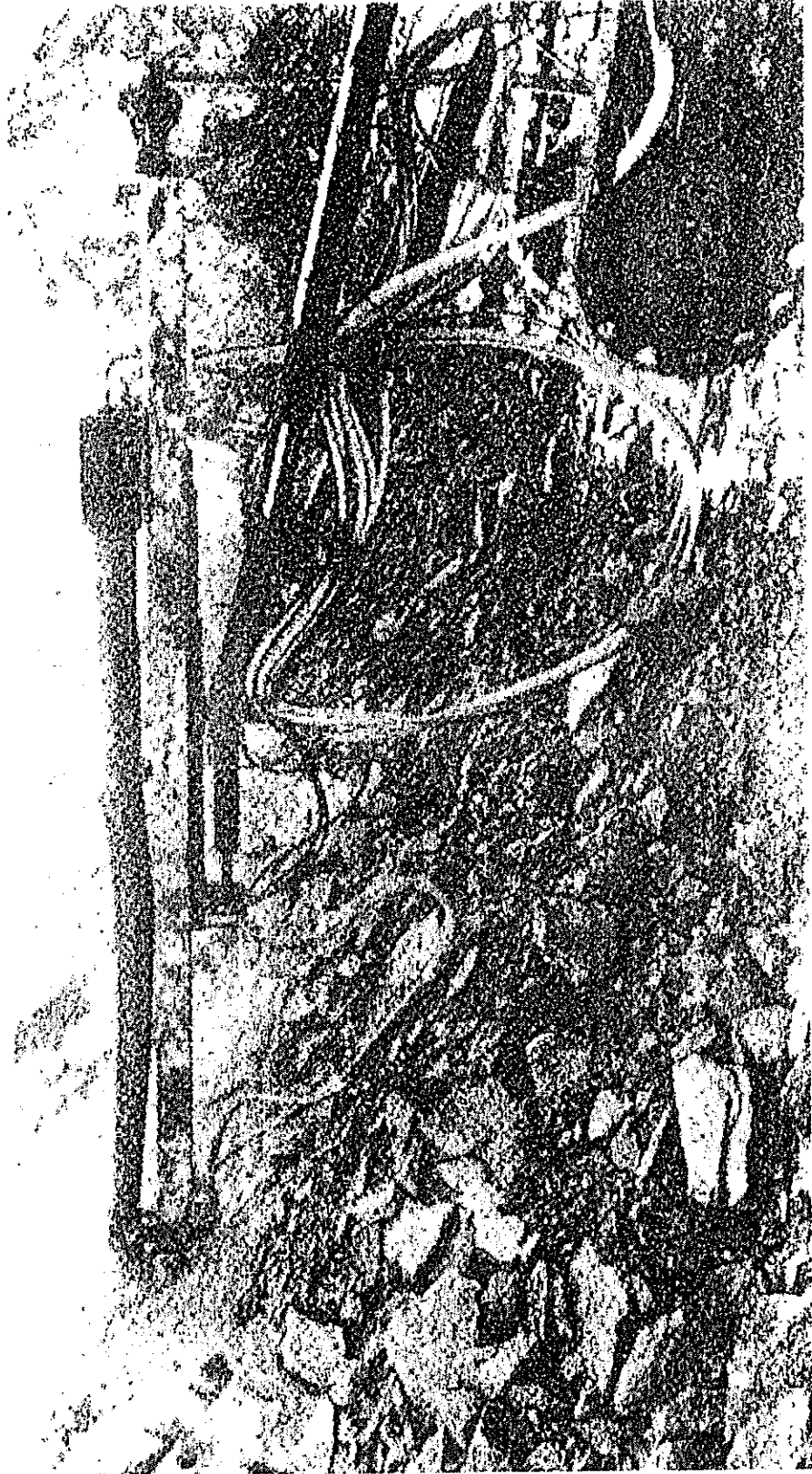


FIGURE 5.4: REDESIGNED DRILL WITH COLLAPSIBLE SHROUD TUBE

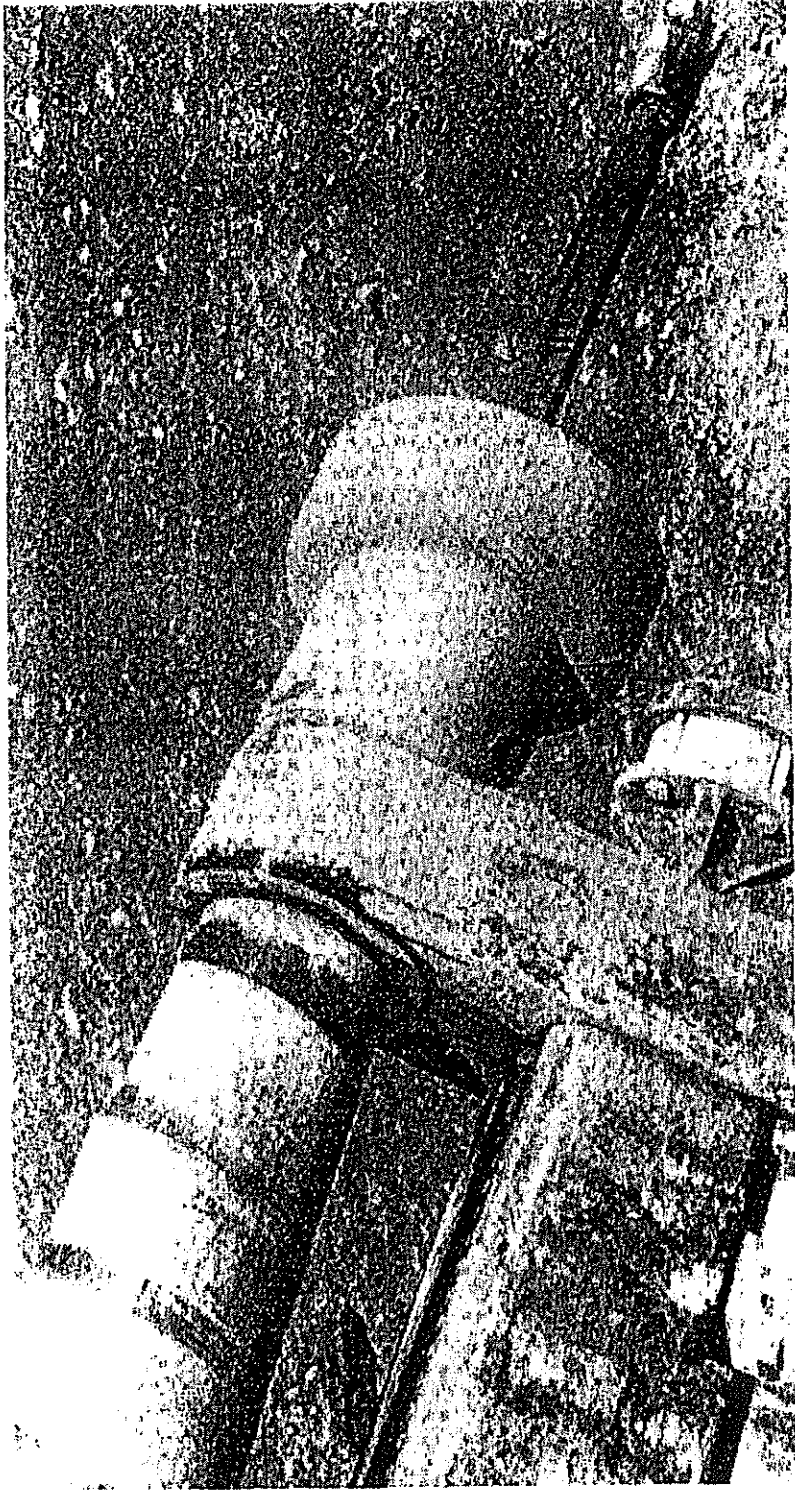


FIGURE 5.5: STINGER ARRANGEMENT

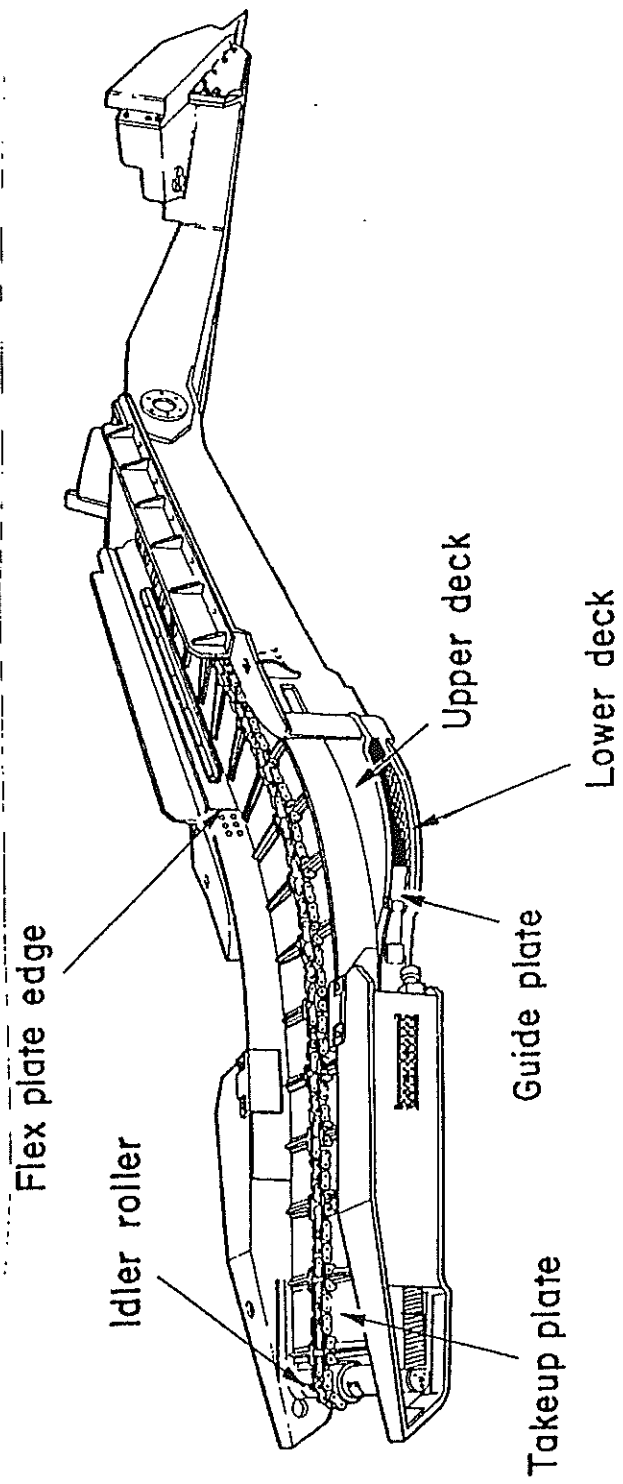


FIGURE 6.1: NOISE PRODUCING COMPONENTS OF A CONTINUOUS MINER CHAIN CONVEYOR

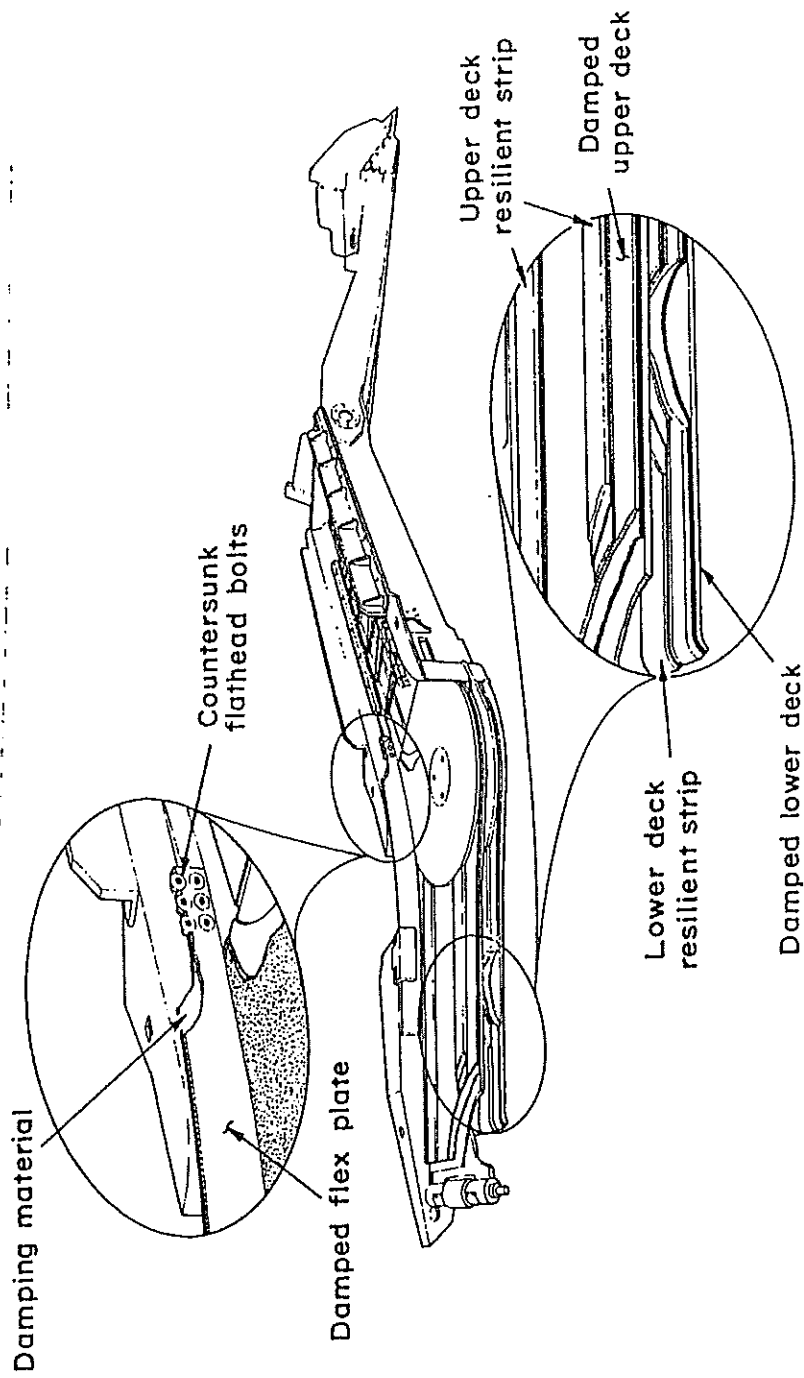


FIGURE 6.2: NOISE CONTROL TREATMENTS ON CONTINUOUS MINER CONVEYOR DECKS AND SIDEWALLS



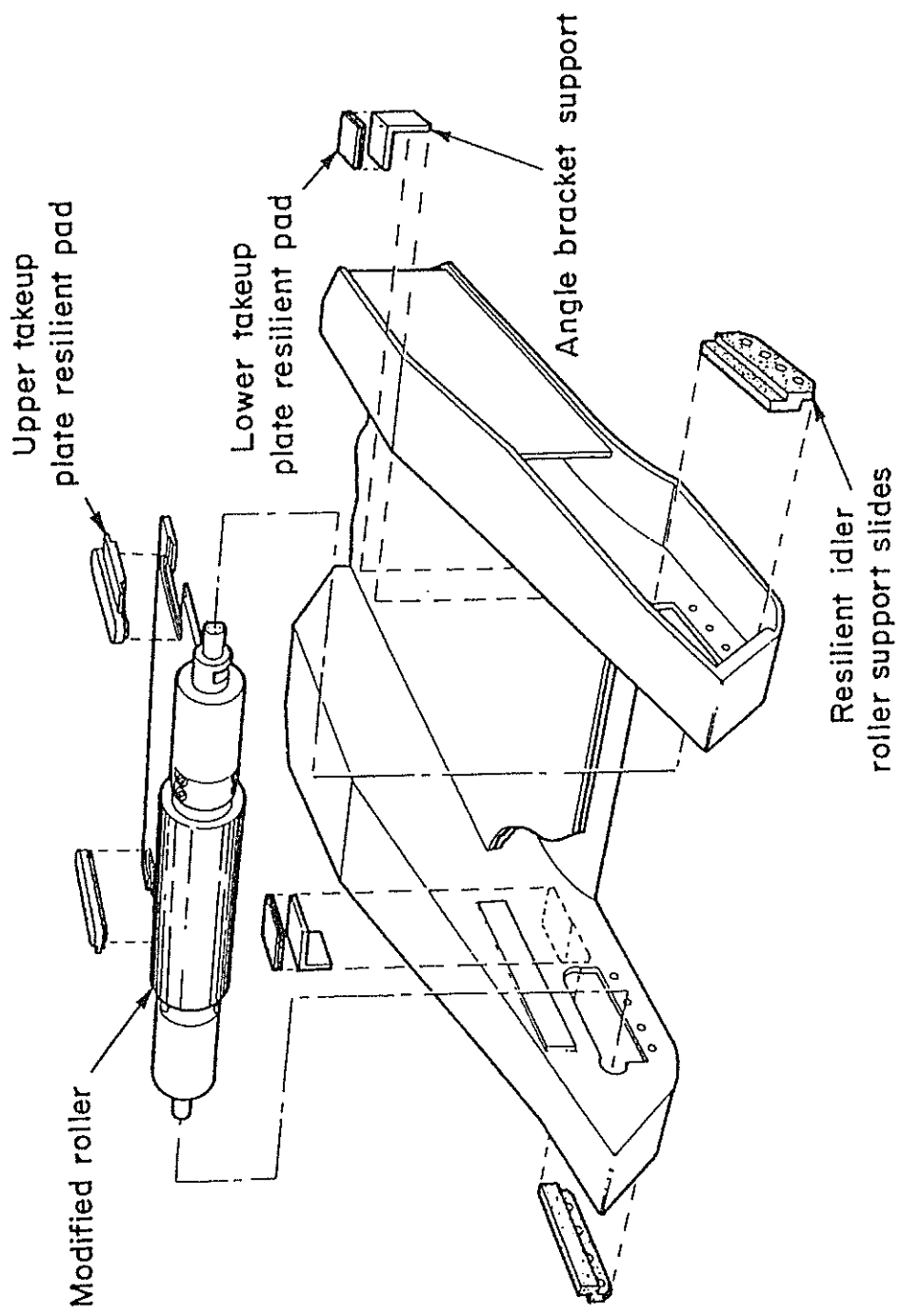


FIGURE 6.3: NOISE CONTROL TREATMENTS ON CONTINUOUS MINER CONVEYOR TAIL ROLLER AND TAKE-UP PLATE



FIGURE 6.4: JOY 12CM11 CONTINUOUS MINER

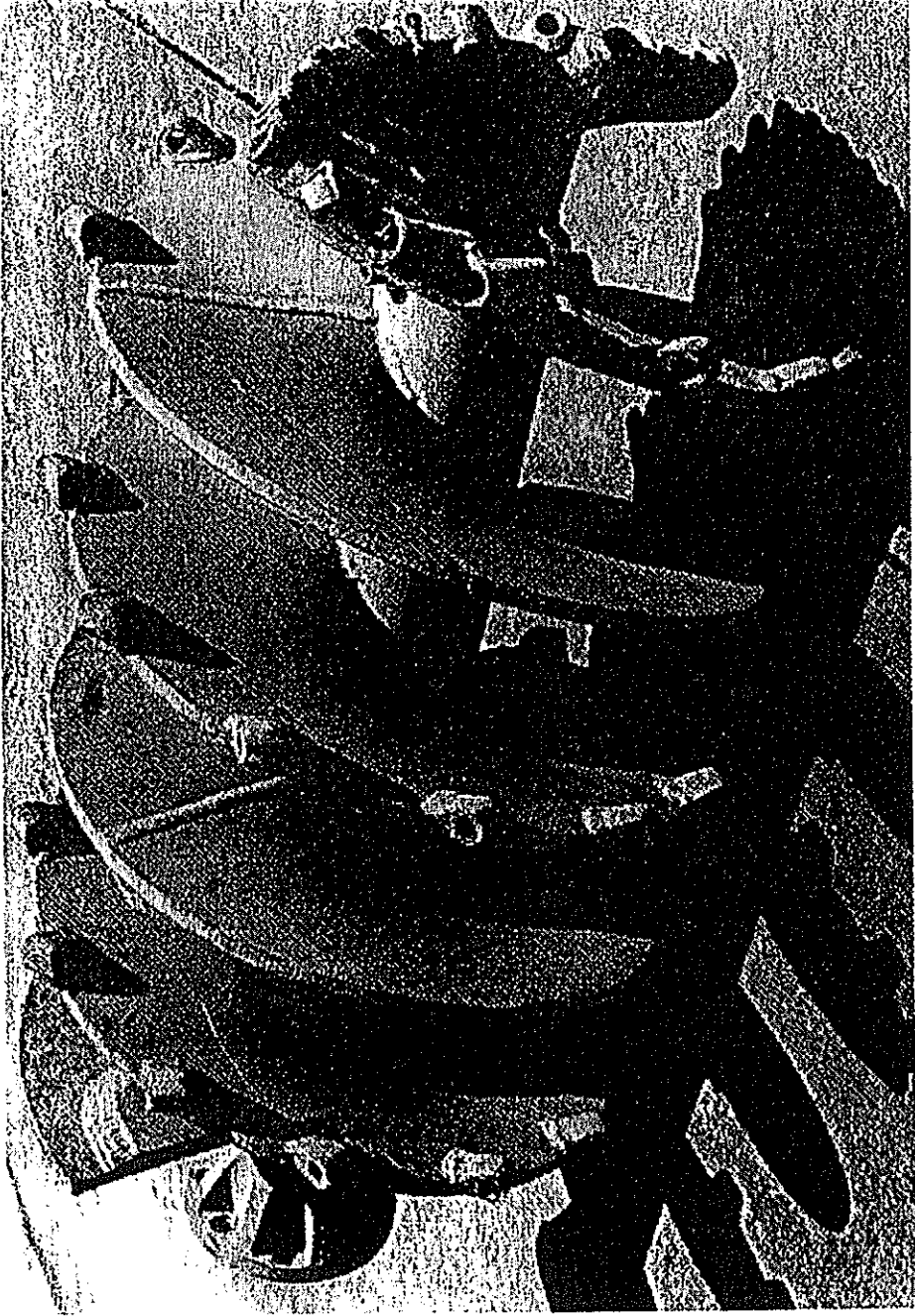


FIGURE 6.5: STANDARD AUGER MINER CUTTING HEAD

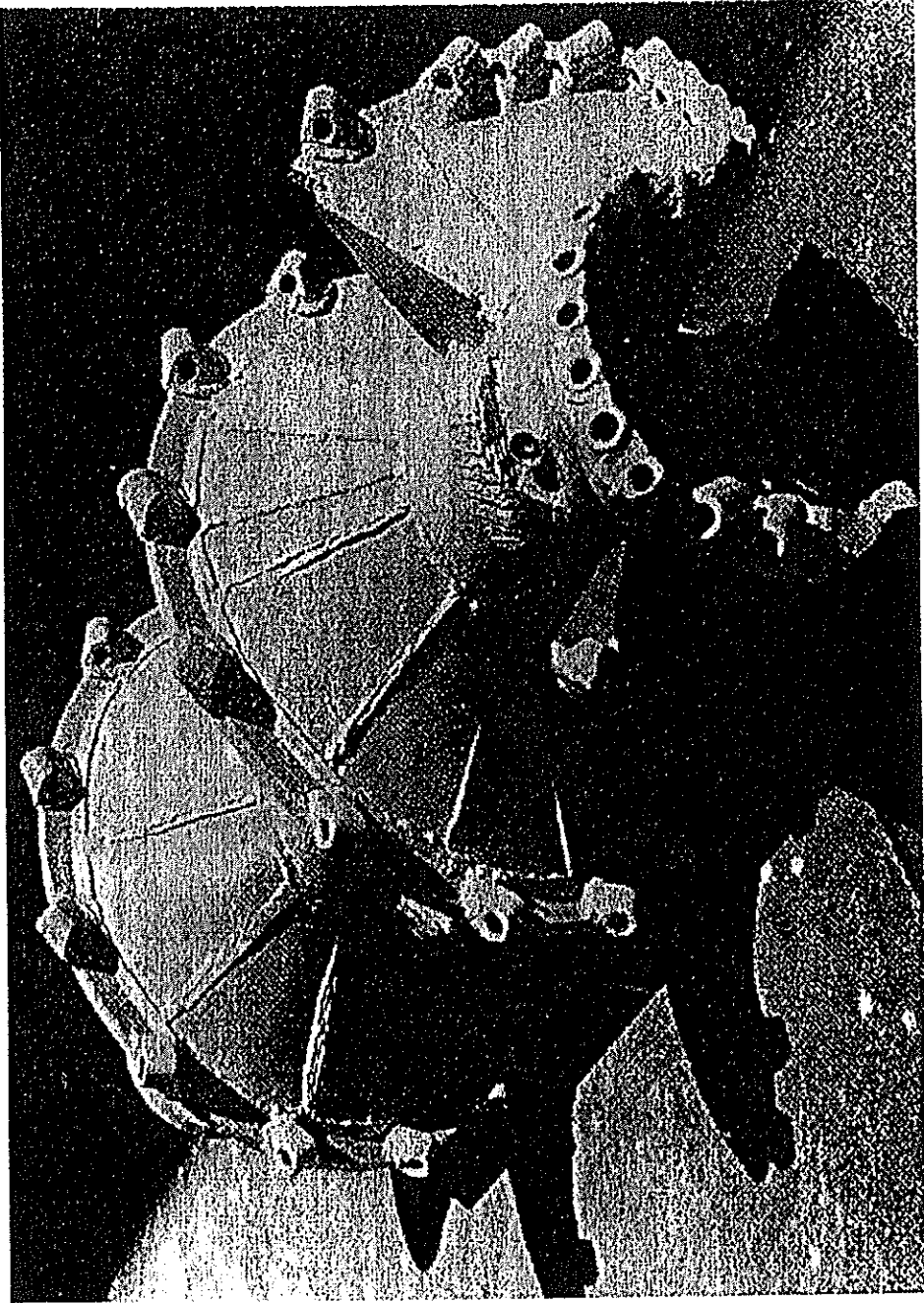


FIGURE 6.6A: REDUCED-NOISE AUGER MINER CUTTING HEAD



FIGURE 6.6B: REDUCED-NOISE AUGER MINER CUTTING HEAD

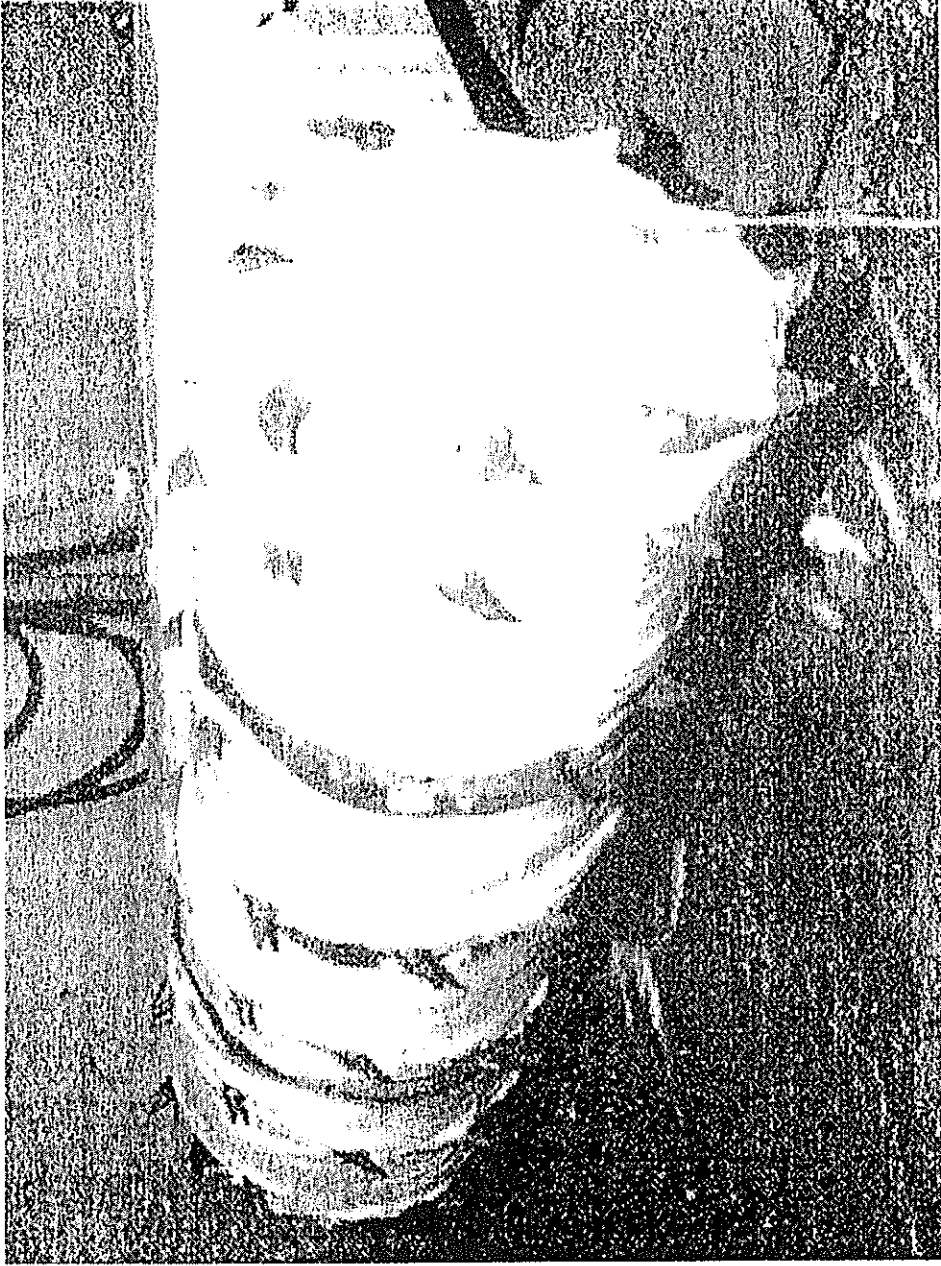


FIGURE 6.7: WRAPPED CUTTER HEAD OF CONTINUOUS MINER

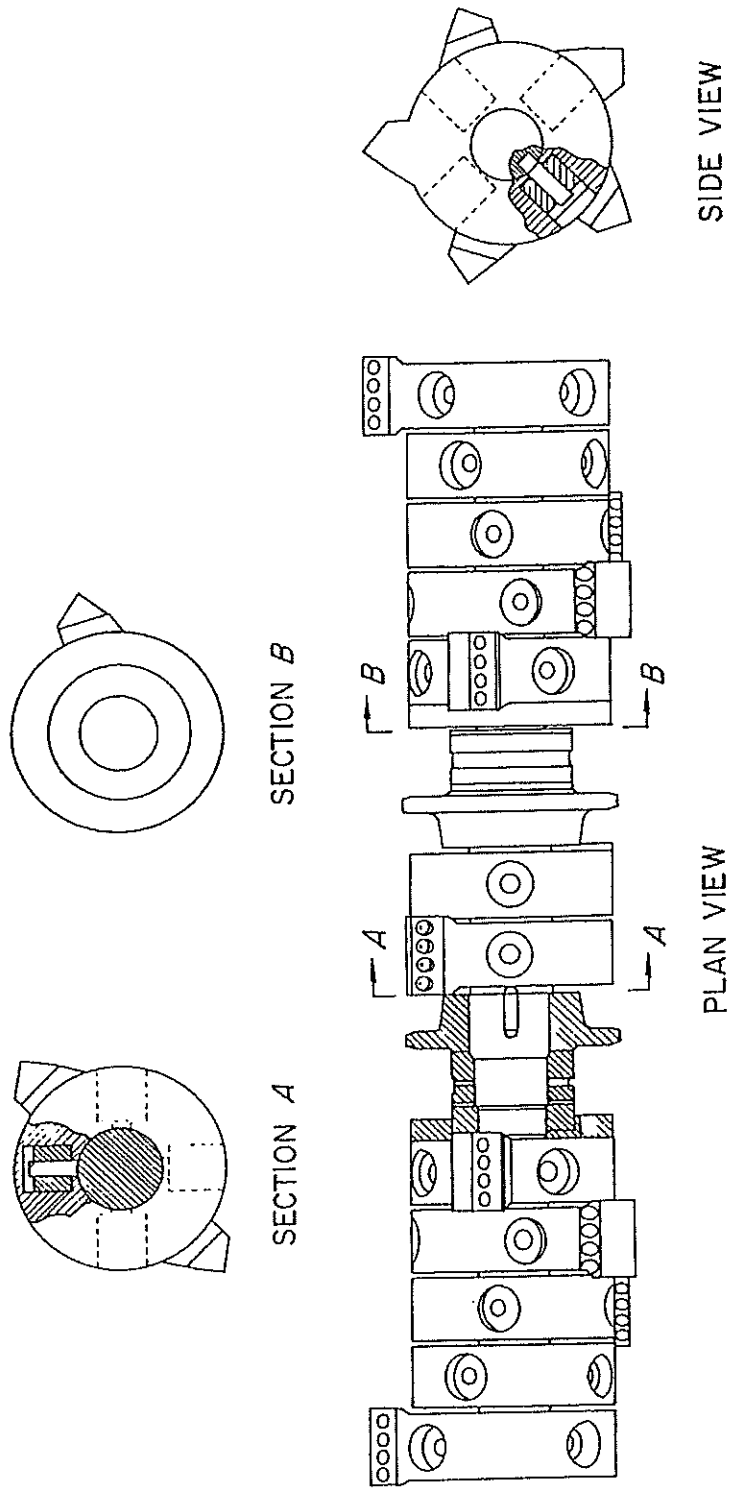


FIGURE 6.8: CONTINUOUS MINER ISOLATED CUTTING HEAD WITH STAGED BIT LACING

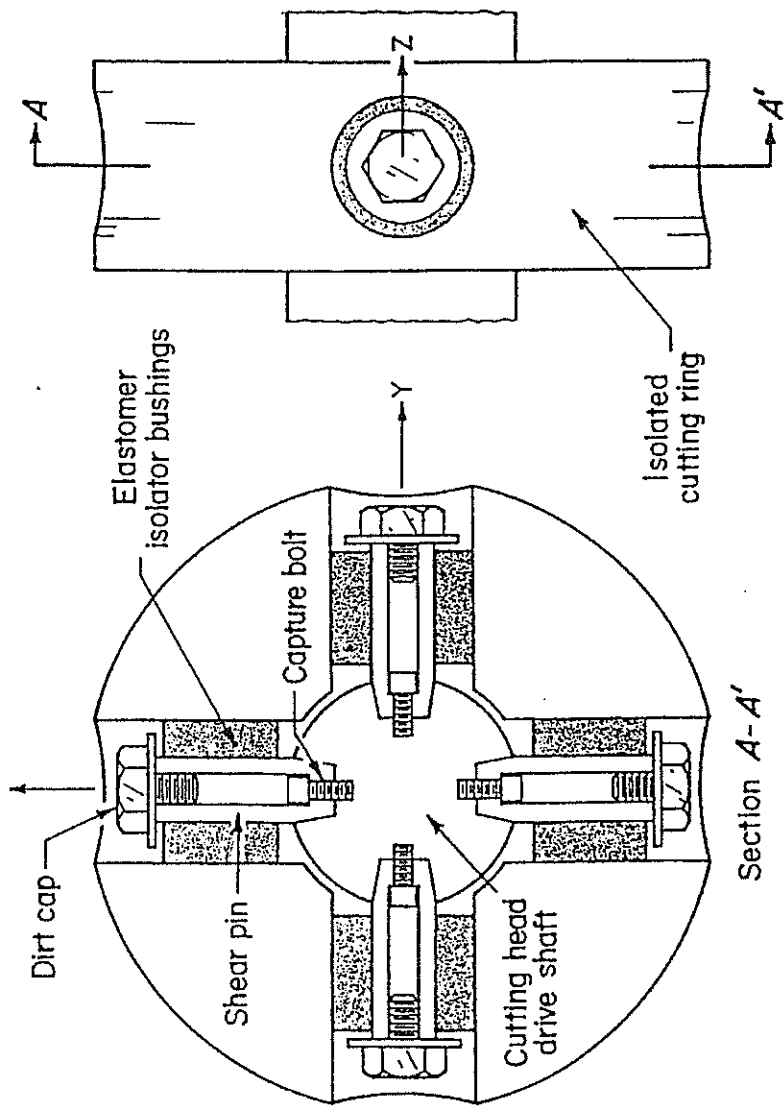


FIGURE 6.9: ISOLATED CONTINUOUS MINER CUTTING HEAD - ISOLATION SCHEME



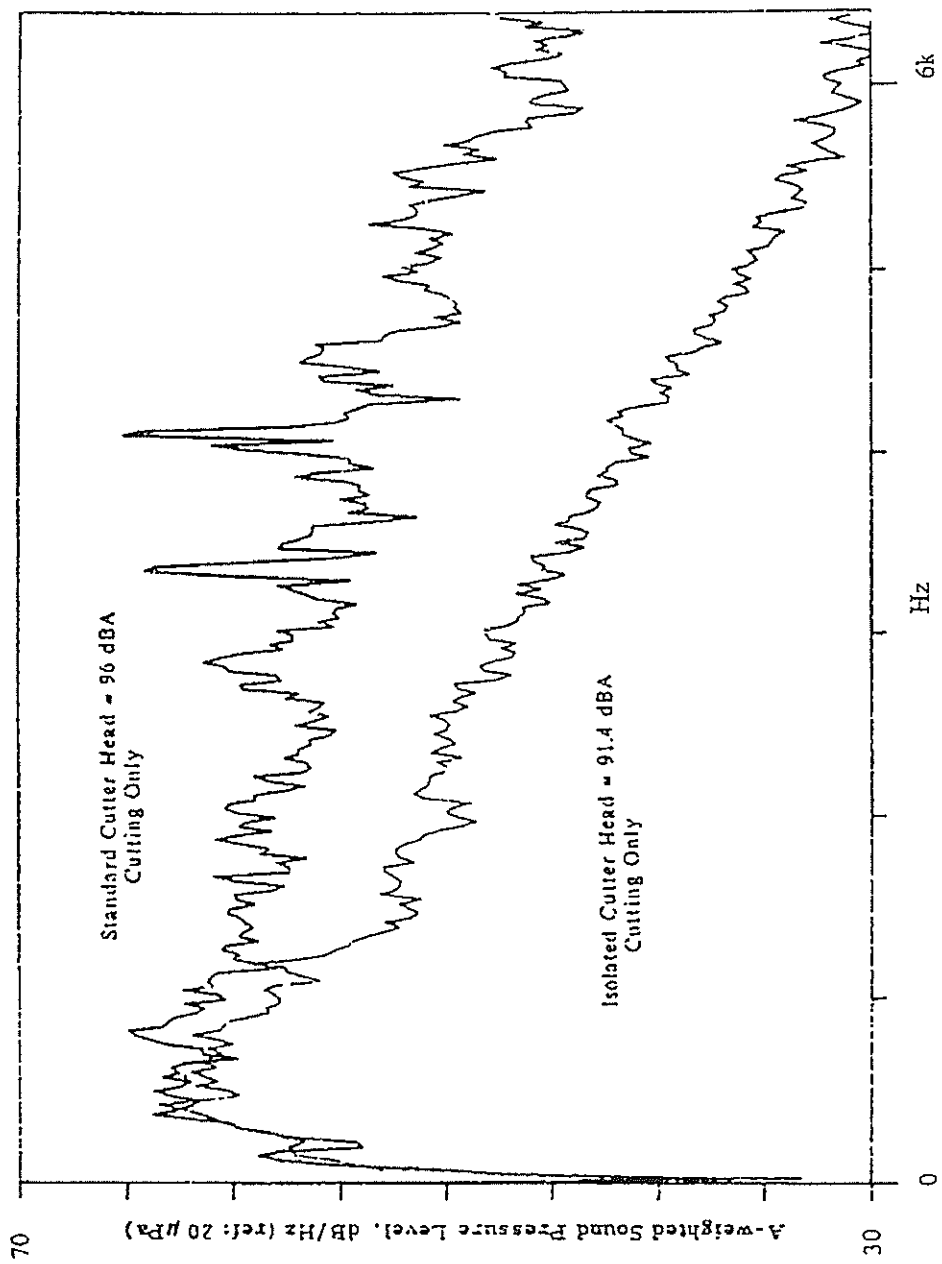


FIGURE 6.10: NOISE CONTROL PERFORMANCE OF ISOLATED CUTTING HEAD - REPRESENTATIVE SPECTRA

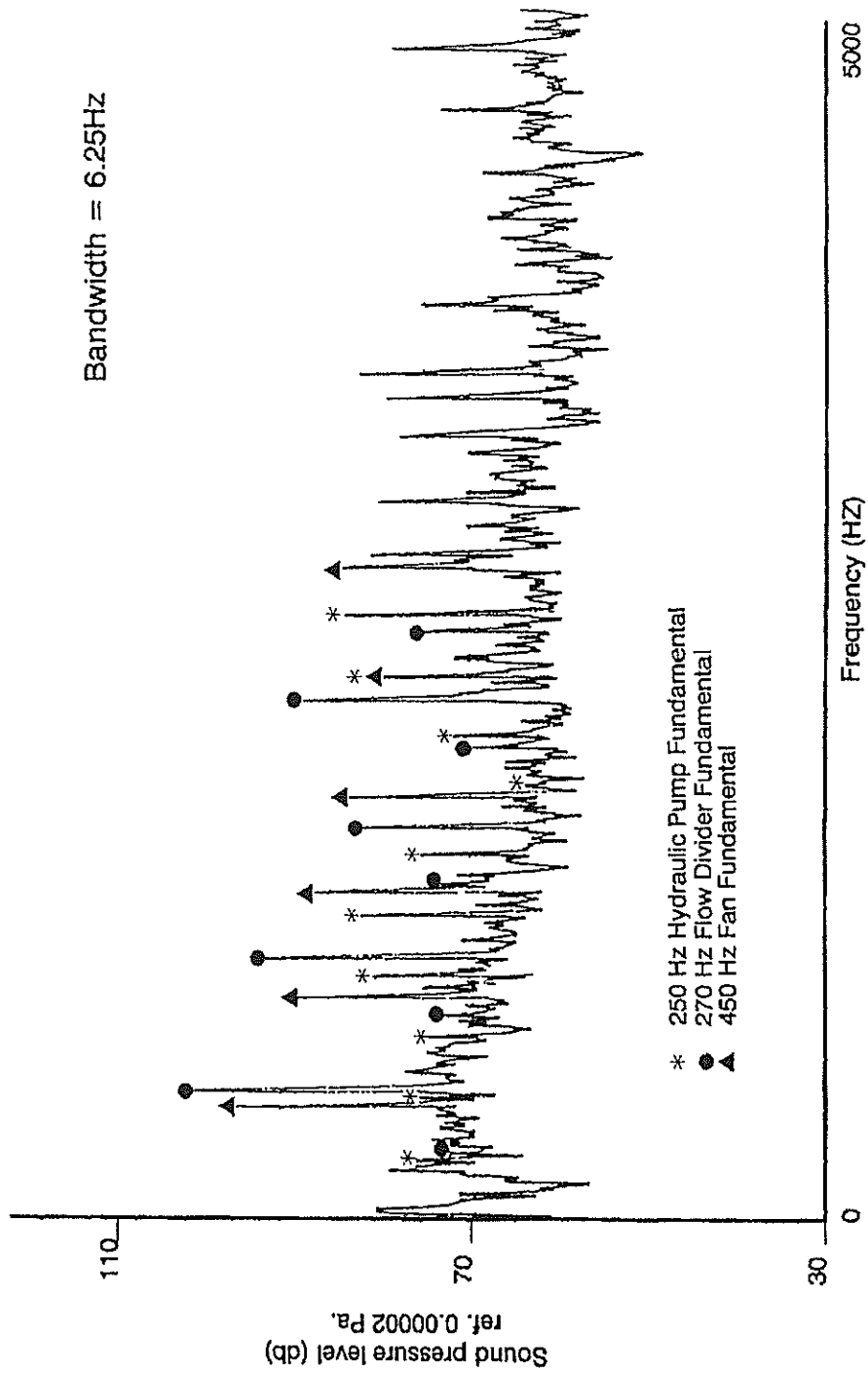
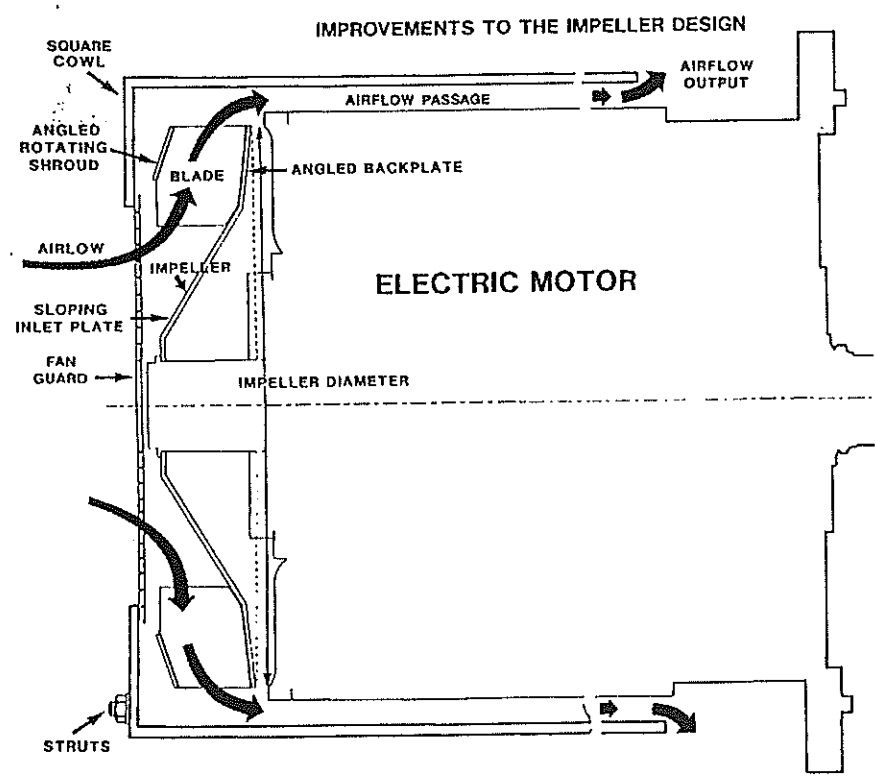
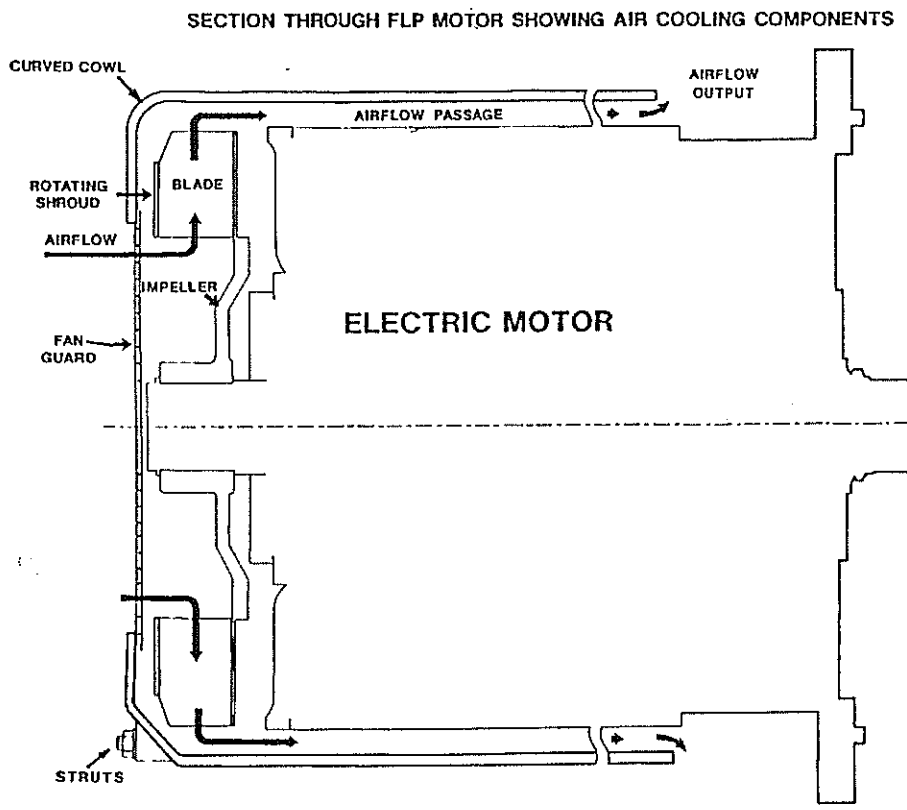
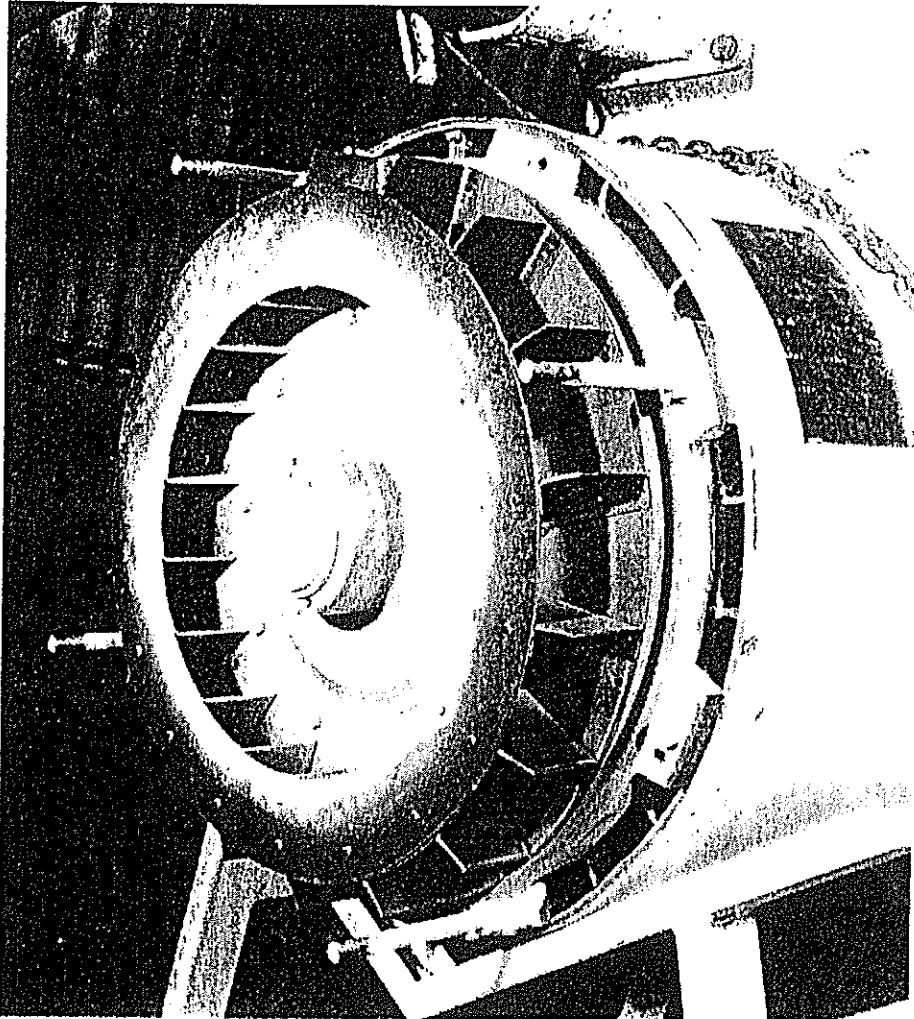


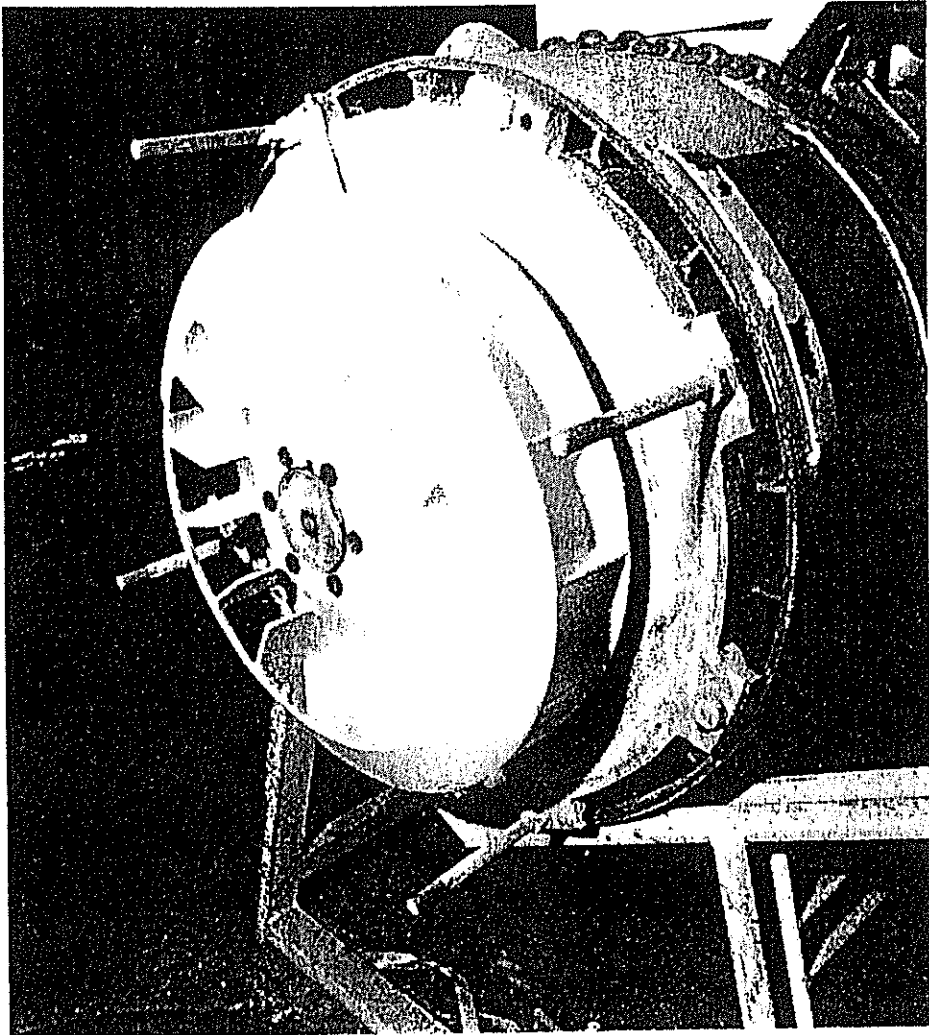
FIGURE 6.11: RH22 NOISE SPECTRUM AT OPERATOR'S POSITION



**FIGURE 6.12: ELECTRIC MOTOR BEFORE AND AFTER NOISE CONTROL RE-DESIGN**



**FIGURE 6.13A: STANDARD FAN IMPELLER ON SPEC 625 MOTOR**



**FIGURE 6.13B: LOW NOISE FAN IMPELLER ON SPEC 625 MOTOR**

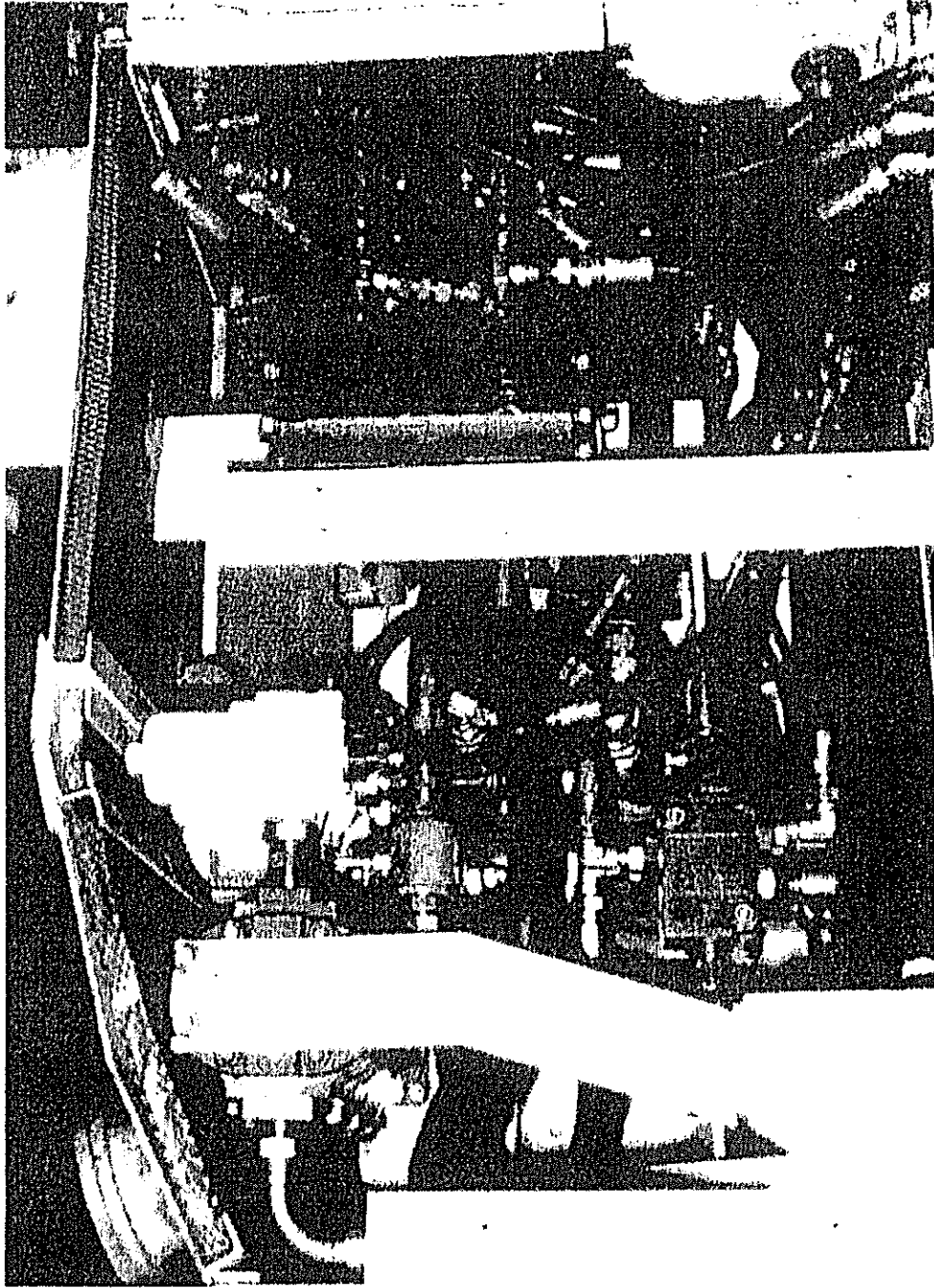


FIGURE 6.14: RH22 HYDRAULICS COMPARTMENT ENCLOSURE

POSITION	OVERALL NOISE LEVEL IN dB(A)					BEST REDUCTION ACHIEVED ie INLET SILENCER & OUTLET SILENCER
	UNTREATED	INLET SILENCER	INLET SILENCER & OUTLET SILENCER	OUTLET DEFLECTOR	INLET DUCTING & OUTLET DEFLECTOR	
1	104	103	96	106	105	8
2	104.5	103.1	99.5	105.2	105	5
3	107.2	101.1	97.4	107.1	106.2	10
4	103.3	101	96.8	104.2	105.7	6
5	108.7	108	107.2	111.3	109.9	2
6	107.8	106.9	104.7	108.4	108.3	3
7	103.4	99.4	96.3	102.9	102.6	7
8	108	98.9	97	108.9	103.1	11
9	107.7	98.3	96.1	108.3	101.9	12

POSITION 1 IS TYPICAL (REMOTE CONTROL) MACHINE OPERATOR POSITION  
POSITION 2 IS TYPICAL SHUTTLE CAR DRIVER POSITION

TABLE 7.1: MEASURED SCRUBBER NOISE LEVELS FROM TESTS ON A MOCK ROADHEADER AT SWADLINCOTE TEST SITETABLE

3. WADLINCOTE SCRUBBER - MEASURED LEVELS IN SIMRAC GALLERY

POSITION 1 - TYPICAL MACHINE OPERATOR (REMOTE CONTROL) POSITION

SCRUBBER WITH INLET SILENCER

	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	
A weighting Correction	-26	-16	-9	-3	0	1	1	-1	
Resultant Noise Level dB(A)	50	64	98	92	96	94	87	78	103

SCRUBBER WITH INLET SILENCER

+ SQUARE SILENCER ON OUTLET

	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	75	78	98	89	91	88	83	74	
A weighting Correction	-26	-16	-9	-3	0	1	1	-1	
Resultant Noise Level dB(A)	49	62	89	86	91	89	84	73	96

TABLE 7.2: MEASURED SCRUBBER NOISE LEVELS  
AT TYPICAL MACHINE OPERATOR POSITION



ENGART SILENCERS NOISE DATA FROM MANUFACTURER'S INFORMATION SHEET

SCRUBBER WITH INLET SILENCER (MEASURED LEVELS AT SWADLINCOTE)  
 POSITION 1 - TYPICAL MACHINE OPERATOR (REMOTE CONTROL) POSITION

CYLINDRICAL SILENCER - SIZE 762 ON OUTLET

D SILENCER ON OUTLET	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
A-weighted Level	50	64	98	92	96	94	87	78	103
Dynamic Insertion Loss 1D *	-4	-3	-9	-17	-14	-12	-13	-14	
Noise Contribution From Outlet	46	61	89	75	82	82	74	64	91
Estimated Noise Contribution From Inlet									89
Resultant Noise Level at Position 1									93

1.5D SILENCER ON OUTLET	OCTAVE BAND (Hz)								TOTAL LEVEL dB(A)
	63	125	250	500	1000	2000	4000	8000	
Position 1 Noise Level dB	76	80	107	95	96	93	86	79	
A-weighting Correction	-26	-16	-9	-3	0	1	1	-1	
A-weighted Level	50	64	98	92	96	94	87	78	103
Dynamic Insertion Loss 1.5D *	-5	-5	-11	-20	-17	-15	-15	-17	
Noise Contribution From Outlet	44	56	86	73	81	83	77	69	89
Estimated Noise Contribution From Inlet									89
Resultant Noise Level at Position 1									92

Silencer length 1D = 1 diameter, 1.5D = 1.5 diameters

TABLE 7.3: ESTIMATIONS OF SCRUBBER NOISE LEVELS USING  
 ENGART OUTLET SILENCER

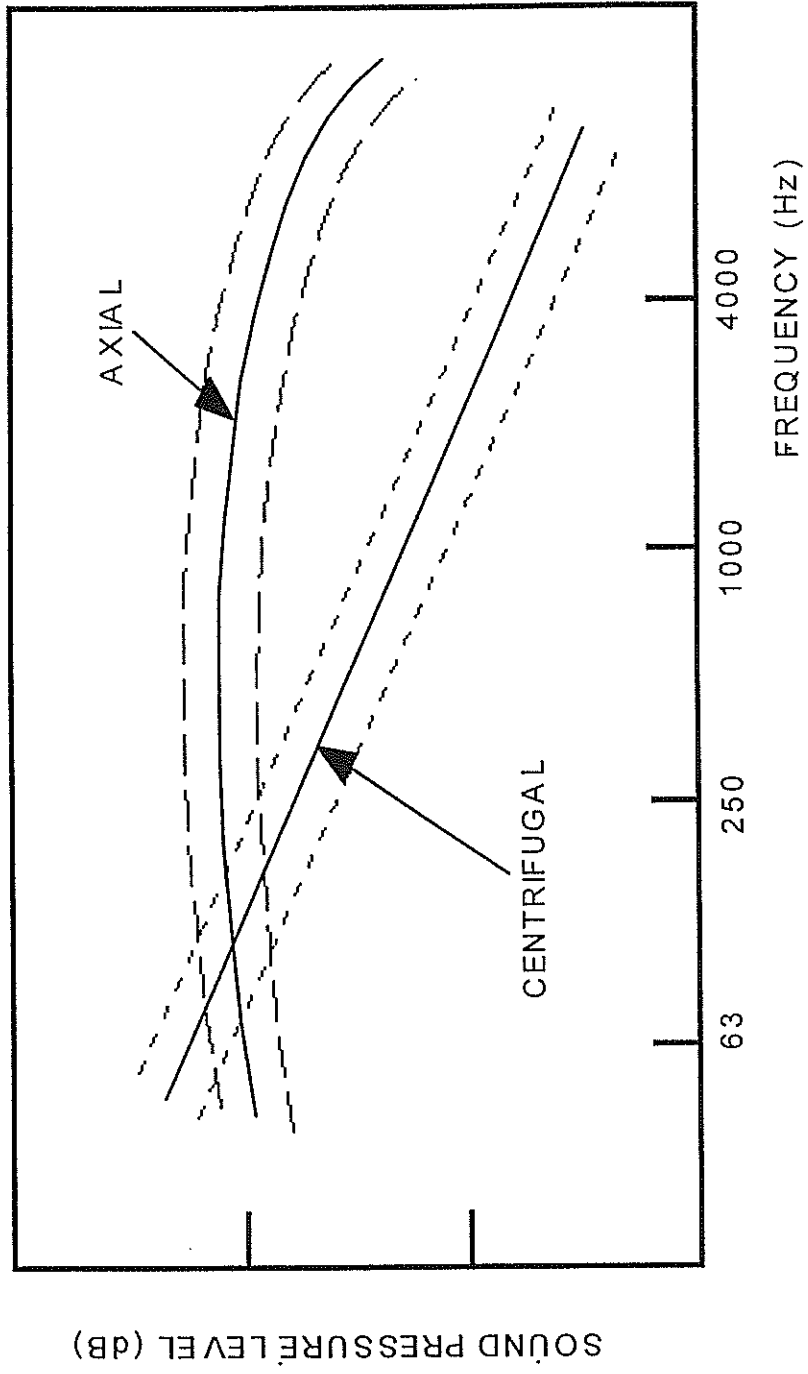


FIGURE 7.1: SOUND POWER SPECTRA OF FANS

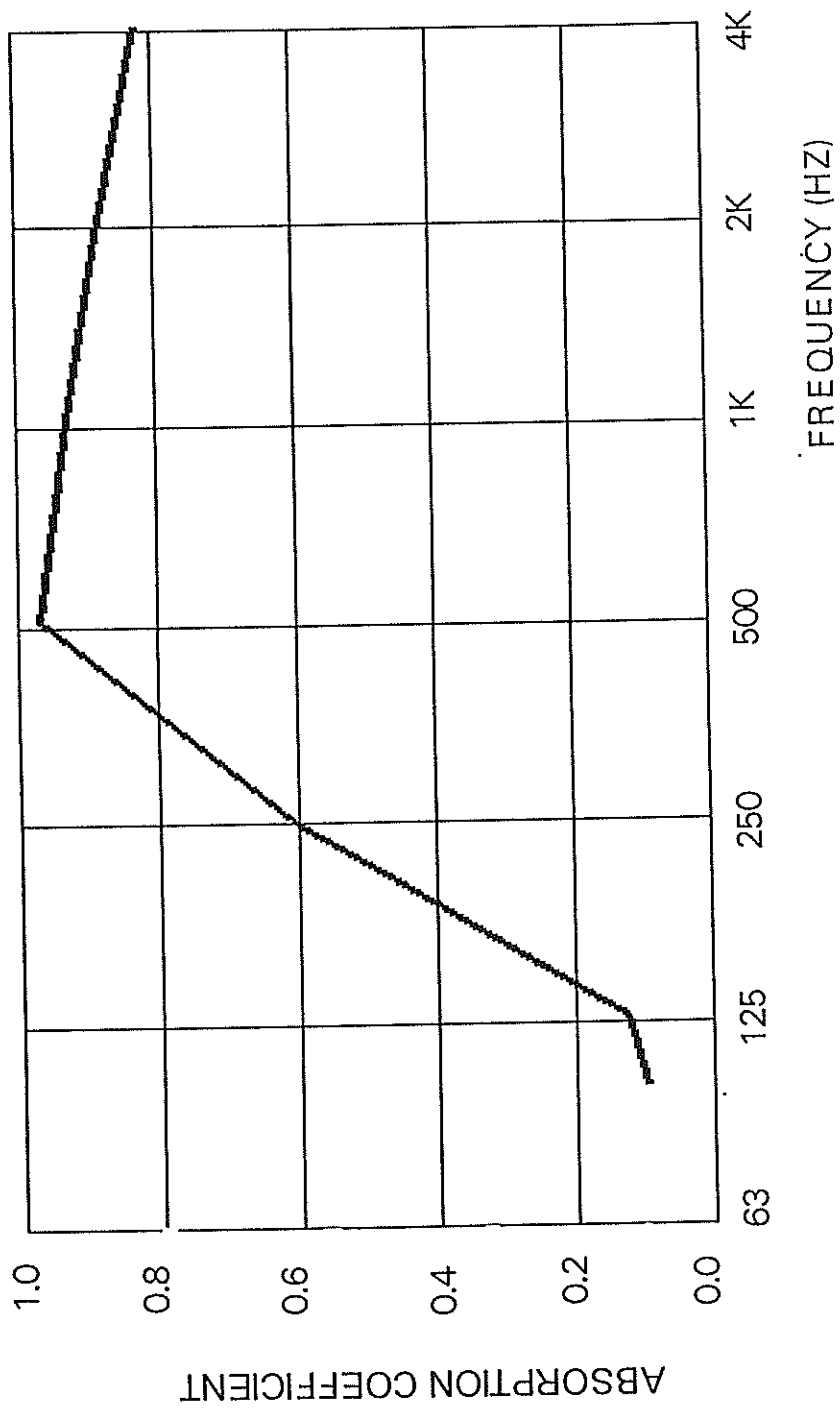
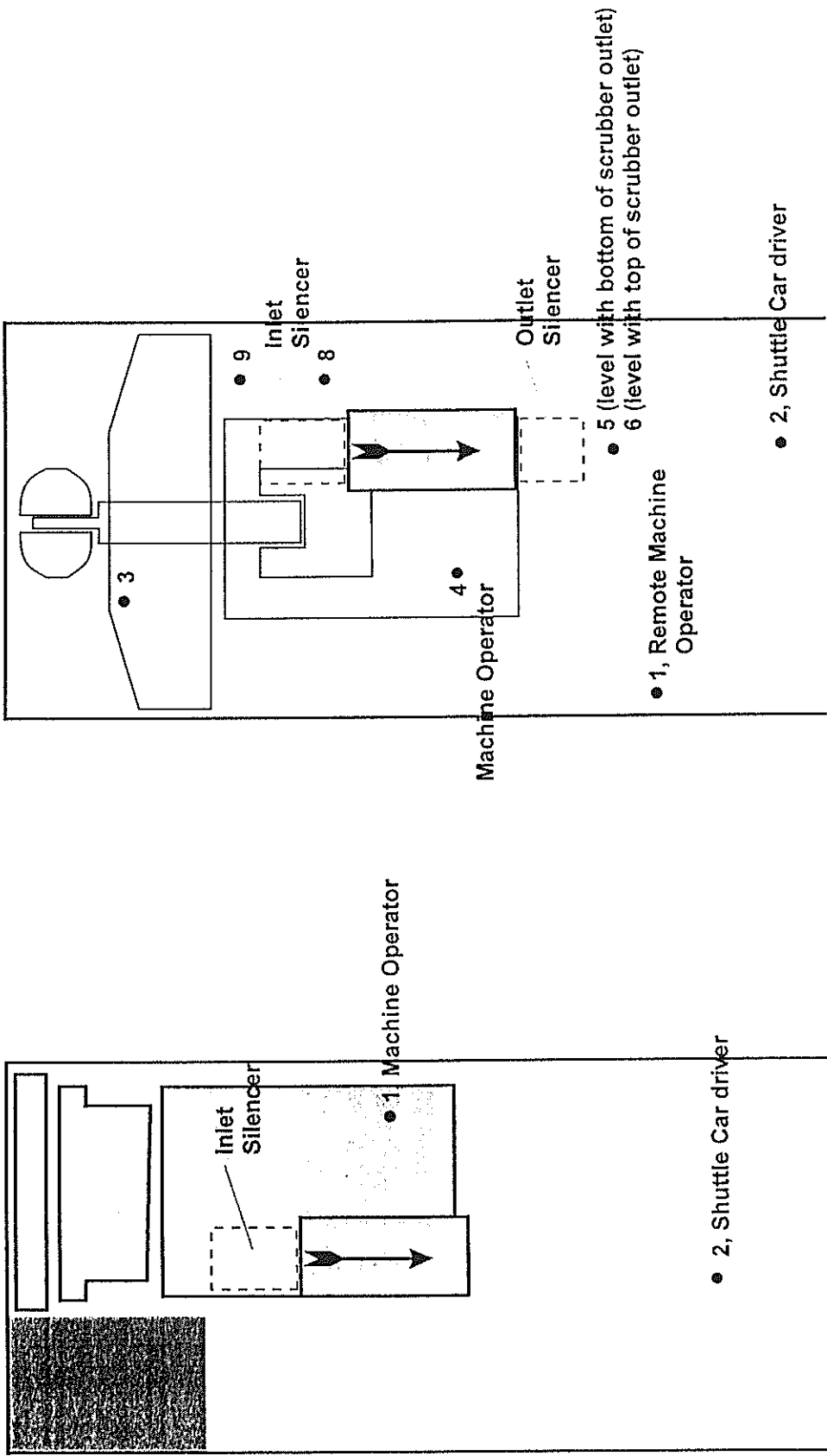


FIGURE 7.2: TYPICAL NOISE ABSORPTION - MINERAL WOOL



a. Continuous Miner

b. Roadheader

FIGURE 7.3: LAYOUT OF THE SCRUBBER TESTS AT SWADLINCOTE SHOWING NOISE MEASUREMENT POSITIONS

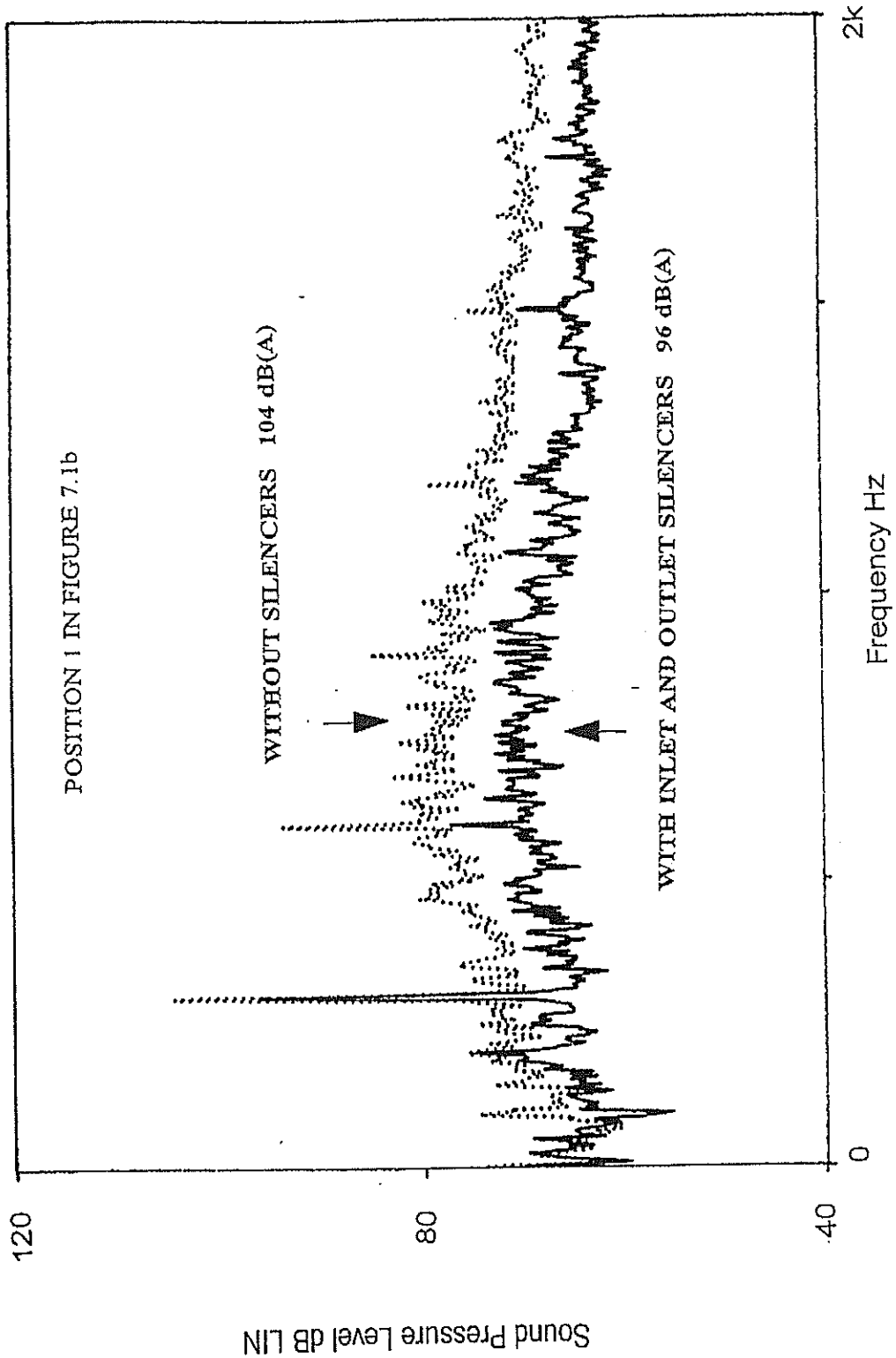


FIGURE 7.4: SCRUBBER NOISE FREQUENCY SPECTRA - WITH AND WITHOUT SILENCERS

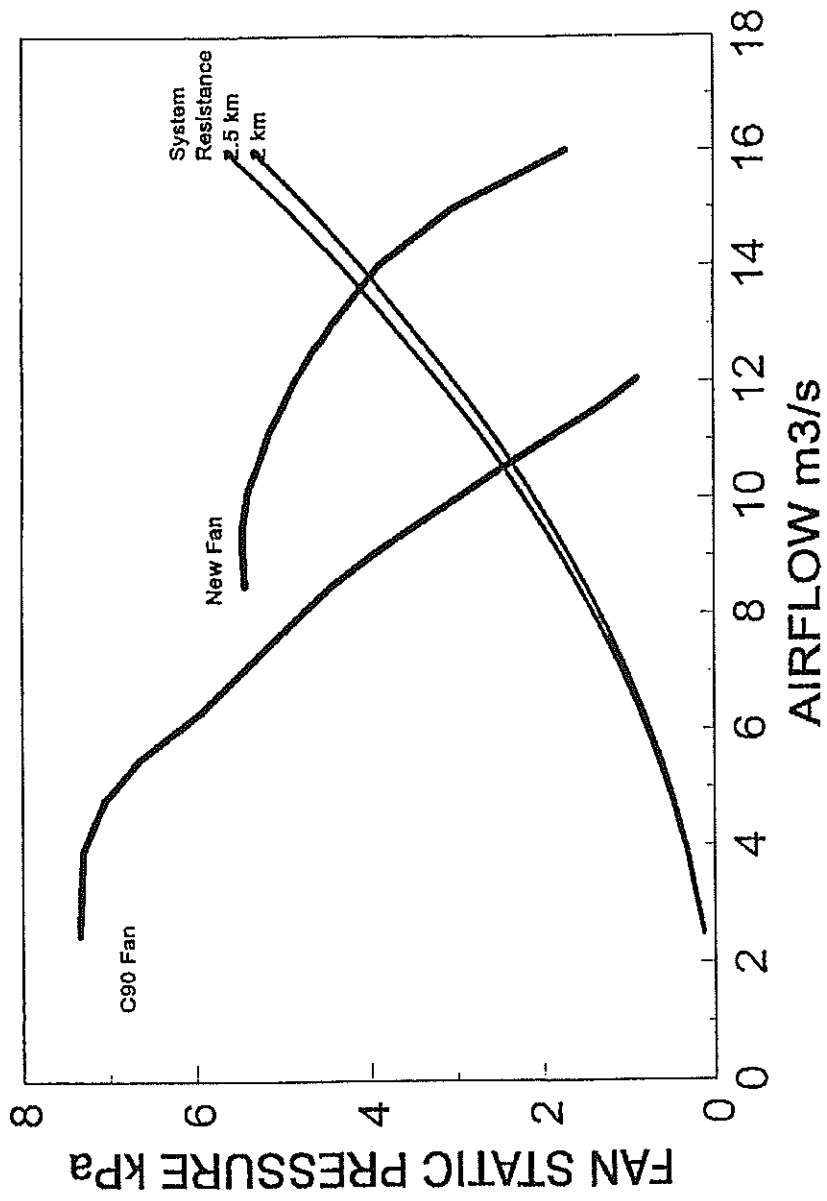


FIGURE 7.5: AUXILIARY FAN CHARACTERISTICS C90 AND NEW FAN

TABLE 8.1: NOISE FROM AFC LINE PANS

CHAIN SIZE (MM)	PAN TYPE	CONFIGURATION	NOISE LEVEL (MID FACE) dB(A) AT 1 METRE	
			NO COAL	COAL
19	190MM IIF	TWIN OUTBOARD	91-95	84-94
22	222MM	TWIN OUTBOARD	92	84
22	222MM	TWIN OUTBOARD (CONT)	95	84
22	222MM TREDOMEN	TWIN INBOARD	94	-
22	222MM (OTHER)	TWIN INBOARD	103	93
26	222MM TYPE A	SINGLE CENTRE STRAND	98	89
26	222MM TYPE B	SINGLE CENTRE STRAND	101-103	90-93
26	222MM TYPE C	SINGLE CENTRE STRAND	109	-



FIGURE 8.1: STANDARD SHEARER CUTTING HEAD





FIGURE 8.2: REDESIGNED SHEARER CUTTING HEAD

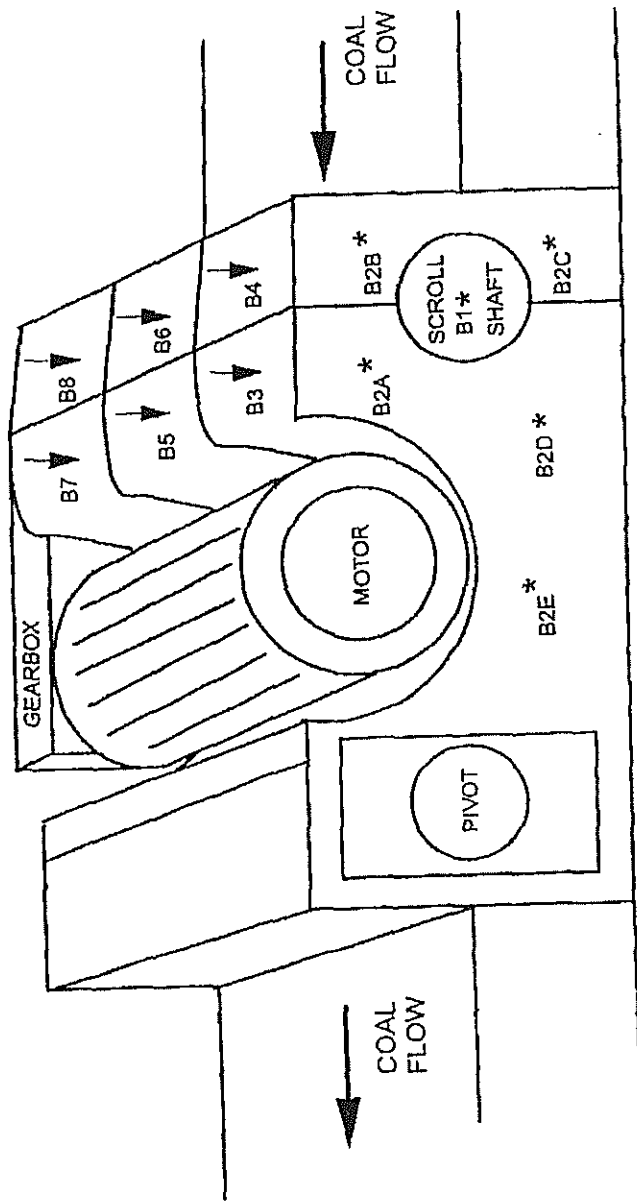


FIGURE 8.3: SCHEMATIC OF BREAKER SHOWING VIBRATION MEASUREMENT POSITIONS

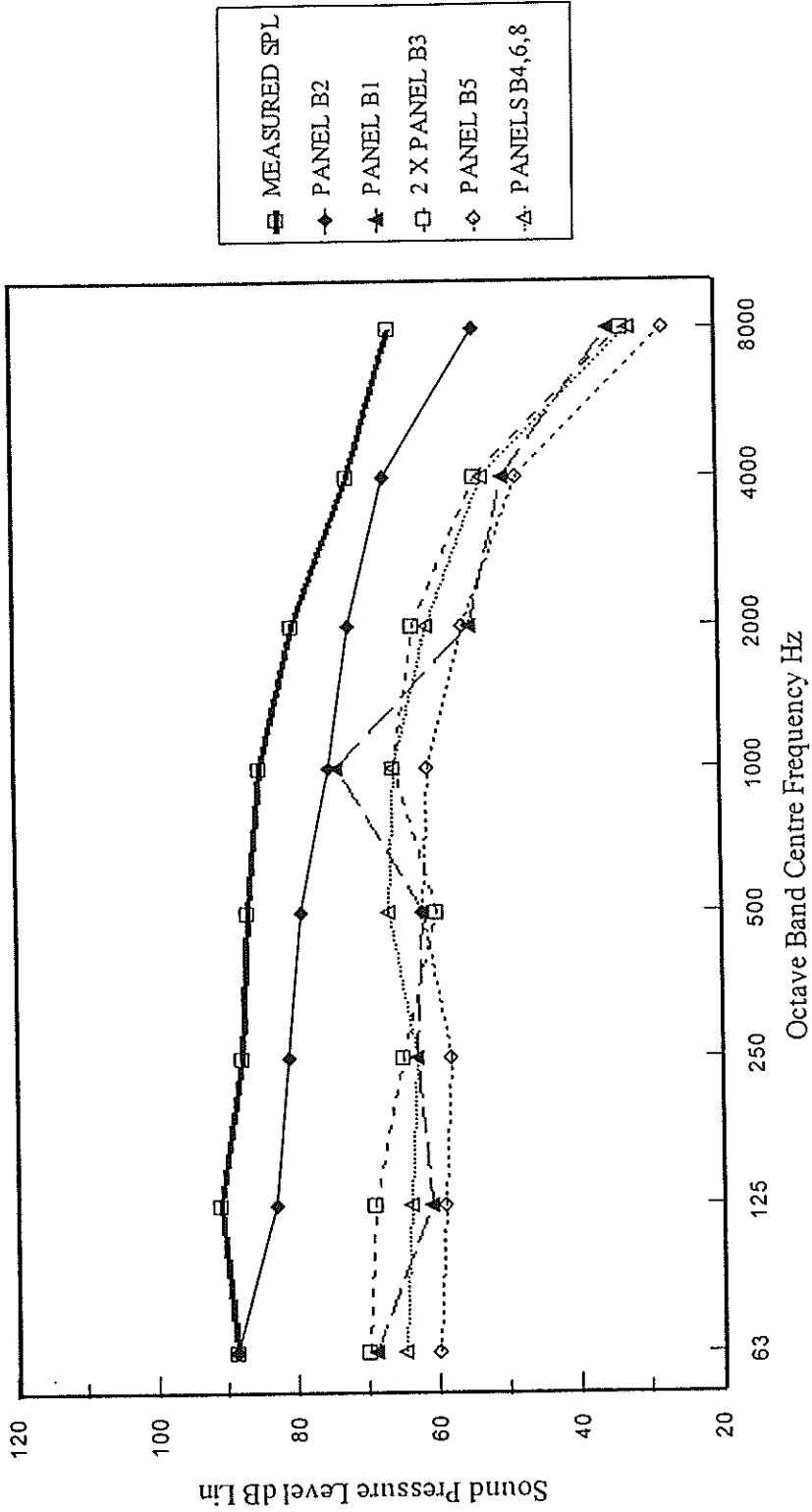


FIGURE 8.4: BREAKER MEASURED AND CALCULATED NOISE LEVELS

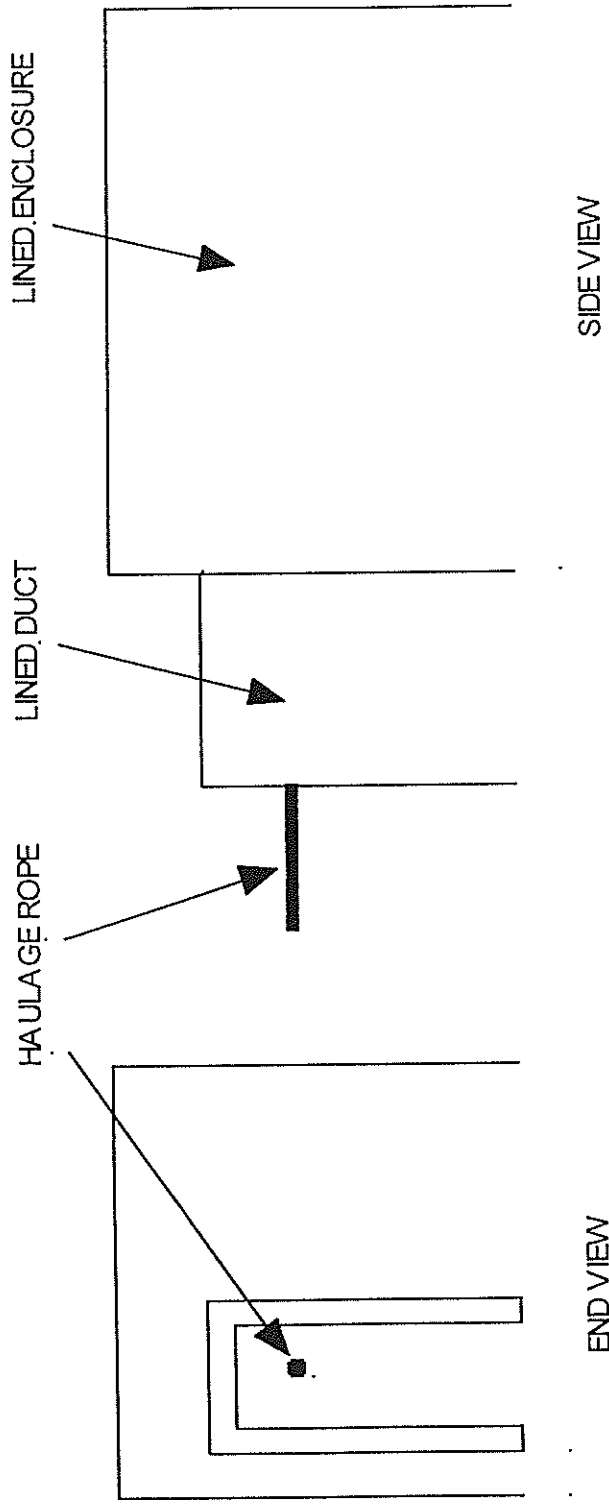


FIGURE 8.5: SIMPLE ENCLOSURE FOR HAULAGE OR WINCH

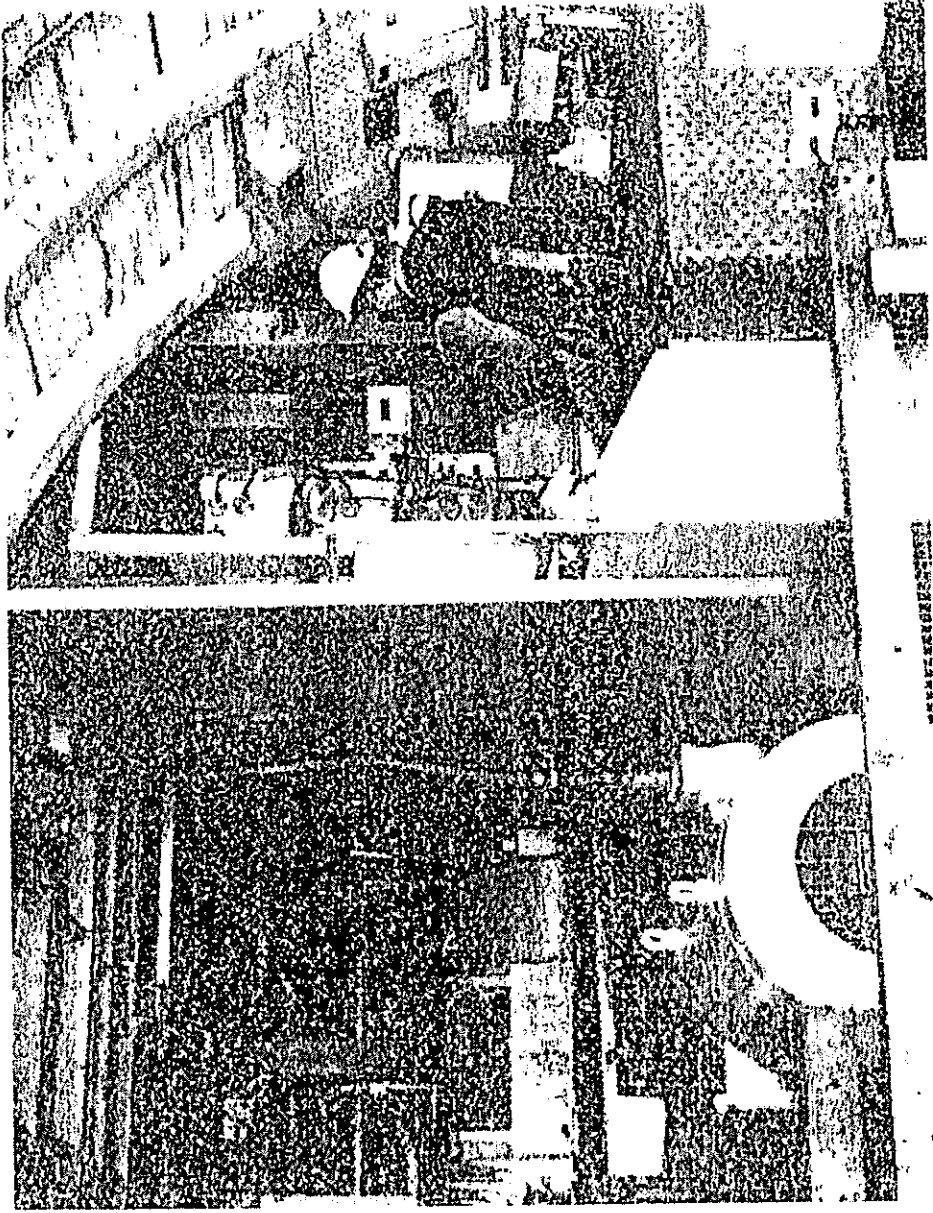


FIGURE 8.6: NOISE BARRIER IN UNDERGROUND INSTALLATION

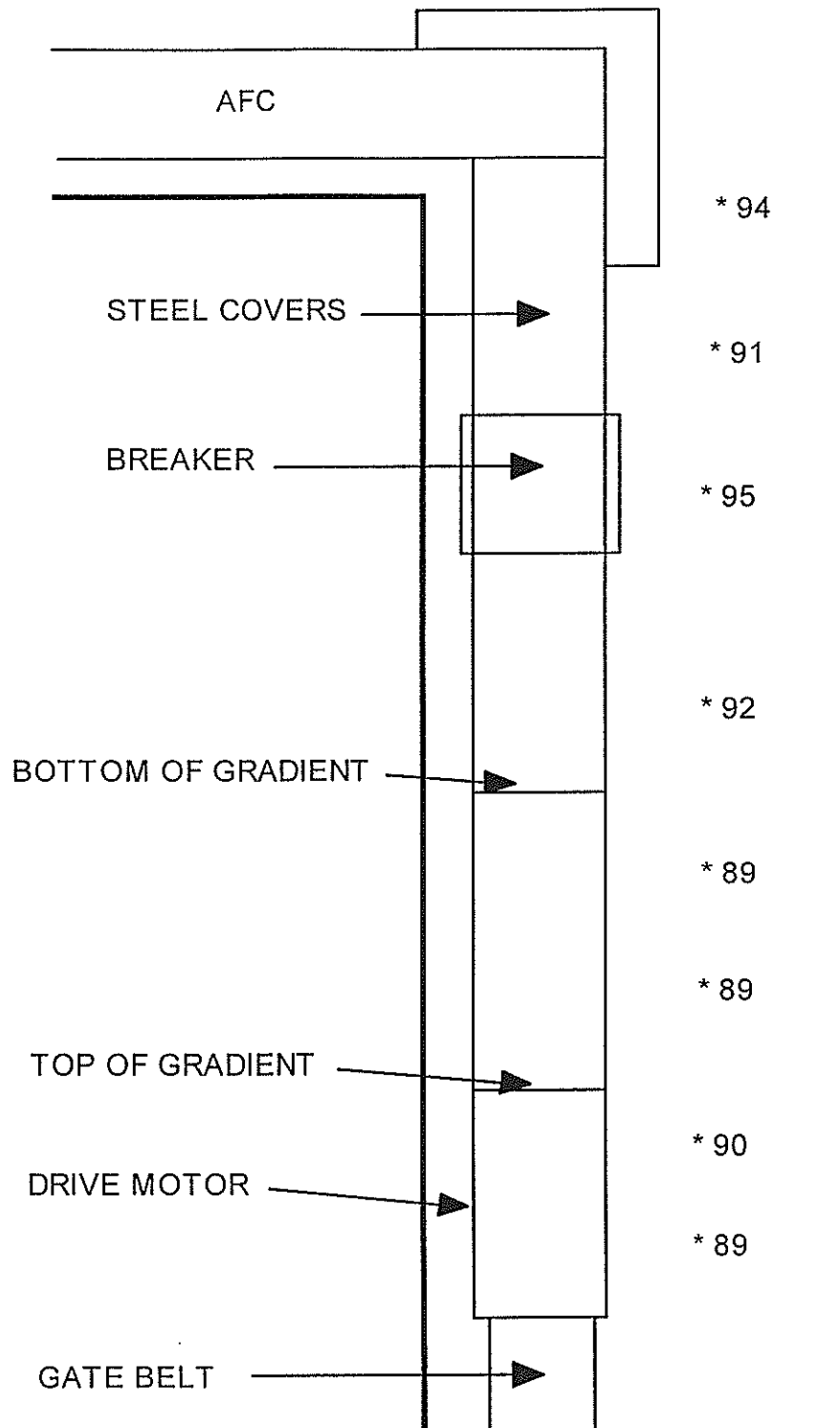


FIGURE 8.7: EXAMPLE FACE END LAYOUT/SITE 1

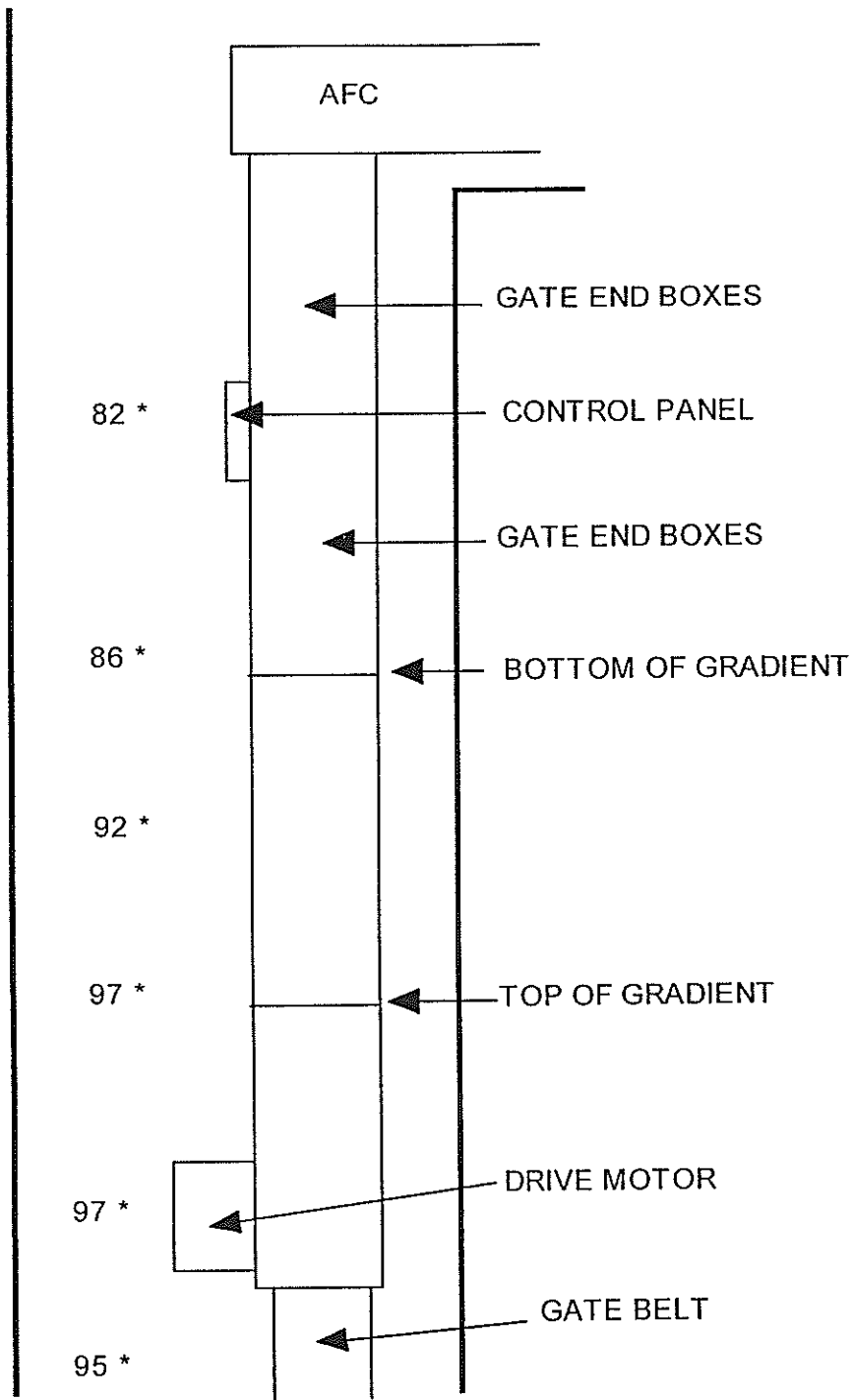


FIGURE 8.8: EXAMPLE FACE END LAYOUT/SITE 2

TABLE 9.1: PERFORMANCE OF NOISE CONTROL TREATMENTS / A-WEIGHTED LEVELS

TEST CONDITION	WAGNER ST-2B	WAGNER ST-2D	EIMCO 912D	WAGNER ST-5A	WAGNER ST-8A	EIMCO 918
SURFACE HIGH IDLE: <sup>(1)</sup> BEFORE AFTER	97 - 98 90 - 91	NA NA	101 91	100 - 101 NA	NA 94	99 90
MINE HIGH IDLE: BEFORE AFTER	98 93 - 95	99 88	NM 91 - 92	NM 93	95 <sup>(2)</sup> 90 - 91	NM <sup>(3)</sup> 94
OPERATING: BEFORE AFTER	102 <sup>(4)</sup> 97 - 99 <sup>(4)</sup>	101 - 103 <sup>(4)</sup> 96 <sup>(4)</sup>	NM NA	NM 89 - 92 <sup>(4)</sup>	NA NA	NM NA

Notes:

- NA Not Available
- NM New Machine Treated. Underground Data Not Obtained Prior To Treatment
- (1) Machine Stationary, In Neutral, Maximum Revolutions
- (2) Data Obtained With Some, But Not All, Treatments Removed
- (3) After 1 Year All Treatments Removed. Resulting Noise Level Was 102 dB(A)
- (4) Corresponding Average Noise Levels Based on Dosimeter Data, LOSHA



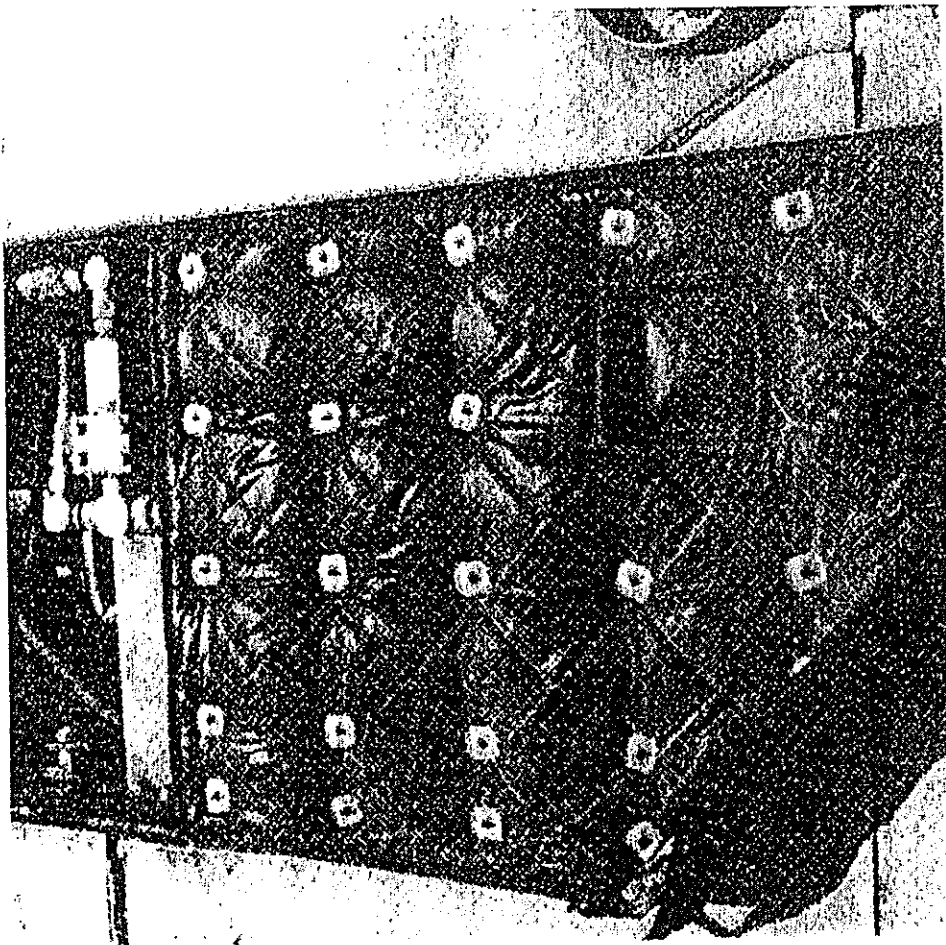


FIGURE 9.2: TREATED TRANSMISSION HOOD

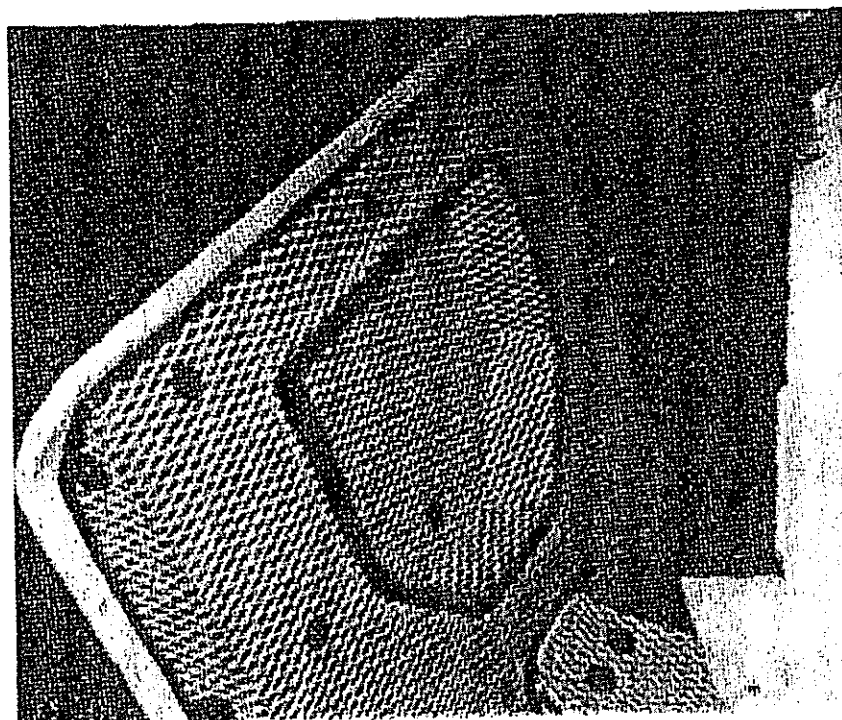
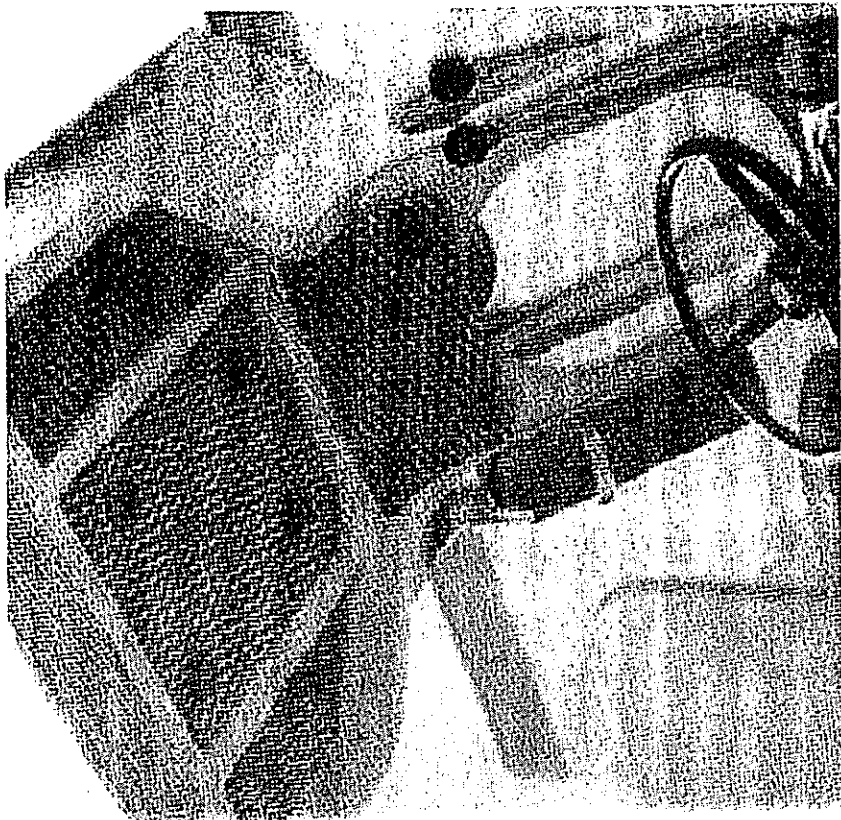


FIGURE 9.3: TREATED OPERATOR CANOPY