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NOISE STUDY OF LONGWALL MINING SYSTEMS

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Noise control of secondary sources was also investigated. A method of reducing full or empty stage conveyor noise (and continuous miner conveyor noise) by 5 to 10 dBA was demonstrated through isolation techniques. Underground mine durability testing, however, is yet to be demonstrated.

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FOREWORD

This report was prepared by Wyle Laboratories, Research & Engineering Division, Huntsville, Alabama, under USBM contract number JO188072. The contract was initiated under the Coal Mine Health and Safety Program. It was administered under the technical direction of the Pittsburgh Research Center with Mr. William Aljoe acting as Technical Project Officer. Mr. Oliver H. Snyder was the contract administrator for the Bureau of Mines.

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CONTENTS

ILLUSTRATIONS

ILLUSTRATIONS — Continued

ILLUSTRATIONS — Concluded

TABLES

NOISE CONTROL OF LONGWALL MINING SYSTEMS

by

Mark R. Pettitt[1] and Robert M. Slone, Jr.[2]

Abstract

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[1] Senior Research Engineer, Wyle Laboratories
[2] Manager, Huntsville Research, Wyle Laboratories

Section 1

INTRODUCTION

The Bureau of Mines chartered a research program to assess the noise generated during longwall mining operations and to develop noise control measures applicable to this machinery at the time of manufacture. This broad objective was pursued in several rather distinct steps or phases:

Phase I - Assess the state of the art in longwall mining. State-of-the-art longwall machinery and industry trends were evaluated to project the types of equipment likely to be in production in the foreseeable future. This was necessary to help assure that program efforts were directed toward equipment that will be dominant in U. S. longwall mines in the future.

Phase II - Investigate longwall system noise sources. Longwall mining operations with the selected types of equipment were surveyed to determine the extent of the noise exposure problem. Noise and vibration measurements were also taken in above-ground longwall facilities to rank and quantify more accurately the problem noise sources.

Phase III - Develop measures to reduce the noise from the problem noise sources. Noise control measures for the predominant noise sources in longwall mining were developed. The new technology was tested, both in the laboratory and in situ.

Phase IV - Generalize the machine design concepts for coal mining noise control. The noise control measures developed for the longwall mining system were based on several machine design concepts that are fundamental to any coal mining system that uses cutting bits. These fundamental machine design concepts were fully developed in general terms to aid original equipment manufacturers in the design of new coal mining systems.

The results of these major program efforts are documented in this report as follows:

- Section 2 presents a summary of the state of the art in longwall mining machinery.
- Section 3 details the results of the longwall system noise source survey.

- Section 4 addresses coal cutting noise, the primary source of noise in the longwall mining system. The nature of coal cutting noise and the machine design concepts necesary for reduction of this noise are explained. The noise control measures developed specifically for the longwall system are presented. Laboratory and field tests that quantify the noise reduction achieved by the noise control measures are also documented.
- Section 5 addresses secondary noise source control.
- Section 6 gives conclusions and recommendations.
- Section 7 lists the references.

Section 2

STATE OF THE ART IN LONGWALL MINING MACHINERY

The objective of the program was the assessment of noise generated during longwall mining operations and the development of engineering noise control measures applicable to this machinery at the time of manufacture. Because of the relatively long lead time required for a machine design concept to find its way into the marketplace, this study was chartered to address state-of-the-art machinery that may still be in production in the foreseeable future. Therefore, the first task of the study was to define the state of the art of longwall mining.

Longwall mining is a highly productive, capital intensive method of extracting coal through underground operations. Virtually all the coal within a well defined panel can be removed by the longwall method since supporting pillars are not required. Additionally, coal production by this method can exceed 4000 tons per shift. In this context, longwall mining might be characterized as representative of the state of the art in coal mining technology. No attempt is made herein to qualify the theory of operation for longwall mining or to describe fully the mechanical systems involved. For a very competent treatment of this subject, the reader is directed to reference 1.

The concept of longwall mining dates back as far as the 17th century in England. Current longwall technology involving integrated mechanical systems has been developed since World War II. Conceptually, these modern-day systems center about a winning machine, which also loads the cut coal onto a face conveyor. Integral to the concept of longwall mining are roof supports, which span the panel face. Additionally, coal is transported from the face area by a haulage system usually consisting of a stage loader and a belt conveyor. These subsystems are further discussed in the following subsections.

2.1 Winning Machines

Two types of winning machines are currently used in longwall mining. The shearing machine is predominant in the United States, in England, and in some eastern European countries. The shearing machine cuts coal with one or two rotating drums. These drums may be mounted on ranging boom arms or may be fixed to the body of the shearer, which houses electrical drive motors, gearboxes, and hydraulic subsystems. Several popular shearing machines are presented in Figure 1. Longwall plows were originally developed in West Germany for application to low seams of friable coal. The plow-type system, which is presented in Figure 2, has several distinct advantages over shearing machines, but it also has serious deficiencies which will be discussed in Section 2.1.2.

2.1.1 Shearing Machines

Longwall shearing machines are favored for use in the United States because of their high production rates and because they are most readily adaptable to prevalent geological conditions. The features most frequently ordered on new shearing machines in the United States include double cutting drums, ranging arms, and chainless haulage. This last feature is a substantial improvement over chain-type haulage systems, which are employed on machines still in use. Haulage refers to the process of traversing the working face. Previously, haulage chains posed a significant maintenance and safety liability. It is reasonably safe to assume that conventional chain-type haulage systems will eventually disappear completely from the marketplace, being replaced by chainless-type haulage systems. Chainless haulage can be accomplished by any of several mechanisms. All apply the haulage force from the shearer rather than from the gate ends of the face.

There are three general types of chainless haulage units, the wheel and rack, chain and rack, and ram propulsion. At present there are at least five different chainless haulage systems in use in the United States, including the Anderson Mavor Roll Rack system, the Joy Traction Rack, the Eickhoff Eicotrack, the Pitcraft Rackatrack, and the Perard Peratrack system. These are based on the first two types of chainless haulage systems. Ram propulsion is not yet in use in this country.

Further refinement of haulage systems is occurring in the area of control systems. The classical approach to provide variable speed control of haulage is to drive this system through a hydraulic pump. By throttling the hydraulic pump, it has been possible to maintain excellent control over haulage rates. This technique, however, involves another hydraulic system, which poses liabilities in terms of noise and maintainability.

At least two manufacturers are currently marketing shearing machines that employ electrical control of motor speed and correspondingly require no hydraulic drive system. This innovation is consistent with developing technology and can be expected to be used more widely in the future.

Another interesting variation from the mainstream of shearing machine design is represented by the off-the-pan shearer concept. Whereas conventional shearer designs place the cutting machine squarely over the face conveyor, this concept places the shearer on the face side of the pan. Instead of riding on guide tracks fore and aft, this design is guided by the pan only along the gob side of the machine. All other factors being equal, this allows the machine to be designed with a lower profile, permitting it to be used in lower seam heights. This system is in limited use in the United States and in the United Kingdom.

Other, more avant garde shearing machine design concepts are under development in Europe. In England, at the Mining Research and Development Establishment (MRDE), a prototype of the previously referenced ram propulsion shearer was observed. It appeared to function smoothly, without apparent anomalies. At the Bergban Forshung, in West Germany, a prototype shearer employing hydraulic motors to drive the cutter drum was observed. This concept also appeared feasible and would allow ranging boom arms the additional flexibility to telescope, a potentially valuable feature. However, both of these latter concepts are unproven in production scenarios.

2.1.2 Plows

The major use of plows is in seams 42 inches or less or where severe problems with dust control exist. The Gleithobel type of plow is guided on a ramp-like face side guidance. A stabilizer arm reaches across the conveyor and runs on a gobside tube guide. With the Gleithobel, the problems of steering, keeping a straight face, and cutting hard coal (all of which are associated with plows) have been reduced. Also models with trapped haulage chains are available.

Few plows are used in the United States; however, much developmental work on advanced plow systems is underway in West Germany where longwall plows produce by far the bulk of coal mined. Among the advanced systems under development are water jet plows and impact plows. The former of these concepts utilizes high pressure water jets to win the coal. This approach generates little dust but is more noisy than the standard plow. The impact plow concept employs a vertical array of large chisels, which are individually driven by hydraulic cylinders. The modulation provided by the hydraulic cylinders is then superimposed upon the plow motion. This system is purported to produce a more desirable coal lump size and should provide improved depth-of-cut control and guidance capability.

It is not anticipated that longwall plows will appreciably increase their small share of the U. S. longwall mining market in the foreseeable future. Hence, the importance of these product development efforts to state-of-the-art longwall mining in this country is minimal.

2.2 Armored Face Conveyors

With the move to high powered double-drum shearers, high capacity, high horsepower face conveyors have followed. Thirty-inch-wide pans with 30-mm single strand or twin inboard chains and 400- to 600-hp drive motors are now in use. The single and twin inboard chain configurations reduce the differential stretch and jamming problems encountered with the outboard chain configuration.

It appears that production at the longwall face is limited in most cases by the capacity of contemporary chain conveyors to remove the extracted coal. In response to this restriction, research is currently underway to develop and demonstrate high speed, high capacity armored face conveyors. A developmental prototype of one such system was viewed under test at the MRDE in England. It is axiomatic that future conveyor systems be of increased capacity. Additional research on armored chain conveyors is being directed at the development of curved conveyor pans. Two such prototype systems under study at the Bergban Forschung are shown in Figure 3. Critical elements of this developmental work center around the design of the flights, races, and chain drive configurations.

Armored chain conveyors are an important noise source at the working face. The most desirable solution to this problem is mass loading of the pan, which results when it is loaded with coal. It is reasonable to expect that high capacity face conveyors may require even more attention with regard to noise control than is afforded contemporary systems. A West German study on this subject is referenced in section 3.2 of this report.

2.3 Roof Supports

The current trend in roof support is toward use of the shield, several examples of which are depicted in Figure 4. The shield's continuous one-piece canopy and caving shield provide excellent roof support and superior protection of the working area. The shield is more stable than the chock, and the new designs with leminscate linkage permit the roof canopy to remain in the same vertical plane relative to the face as the support is raised or lowered. This continuous construction greatly increases the side loading capability of the shield in comparison with the older chock. Also shields have a greater operating range, are simpler hydraulically, and provide excellent travelways.

2.4 Hydraulic Pumps

With the advent of shield roof supports, larger capacity hydraulic pumps are necessary to obtain the setting load for the shields. Pumps with capacities of 50 gpm at 5500 psi are being used to speed movement of shield supports and conveyors. High horsepower and increased capacity, variable displacement pumps, which function on demand to supply only the volume and pressure required, are under development and may provide reduced noise levels in the headgate area.

2.5 Automation

To increase safety and productivity, industry and government have been working over the last few years to develop automated longwall mining technology. The efforts have focused on automated shield advance, shearer haulage rate, and shearer ranging arm control. An automated system applicable in the general case has yet to be developed. The above listed functions, however, have been automated in one form or another with some in-mine success. Examples of efforts along this line include Dowty, Hemscheidt, and Westfalia Lunen semi-automated shield advance systems. Eickhoff developed a haulage controller that optimizes the shearer haulage rate. Eickhoff has also tested a system for shearer ranging arm height control. The development of better, more reliable automation systems will probably continue and have a significant effect on future longwall mining practices.

2.6 Conclusions on the State of the Art in Longwall Mining Machinery

It is apparent that longwall mining in the United States is in a state of flux in terms of machinery and the technology behind that machinery. Nevertheless, an attempt has been made to discern trends in the evolution of longwall mining systems. The following sums up the conclusions of this evaluation.

1. Double-drum shearing machines will probably predominate in the American marketplace over competitive plows and single-drum longwall shearers for the foreseeable future.

2. Chainless haulage drive systems will be used exclusively on shearing machines of the future.

3. Plows may find application primarily to extraordinary mining scenarios for reasons of dust and methane control or especially suitable geological conditions.

4. Longwall equipment of novel design, such as ram propulsion haulage, water jet plows, etc., are several years away at best. Even in-web shearers, several of which are currently in use, have many unsolved problems associated with their operation.

5. The next generation of longwall mining machinery will probably employ electronic control of motor speed in lieu of the hydraulic pumps and motors that are currently used where speed control is required.

6. Remote controlled operation of shearing machines is a foreseeable reality. Remote control via umbilical cord is now standard on at least one model of shearing machine and fully automated shearer operation is currently under development (2).

7. Increased motor horsepower for shearer drives is distinctly evident. Up-to-date machinery employing motors up to 500 horsepower are likely to be surpassed in attempts to take deeper cuts in order to produce less respirable dust.

8. Conveyor technology will probably realize substantial gains in the areas of high speed, high power face conveyors, larger capacity stage loaders, and roller curve face conveyors, which eliminate the need for stage loaders.

9. Heavy-duty conveyor pans will probably find increasing application with the gradual implementation of shield supports, which often require massive pan line connections.

10. Twin inboard and single center-strand conveyor chains will become increasingly prevalent over twin outboard or triple-strand conveyor drives. Also conveyor chain tension will be more precisely controlled with hydraulic- or air-operated auxiliary tensioning devices.

Section 3

SYSTEM NOISE SOURCE IDENTIFICATION

A primary objective of the program was the identification and ranking of noise sources associated with the operation of longwall mining machinery. The mechanism for accomplishing this objective includes discussions with longwall machinery manufacturers, longwall technology researchers, and longwall system operators in addition to a noise and vibration measurement program. Information gathered through these sources is presented in the following subsections.

3.1 Manufacturers' Views

During the course of the current study, program personnel traveled to England and West Germany. In Bochum, West Germany, discussions were held with Eickhoff design engineers. Their attention to noise control has been directed primarily at the haulage system of their shearing machine. Previous designs of the Eickhoff shearer had employed a hydraulic pump to provide haulage power. This pump was driven by an electric motor through a gearing arrangement, and the pump in turn powered the haulage drive motor. During noise and vibration measurements on the shearer, Eickhoff engineers identified a ringing in one of the drive gears to the hydraulic pump. This ringing was apparent not only in vibration measurements on the gear housing but also in measurements of radiated noise. A design modification that remedied this condition was made. The modification consisted of eliminating the hydraulic pump and hydraulic motor and providing electronic speed control for the haulage motor. This change allowed the chainless haulage (Eickotrack) system to be driven directly by an electric motor. The net result is a more efficient, more reliable, and quieter shearing machine.

Invariably, when shearer manufacturers are approached regarding noise of their equipment, a gear design specialist is called into the discussions. Such was the case at Eickhoff and at Anderson Strathclyde in Motherwell, Scotland. In both cases, gear noise was acknowledged to be present in their products, and both manufacturers had reasonable explanations to support the design of their gearing systems.

Briefly, a double-drum longwall shearing machine is designed to cut in either direction. The classical design for this machine involves utilization of a single electric motor to drive both cutting drums and the hydraulic haulage system. This arrangement calls for the drive motor to be oriented in such a way that its shaft is situated longitudinally in the shearer body, with gear boxes on each end of the electric motor. The gear boxes provide speed reduction and, also by virtue of bevel spur gears, redirects the motor power into a plan that allows transmission of this power to the drums through a series of spur gears in the boom arms. This allows for an optimum apportionment of available power, which could not be accomplished with multiple motors, to satisfy the three power requirements. The leading drum of a double-drum shearing machine requires a majority of the available power with less power required for the trailing drum and for haulage. A predominant noise source in this gearing arrangement is the bevel spur gears. It is well known that spiral bevel gears are relatively much quieter than bevel spur gears. However, both manufacturers felt that replacing these bevel spur gears with spiral gears was a manufacturing impracticality.

A most interesting and substantially different perspective on this probelm was afforded by a visit to British Jeffrey Diamond (BJD) in Wakefield, England. BJD, like Anderson Strathclyde and Eickhoff, manufactures the gears used in their shearing machines. However, BJD does employ spiral bevel gears in the application described above because of their superior power transfer characteristics and because of the lower noise that results from their use. The significance of this innovation is not in how BJD manufactures their gears but in the fact that it is possible to build competitive longwall shearing machines using bevel spiral gears. The BJD shearer has not penetrated the American market. The noise reduction accomplished by this means has not been quantified.

3.2 Summary of Data from Other Sources

Very little work has previously been done to quantify, assess, or control noise from longwall mining systems. Perhaps the broadest appraisal of noise levels in U. S. longwall operations lies in the Mining Safety and Health Administration (MSHA) reports and records. Unfortunately, many of these data are published as confidential, in-house reports and cannot be referenced properly. Nevertheless, when viewed as an ensemble, MSHA noise exposure measurements can provide some insight into noise sources in longwall mining.

MSHA survey data are most often based upon noise dosimeter measurements. These data are reinforced by time-resolved dosimeter measurements and sound level meter measurements when appropriate. It is frequently the case that noise exposure levels are determined from portal to portal, or the entire time a worker is underground. In this regard, individual noise sources would seem to lose definition; however, certain conclusions can be drawn based upon differences between various crew members. For example, one MSHA survey (3) indicates that of six mines studied, only one headgate operator was exposed to noise in excess of allowable limits. Of that same group, both shearing machine operators in two of the mines received excess noise doses, and no other crew member in any of the six mines surveyed experienced noise doses in excess of 100 percent.

Another, more comprehensive study of 56 U. S. longwall operations (4) indicated that 96 percent of longwalls were in compliance with the 90 dBA, 8-hour noise exposure standard for all crew occupations. Of the workers out of compliance, the majority were shearing machine operators. Only an extremely small percentage of other job categories had noncompliance.

Yet another MSHA study of noise exposure in a longwall mine is documented in reference 5. This study quantifies the equivalent portal-to-portal noise level to be 100 dBA, or 400 percent of the allowable dose, for two shifts of shearer operators. For the same shifts, the chockman experienced an equivalent noise level of 90 dBA (within allowable limits) and the headgate operator realized an equivalent level of 92 dBA. These data would seem to suggest that machinery along the longwall face, perhaps especially the shearing machine, is most significant in its contribution to worker noise dose.

The armored chain conveyor may also be a significant noise source on a longwall face (6-8). The greatest potential contribution of this face equipment to noise in a longwall section results when the conveyor is run in an empty or lightly loaded condition. Also, since the face conveyor spans the full face length, most of the working crew is exposed to this distributed source.

The stage loader is similar in construction to the armored chain face conveyor. It is used to move the coal from the face to the beltline. Noise at the lead gate operators position from this source can reach 94 dBA.

3.3 <u>Summary of Data from the Current Study</u>

Noise and vibration measurements were taken in five mines during the initial survey stage of the study. These measurements are source specific and are useful for diagnostic purposes in addressing noise source mechanisms. A description of the longwall operations and an identification of the machinery that qualifies this data was presented in appendix A, entitled Mine Visit Reports. These particular mines were selected for study because they employ machinery that is representative of the state of the art in longwall mining. The acoustic data do not independently define noise exposure and **do not** suggest the existence of any noise-related problem associated with these operations and should not be considered in that regard. Rather, the cooperation of these operators can only be regarded as a positive statement about their concern for health and safety. These contributions were fundamentally prerequisite to the development of any meaningful results from the study.

Additionally, noise and vibration measurements have been made on shearing machines and armored chain conveyors above ground during simulated operation. These measurements were performed in Germany, at the Bergban Forschung, in England, at the National Coal Board Surface Test Facility in Swadlincote, and in Pittsburgh, Pennsylvania, at the USBM Mining Equipment Test Facility.

Data from these measurements are presented in the following subsections. These one-third octave-band spectral plots have been reduced from acoustic data recorded on magnetic tape. The spectra have been processed through an A-weighting filter in order to emphasize those frequencies that are most significant in their effect on worker noise exposure. Direct comparison of data from different mines is difficult, as the mining cycle varies just as the machinery does. For this reason, the data should be regarded as qualitative and significant only with regard to trends they reveal.

An order of priorities for consideration of noise sources was assumed. First priority was assigned to noise sources that logically should be controlled at the source, second priority was assigned to sources that could be effectively controlled in the transmission path, and the lowest priority was given to those sources that either are ordinarily or could reasonably be located remotely, away from workers. Also, data resulting from anomalous operation of equipment is so designated with an indication of the equipment or operational condition that contributed to the measured levels.

3.3.1 In-Mine Acoustic Measurements

Winning Machines

Noise from operation of shearing machines usually exceed 90 dBA even on cleanup passes. Figures 5 and 6 show A-weighted one-third octave-band sound pressure level spectra near the leading shearer operator's position for the two doubledown shearers described in Appendix A. The overall sound levels associated with these specific spectra were 94 dBA and 103 dBA, respectively. The lower noise level was measured at a rather slow advance rate. Review of the data tape also indicates the lower measurement was taken somewhat behind the lead operator and possibly as far back as the shearer machine mid-body. Noise measurements at the operator's position taken on an additional three shearers during the prototype testing phase of the program were all in the 99 to 103 dBA range. Figure 7 depicts noise at the trailing operator's position for the same shearer represented by figure 6. The overall sound level was 3 dBA lower at this position. These data were typical of the popular double-drum shearing machines studied in this survey. The broad spectral peak between 315 Hz and 1.25 kHz suggests complex noise source mechanisms which were closely studied in above ground tests described in Section 3.3.2. Overall A-weighted noise levels in excess of 110 dBA have been measured for this class of machine during the anomalous operation of cutting top rock. This cannot be considered a routine part of normal operations.

Noise from cutting operations of the single-drum shearing machine studied could not be measured because of equipment malfunction during the visit to that mine. The principal difference between this machine and the double-drum shearers previously addressed lies in the absence of a second trailing drum to perform the clean-up operation. The significance of the trailing drum as a noise source depends primarily on the amount of bottom coal it cuts. Generally, the lead drum noise dominates at the operator's position. A return visit was, therefore, not considered necessary.

Noise levels associated with the operation of a longwall plow system are represented by figure 8. Since no single worker remains with the plow for any great length of time, the measured 90 dBA noise level is unobtrusive.

Conveyor Systems

Three types of conveyor systems were examined. Face conveyors were always of the armored chain variety and spanned the entire length from the headgate to the tailgate. All the face conveyors studied underground were 30 inches wide, and chain speed ranged from 180 fpm to 226 fpm. All the face conveyors studied employed inboard chains, with three of those studied using a single chain and the remaining two using twin inboard chains. Drive motors for face conveyors were situated at both the headgate and tailgate ends, and total drive horsepower ranged from 250 hp to 400 hp for the systems studied.

Noise levels for chain conveyors depend strongly upon the amount of coal on the pan and upon the speed of the chain drive (6). Figure 9 shows a typical one-third octave band noise level spectrum for a twin inboard chain drive face conveyor. The overall noise level associated with this plot is 84 dBA, and this data was taken with the conveyor partially loaded, away from the headgate and tailgate areas. Additional noise sources associated with the face conveyor were apparent in the headgate and tailgate areas. These noise sources related to the drive motors and gearboxes and to the drive sprockets for the conveyor chains. Noise from these sources is reflected in the data for headgate operations.

Stage loader conveyor systems were used in all five mines studied to connect the face conveyor with the belt conveyor. This is common in most mines. The stage loader is similar in construction to the face conveyor, using chain drive and an armored pan line. However, the stage loader is much shorter, usually on the order of 30 to 100 feet, and uses a single drive motor of from 40 to 125 horsepower. This conveyor system is of potential significance from an acoustic standpoint because it is situated very near the headgate operator, chain speed is frequently higher than for face conveyors and the stage loader is often fitted with a rock crusher intended to eliminate large lumps of coal.

Noise measurements on stage loaders are highly variable, depending upon whether or not rock is present in the coal being handled and whether the crusher is operating. Spectral plots for noise measurements on four stage loaders are presented as figures 10 through 13. Overall A-weighted noise levels for these stage loaders ranged from 93 dBA to 97 dBA.

The third type of conveyor system used in longwall mines is a continuous belt conveyor. This system is physically removed from the face area by the length of the stage loader. The sole measurement of noise from a belt conveyor is presented in figure 14. This conveyor was 36 inches wide, with a belt speed of about 300 fpm. Overall noise two feet from the beltline was only 85.8 dBA, well below the level that would call for corrective action.

Other Noise Sources

Much of the powered equipment used in longwall mining is situated in the headgate area, near the headgate operator. In addition to the occasional proximity of the coal winning machine, the stage loader is found there, and hydraulic pumps and motors are frequently located in this area. Consequently, noise in the headgate area is often high and usually difficult to relate to a particular source. Figures 15 and 16 show noise measured at the headgate operator's position for two of the mines surveyed. These data are strongly dependent upon the operational scenario in effect in each mine. Not only is the equipment selection and maintenance condition critical to these measurements but also to the time of measurement. When the coal winning machine cuts through into the headgate area, noise levels at that position may be in excess of those measured for the shearer operator. However, the winning machine is in that operational mode only a small fraction of the mining cycle. Thus, these noise measurements cannot be considered to be typical of the noise exposure for that worker.

Hydraulic pumps in some mines are situated in the headgate area and in other mines are remotely located. Noise levels from hydraulic pumps depend upon pump design, maintenance condition, and pump capacity. Figures 17 and 18 depict noise measured on 75-hp pumps in two different mines; figures 19 and 20 characterize noise from two 125-hp pumps, again in different mines. Figure 21 is yet another measurement from the fan end of the pump motor represented by figure 20. To limit the influence of other sources, all measurements were taken within two feet of the pumps.

The measured levels ranged from 96 to 108 dBA. If not remotely located or shielded from normal worker positions, therefore, hydraulic pumps could contribute to worker overexposure.

3.3.2 Above-Ground Acoustic Measurements

Above-ground noise measurements were made on a double-drum shearing machine at the National Coal Board Surface Test Facility at Swadlincote. Also, extensive noise and vibration measurements were made on a Joy double-drum shearer at the USBM Mining Equipment Test Facility longwall mockup at the Pittsburgh Research Center. It should be emphasized, in considering these data, that the in-mine environment was not simulated; therefore, the magnitudes of the presented data are somewhat arbitrary. However, the frequency information conveyed by these data are valid. For safety reasons, these data could not have been acquired in operating mines.

Figure 22 characterizes near-field noise from the Swadlincote double-drum shearing machine about three feet from the leading drum. Since this machine utilized an experimental ram propulsion haulage system, near-field measurements (within one foot) in search of discrete fluid dynamic noise were also made on the hydraulic system. The spectrum in figure 23 exhibits a relatively broadband characteristic from the hydraulic pump with a high frequency spectral component in the 10-kHz one-third octave band. Fortunately, this hydraulic pump is usually placed at some distance from normal worker positions.

The test procedure and results of the intensive noise and vibration measurements made on the double-drum shearer at the USBM Mining Equipment Test Facility longwall mockup were published in a 163-page test report (9). Certain portions of the report are repeated here. The instrumentation is described in table 1 and the measurement locations indicated in figure 24. Figure 25 shows the USBM longwall shearer during cutting operations. A summary of the overall A-weighted sound pressure levels at the various measurement locations and for several operating conditions is given in table 2.

TABLE 1. Measurement locations

Type and designation	Location	Sensitivity direction
Microphones:		
M1..........	Lead operator's position	Not applicable
M2..........	Machine center........................	.. do........
M3..........	Lead shearer drum, near-field	.. do........
M4..........	Trailing shearer drum, near-field	.. do........
Accelerometers:		
A1	Lead shearer drum core by A9 (telemetry).	Radial
A2	Lead shearer face plate (telemetry)	Normal to back face surface
A3	Lead shearer drum cowl	Normal to cowl surface
A4	Lead shearer ranging arm end	Vertical
A5	.. do.................................	Horizontal
A6	Coal face.............................	.. do.....
A7	Machine center by operator	Vertical
A8	.. do.................................	Horizontal
A9	Lead shearer drum core by A1 (slip ring) ..	Radial
Switch: S1.....	Lead shearer ranging arm end	Drum rotation trigger

TABLE 2. Overall A-weighted sound pressure levels (dBA, ref. 20 μPa)

Run No.	Condition	Operator position, M1	Mid-machine, M2	Lead shearer near-field, M3	Trailing shearer near-field, M4
4	Baseline ambient noise	82.4	72.6	74.7	74.5
5	Hydraulics on	*	78.3	75.5	75.4
6	Both drums rotating, not cutting .	*	82.1	84.3	83.7
7	Haulage on (drums off, hydraulics on)...............	*	79.8	77.2	75.5
8	Lead drum cutting	93.9	93.7	101.5	87.6
9	Lead drum cutting (instrumentation adjusted and trailing drum cutting coal face lightly).......	95.3	97.6	103.9	98.0
10	Lead and trailing drums cutting..	93.9	96.2	101.9	97.6

*Although comparative information can be obtained from the spectral plots, overall levels are controlled by the higher ambient noise levels for M1.

It is clear from this table that the noise related to coal cutting is predominant. The near-field measurements at the cutter drums show the source of the objectionable noise is in that area.

The dynamic forces generated during coal cutting cause the coal face, the cutter drum, and the mining machine as a whole to vibrate and radiate noise. Identification of the predominant noise source is a prerequisite to the development of appropriate coal cutting noise control measures. Such a determination can be made from a detailed narrow-band analysis of the noise and vibration measurements made at the USBM longwall mockup. Since A-weighted noise is the measure of interest, the analysis was conducted on A-weighted noise and vibration spectra shown in figures 26 through 31. From the one-third octave band level plot of figure 26, it can be seen that significant noise was generated out to the 3150-Hz band. The narrow-band data was reduced in the 0- to 1600-Hz band to aid in noise source identification analysis.

Combining the graphs of figures 27 through 30 into a single graph (figure 32) produced a somewhat crowded but very informative figure. The figure can be used for "peak picking," a simple and sometimes very effective method of noise source identification. Lines of constant frequency are shown in figure 32 at each prominent peak in the noise spectrum. Since structurally radiated noise and surface velocity are directly related, the noise peaks were caused by the resonant peaks of the structural vibration. In "peak picking" one attempts to determine the primary radiator by matching the noise spectrum peaks to the velocity spectrum peaks of different parts of the longwall shearing machine, cutter drum, and the coal face.

An upper bound on noise radiated by the Joy drum core can be estimated by careful examination of figures 27 and 32. In figure 32, it is clear that, although the velocity of the cutter drum core was greater than any other measured surface, it was not the source of the very peaked noise spectrum below 800 Hz. Only three noise and vibration spectra peaks match above 800 Hz, but each of these peaks is also evident in the velocity spectra of other structures. The noise in the 0 to 800 Hz band and the noise in the 800 to 12 KHZ band are each about 100 dBA as determined by a spectum analyzer. The noise in the 0 to 800 Hz band and the noise in the 800 to 12KHZ band are each about 100 dBA as determined by a spectrum analyzer. Assuming the other noise sources and the core noise are equal above 800 Hz yields the following estimate of the core noise limit.

	Noise (dBA)				
Band (Hz)	Core		Other		
0-800	~0		100		
800-12000	97		97		
0-12000	97	+	101.75	=	103 dBA

If one also subtracts the noise in the 800 to 1600 Hz band that is controlled by the peaks that are definitely not related to the core velocity spectrum (at 1100 and 1500 Hz), an estimated maximum core noise of 95 dBA results. Hence, the core vibration noise was probably 7 dBA lower than the noise radiated by the dominating noise sources. The velocity spectrum of the drum surface can probably be considered the vibration input to the rest of the longwall shearing machine. The cowl, cut coal, and the helix all shield the drum surface from the operator's position and thereby reduce the importance of drum core vibration as a noise source.

The helix, which includes the face plate, is obviously a relatively flimsy, highly resonant structure that could be the primary noise source. Unfortunately, complete data for the helix vibration during the cut could not be obtained because of the interference of cut material. Figure 33 shows the helix and core vibration measured when the cutter drum was only lightly cutting. The helix vibration clearly corresponds to the very high noise levels in the frequency region below 1000 Hz. Although this strongly indicates that helix vibration was important, the relative level of the helix-radiated noise to the noise radiated by the other shearing machine structures is not apparent from the test data because both exhibited highly resonant structural response.

In respect to this other equipment, another striking feature of figure 32 is the almost perfect peak alignment of the noise spectrum with the velocity spectra of all the secondary equipment. Measurements were taken on the cowl, ranging arm, and machine body. The close alignment of the velocity peaks at each measurement location on the secondary equipment indicates a very strong resonant response of the whole ranging arm structure to the excitation passed to it through the cutter drum. The large surface area and high radiation efficiency of the plate-like structures of the secondary equipment may make the vibrations of this equipment an important noise source.

The final coal cutting noise source of interest is the coal fracture and face radiation noise. The level of this particular source is very difficult to establish. The Bureau of Mines funded a program to investigate the mechanics and noise associated with coal cutting (10). This study determined that the coal fracture noise is small relative to the local coal face radiation. The experimental apparatus for the coal cutting study is shown in figure 34.

In situ coal face radiation could be dominated by either the vibration of the coal face very near the cutting bit or the vibration of a larger area of the coal face. Vibration of the simulated coal face within 45 inches of the cutting was measured during the test at the USBM longwall mockup (figure 31). The most evident characteristic of the vibration is the lack of prominant velocity peaks. The spectral characteristics of face radiation from material close to the bit should be somewhat similiar although the vibration level could be significantly higher. Nevertheless, it is clear that the flat coal face vibration cannot control the operator noise level because the noise is characterized by many strong peaks over the frequency spectrum.

An upper limit on the fracture and face radiation noise level can be reasonably estimated by selecting the lowest noise level in dBA/Hz from the M3 spectrum in figure 27 (about 62 dBA/Hz) and assuming that level to be constant over the analysis band of 200 to 1600 Hz. This assumes the lowest noise level was caused by the coal face vibration. Figure 31 (trace A6) shows that the coal face vibration was flat over the analysis band. The noise spectrum from the coal face should follow suit. The calculated result is 93.5 dBA or almost 10 dBA down from the measured overall noise level. The coal face radiated noise level could, of course, be lower if other sources control the minimum overall noise level.

3.4 Conclusions on Longwall Noise Sources

The extensive series of laboratory and in situ noise and vibration measurements documented in this section provide an adequate data base to support the following conclusions.

1. The shearing machine is the predominant noise source in longwall mining, generating operational noise levels frequently in excess of 100 dBA at the operator's position.

2. The predominant noise source on the shearing machine is the cutter drums. Further, the primary radiating structure of the cutter drum is the helix.

3. The vibration of the remainder of the shearing machine structure (primarily in the structures closest to the cutter drums) is the next most important source of noise caused by coal cutting.

4. Coal fracture and face radiation noise is probably less than 90 dBA at the operator's position.

5. Headgate equipment can collectively generate high noise levels, sometimes in excess of 100 dBA. Principal noise sources in the headgate area are conveyors, drive motors, gearboxes, chain drives, hydraulic pumps, crushing machines, and, periodically, coal cutting machines.

6. Longwall plow systems generate less noise than shearing machines.

7. Armored chain conveyors generate potentially high noise levels in unloaded or lightly loaded conditions. However, away from the end roller/drive mechanism and with a measurable amount of coal on the pan, this noise source has been measured to be consistently less than 90 dBA.

8. Noise from hydraulic pump packages has been measured in excess of 95 dBA for normally operating pumps and approaches 110 dBA for pumps requiring maintenance attention. However, in no event was fluid dynamic noise away from the pump observed to pose a noise problem.

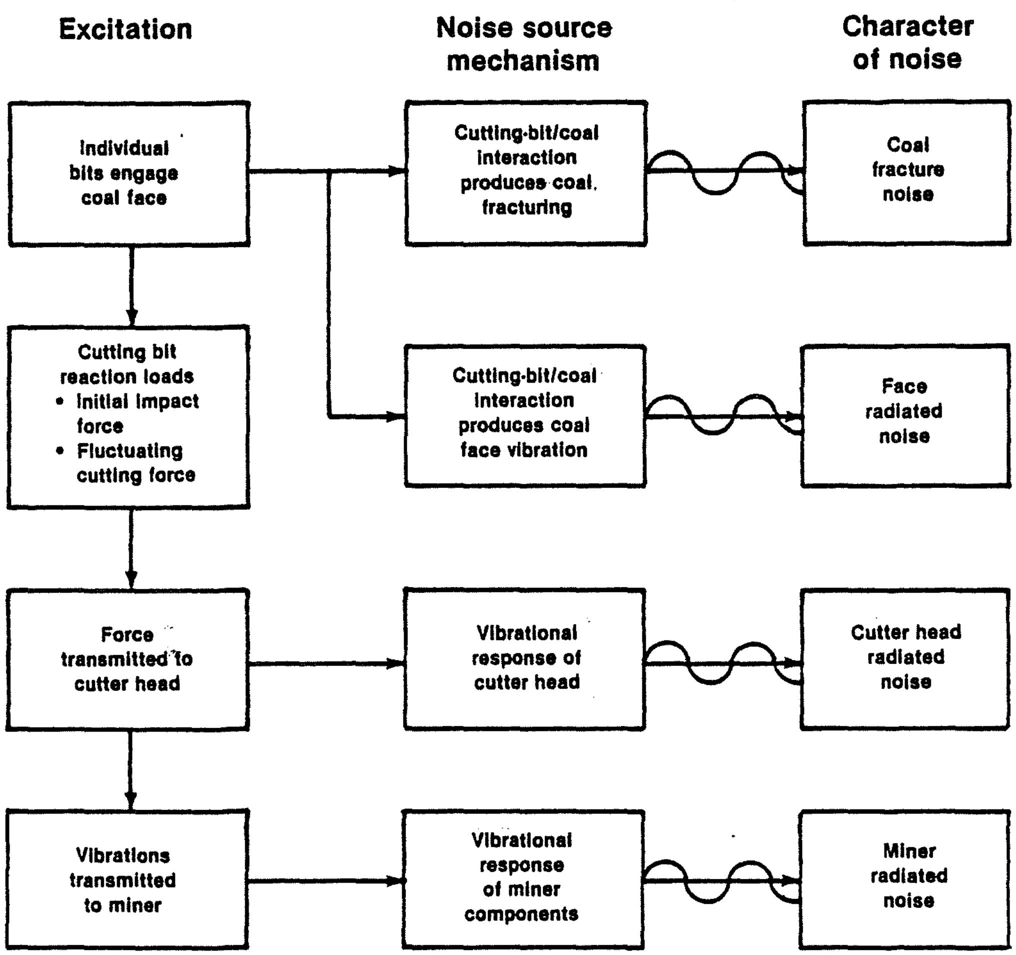
The noise sources that may require treatment, therefore, are coal cutting noise, stage conveyor noise, and pump/motor noise. Solutions to these noise problems are presented in sections 4 and 5.

Section 4

COAL CUTTING NOISE CONTROL

The noise sources directly associated with the cutting of coal are diagrammed below. The order of importance for these sources is primarily determined by the design and operation of the cutter drum.

Coal Cutting Noise Sources



The fundamental design and operational concepts for low noise coal cutting apply to all types of continuous miners, including longwall, shortwall, and auger. Taking advantage of the nature of the dynamic forces generated during coal cutting is the key to each coal cutting noise control concept presented in this section.

Continuous mining machines exploit the brittle fracture characteristics of the coal. When a cutting bit engages the coal face, the coal resists the bit's advance in a manner analogous to a spring. The applied force increases until the stress in the coal initiates localized brittle fracture. The coal fracture propagates for some characteristic distance from the point of force application. The cutting bit continues to advance, meeting steadily decreasing resistance as it clears out the fractured coal. When the bit again contacts unfractured coal, the process is repeated.

A representative force time history and PSD of a rigidly mounted bit cutting coal (figures 35 and 36) (10) clearly show the highly impulsive nature of the process. It is interesting to note that the impact force generated as the bit initially enters the coal is no more impulsive than the subsequent individual fracture events. The extensive experimental program from which figures 35 and 36 were taken demonstrated that both the stress required to initiate fracture and the characteristic distance of fracture propagation are relatively independent of cutting bit velocity. Hence, for a given type of coal, depth of cut, and bit configuration, the peak force that initiates coal fracture and the total number of fracture events that occur in a prescribed length of cut are velocity independent.

All continuous mining machines experience this type of excitation. In the frequency domain, the cutting force is flat at first and then at some frequency begins to decline with frequency at the rate of slightly more than $1/f^2$. The frequency at which this decline begins, called the cutoff frequency, is primarily a function of the coal type and the cutting bit velocity.

The peak and mean cutting forces are a function of the material being cut, bit type, and depth of cut. While not extensive, cutting force data have been obtained for a variety of coal, shale, and rock types at several different depths of cut (10-13). Table 3 summarizes the generally available data from both laboratory and in situ cutting tests.

TABLE 3. - Laboratory and in situ cutting force data

Material	Bit type	Depth of cut, in.	Mean force, lbf	Standard deviation, lbf	Peak force, lbf
Blue Creek coal	Plumb bob	.5	607	325	1,608
	. . . do	1	925	541	1,994
	. . . do	2	1,500	950	3,309
Pittsburgh No. 1 seam		.5	515	245	1,082
coal	. . . do	1	958	590	2,440
Pittsburgh No. 2 seam		.5	773	406	2,088
coal	. . . do	1	1,037	646	2,727
	. . . do	2	1,489	929	3,679
Illinois No. 6 seam coal	. . . do	.5	694	329	1,704
	. . . do	1	1,112	525	2,495
	. . . do	2	1,310	642	2,555
Illinois shale	Wedge	1	866	673	2,918
Bruceton synthetic shale	. . . do	.5	616	472	3,163
	. . . do	1	1,739	1,127	6,780
Blue Creek shale	Plumb bob	1	925	541	1,994
	Auger point attack	1	650	337	1,494
	Wedge type 1	1	689	533	3,330
	Wedge type 2	1	667	511	2,303
Sandstone	Point attack	.5	1,385	NA	NA

NA - Not available.

Recall the essential characteristics of coal cutting: First, coal resists advance of the bit in a manner analogous to a spring. Second, localized brittle fracture occurs when the tensile stress reaches a critical value. Third, the brittle fracture propagates for some characteristic distance from the point of force application. Fourth, bit velocity does not affect the first three characteristics. Of course, since real coal is not homogeneous, the above is only true in a very rough statistical sense. These characteristics are represented in the very simple but quite useful single-degree-of-freedom coal model illustrated in figure 37.

Assuming a constant bit velocity, V_b, the model results in a sawtooth cutting force, F_c, time history. A more accurate model is achieved if, instead of allowing the spring to go discontinuously to zero when the cutting force equals the fracture force, F_f, a steadily decreasing clean-up force is provided until the next "coal spring" is encountered by the bit ($F_c = F_f - K'_c * \delta x'$, where K'_c is the "spring constant" after fracture and $\delta x'$ is the distance of the bit tip from the point of fracture). The modified model produces the more triangular pulse shape seen in the actual cutting force time history. The model with experimentally determined "constants," K_c, K'_c , ℓ_c and F_f, can be used to evaluate certain cutter drum operational parameters and design concepts.

4.1 Noise Control by Reducing Bit Velocity

The dynamic coal cutting forces are the source of the coal cutting noise. The first principle of cutter drum design and operation is:

Reduce the bit velocity as much as possible.

The characteristic fracture length as described in the coal cutting model is independent of velocity. Therefore, the impulse time (or duration) of an individual fracture event is inversely proportional to the bit velocity. The typical impulse time determines the cutoff frequency of the force spectrum. The shorter the impulse time, the higher the cutoff frequency. Experiments with a wide range of coal, shale, and synthetic coals have shown that the one-third octave-band force level decreases about 1.5 to 2 dB per band above the cutoff frequency (figure 38). This is typical of the spectrum associated with a triangular pulse shape. Slower cutting, therefore, lowers the high frequency force levels by reducing the cutoff frequency.

Several factors combine to reduce the importance of low frequency dynamic cutting forces as a source of harmful radiated noise. Below 200 Hz, most cutter drum and miner structures are generally in the stiffness-controlled region of their structural response. In this region, the magnitude of the structural vibration due to a given unit of force is not amplified by the resonance phenomena. The transfer of energy from the vibrating structure to the air is also very inefficient in the frequency range below 200 Hz for the typical cutter drum, miner size, and configuration. Finally, as the frequency of the noise decreases below 1,000 Hz, the noise sound pressure level becomes less and less damaging to the ear. The A-weighting of noise spectra (figure 39) is typically used to represent this fact. MSHA workplace noise dose regulations are written in terms of A-weighted noise.

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. The hypothetical characteristics given in table 4 and the following paragraph illustrate and explain the noise control advantage of reduced bit velocity. The three hypothetical drums are configured to have the same production rate.

TABLE 4. - Hypothetical drum characteristics

		A	B	C
Cutting diameter	in	40	40	40
Speed	rpm	60	30	30
Bits per cutting line		1	2	1
Depth of cut (max.)	in	1	1	2*

*Doubles force level on the bit over the force level of a 1-in-deep cut.

The force spectrum acting on the bits of each drum type is postulated in figure 40. The force scale of the figure is arbitrary since it is the relative force magnitude of the three drums that is of interest. The dynamic force on each bit of drum B is 6 dB less than on each bit of drum A in the frequency region above the cutoff frequency of drum A. Reduced force on a cutting bit translates directly to reduced miner structural vibration caused by that bit. The total vibration of a continuous mining machine is the sum of the vibration caused by each bit that is in contact with the coal. Since drum B has twice the number of bits as drum A, a 6-dBA reduction in the force on each bit would result in a 3-dBA overall structural vibration and, hence, noise reduction above the cutoff frequency.

Drum C has the same number of bits as drum A. Drum C operates at one-half the cutting speed but twice the cutting depth to maintain the same production rate as drum A. As noted on the description of drum C, the cutting force magnitude is assumed to double with a doubling of cutting depth. This is actually somewhat conservative because data (table 3) often indicates only a 1.5- to 1.7-fold increase in force per doubling of cutting depth. The 3-dB force reduction (figure 40) achieved above the cutoff frequency of drum A should result in a 3-dBA reduction in coal cutting related noise.

Reference 9 contains quantitative experimental support for the above analysis method. figures 41 and 42 show the average measured cutting force spectra from LCA cutting tests at 16 and 96 inches per second, respectively. The lower bit velocity resulted in a 10 to 20 Hz cutoff frequency compared to the 70 to 140 Hz for the faster cut. Because of the small number of samples, the exact values are very suspect, but the general trend is clearly evident.

Although an overall noise reduction of around 3 dBA is not sufficient to solve all cutter drum noise problems, the reduction can be achieved for very little cost. In addition, there is strong evidence that slower and/or deeper cutting produces less dust and also reduces the likelihood of spark ignitions of methane (14-16).

4.1.1 Application to Longwall Shearer Cutter Head

The cutter drum torque and operational speed are factors that must be established at the factory. It was not feasible, therefore, to implement a change in cutting speed in the test program of a new low-noise longwall shearer cutter drum.

4.2 Noise Control Through Cutter Head Structural Response Alteration

The dynamic forces at each bit-coal interface are reacted by the cutter drum structure. As shown in section 3, the resulting vibration of the cutter drum structure is generally the major coal cutting noise source. The second principle of cutter drum design is:

Shift the frequency of the first structural mode as high as possible.

The third principle of cutter drum design is also related to the response characteristics of the cutter drum structure:

Increase the structural damping as much as possible.

Because the excitation also passes through the cutter drum to the other miner components, these rules apply to all miner structures. The objective rules is to reduce the resonant response of the structures to the dynamic excitation of the coal cutting forces.

Because the frequency domain cutting force level decreases rapidly as the frequency increases, the coal cutting noise above about 2000 Hz is generally below the level of concern. Indeed, noise below the 1000 Hz band generally controls the overall level. The second design principle limits the structural response to the cutting forces primarily by eliminating structural resonances in the "problem" frequency range. Those structural resonances that cannot be eliminated from the band of concern should be well damped (design principle number three). Damping, of course, limits the magnitude of the structural response at the resonances.

4.2.1 Application to Longwall Shearer Cutter Drum - Stiffening and Damping

In section 3, it was shown that the helix is the primary coal cutting noise source of the Joy longwall shearer cutter drum. Following the precepts of section 4.2, the helix of the cutter drum was stiffened and damped. Figures 43 and 44 show the stiffening and damping technique used. The first resonance of the helix was increased from about 200 Hz to 700 Hz. The modal density (or number of resonances per frequency band) increased rapidly above 700 Hz. Because of the stiffness and mass of the stiffened-helix configuration, the damping technique only slightly increased the helix damping.

Cutting tests at the USBM synthetic longwall seam were conducted to document the effectiveness of the noise control techniques. To establish a baseline, the noise radiated during cutting operations with a standard cutter drum was measured (figure 45). Unfortunately, the noise radiated during the cutting tests with the stiffened and damped cutter drum was incorrectly recorded and the data lost. Structural response transfer functions relating applied force to helix vibrational velocity were obtained for both the standard cutter drum and the stiffened and damped cutter drum. Radiated noise is, of course, proportional to the structural velocitiy. The transfer functions, when multiplied by the same A-weighted cutting force spectrum, yields the A-weighted structural velocities shown in Figure 46. This procedure indicates the stiffened and damped cutter drum should be 3 to 4 dBA quieter than the standard cutter drum.

Consolidation Coal Company allowed two of their cutter drums to be stiffened and damped for testing on an Eickhoff 300 shearing machine in Consol Mine #95 near Fairmont, WV (figures 47, 48). The stiffened and damped cutter drums achieved a 3 dBA noise reduction at the lead operator's position. While the tail drum functioned adequately, clogging problems were experienced with the lead drum and it had to be removed. The stiffened and damped tail drum was used until completion of the panel for a total of about eight months. The cause of the clogging was probably the obstructions indicated in figure 49. Both the water supply and the handling mount were removed from the conveying path. The modified lead stiffened and damped drum has not been retested.

Several steps could be taken to achieve somewhat greater cutter drum noise reductions from the basic noise control technique shown in figures 43 and 44. The helix damping could possibly be increased by the use of a different damping material or a different sand cavity design. The number of helix resonances within the frequency range of concern could be decreased by segmenting the stiffened and damped helix. The stiffened and damped helix is segmented by cutting the helix and stiffener from the outer edge to the core in several places along the helix length. Additional closure plates are added at each cut to create several closed and separated sand cavities. For this arrangement to provide optimum results, the cutter drum core must be very rigid and/or well damped so that significant vibrations cannot pass from one segment to the next through the core. Figures 50 and 51 this alternative stiffened and damped helix cutter drum design. Fabrication of the segmented helix would, however, be very involved and expensive. A simpler method was devised to achieve the same result and is described in the following section.

4.2.2 **Application to Longwall Shearer Cutter Drum - Staged Bit Lacing**

The helix of the traditional cutter drum performs two functions: It supports the cutting bits in the standard helical pattern, and it moves the cut coal to the conveyor. The dual function forces a configuration that is inherently highly resonant and lowly damped in the important frequency range below 2000 Hz. As previously discussed, attempts to stiffen and damp the helix result in only moderate noise reductions. The basic geometry limits the increase in stiffness and resonant frequency that can reasonably be achieved. Increases in damping are limited by both geometry and material properties.

A different concept in bit lacing was developed to eliminate the necessity of a continuous, two-function steel helix. In this lacing concept, called "staged bit lacing," the bits are arranged in sets of cutting bit arrays. Each cutting bit array consists of one or more bits mounted on a single stand. The bits of each cutting array are set adjacent to one another. The number of bits on each stand is a function of the bit spacing desired. The individual stands have no natural frequencies within the range of interest. Significant structural modal response and the attendant noise is thus avoided.

The "bit stands" design allows the complete removal of the continuous steel helix, thus eliminating the most significant noise-radiating structure on the cutting drum. At first it was believed that this would hamper the conveying function of the drum, so an abrasion-resistant polymer sheet (1-inch thick) was bolted to the bit stands to provide a helical conveying surface. However, subsequent underground tests at Consol mine No. 95 showed that the polymer helix was unnecessary. In fact, the polymer wore away completely due to abrasion from coal and rock, yet the conveying function of the drum remained unimpaired. This showed that the large size and tapered shape of the bit stands provided an adequate conveying surface for the Mine No. 95 situation (60-inch drum in a 90-inch coal seam).

Figures 52 and 53 illustrate a cutter drum design that implements these concepts. It will be referred to as the "stage-laced" longwall shearer cutter drum. The cutter drum core design is as shown in figure 50.

A stage-laced cutter drum with a nonresponsive core was fabricated (figure 54) for testing at the USBM longwall mockup. While the bit lacing was not exactly as detailed in the previous drawings, the fundamental configuration of the test cutter drum was as shown. Shaker tests of the new bit stand/helix configuration demonstrated a very large decrease in structural response to a given excitation as compared to the $1\frac{1}{2}$-inch-thick steel helix of the Joy longwall cutter drum (figure 55). Cutting tests with the USBM longwall shearer showed a system noise reduction at the operator's position of 4 to 5 dBA (figures 56 and 57).

These cutting tests included a running conveyor. The lightly loaded conveyor noise was about 93 dBA for all cutting tests. The unusually high conveyor noise was possibly due not only to the lightly loaded condition but also to lack of coal damping underneath the trough and perhaps to pronounced conveyor trough discontinuies. Given the overall noise and the conveyor noise level, the noise due to cutting can be calculated. For the standard cutter drum, the cutting noise calculates to be about 97.5 dBA. The stage laced cutter drum cutting noise calculates to about 91.5 dBA. Hence, a 6 dBA cutting noise reduction is indicated. At this level, it is probable that the cutting noise level is dominated by a combination of secondary equipment and coal face vibration noise. For example, if the noise from coal face vibration were in the previously estimated 90 dBA range, cutter drum radiated noise would have to have been about 88 dBA. It is difficult, however, to come to any firm conclusions from these data because cutting noise was in the region dominated by conveyor noise.

A stage laced cutter drum was fabricated for in-mine testing on an Eickhoff 300 shearing machine in Consol Mine #95 (figure 58). Noise measurements at the lead drum operator position showed that the noise-controlled drum provided a 5 to 6 dBA noise reduction compared to a standard drum on the same shearer under the same cutting conditions. Overall noise levels were about 95 dBA with the noise-controlled drum and 100-101 dBA with the standard drum; this increased the allowable operating time per shift from 2 to 4 hours. Frequency analysis of the noise data showed that helix noise (peaks in the 200-1,000 Hz range) was essentially eliminated and bit stand noise did not control the overall noise level. Operationally, the noise-controlled drum performed quite well, despite the initial skepticism of the miners when first introduced to this "radical" change from the standard drum design. Although the polymer helix wore away, the drum operated almost continuously for a period of more than six months without failure, and has been able to cut and load roof rock when necessary. The stage laced cutter drum was removed from the mine after completion of the panel. A comparison of the mid-body noise to the noise at the lead cutter drum indicates that the overall noise level at the operator's position is controlled by machine radiated noise rather than by cutter drum radiated noise (see figure 59). Of special note are the strong peaks below 800 Hz. These noise peaks probably emanate from the haulage system (Eicomatic) and could be reduced by standard enclosure noise control techniques if greater noise reduction is desired.

4.3 Noise Control Through Isolation of Radiating Structures from the Dynamic Cutting Forces

The dynamic cutting forces cause vibration not only of the cutter drum but also of the other miner components. The vibrations propagate throughout the mining machine. The fourth principle of cutter drum design is

Isolate high frequency cutting forces from the miner structure.

As previously discussed, the higher frequency coal cutting forces cause the unacceptably high noise levels. The higher frequency coal cutting forces can, in principle, be isolated from the major radiating surfaces of the cutter drum and the miner. This has been demonstrated by both analysis and experiment. Two analytical models were developed. The first model was capable only of evaluating the feasibility of the concept. The second model allowed the detailed evaluation of candidate isolated cutting tool designs.

The peak forces seen in the force time history of figure 35 produce the stress required to initiate brittle fracture of coal. The peak force, not the mean force, cuts the coal.

Therefore, if coal cutting is to occur, the force time history at the tip of an isolated cutting tool should be very similar to that of a rigidly mounted tool. The force peaks must be reached. The initial loading rate of the bit tip at each fracture event may be somewhat smoothed by the softer response of the isolator. This effect is, however, limited by the number of fracture events that must occur in a given length of cut. Therefore, the force on the bit tip can be considered, in this rough analysis, to be independent of the response characteristics of the cutting tool.

A simple single-degree-of-freedom isolated cutting tool is shown in figure 60. Assume that

- The mining machine has sufficient power to maintain a constant cutter drum velocity, V_h, regardless of the force generated at the bit-coal interface, F_c.
- The cutter drum structure is rigid compared to the isolator.
- The cutting bit and tool holder are rigid compared to the isolator.
- The isolator is massless.

A force balance on the isolator yields

$$F_b(t) = K_i(X_b(t) - X_h(t)) + C\, d/dt\, (X_b(t) - X_h(t)), \tag{4}$$

where

$F_b(t)$ = force on bit side of the isolator at time, t;

K_i = isolator spring constant (lb/in);

$X_b(t)$ = position of point b at time, t;

$X_h(t)$ = position of point h at time, t.

$F_h(t)$ = $F_b(t)$, since isolator is massless.

A force balance on the bit tip yields

$$F_c(t) = F_b(t) + m d^2/dt^2\, X_b(t); \tag{5}$$

m = mass of the isolated cutting tool;

C = viscous damping coefficient.

Take the Fourier transform of equations 4 and 5 to convert to the frequency domain,

$$\int_{-\infty}^{\infty} (F_b(t) - K_i(X_b(t) - X_h(t)) - C\, d/dt(X_b(t) - X_h(t))e^{-i\omega t}\, d\omega = 0$$

$$\int_{-\infty}^{\infty} (F_c(t) - F_b(t) + md^2/dt^2\, X_b(t)e^{-i\omega t}\, d\omega = 0$$

For these equations to hold, the integrands must also be equal to 0. Hence,

$$F_b(\omega) = K_i(X_b(\omega) - X_h(\omega)) + iC\omega\,(X_b(\omega) - X_h(\omega)) \tag{6}$$

$$F_c(\omega) = F_b(\omega) - m\ \ ^2X_b(\omega) \tag{7}$$

Solve for $X_b(\omega)$ in equation 6.

$$X_b(\omega) = \frac{F_b(\omega) + X_h(\omega)(K_i + iC\omega)}{(K_i + iC\omega)}$$

Substitute into equation 7.

$$F_c(\omega) = F_b(\omega) - m\omega^2 \quad \frac{F_b(\omega) + X_h(\omega)(K_i + iC\omega)}{(K_i + iC\omega)}$$

$$F_c(\omega) = F_b(\omega) \quad \frac{m\omega^2 F_b(\omega)}{(K_i + iC\omega)} - m\omega^2 X_h(\omega) \tag{8}$$

Recall that

$$m \cdot d^2/dt^2\ (X_h(t)) = -\ m\omega^2 X_h(\omega)$$

By constraint, the cutter drum moves at a constant velocity

$$d^2/dt^2\ (X_h(t)) = -\ m\omega^2 X_h(\omega) = 0$$

Then $$F_c(\omega) = F_b(\omega) \quad 1 - \frac{m\omega^2}{(K_i + iC\omega)} \tag{9}$$

Force transmissibility is defined as the ratio of force at the base of the isolated cutting tool to the force at the bit tip. After a little algebra, the result is

$$T = \left[\frac{(K_i^2 + \omega^2 C^2)^2}{(K_i^2 + \omega^2 (C^2 - mK_i))^2 + (\omega^3 mC)^2} \right]^{1/2}$$

This equation can be used to evaluate the gross feasibility of the isolated cutting tool. It also demonstrates the concept and advantages of force isolation. Figure 61 gives the transmissibility at various levels of damping. The isolated cutting tool simply allows the mass and acceleration of the tool to react a portion of the coal cutting force.

A mine-worthy isolated cutting tool must have the following characteristics.

- The highest natural frequency of the tool isolation system must be less than 170 Hz to assure that isolation occurs above 250 Hz.
- Elastomeric isolator elements must not be allowed to deflect more than 10 percent of the isolator thickness under shale cutting force levels.
- Cutting tool space constraints must not be violated.

The first characteristic assures that the frequency region of force isolation begins before the problem frequency band. The second characteristic is required to extend the life of the elastomeric isolator elements. The third characteristic is an obvious requirement. It is most difficult to meet when the prototype reduced-noise cutter drum must fit on present machinery.

For the purpose of rough feasibility calculations, assume a maximum allowed isolated cutting tool deflection, d_m, of 0.15 inch, damping, ζ, of 8 percent and natural frequency of 170 Hz. Table 5 gives details on shale cutting forces (13), which are considered "worst-case" conditions. The design values are the maximum force levels measured during the experimentation. Since the excitation of the isolated cutting tool is a continuous dynamic process, the peak deflection of the isolator will occur at the resonance. The transmissibility equation can be used to help calculate the peak deflection of the isolator at resonance where the equation reduces to

$$T \simeq \frac{2\sqrt{K_i\, m}}{C} = \frac{1}{2\zeta}$$

Hence, the total equivalent peak static force on the isolator at resonance is

$$F_E = (F_B * CF * 1/2\zeta) + F_m$$

where the variables are as described in table 5.

TABLE 5. - Shale cutting parameters for feasibility calculations

Parameter		Measured[1]	Design
Peak force, F_p	lbf	2,900	7,800
Mean force, F_m	lbf	900	1,500
Standard deviation of resultant cutting force,	lbf	670	1,400
Peak force to root-mean-square force ratio, CF		3.0	3.6
Root-mean-square force in band of width (2 f_n centered at f_n, F_B	lbf	135	525
Equivalent static force, F_E	lbf	3,800	15,000

Assumed damping, ζ, 8 pct. Assumed resonant frequency, f_n, 170 Hz. [1]Average.

The isolated cutting tool mass needed to meet the dynamic conditions assumed above can be calculated by

$$m = \frac{(F_E/d_m)}{(2\pi f_n)}$$

$$m = \frac{(15000/0.15)}{(2\pi\ 170)^2}$$

$$m = 0.09 \text{ lbf sec}^2\text{/in. (33 lbm)}$$

These simple calculations indicated that the isolated cutting tool concept was feasible and deserved much more rigorous scrutiny.

A more complex model was developed to aid in the detailed design and evaluation of isolated cutting tools. The analytical model includes rigid body cutting tool motion in all translational and rotational directions. Any number of isolators are allowed by the computer program that implements the model. The user simply tells the program where the elastic center of each isolator is located and supplies the stiffness and damping of the isolator in the coordinate directions best suited to the isolator. The program calculates the required global stiffness and damping vectors from the local isolator specific values. The total mass and the rotational inertia of the cutting tool can be approximated by the combination of several simple geometric shapes to match the more complicated shape of the cutting tool. The program combines the known mass and rotational inertias of the simple shapes to obtain the total cutting tool mass and global coordinate system inertias. With these inputs, the program calculates the rigid body natural frequencies, the frequency domain cutting force transmissibility, and the maximum displacement of the cutting tool. The program is a fast and effective way of evaluating and optimizing the response characteristics of an isolated cutting tool design.

The first experimental evidence that isolated cutting tools could cut coal and achieve significant isolation was obtained with a linear cutting apparatus (LCA). The LCA (figure 34) was designed specifically for controlled coal cutting force measurements. An isolated cutting tool (figure 62) was built for mounting on the LCA. A rigid cutting tool was also built (figure 63). Coal was cut at 60 ips from opposite sides of the same coal sample (figure 64). One side was cut with the rigid tool and the other side with the isolated tool. Cuts were made on several blocks of coal in this manner. Figure 65 gives the force time histories experienced at the cutting tool mount for both tools. The force transmitted to the mount by the isolated cutting tool is clearly smoother. Figure 66 gives the frequency domain results in the form of isolator transmissibility (force at base for isolated tool divided by the force at the base for the rigid tool). For this particular tool, the isolation begins at 175 Hz. At 300 Hz, only one tenth of the applied dynamic force is passed on to the cutter head. Note also that the dynamic forces below 175 Hz are amplified due to the isolator resonance. The potential overall noise control effect of an isolated cutting tool with these dynamic characteristics is estimated by multiplying a typical longwall shearer A-weighted noise spectrum by the transmissibility curve shown in Figure 66. The result of this calculation on a spectrum analyzer indicates a potential noise reduction of over 10 dBA. This, of course, assumes the isolated tool is mounted on the drum without altering the structrual resonse characteristics of the drum. It also assumes all the noise is from vibrations induced by the cutting forces. The weight and particle size distribution of the coal cut by the isolated bit was not statistically different from that of the rigidly mounted bit. These analytical and experimental results clearly indicate that coal cutting forces can be isolated to achieve significant A-weighted noise reduction.

4.3.1 Application to the Joy Longwall Shearer Cutter Head

The concept of an isolated cutting bit (ICB) calls for the bit to be isolated from the bulk of the cutter drum structure. The isolated cutting tool computer model was used to evaluate the candidate ICB designs. After many design iterations, it was concluded that an ICB with adequate response characteristics would probably disrupt the conveying function of the cutter drum. Fortunately, the cutter drum structural modifications described in section 4.2.1 provide adequate relief from the cutter drum radiated noise problem. Isolation of the cutting forces from the cutter drum structure is therefore not required. The cutter drum can be isolated from the shearing machine, however, to reduce the noise radiated by the shearer body.

The stage-laced cutter drum shown in figures 52-54 was also isolated from the shearer body by the technique shown in figure 67. Vibration measurements were taken on the cowl, ranging arm and the machine body during full cutting tests with both the standard and the isolated cutter drum at the Bruceton Research Center. The overall vibration of these machine body parts was reduced about 10 dBA by the isolated cutter drum. A good indication that this reduction was indeed due to isolation and not operational differences is obtained by dividing the vibration spectrum produced using the isolated cutter drum by the vibration spectrum produced using the standard drum. The resulting curve is the transmissibility of the isolation system. The result of this operation (figure 68) displays some of the typical transmissibility curve characteristics (see figures 59 and 66). The curve does depart from the transmissibility curve characteristic in the higher frequency region where some vibration amplification is indicated. A possible explanation for this behavior is that dominating higher frequency vibration may reach the machine through paths other than through the drum. One such path could be through steel on steel impacts of the bouncing shearer with the track and face conveyor. The stage-lacing of the Bruceton drum may well have increased the bouncing of the shearer. This could account for the amplification in the higher frequency region.

Because the isolation was also combined with stage-lacing, the overall noise reduction attributable to isolation alone could not be determined.

The stage-laced, isolated cutter drum was not given an in-mine test because a Joy longwall machine was not readily available for the test program. A similar isolation scheme, however, was tested on a continuous miner (figure 69). (18) This test demonstrated the long term, in-mine durability of the isolators in a more severe service than the isolators in the longwall cutter drum would experience. One of the critical unknowns of the isolation concept was the in-mine survivability of elastomeric isolators. The continuous miner results therefore, serve as a fundamental validation of the isolated coal cutting tool concept.

Section 5

SECONDARY NOISE SOURCE CONTROL

The intensive investigation of longwall mining systems identified the shearer cutter drum as the primary longwall noise source. The bulk of the current program effort was directed toward the development of quieter cutter drums. This concentration was well directed in that significant system noise reductions were achieved by redesign of the cutter drums.

Two other types of noise sources were also identified as being potentially harmful. The program scope called for the identification of noise control concepts for these additional sources. These sources can be categorized as pump/motor noise sources and as conveyor noise sources. Noise close to (within 2 feet of) a hydraulic pump can reach 100 dBA. Electric motor noise can also be excessive. Conveyor noise includes the armored chain face conveyor and the stage loader. The stage loader is usually noisier than the armored chain face conveyor because of higher chain speeds and the proximity of the chain drive and idler rollers. The stage loader may also have somewhat less damping from coal underneath and on the sides of the pan.

Fortunately, most pumps and motors can be rather easily enclosed and/or remotely located to reduce worker noise exposure. The in-mine tested electric motor enclosure illustrated in figure 70 achieved an 8.5 dBA noise reduction with no evident durability problems (17). Hydraulic pumps can be treated in a similar manner to achieve an equivalent result. Further research on the subject is therefore not warranted. Mine equipment manufacturers need simply apply the proven technology.

Noise control of chain conveyors presents a more difficult problem. A significant amount of noise control work has been done on the continuous miner chain conveyor (18-22). The longwall armored face conveyor and bridge conveyor have similar noise source mechanisms as the continuous miner conveyor. Hence, the noise control measures recommended in the referenced reports could be implemented to reduce the longwall face conveyor and bridge conveyor noise. The reports primarily describe methods of increasing the conveyor structural damping and of smoothing the path of the chain and flights.

Steel-on-steel impacts are the root cause of most of the conveyor noise. The impacts occur along the conveyor panline, at the conveyor drive sprocket, and at the conveyor tail roller, or sprocket (figure 71). Especially violent impacts of the conveyor flight and chain with the panline often occur as the chain changes direction at the conveyor ends. The conveyor structure vibrates in response to each impact and radiates noise. Noise is radiated by the initial motion of the structure as it deflects under the impact force. Noise is also generated as air is forced from between the flight and the structure. The combined result is called impact noise. The noise radiated by the resonant vibration of the structure following the removal of the impact force is called resonant noise. Structural damping reduces resonant noise. Smoothing the path of the chain and flights reduces the initial impact force and, hence, both the impact and resonant noise.

Increased damping of the top and bottom plates of a typical continuous miner chain conveyor reduced the conveyor noise by up to 5 or 6 dBA when empty, but only 2 to 3 dBA when full (21). It is very difficult to significantly increase the damping of a loaded conveyor. Another problem is the difficulty often encountered in applying damping to all the structures to which the impact-induced vibration propagates. If a full conveyor noise reduction greater than 2 to 3 dBA is needed, it seems that a different approach is required. The conveyor noise control technique presented in this section significantly reduces both impact and resonant conveyor noise in a very simple manner. The in-mine durability of the concept, however, is yet to be demonstrated.

5.1 Concept Presentation

When a conveyor flight (or chain) impacts a panline discontinuity, a reactive impulse causes the flight to jump the obstruction. The magnitude of this impluse is a function of the discontinuity geometry, and the flight (and chain) shape, mass, and velocity. Significant noise and vibration reduction can be achieved by providing a means for this impulse to be imparted to the flight over a longer impact time. Consider an impact that takes the shape of a half-sine pulse of duration, T_1, and peak force, F_1, (see Figure 72). The total impulse is $(2/\pi)T_1F_1$. If the impulse time were doubled, the peak force required to maintain the same impulse would be only $F_1/2$ (a 3 dB peak force reduction). Of particular importance to noise control, however, the frequency domain force components above the frequency, $1/T_1$, would be 6 db lower (see again Figure 72). The drawing in figure 73 illustrates a quiet conveyor chain and flight design. The new flight design eliminates steel-on-steel impacts along the panline. All impacts of the moving flights with the miner structure occur at a isolated interface. The chain floats above the surface and never contacts the panline. The miner structure would experience significantly lower high frequency (greater than 200 Hz) excitation from isolated flight impacts. Since the isolated conveyor flight alters the excitation and not the structural response of the conveyor or miner structure, the loaded conveyor noise reduction should be as large as the empty conveyor noise reduction.

5.2 Concept Testing and Evaluation

To further evaluate the concept, a continuous miner conveyor chain was equipped with isolated flights and tested on the Lee Norse HH105 miner at the Wyle Mining Noise Test Facility. The Lee Noise miner equipped with isolated continuous mine conveyor flights is pictured in Figures 74 and 75. The height of the miner conveyor tunnel had to be increased in a few places to allow passage of the new three inch tall conveyor flight. Two types of noise tests were conducted, free field and reverberant field. Noise measurements were taken two feet from the miner and three feet from the ground at three positions along the miner length; at the tail roller, at the operator's position, and at the head. The conveyor tail section was positioned straight for one series of tests and the tests repeated again with the conveyor tail section at a 45° angle. All tests were conducted with the conveyor running at full speed (340 ft/min). A stop watch was used to verify that there were no significant chain speed differences between the tests. All tests were conducted on the same day. The results were as follows.

Conveyor Noise

Condition	Noise With Steel Flights dBA	Noise With Isolated Flights dBA
Free field, straight conveyor		
at operator	100	94
at tail	103	98
at head	97	93
Free field, 45° angle conveyor		
at operator	97	90
at tail	101	95
at head	96	96

Conveyor Noise

Condition	Noise With Steel Flights dBA	Noise With Isolated Flights dBA
Reverberant field, straight conveyor		
at operator	106	99
at tail	108	101
at head	96	96
Reverberant field, 45° angle conveyor		
at operator	102	96
at tail	104	98
at head	102	98

While the data brings up several interesting facts, the basic result was that simple substitution of the isolated flight for the steel flight achieved about 6 to 7 dBA noise reduction at the operator's position regardless of operational mode or acoustic field conditions. During the straight conveyor, isolated flight tests in the free field, it was evident to the ear that noise radiating from the discharge section was dominating the noise at the operator's position. The noise seemed to be from the impacting of the conveyor chain on the steel tail roller. This fact is somewhat supported by the data of this first series of tests. When the tail was swung to the 45° angle position, a 3 dBA noise reduction occurred at both the tail and the operator's position (but not at the head). It is postulated that the reduced chain tension of the 45° angle position may be responsible for the reduction. As an aside, previous noise studies of this miner indicated an increase in noise when the discharge section was swung from straight to the 45° angle position (18), conveyor chain and flights banged heavily on the bottom entry deck when in the loose, 45° angle position. The banging was significantly less severe in the straight position. When the conveyor chain return tunnel clearence was increased to make room for the isolated flight, the clearance between the tail roller and the bottom deck at the return tunnel entry was also significantly increased. Hence, chain banging at the tunnel entry did not increase as the tail section was swung from straight to the 45° angle position.

A second series of tests were conducted with a modified discharge section to quantify the influence of the standard discharge section on the noise at the operator's position. The steel tail roller was replaced by a urethane clad tail roller and the take-up plate was removed. The results were as follows.

Conveyor Noise with
Modified Discharge Section

	Noise with Steel Flights (dBA)	Noise with Urethane Sleeve Isolated Flight (dBA)
Free Field, Straight Conveyor		
Operator	101	90
Tail	102	93
Head	98	91
Free Field, 45^{o} Angle Conveyor		
Operator	99	89
Tail	99	90
Head	99	96

Hence, after the discharge section noise was reduced, the isolated conveyor flight achieved over 10 dBA noise reduction at the operator's position.

Consider a two source model, one source being the conveyor pan line plus the conveyor drive noise and the other source the discharge section noise (that noise generated by the interaction of the chain and the tail roller). Assume that the straight conveyor, free field noise at the operator's position with the modified discharge section and isolated flight is due solely to conveyor panline and drive noise (90 dBA). Recall the noise generated with the isolated flights and the standard discharge section (94 dBA). Calculate the noise from the standard discharge section.

$$90\text{dBA} + X = 94\text{ dBA}$$

and X=92 dBA, the noise at the operator's position due solely to the interaction of the chain/flights and tail roller.

Two aspects of the urethane sleeve isolated conveyor flights remain to be fully evaluated. First, the shape of the flight may not adequately prevent the build up of mud on the panline. Above-ground tests should be conducted to evaluate this performance factor. The shape of the flight is not super-critical and may be altered to improve performance in this area if required.

Secondly, the in-mine duribility of the urethane material in this specific application is yet to be established. The material has, however, been used with very good success as a bulk material chute liner. The material has a very good chance of providing adequate service life, especially if the isolated flight configuration allows in-mine sleeve replacement.

Because of the questional durability of the urethane sleeve isolated flight, a steel shell isolated flight was designed for testing (figures 76, 77). The specific configuration of the prototype steel shell isolated flight should not be considered ideal. The design was chosen merely to prove the steel shell isolation concept in the form of an inexpensive retrofit to the steel portion of the urethane sleeve isolated flight. The test procedure followed that previously outlined. The test was conducted with the modified tail section to get an uncontaminated comparison of the urethane sleeve versus the steel shell isolated flight generated noise. The results were as follows (previous urethane sleeve isolated flight results are also included for comparision).

Conveyor Noise with
Modified Discharge Section

	Noise with Steel Flights (dBA)	Noise with Urethane Sleeve Isolated Flight (dBA)	Noise with Steel Shell Isolated Flight (dBA)
Free Field Straight Conveyor			
Operator	101	90	96
Tail	102	93	95
Head	98	91	95
Free Field, 45° Angle Conveyor			
Operator	99	89	95
Tail	99	90	97
Head	99	96	95

To get an estimate of the noise reduction achievable with the steel shell isolated flights without discharge section modifications one can simply add in the noise at the operator's position that was calculated to be from the chain/flight and steel tail roller interaction (92 dBA). Thus, 96 dBA plus 92 dBA equals 97.5 dBA, or about 3.5 dBA noise reduction. The steel shell isolated conveyor flight will therefore, result in a 3 to 4 dBA noise reduction at the operator's position. Again, this reduction should be independant of conveyor loading conditions (i.e, panline damping).

Section 6

CONCLUSIONS AND RECOMMENDATIONS

The longwall shearer cutter drum, specifically the cutter drum helix, is the primary noise source in longwall mining. The new stage-laced cutter drum design presented in this report reduced noise at the lead-drum operator position by 5 to 6 dBA. Additional system noise reduction could be achieved by treatment of the shearing machine motor and hydraulic noise sources with standard enclosure methods.

Attempts to isolate the cutter drum structure from the cutting forces were unsuccessful. The primary difficulty was providing sufficient isolation mass and isolator stiffness while maintaining the conveying function of the cutter drum. Isolation of the entire cutter drum from the shearer proved successful in laboratory tests, but was not in-mine tested. The concept should be pursued further for noise control purposes only after hydraulic and motor noise have been reduced.

Stage loader noise was also identified as a potential problem. Because increased damping of a conveyor loaded with coal is very difficult to achieve, some form of conveyor chain and/or flight isolation is needed to significantly reduce chain conveyor noise. Hence, isolation of the conveyor chain impacts from the conveyor structure was investigated. Two types of isolated conveyor chains were tested on a continuous miner conveyor. The urethane sleeve isolated conveyor flights achieved a 6 to 7 dBA noise reduction with the standard steel conveyor tail roller and a 9 to 11 dBA noise reduction with a urethane clad tail roller. The mud cleanup capability and the in-mine druability of the urethane sleeve isolated flights remain to be tested. Isolated conveyor flights with steel shielded isolators achieved a 3 to 4 dBA noise reduction with relatively low durability failure risk. The results of these tests provide ample evidence that the chain conveyor noise can be reduced to acceptable levels through isolation of the conveyor chain from the bulk of the conveyor structure.

Section 7

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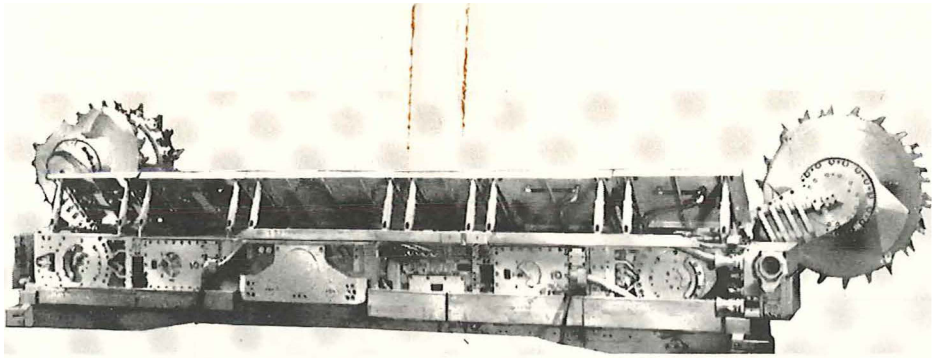
FIGURES



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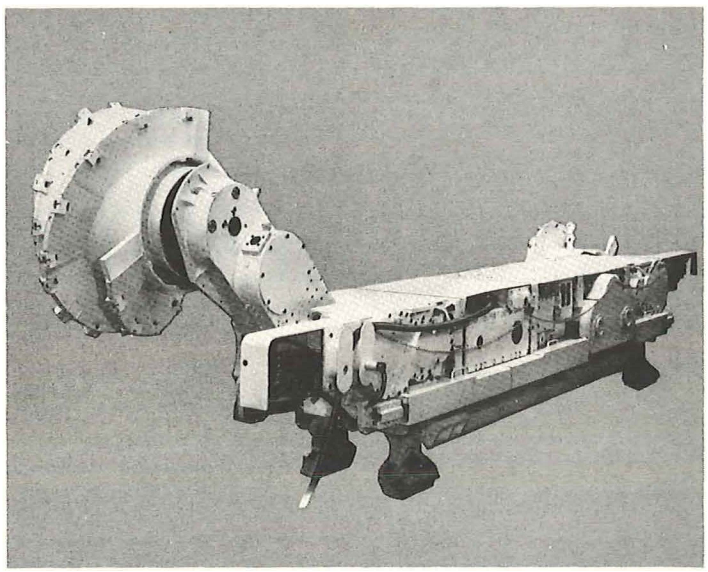

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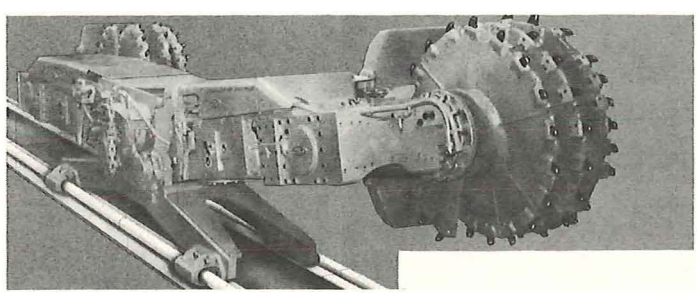
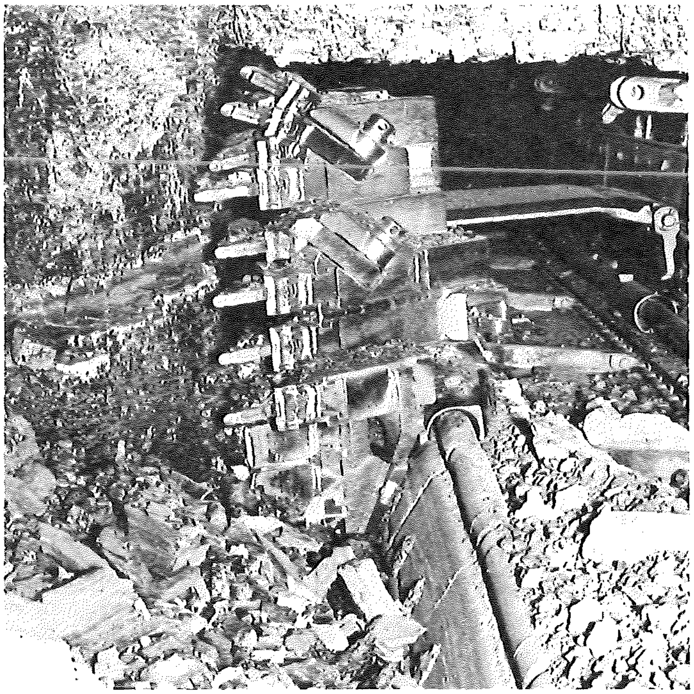
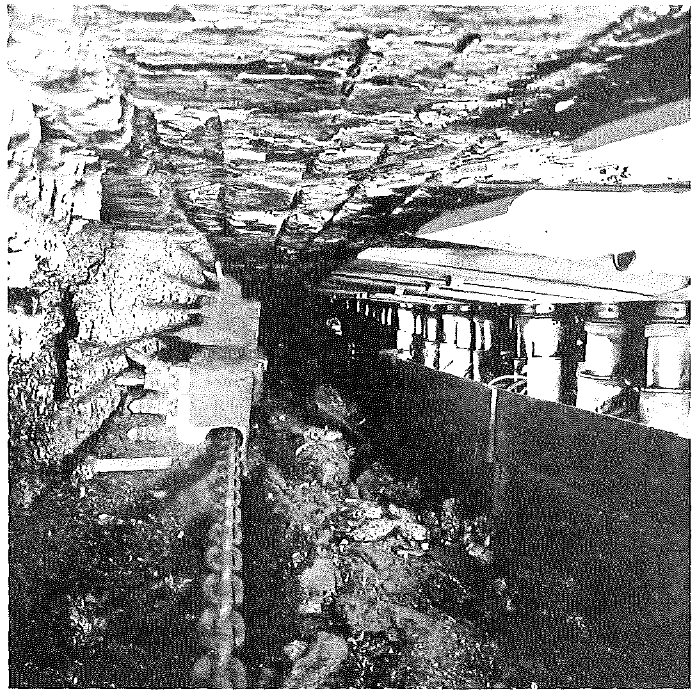


(Courtesy Sagem)

FIGURE 1. Contemporary longwall shearing machines





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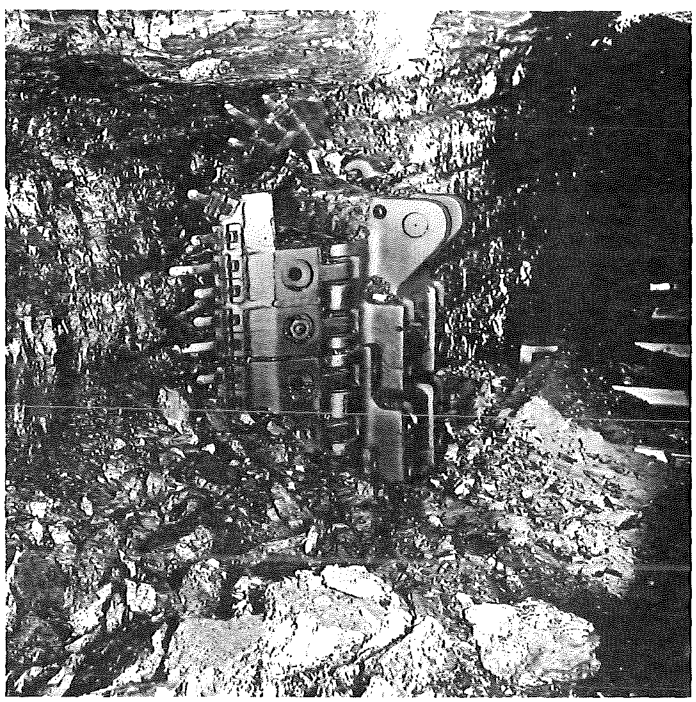


FIGURE 2. Longwall plows

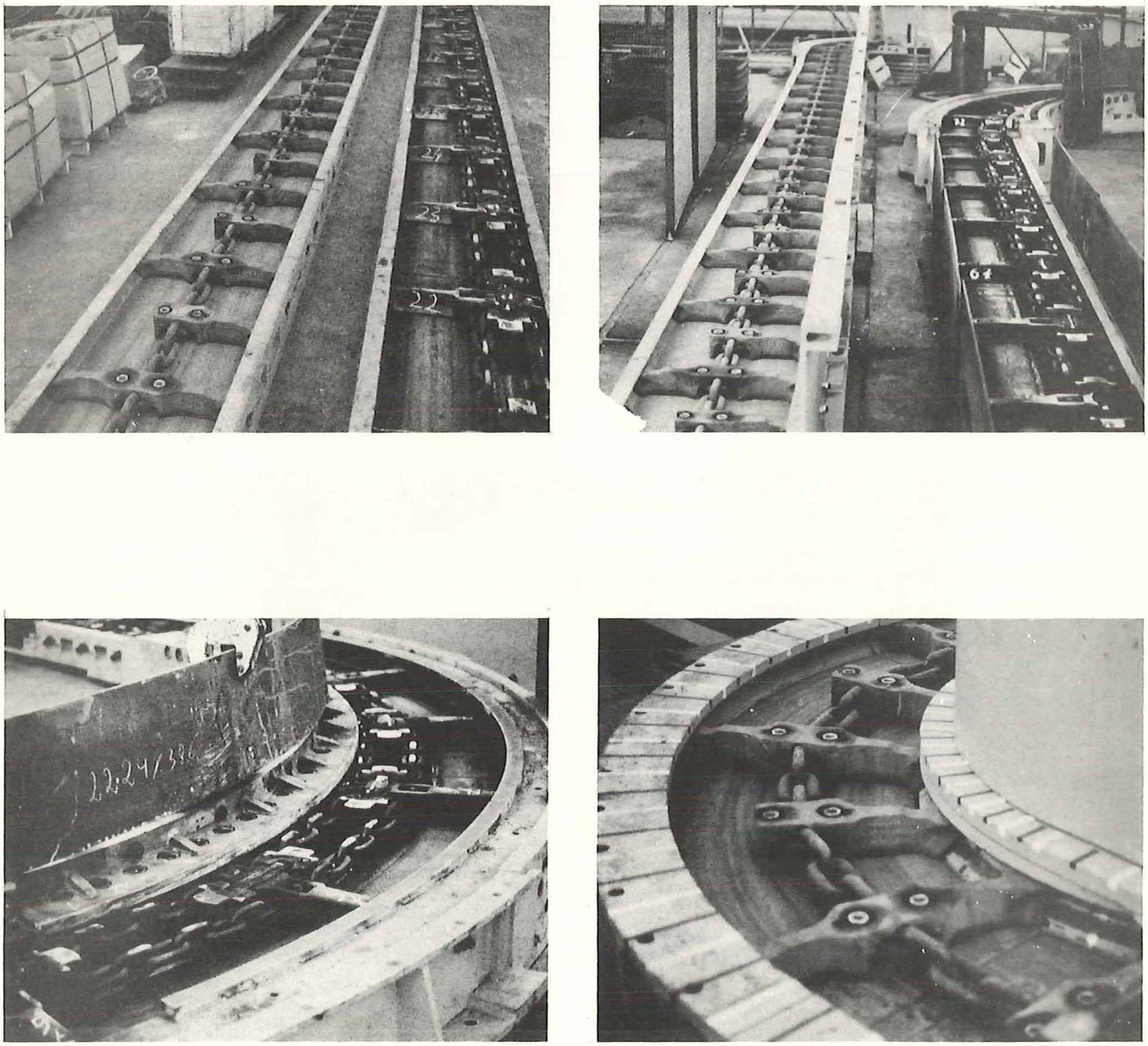
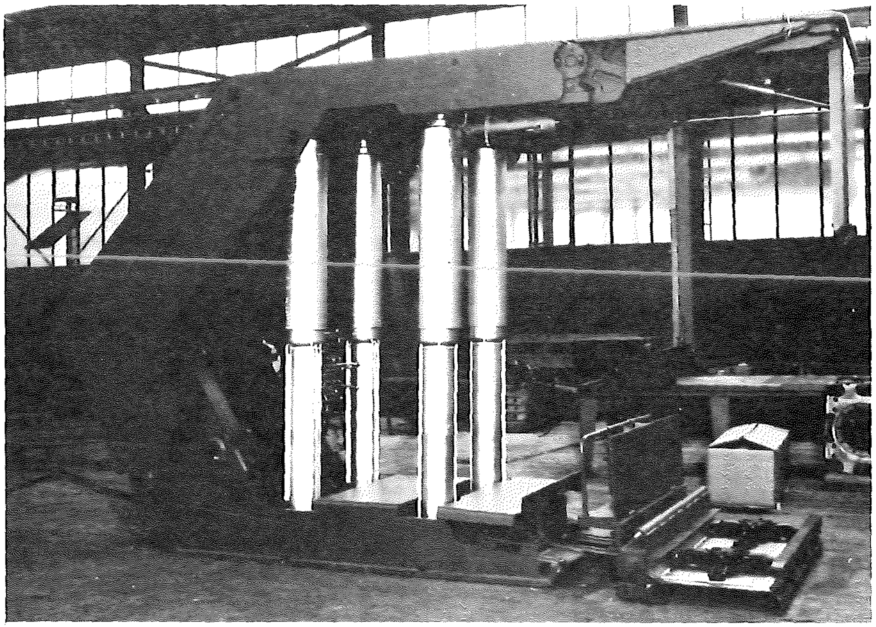
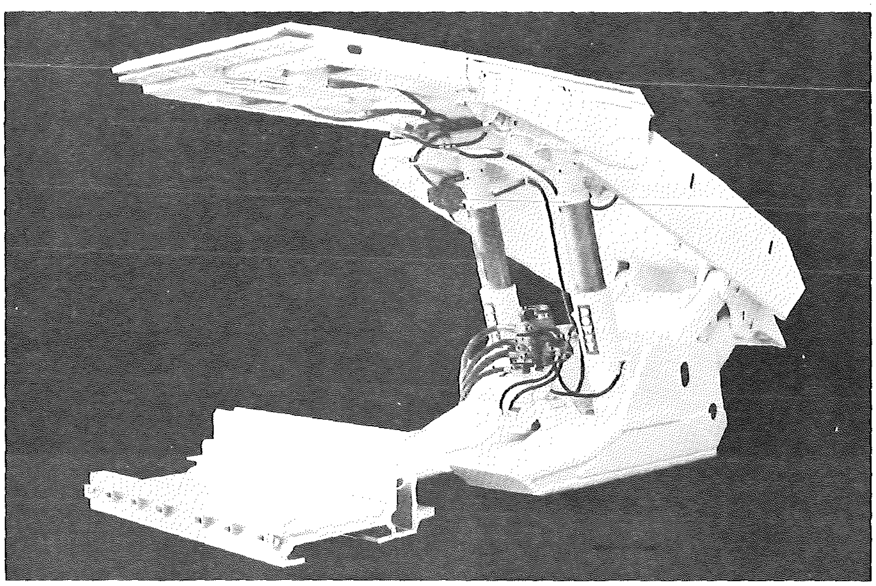


FIGURE 3. Experimental conveyor prototypes



(Courtesy Klöckner-Becorit/National Mine Service)



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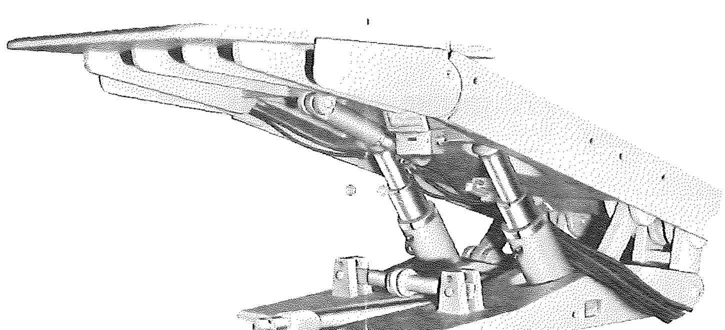


(Courtesy Westfalia Lünen)

FIGURE 4. Longwall shields

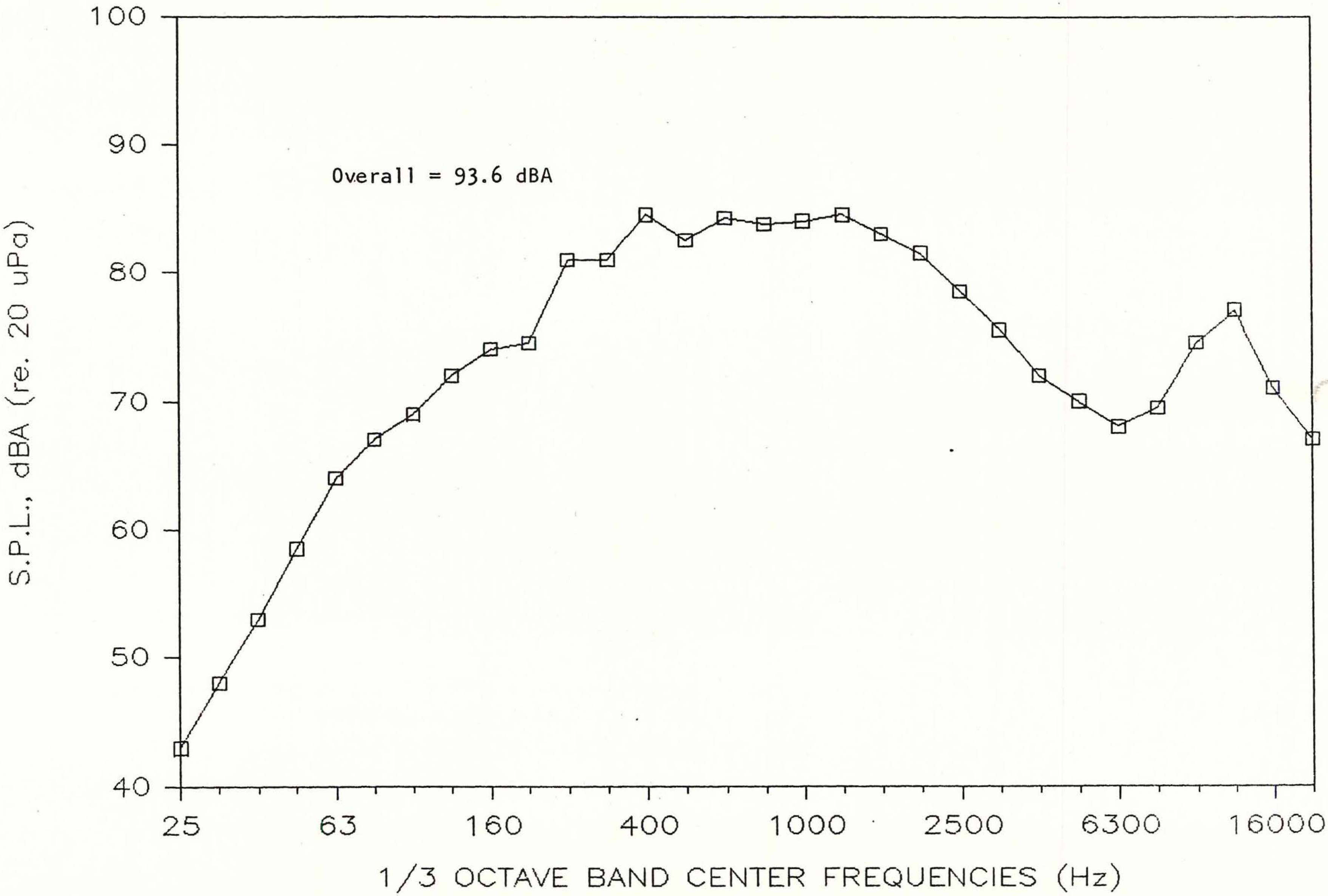


FIGURE 5. Noise at leading operator's position of double-drum shearer

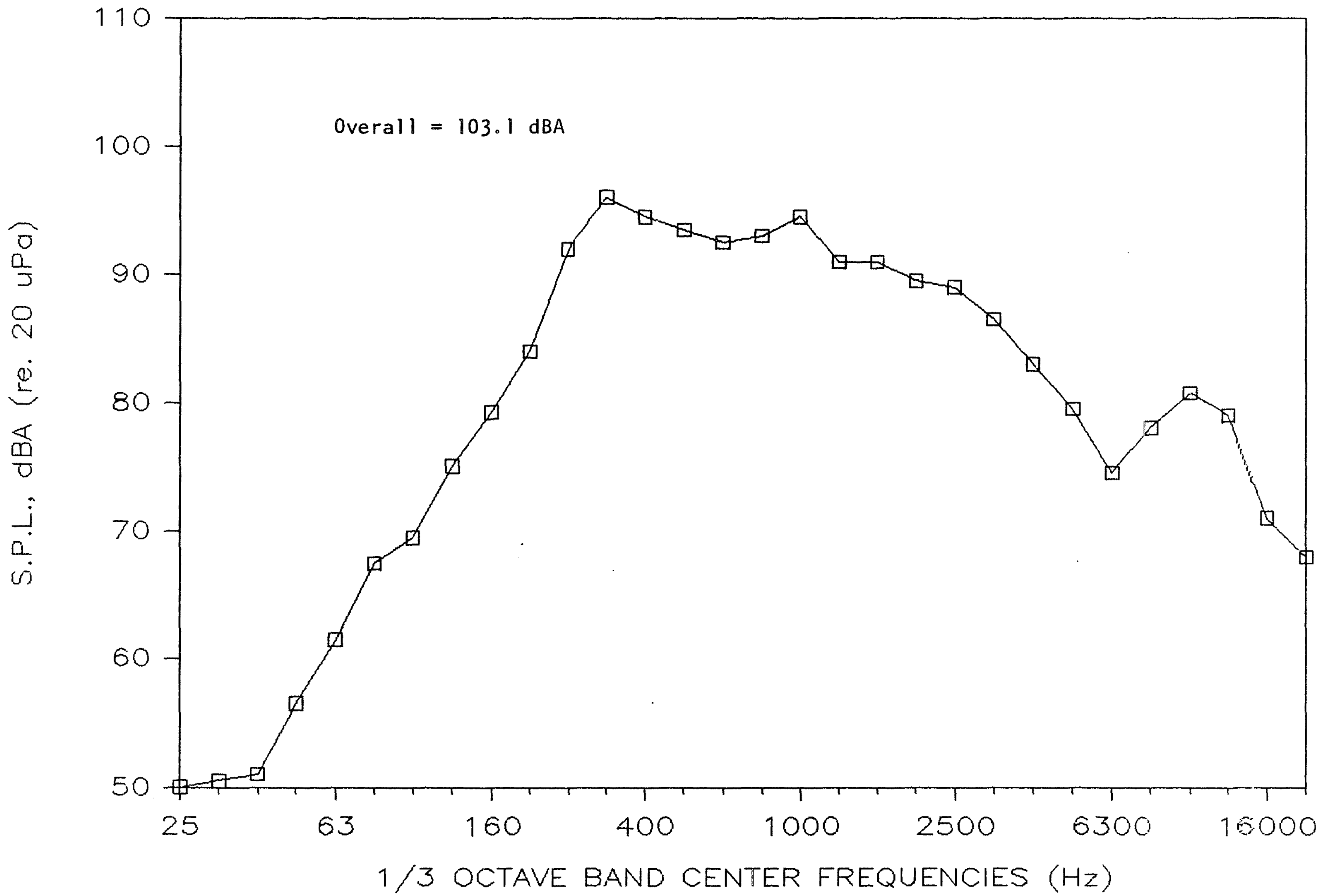


FIGURE 6. Noise at leading operator's position of double-drum shearer

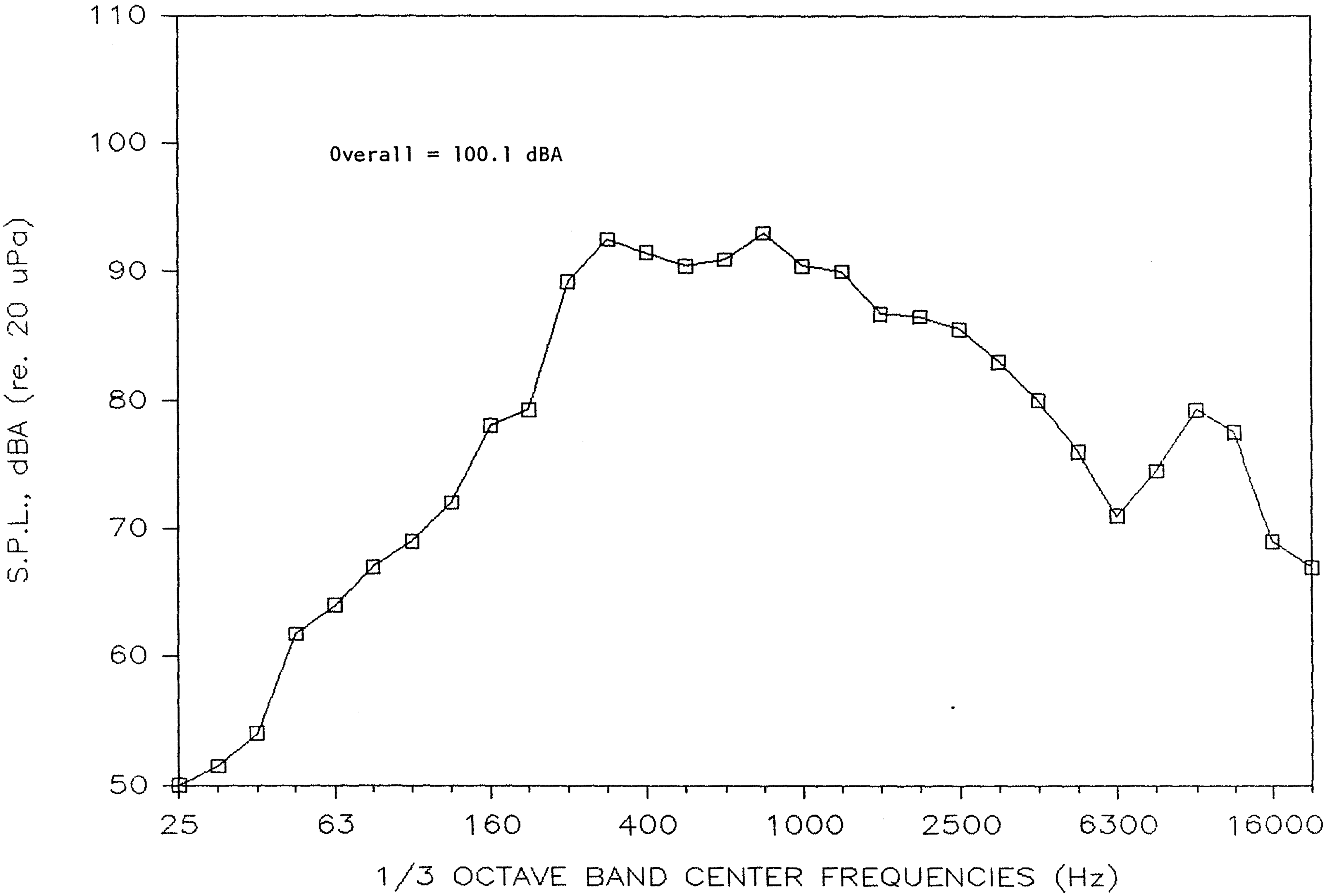


FIGURE 7. Noise at trailing operator's position of double-drum shearer

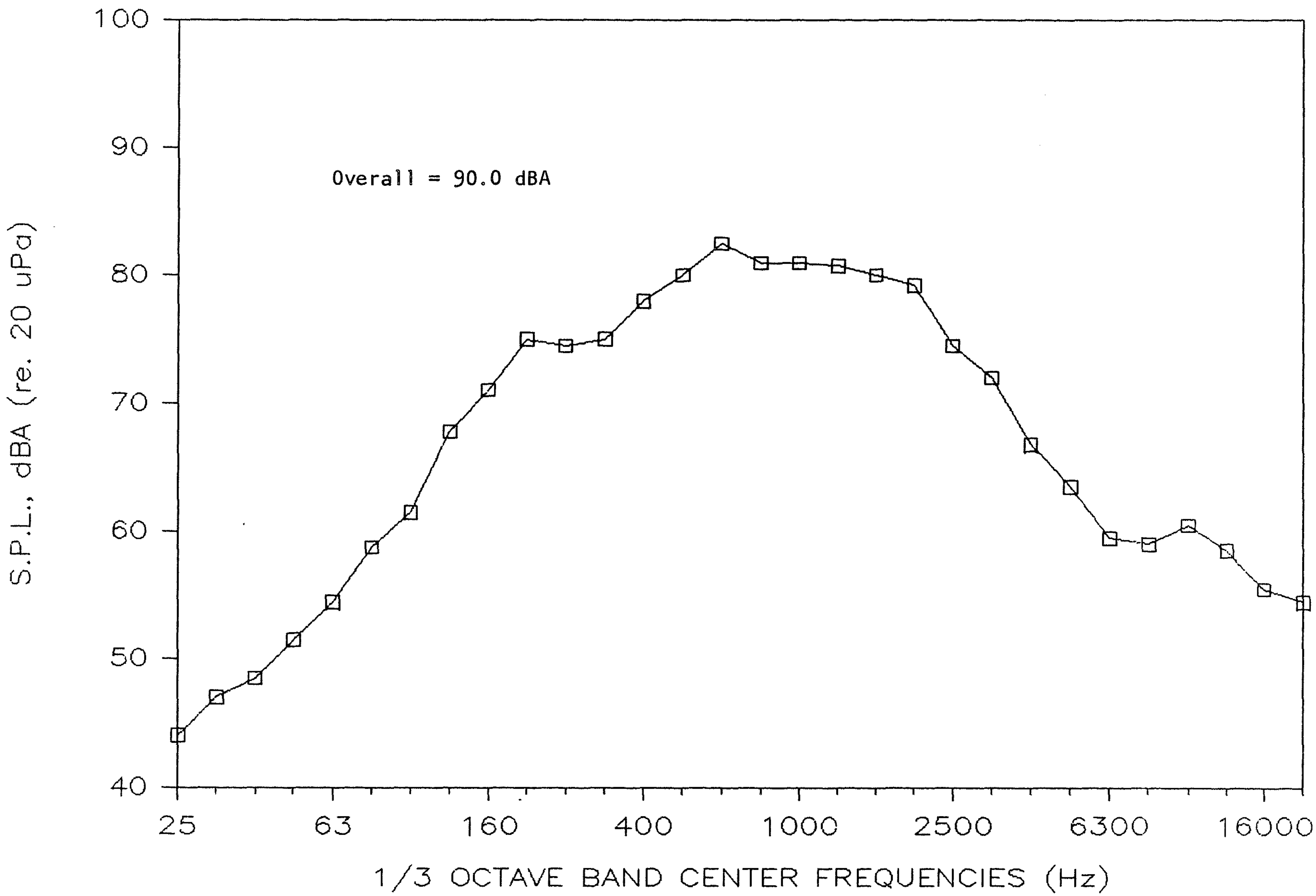


FIGURE 8. Noise of plow pass-by

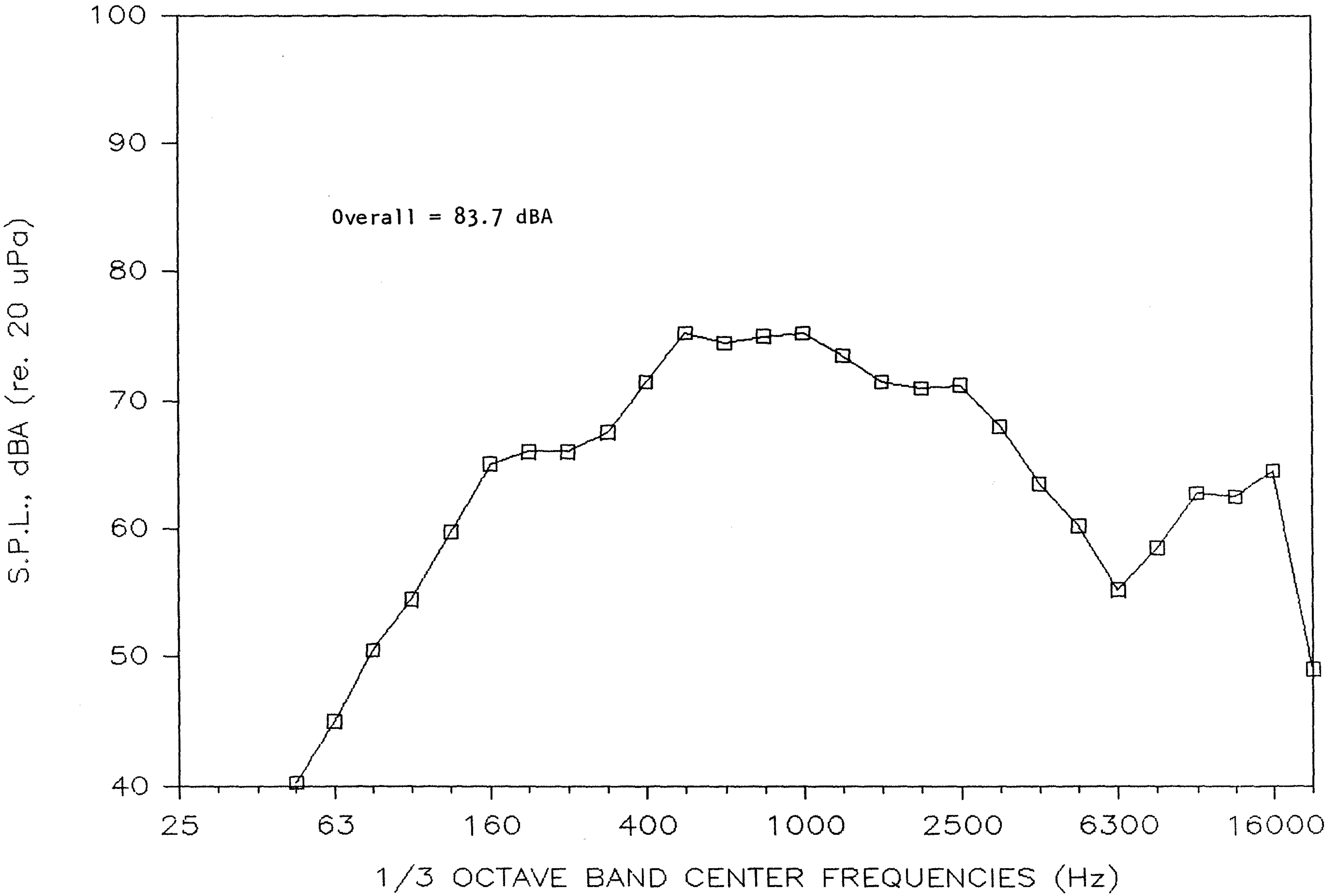


FIGURE 9. Noise from armored face conveyor partially loaded, shearer off

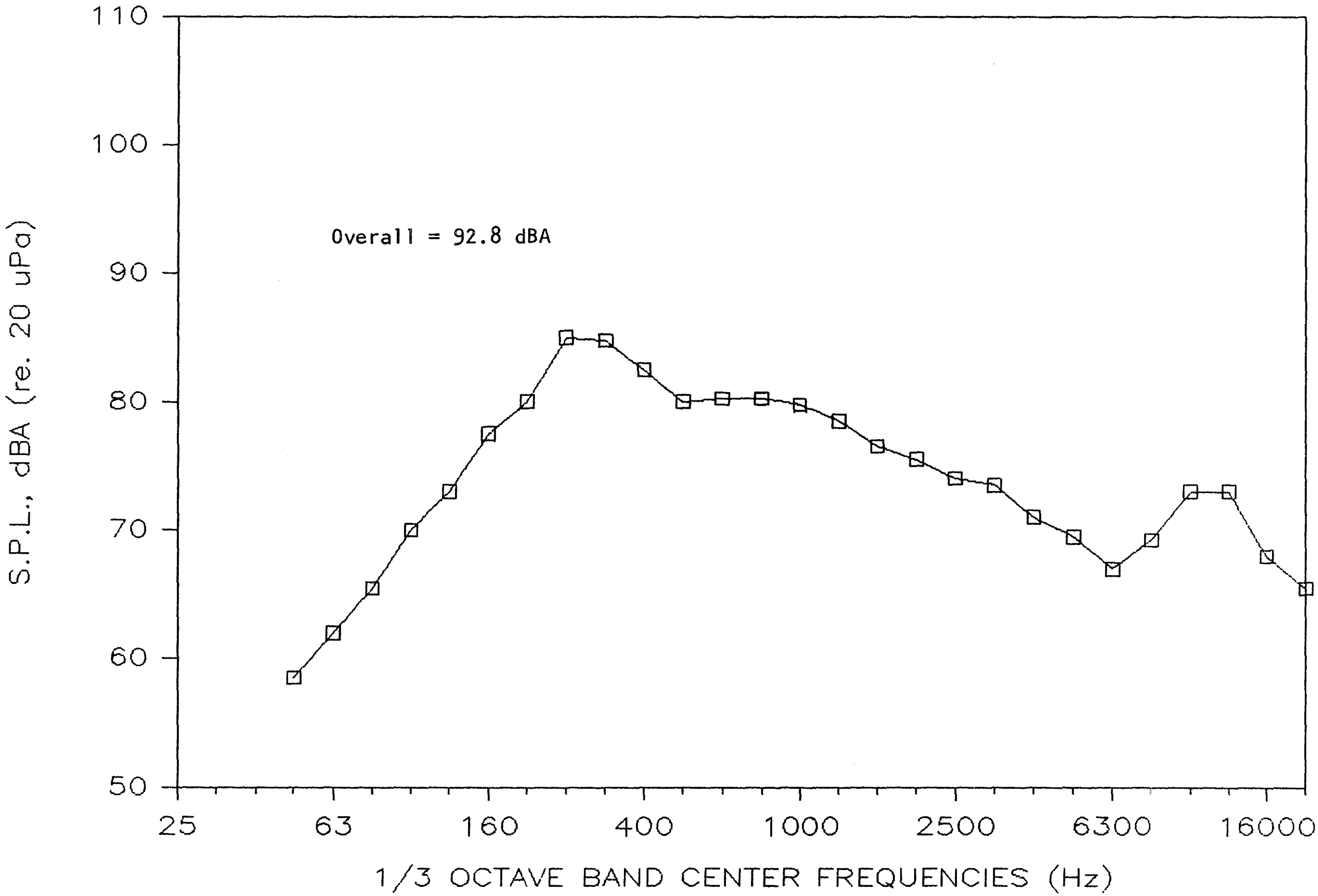


FIGURE 10. Noise from stage loader

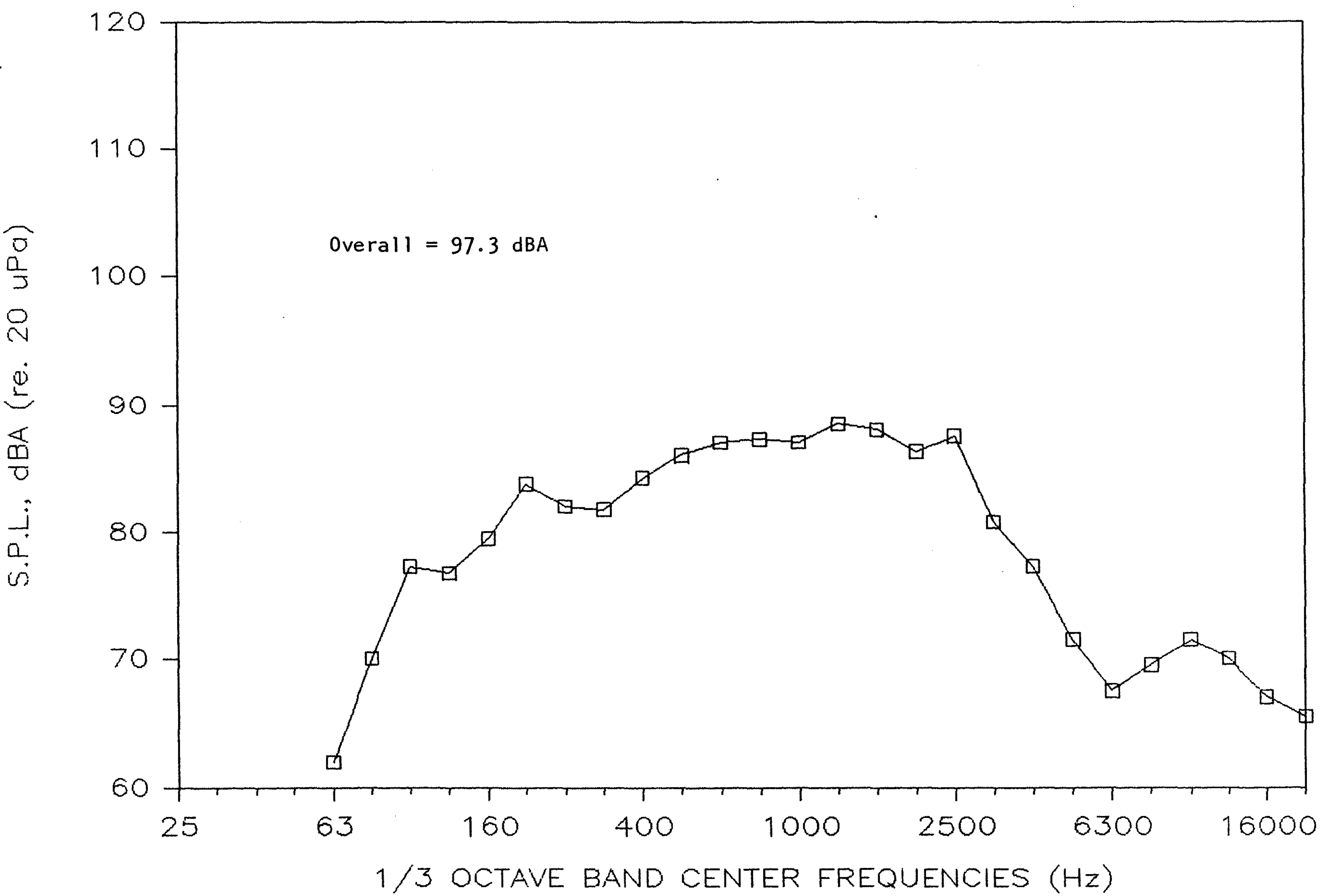


FIGURE 11. Noise from stage loader, belt transfer point

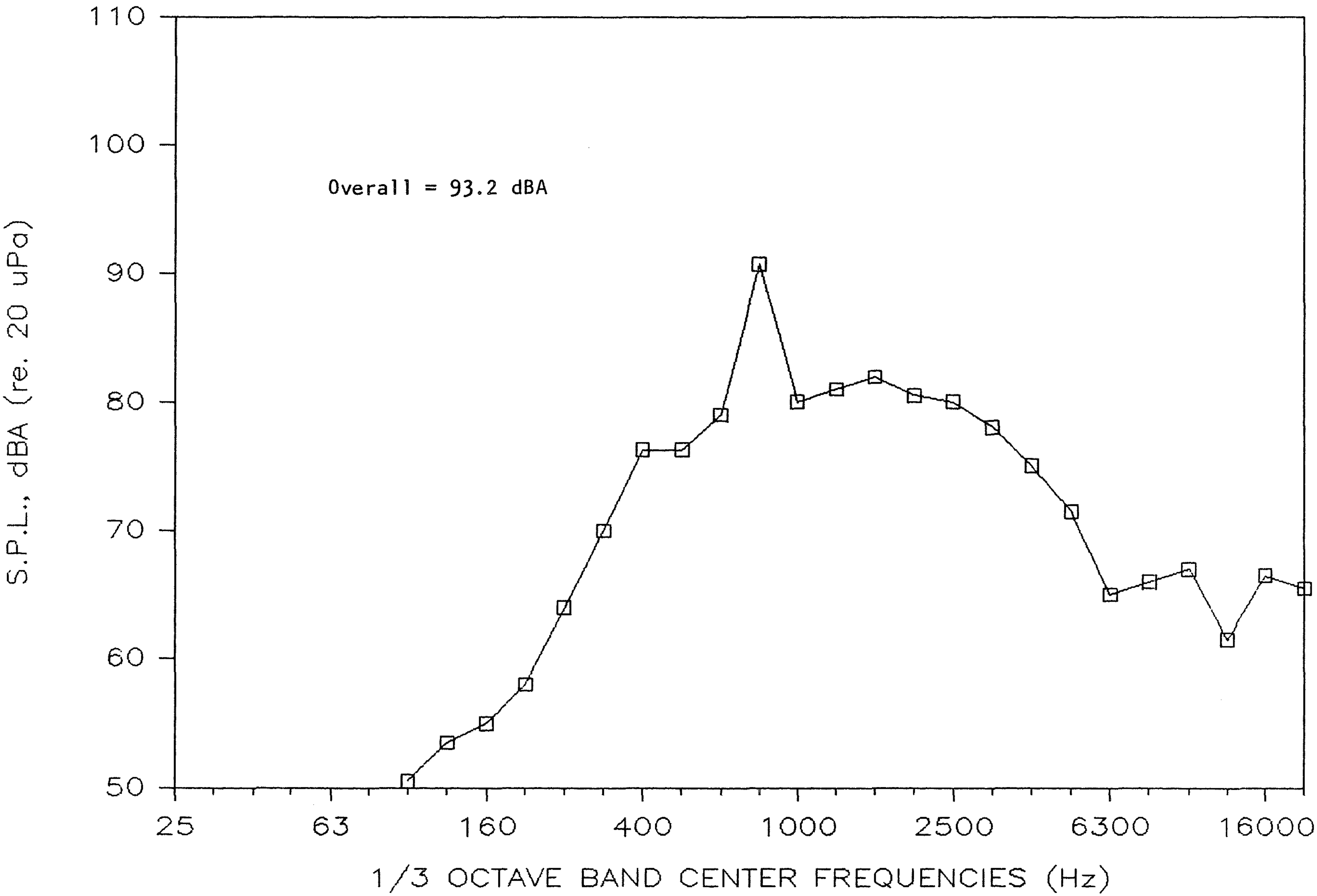


FIGURE 12. Noise from stage loader

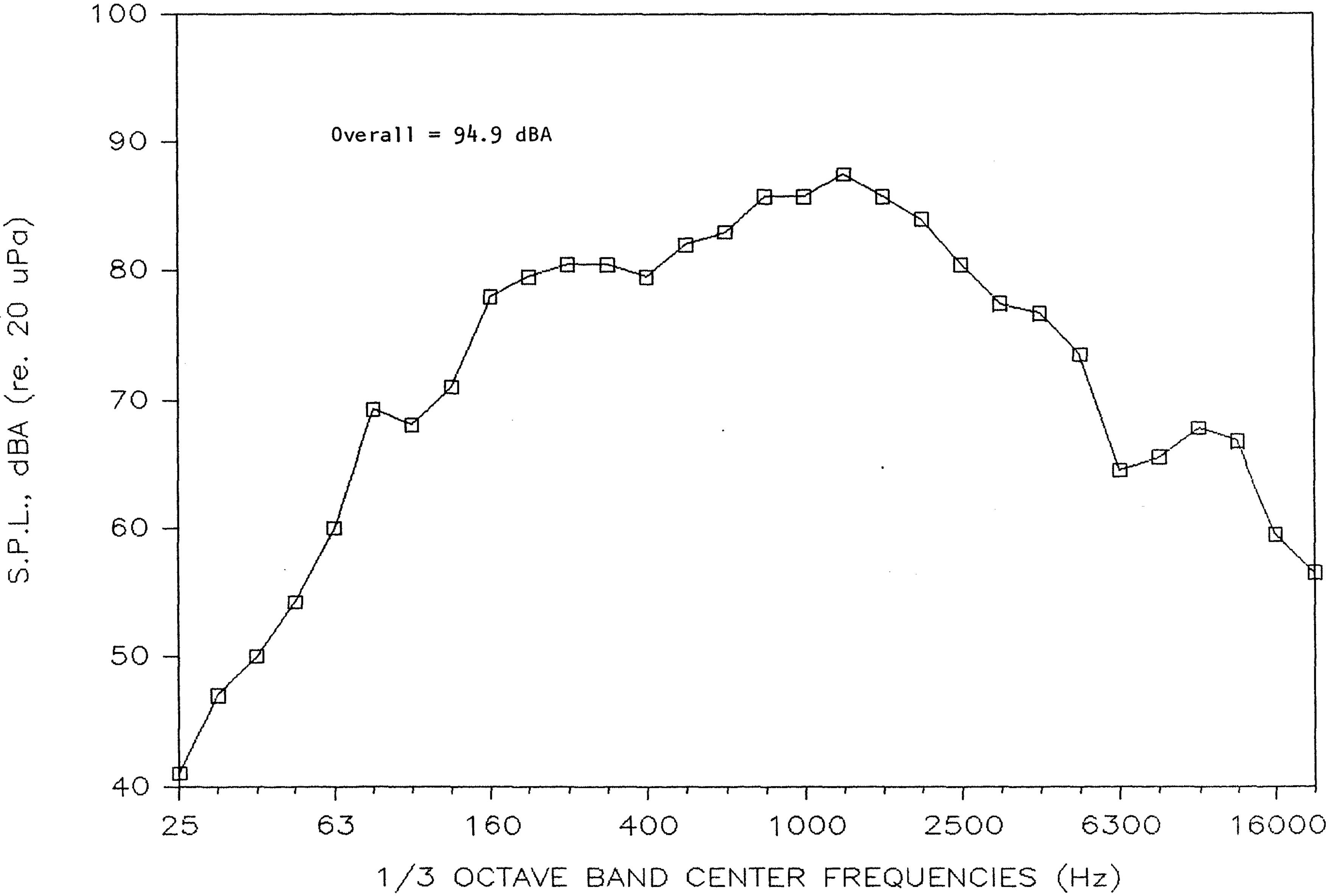


FIGURE 13. Noise from stage loader

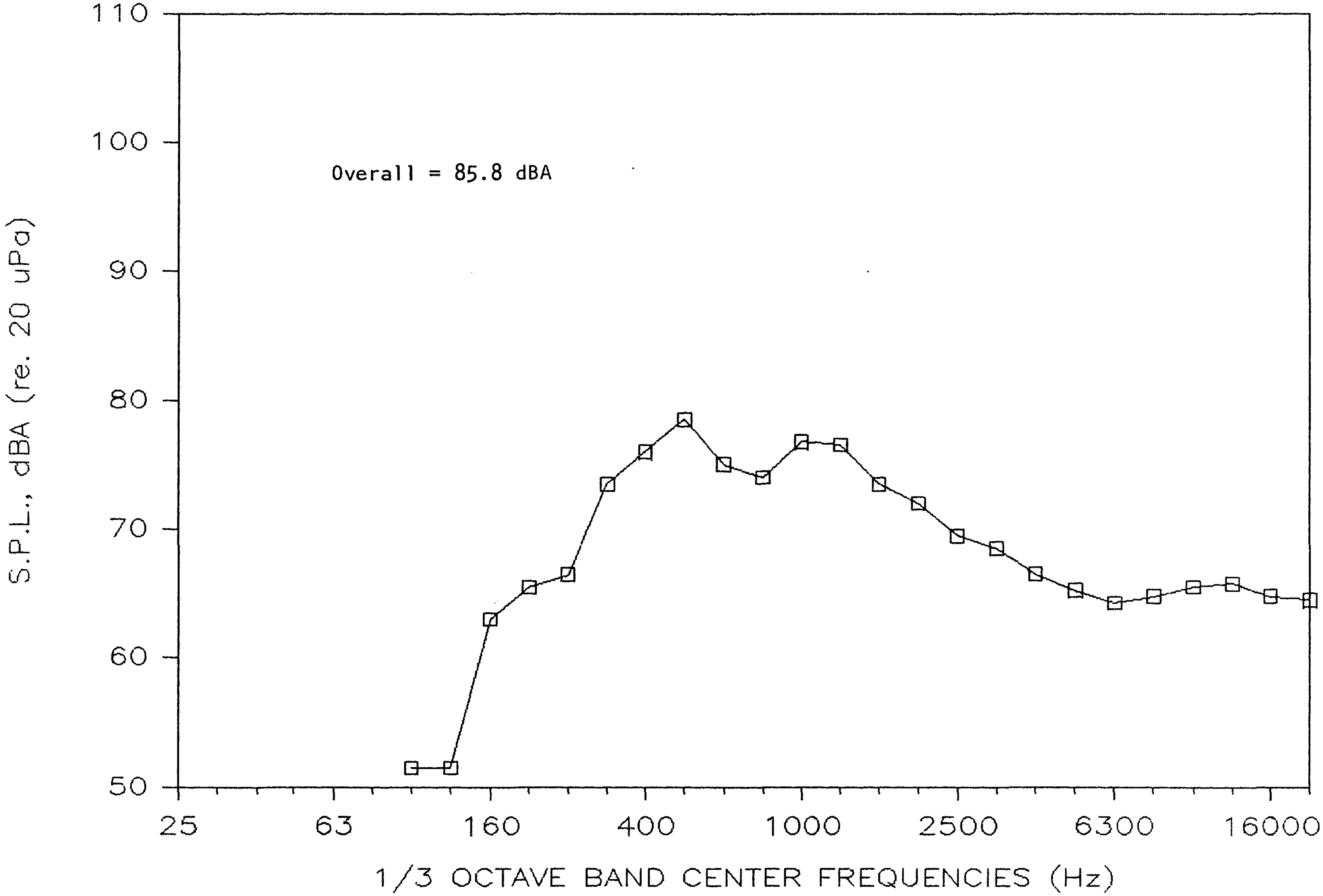


FIGURE 14. Noise from belt conveyor

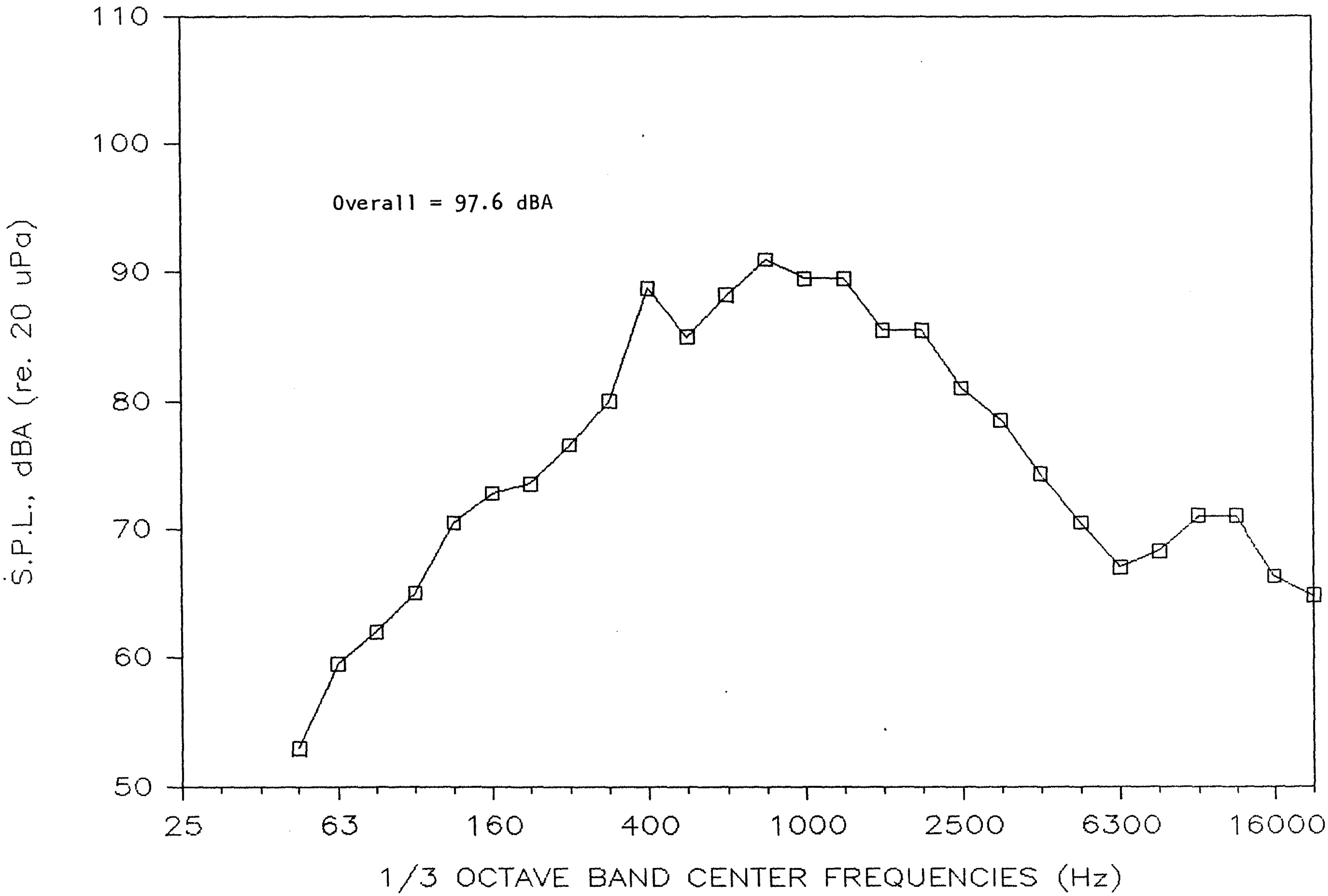


FIGURE 15. Noise at headgate operator's position

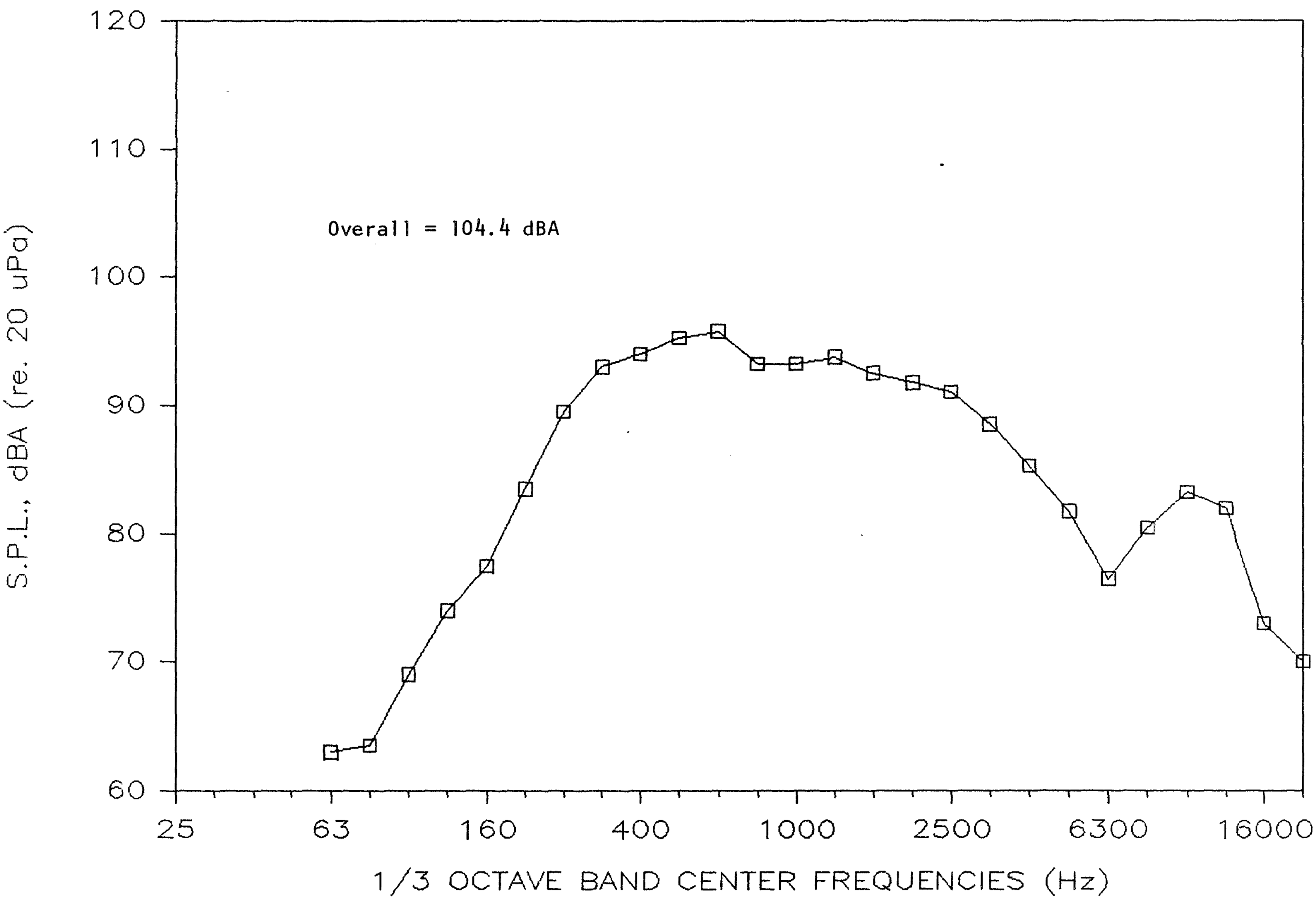


FIGURE 16. Noise at headgate operator's position

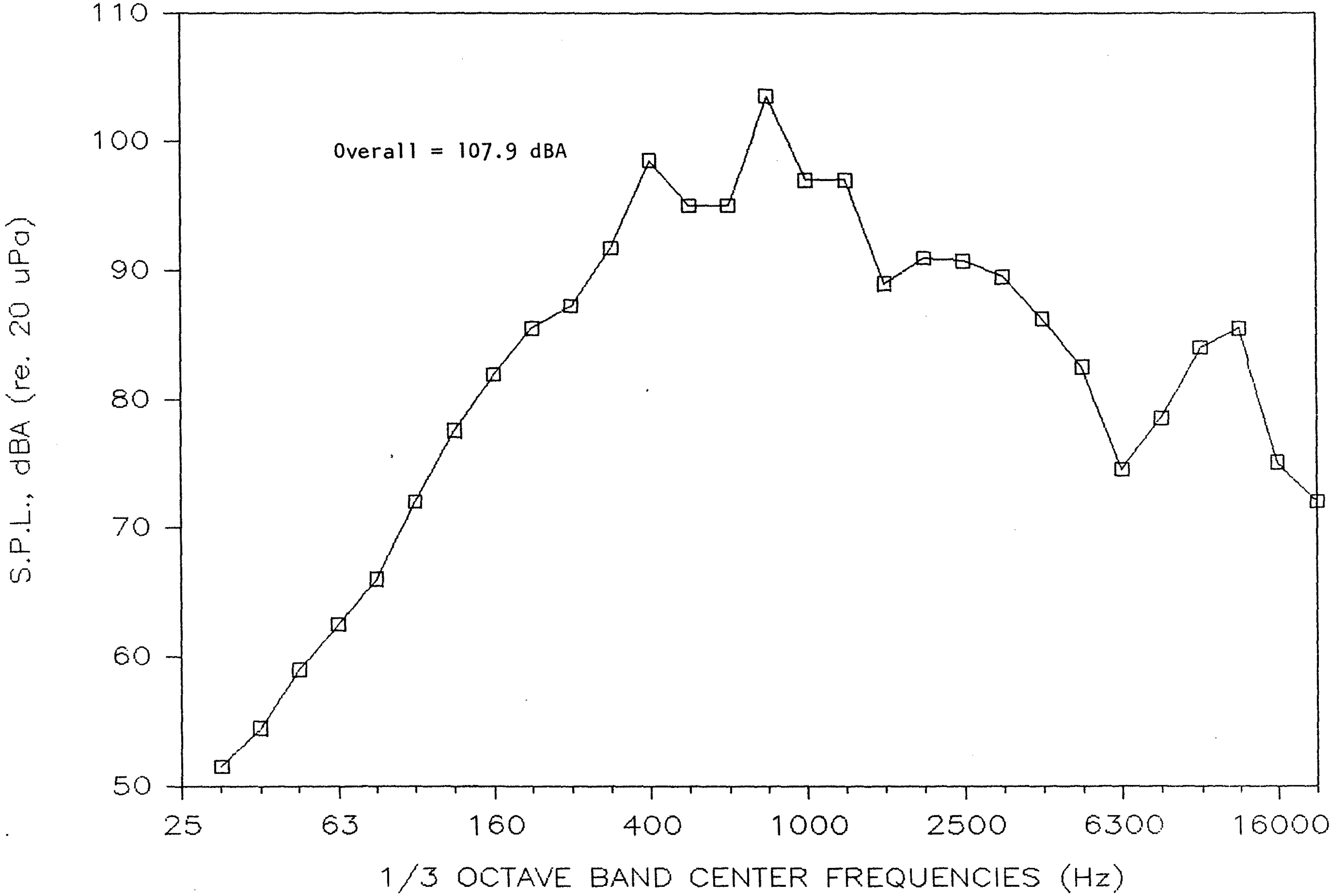


FIGURE 17. Noise from 75-hp hydraulic pump

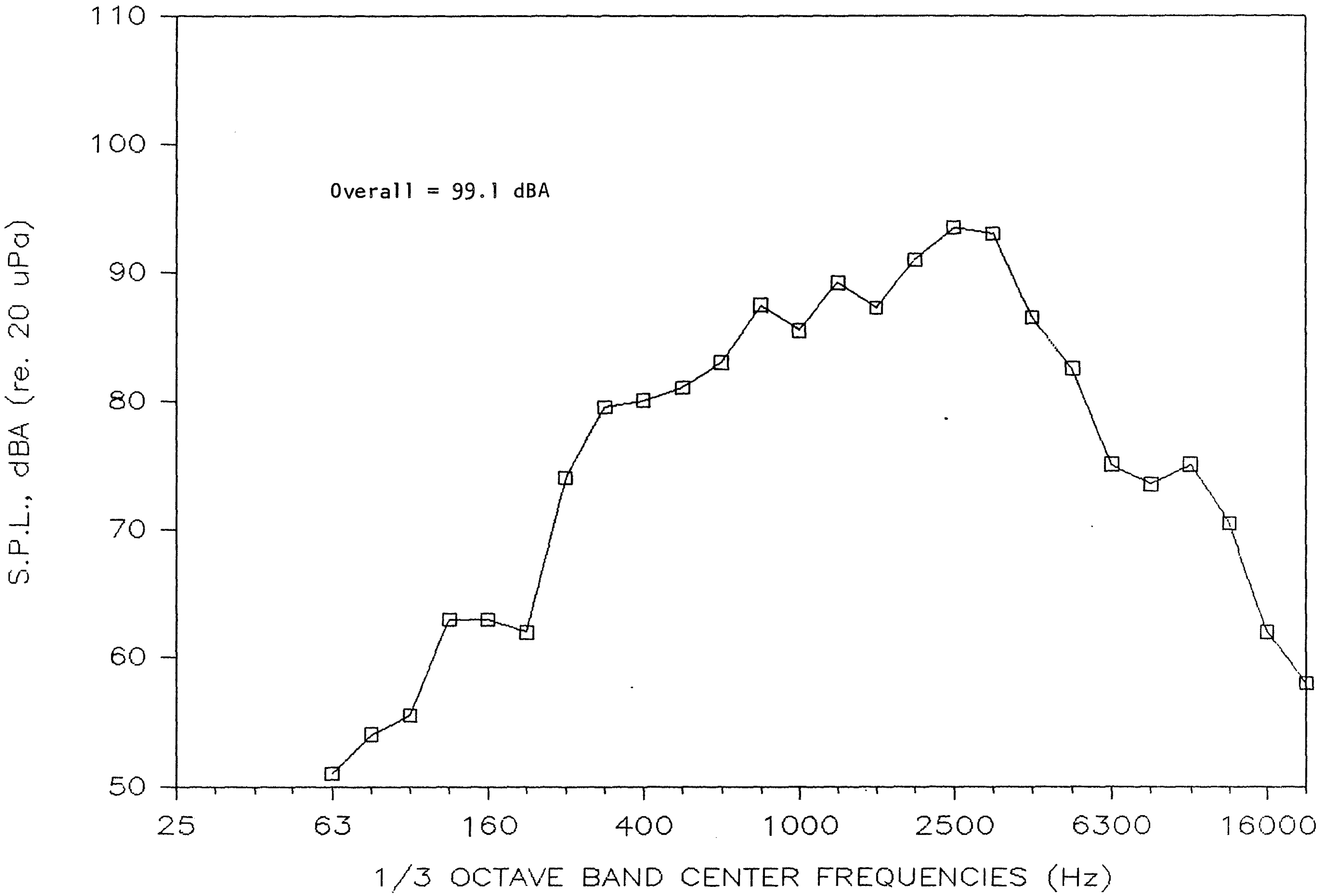


FIGURE 18. Noise from 75-hp hydraulic pump

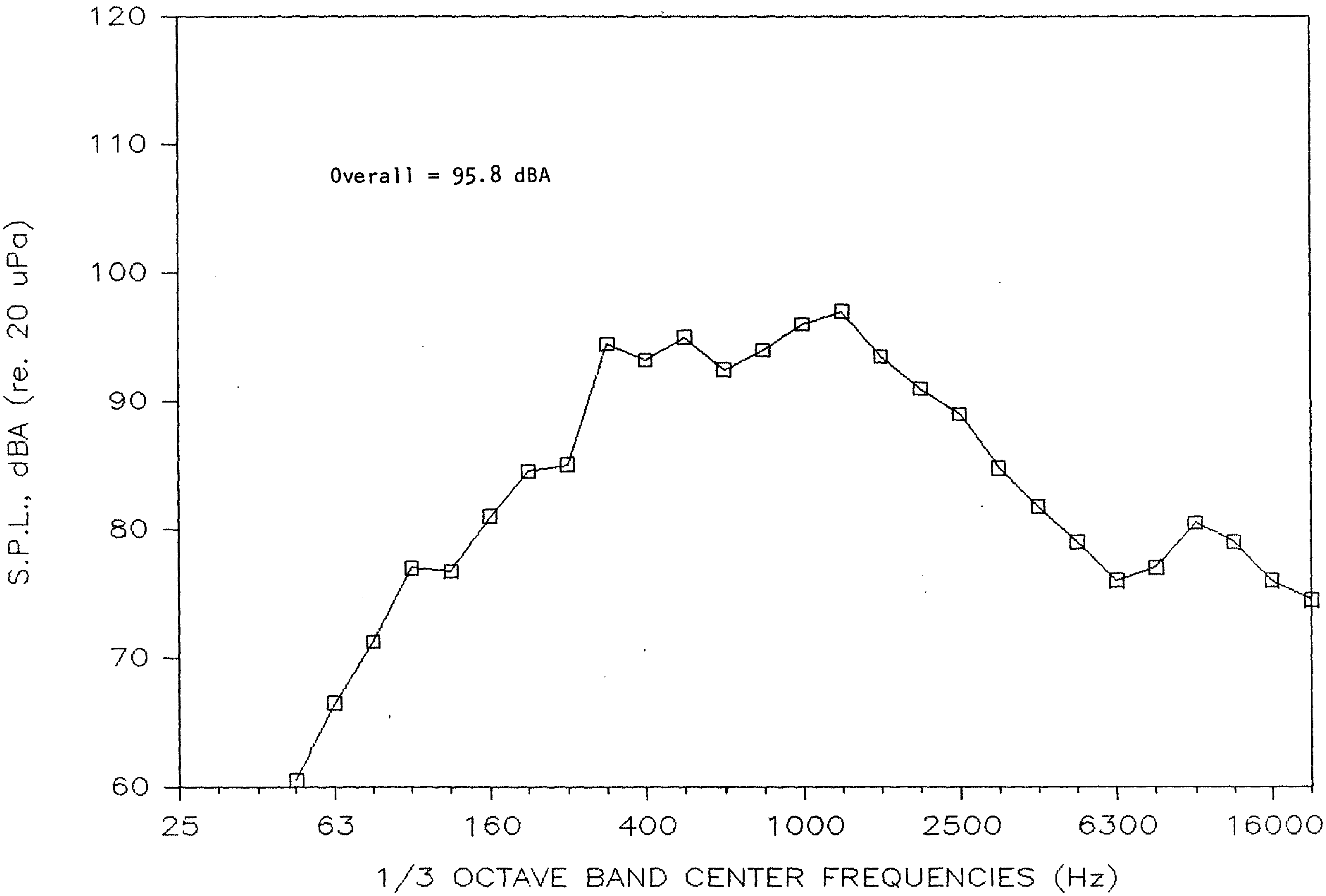


FIGURE 19. Noise from 125-hp hydraulic pump

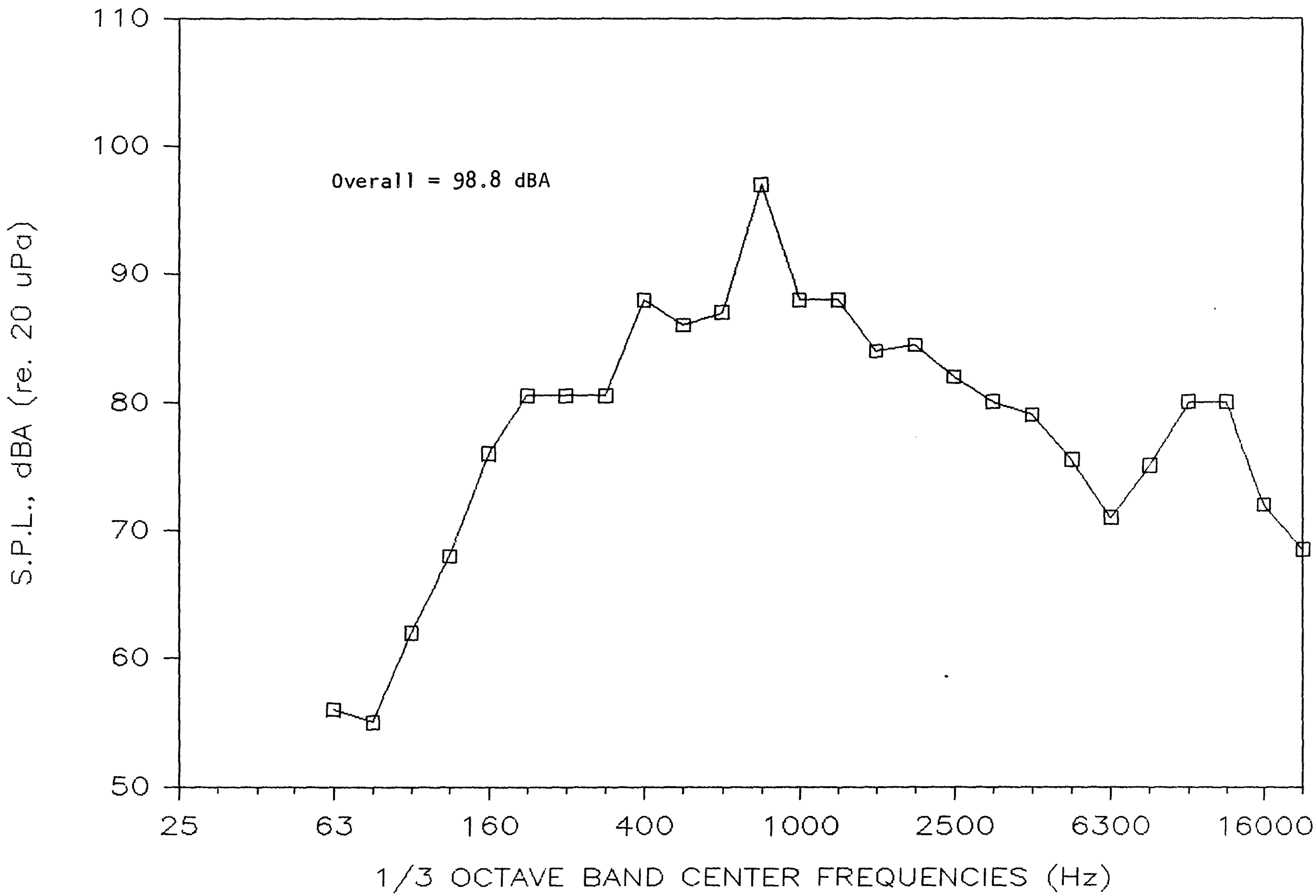


FIGURE 20. Noise from 125-hp hydraulic pump

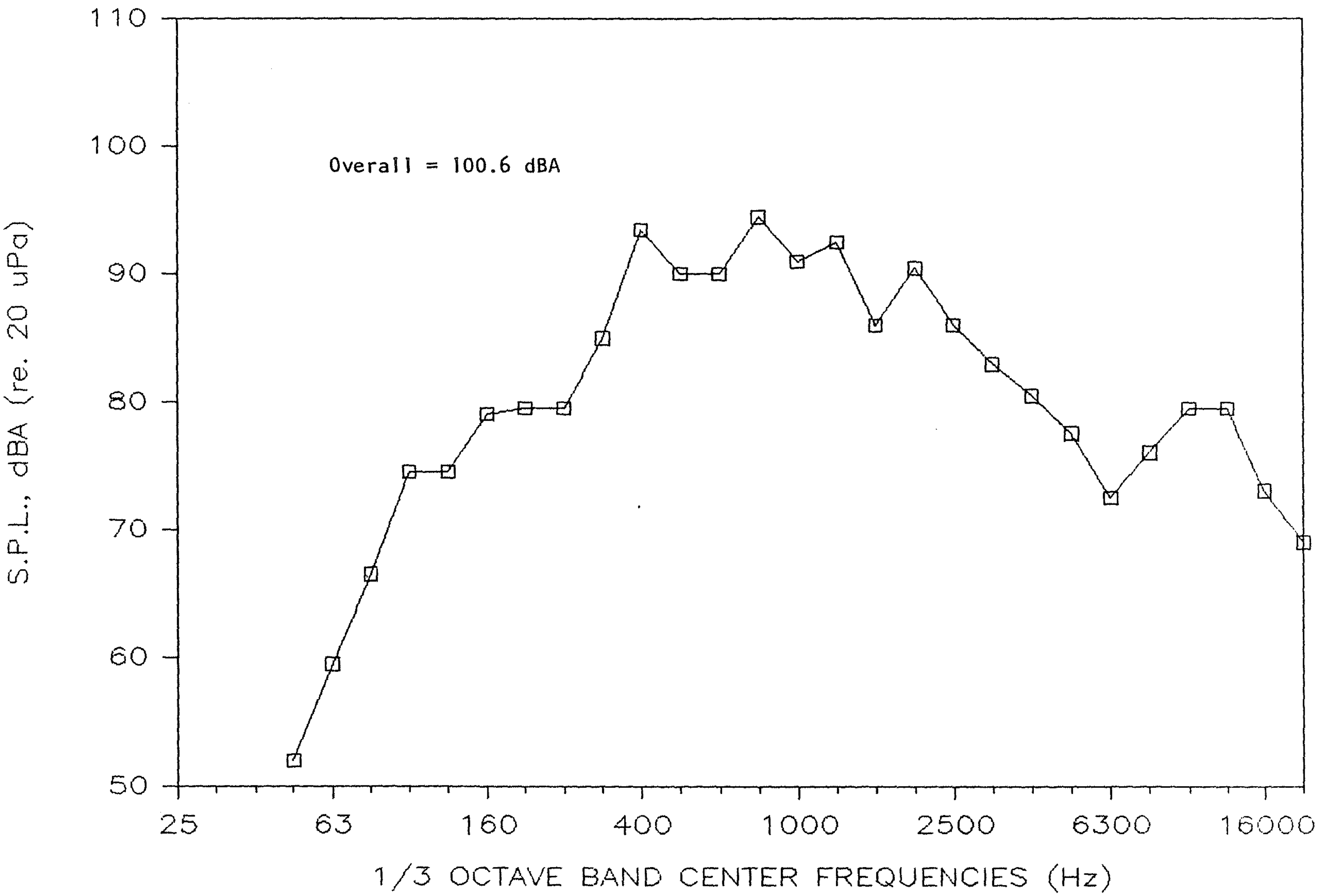


FIGURE 21. Noise from 125-hp hydraulic pump motor

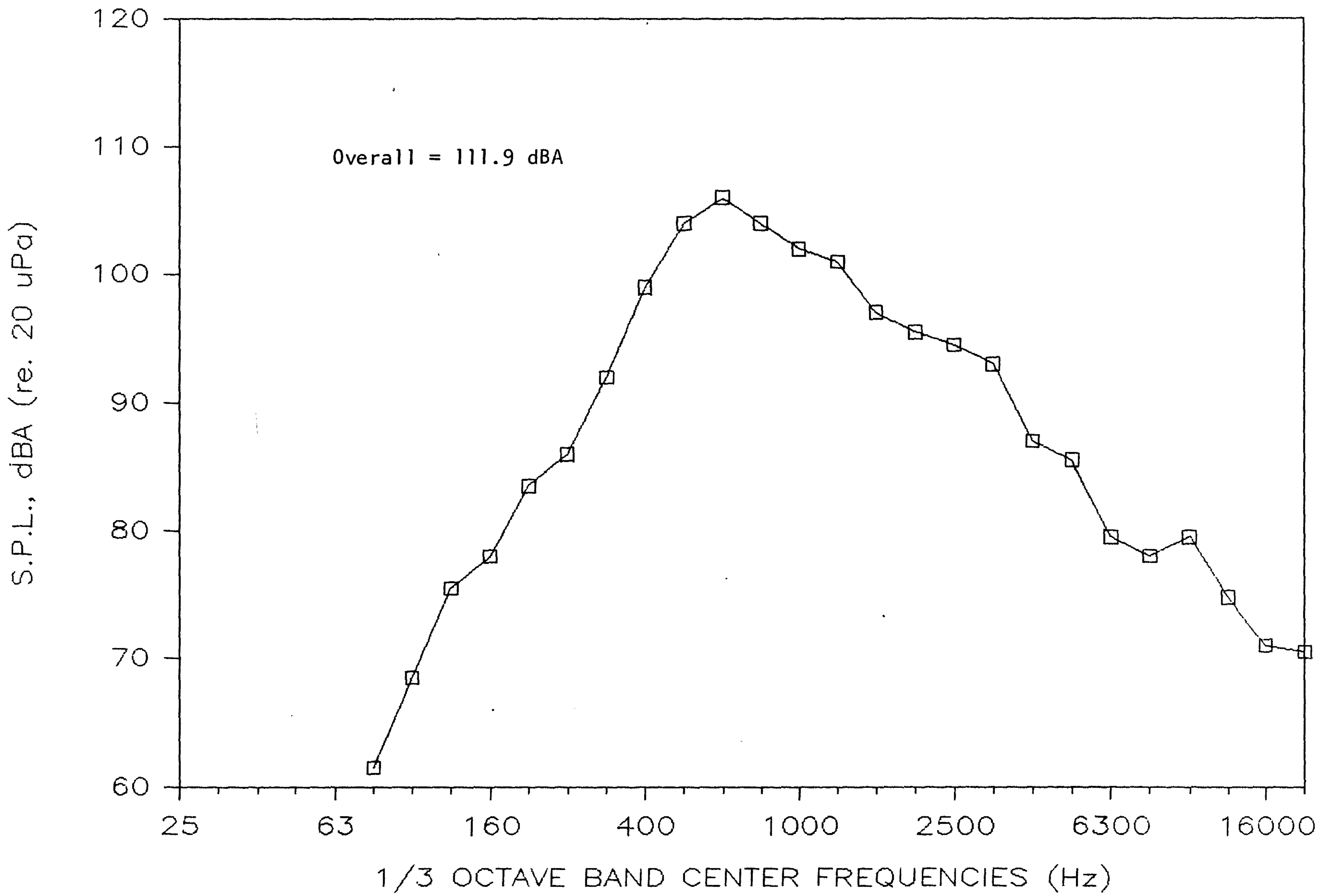


FIGURE 22. Acoustic measurements near the leading drum of a shearing machine during simulated coal cutting operations

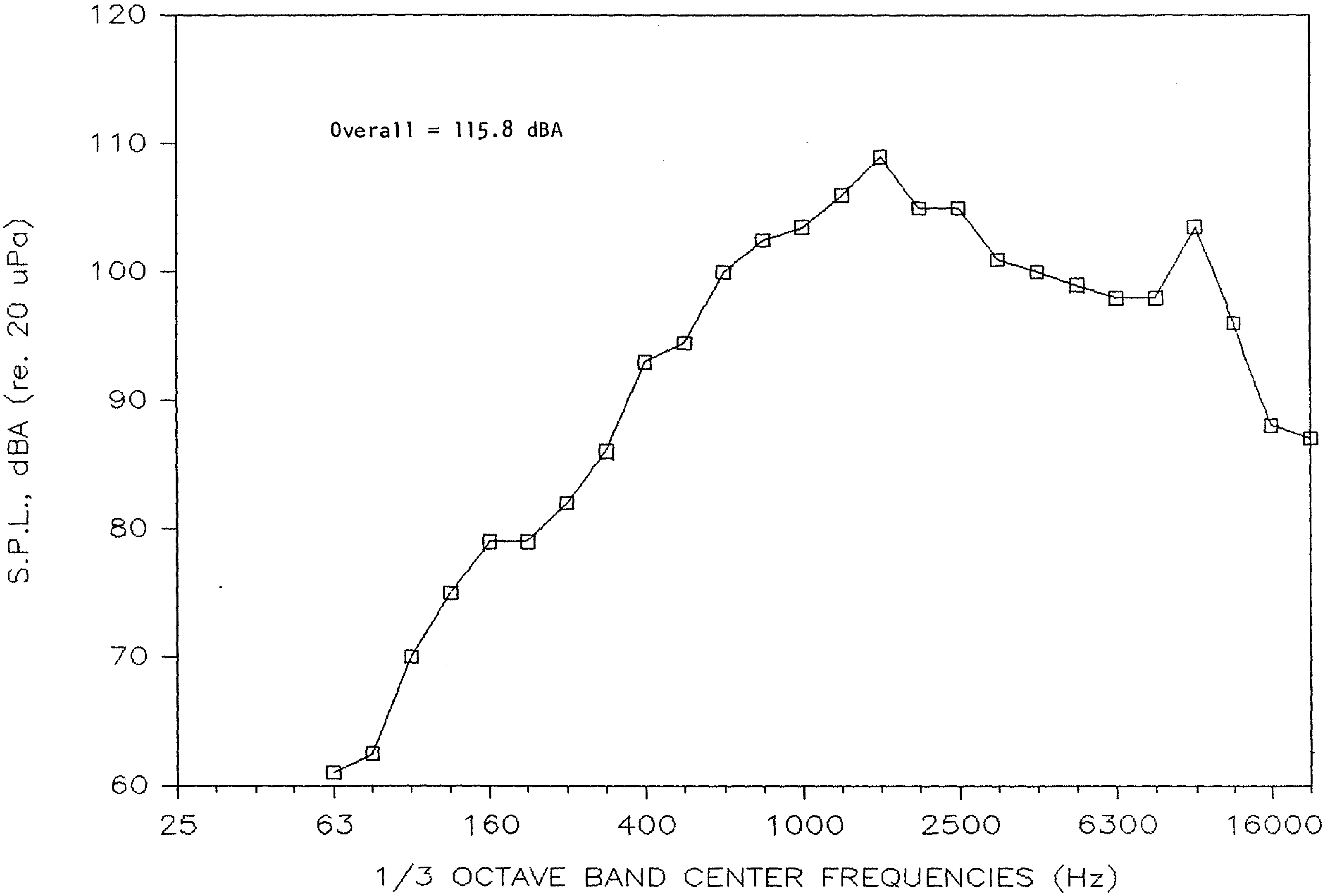


FIGURE 23. Acoustic measurements near the hydraulic pump of a shearing machine during simulated coal cutting operations

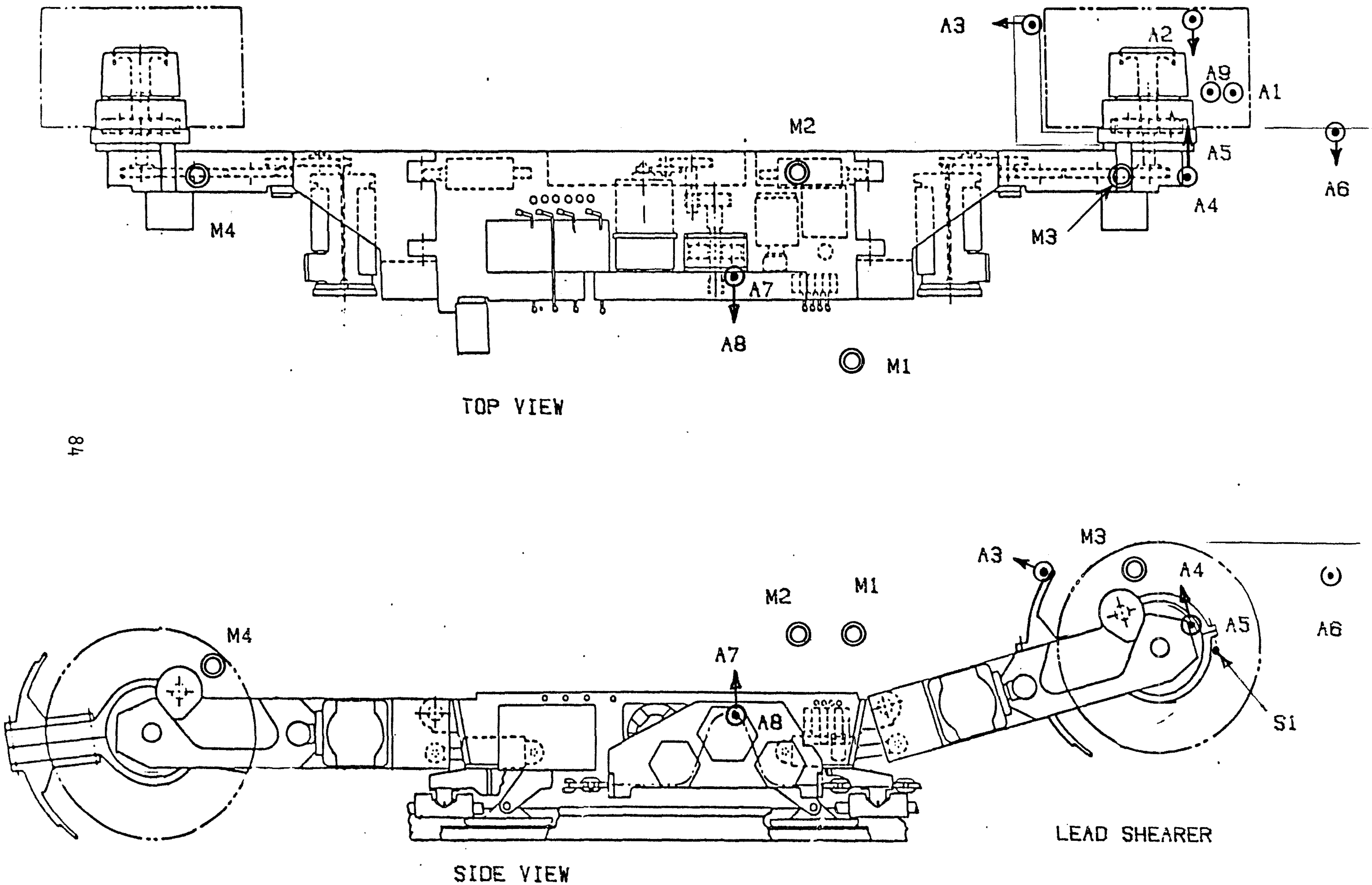


FIGURE 24. Instrumentation locations for tests using USBM longwall mockup

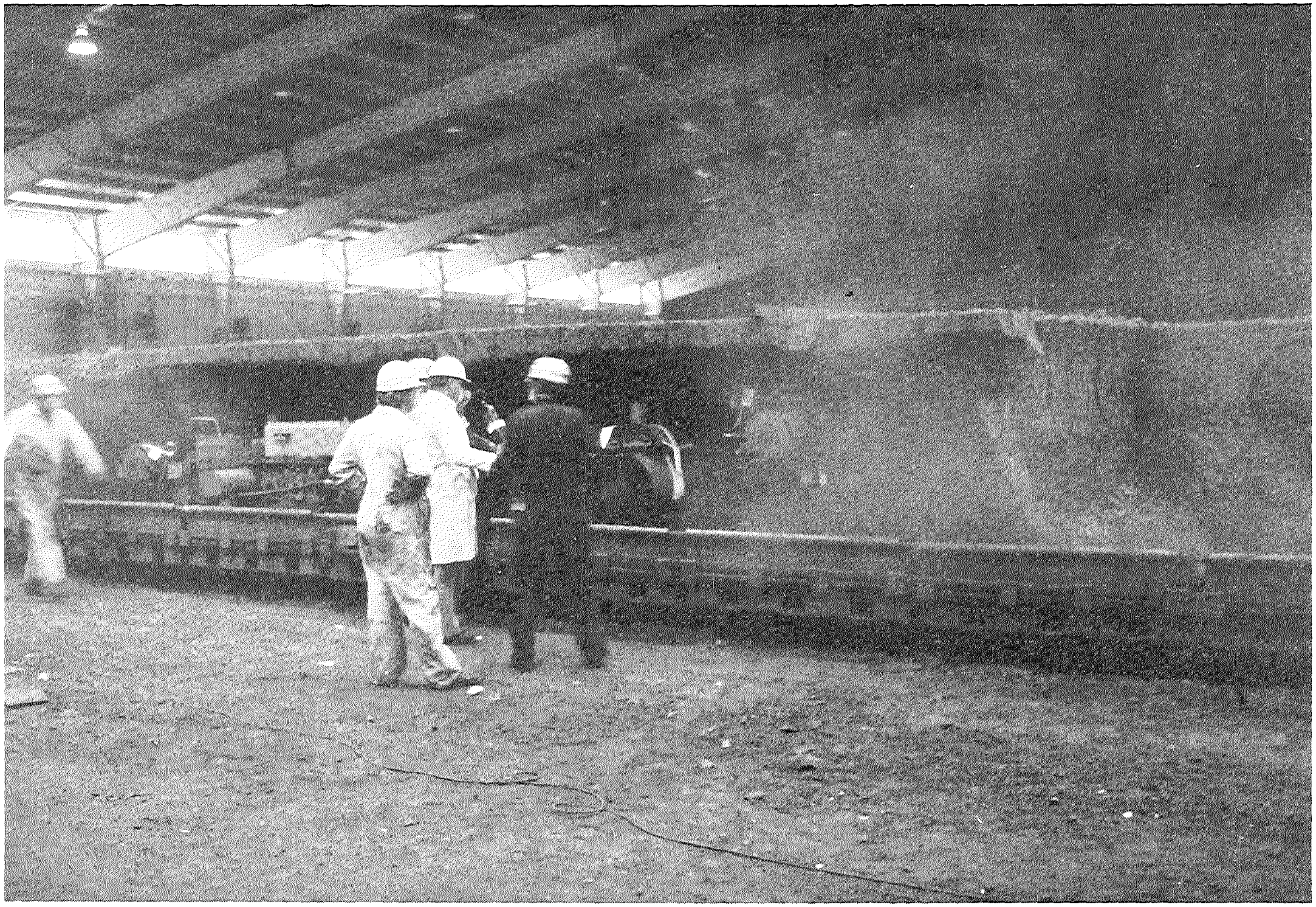


FIGURE 25. USBM longwall machine during cutting operations

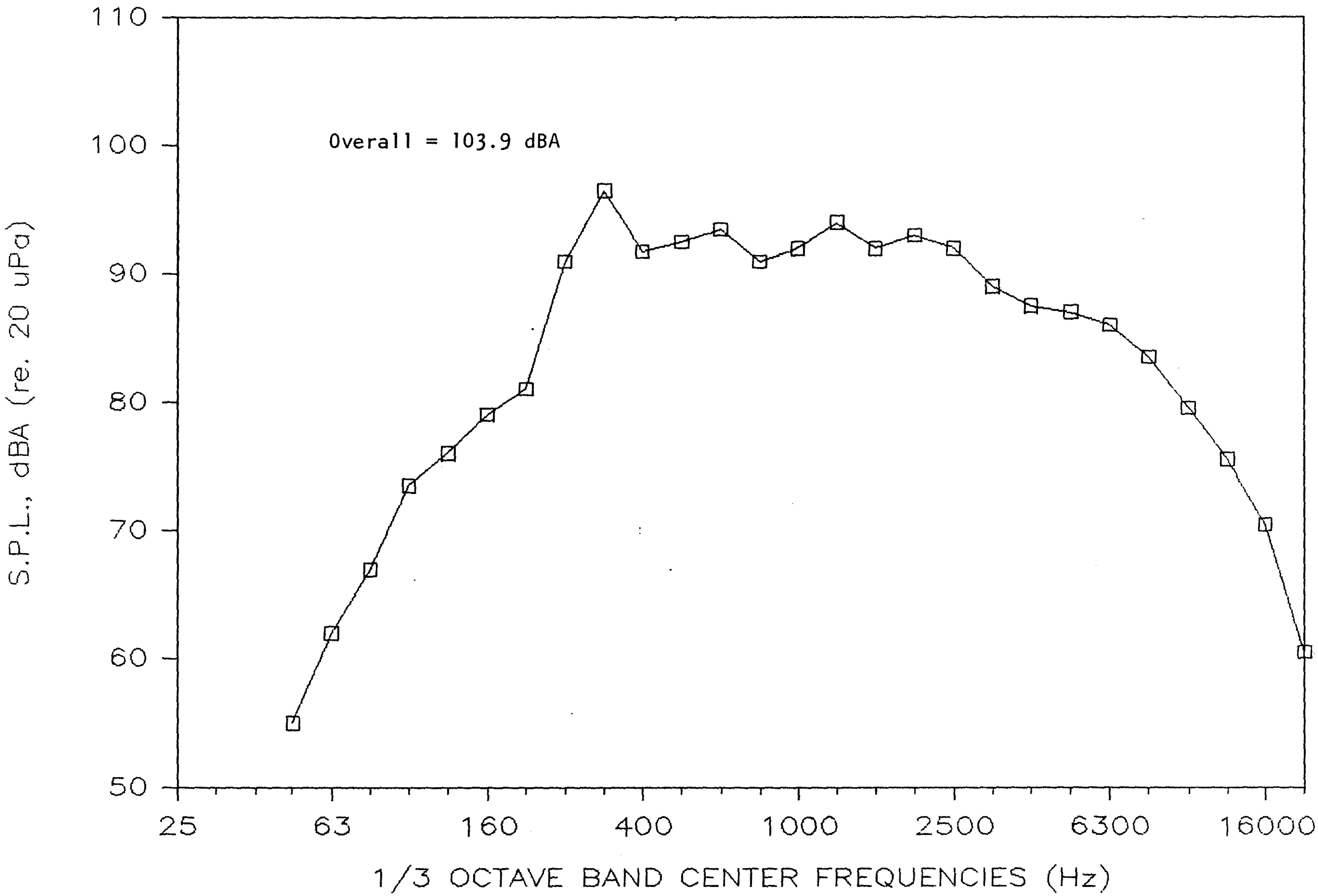


FIGURE 26. Noise from USBM longwall test

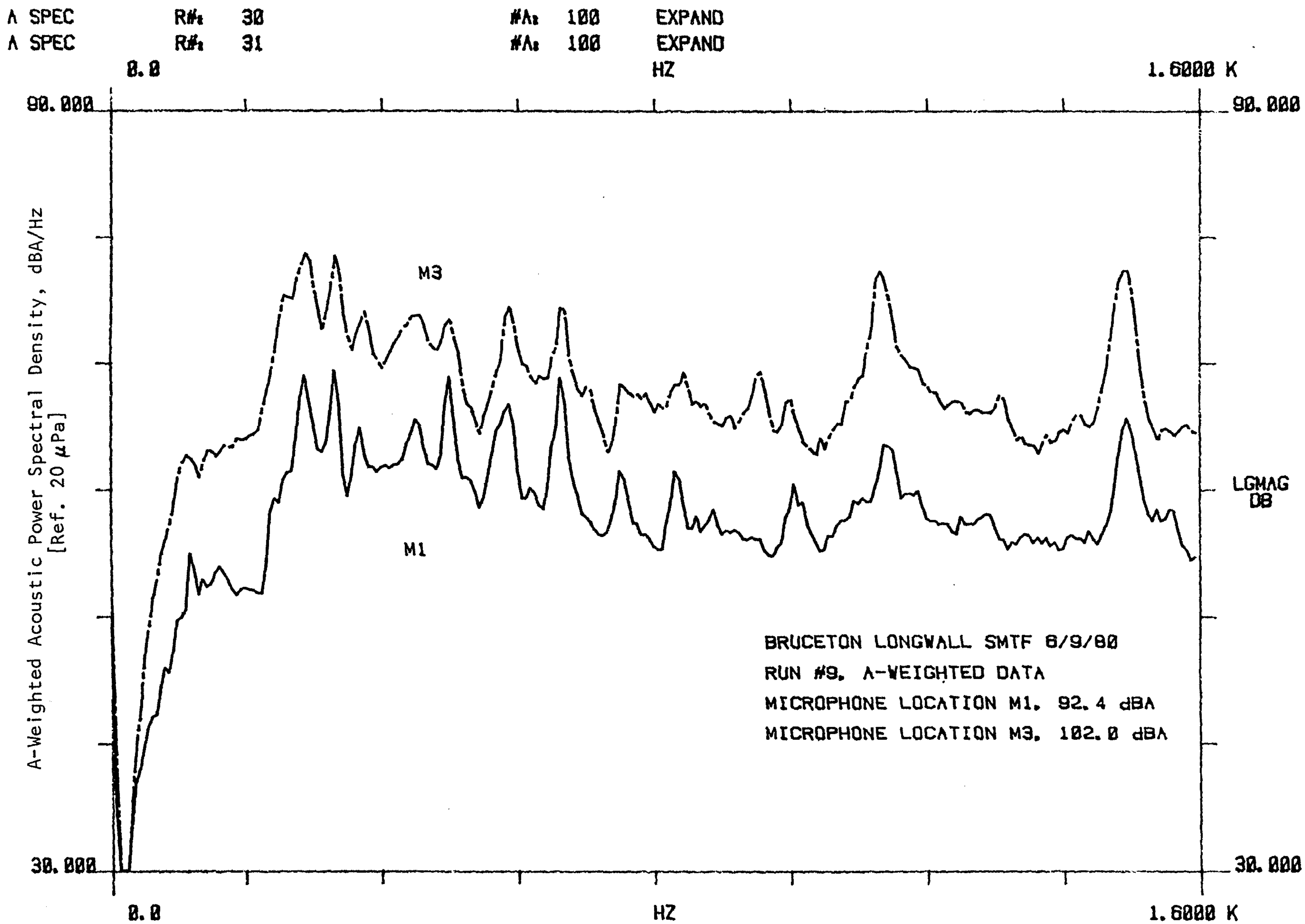


FIGURE 27. Noise from USBM longwall test

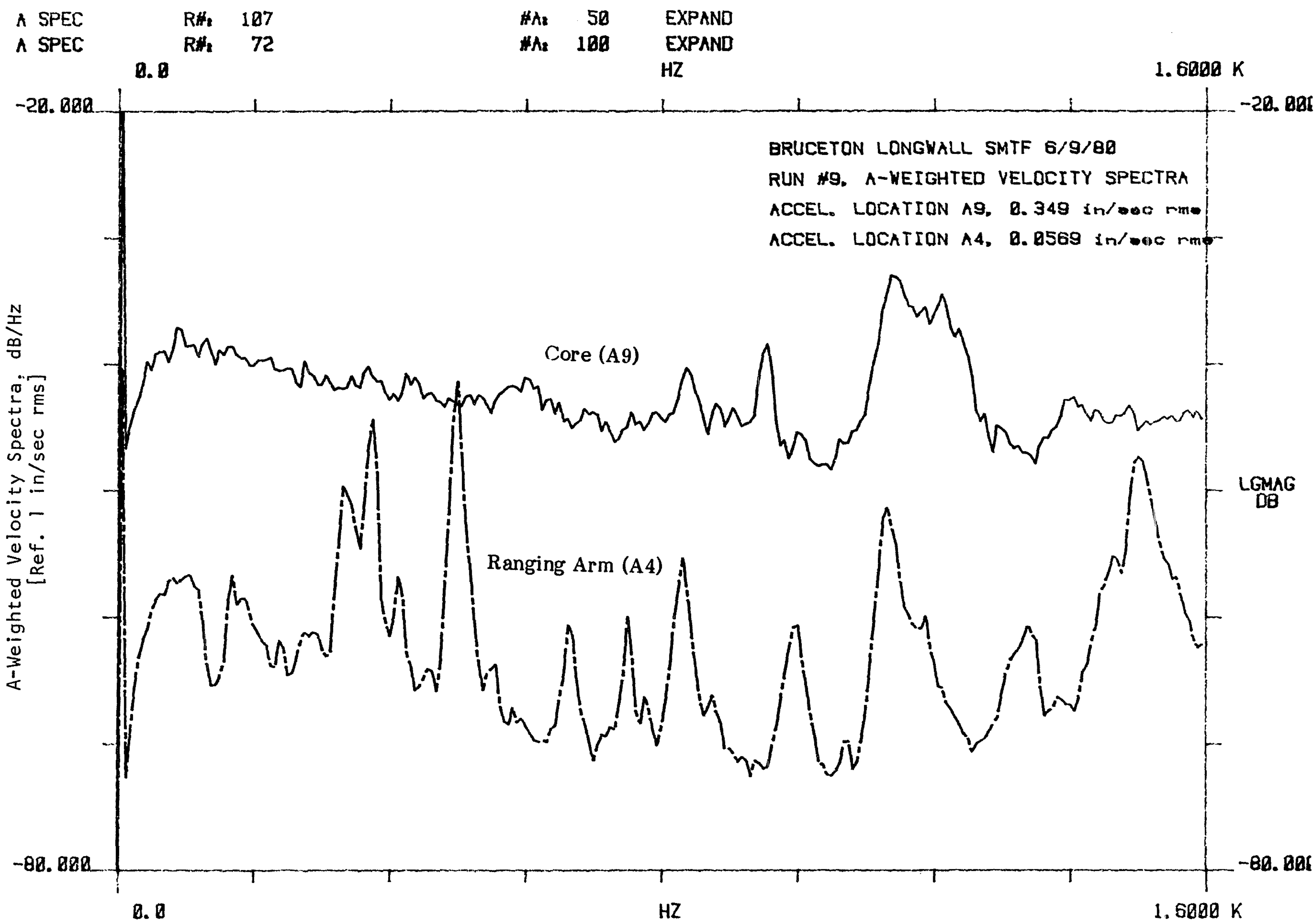


FIGURE 28. Vibration from USBM longwall test

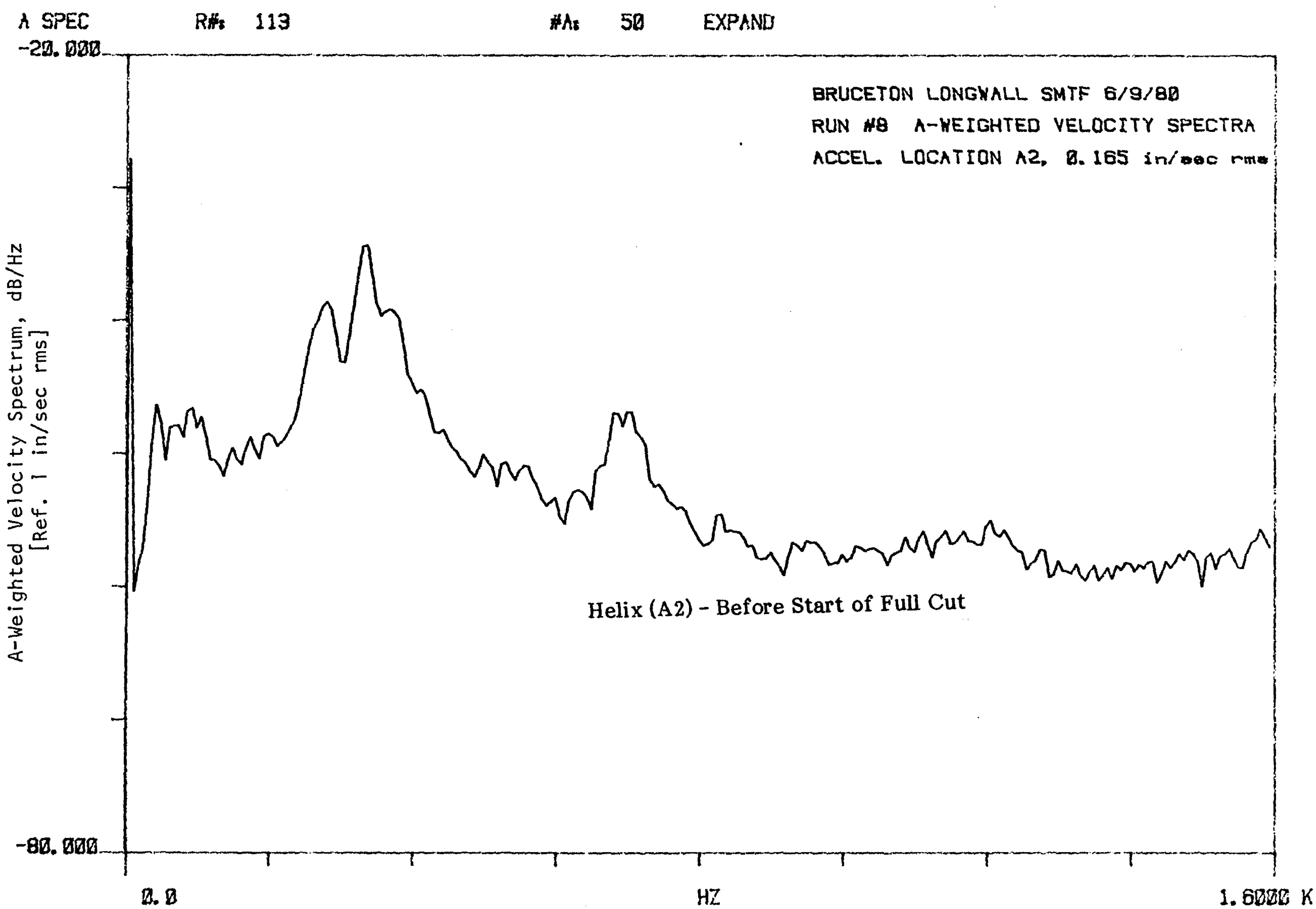


FIGURE 29. Vibration from USBM longwall test

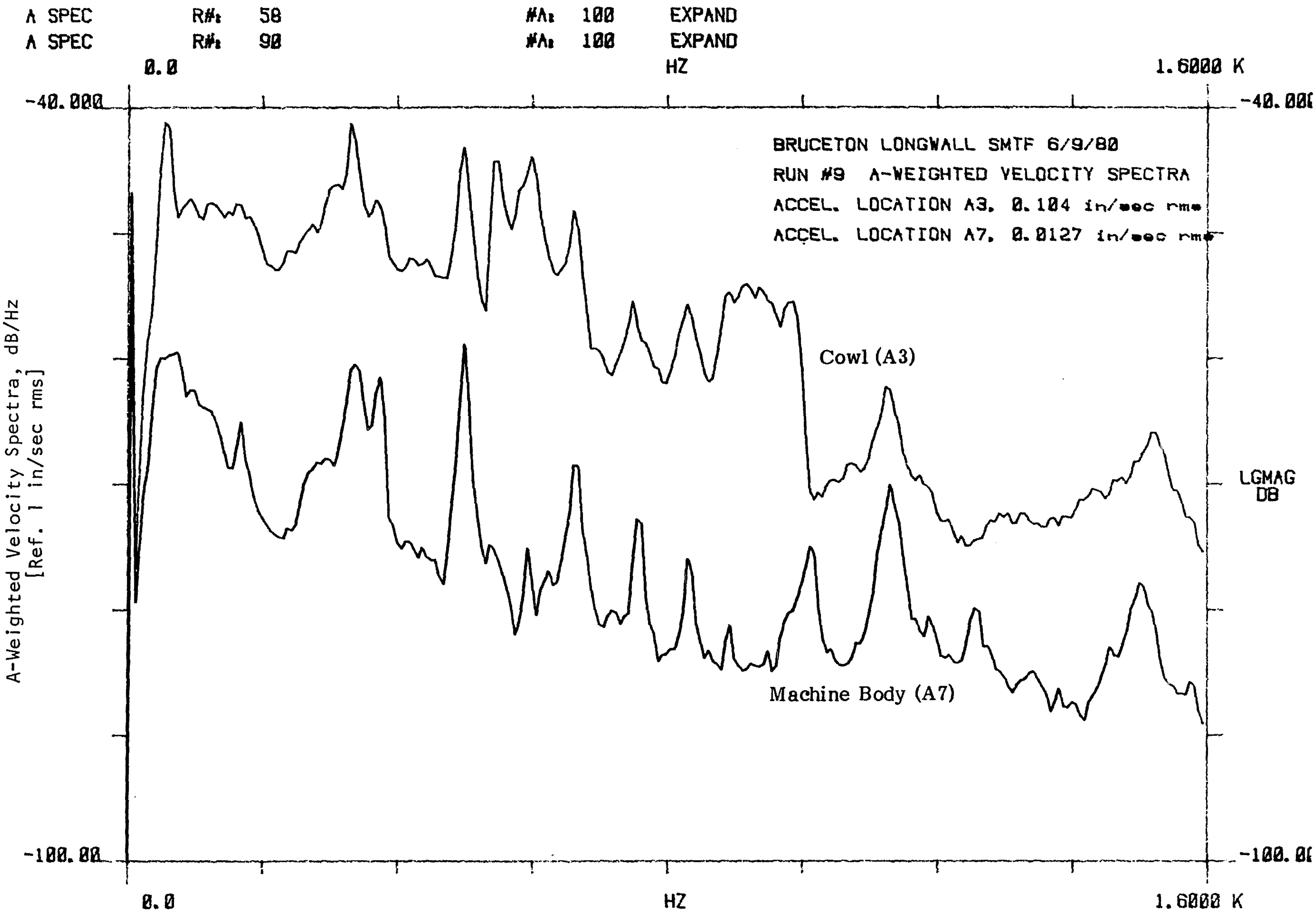


FIGURE 30. Vibration from USBM longwall test

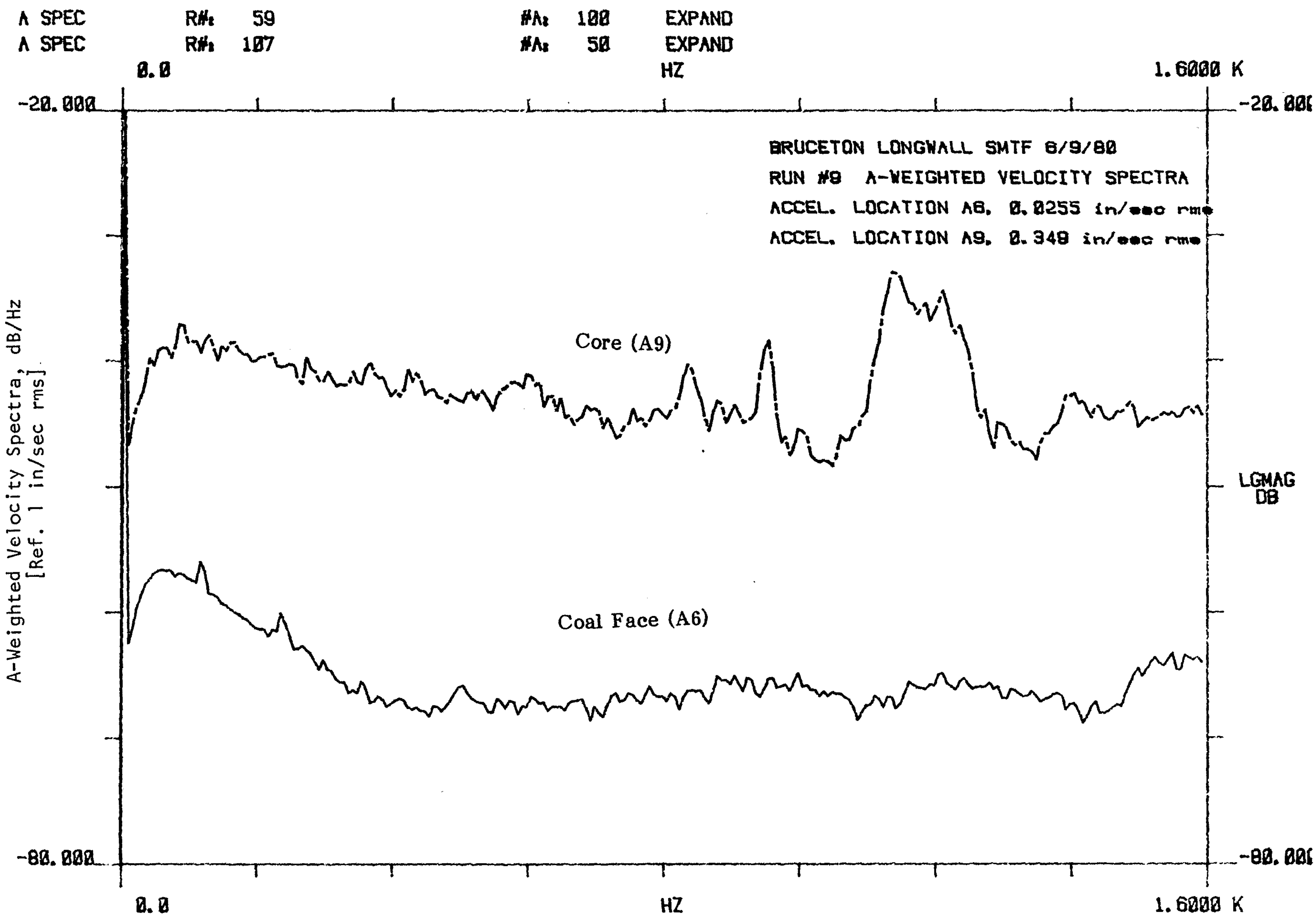


FIGURE 31. Vibration from USBM longwall test

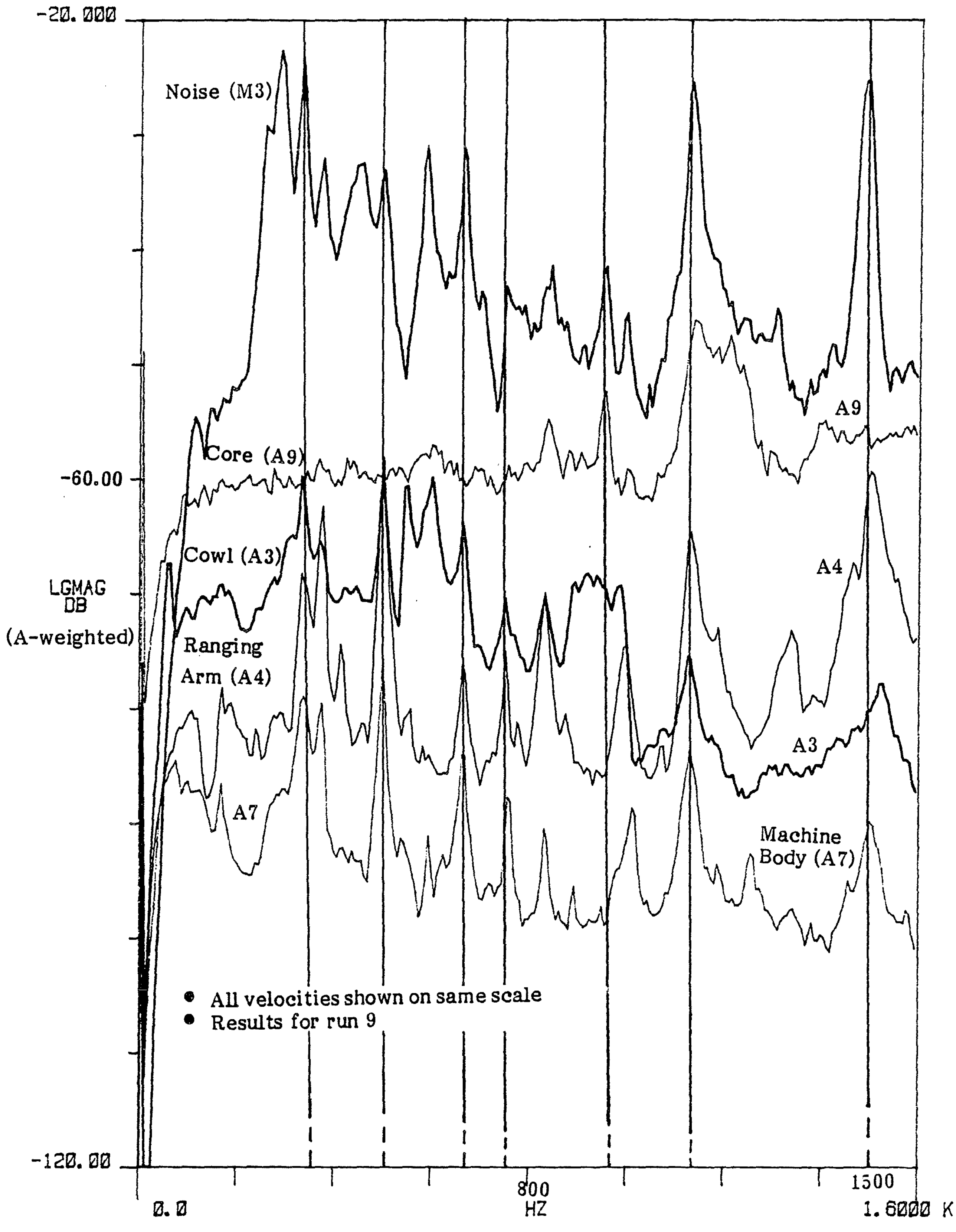


FIGURE 32. Noise and vibration from USBM longwall test

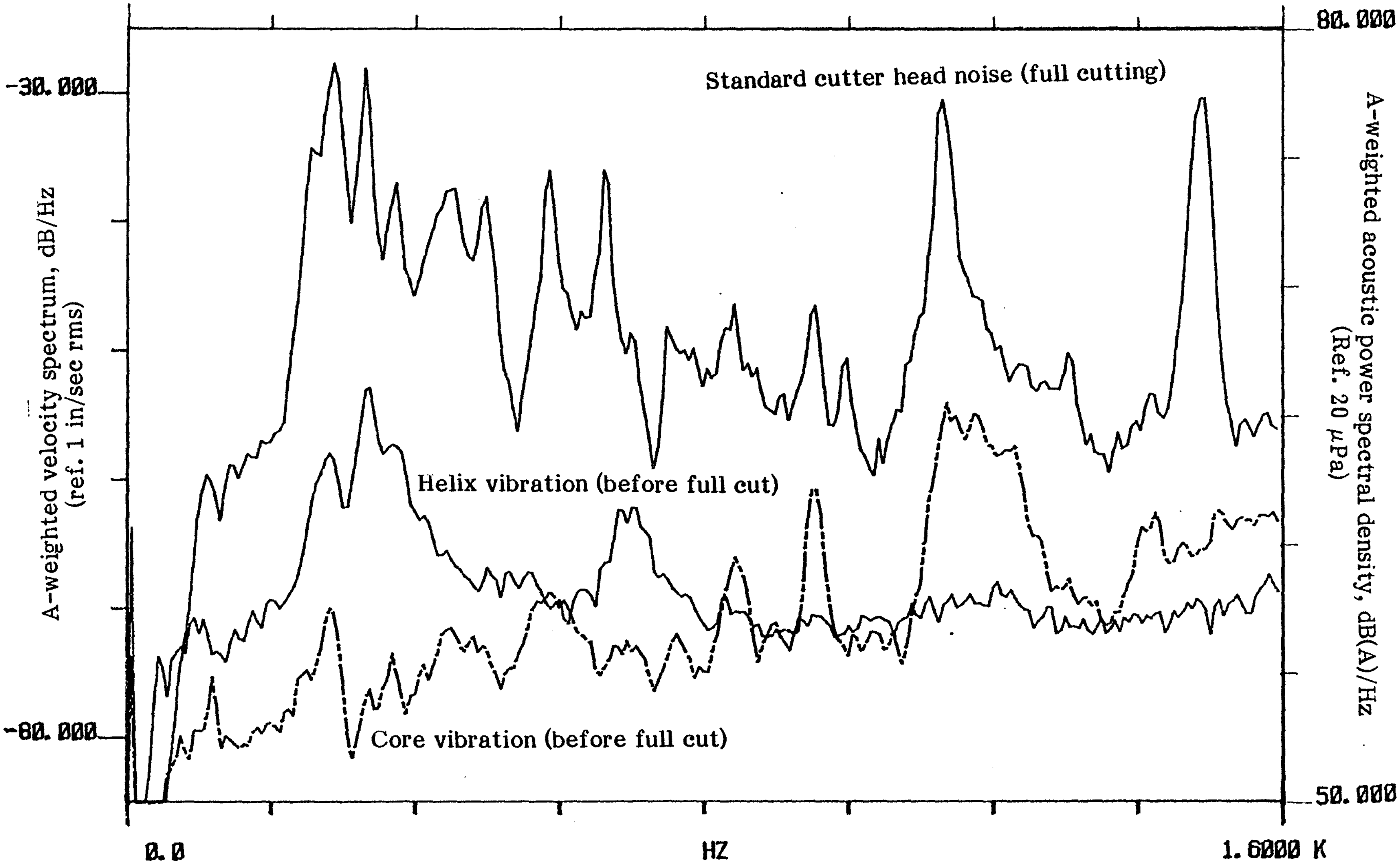


FIGURE 33. Noise and vibration from USBM longwall test

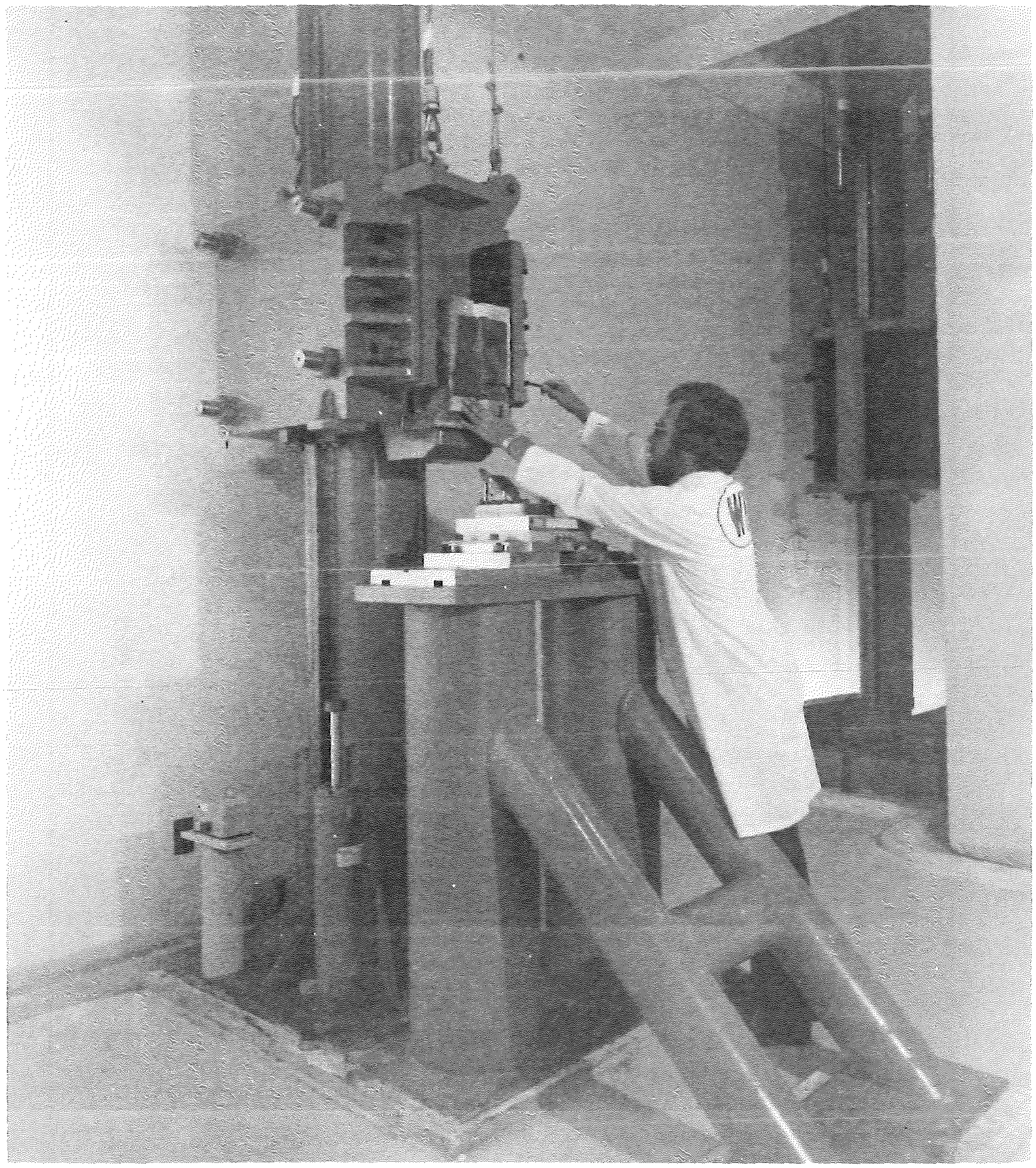


Figure 34. Linear cutting apparatus (LCA)

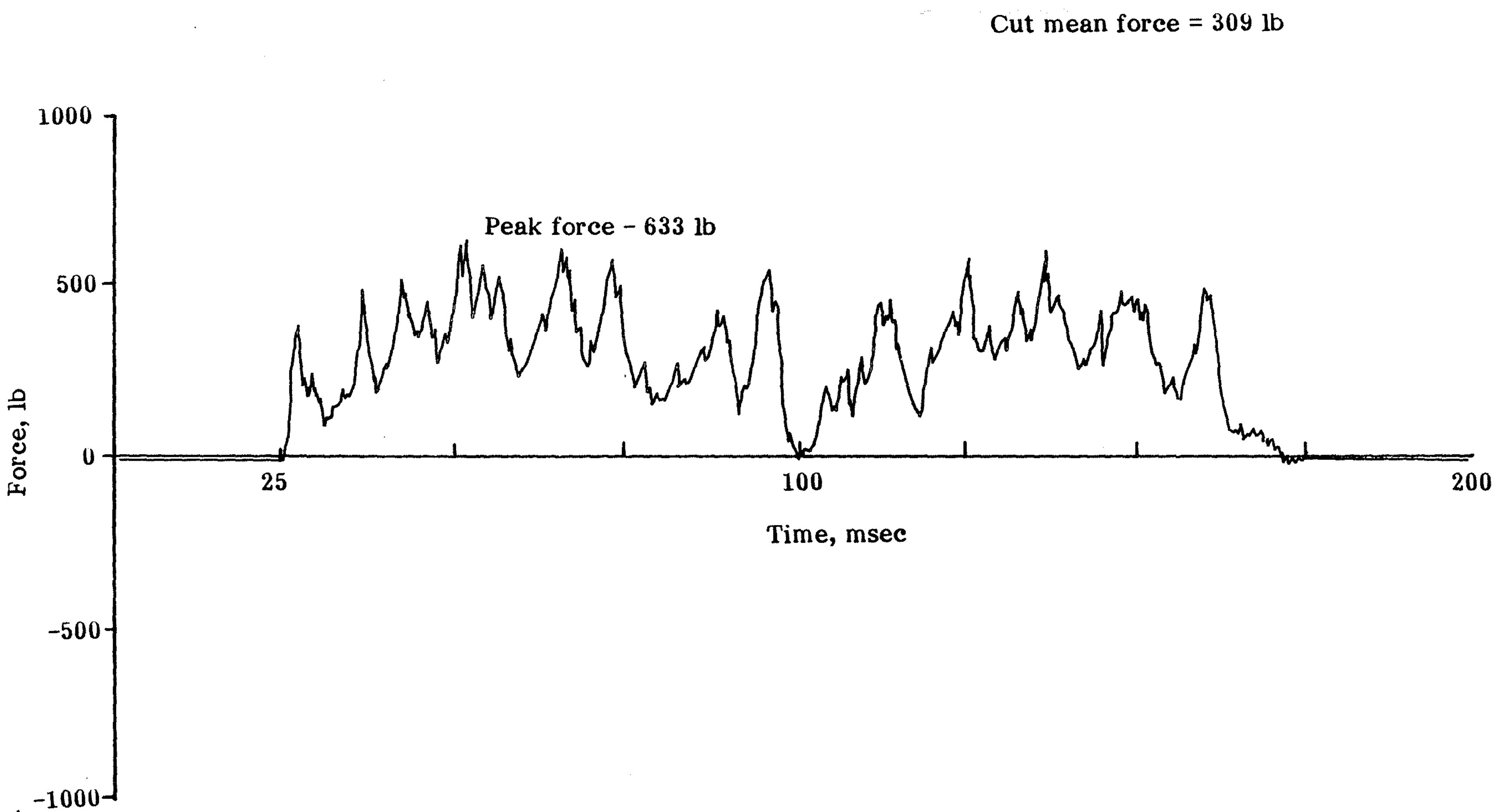


FIGURE 35. Force time history of single bit cutting coal at 60 ips

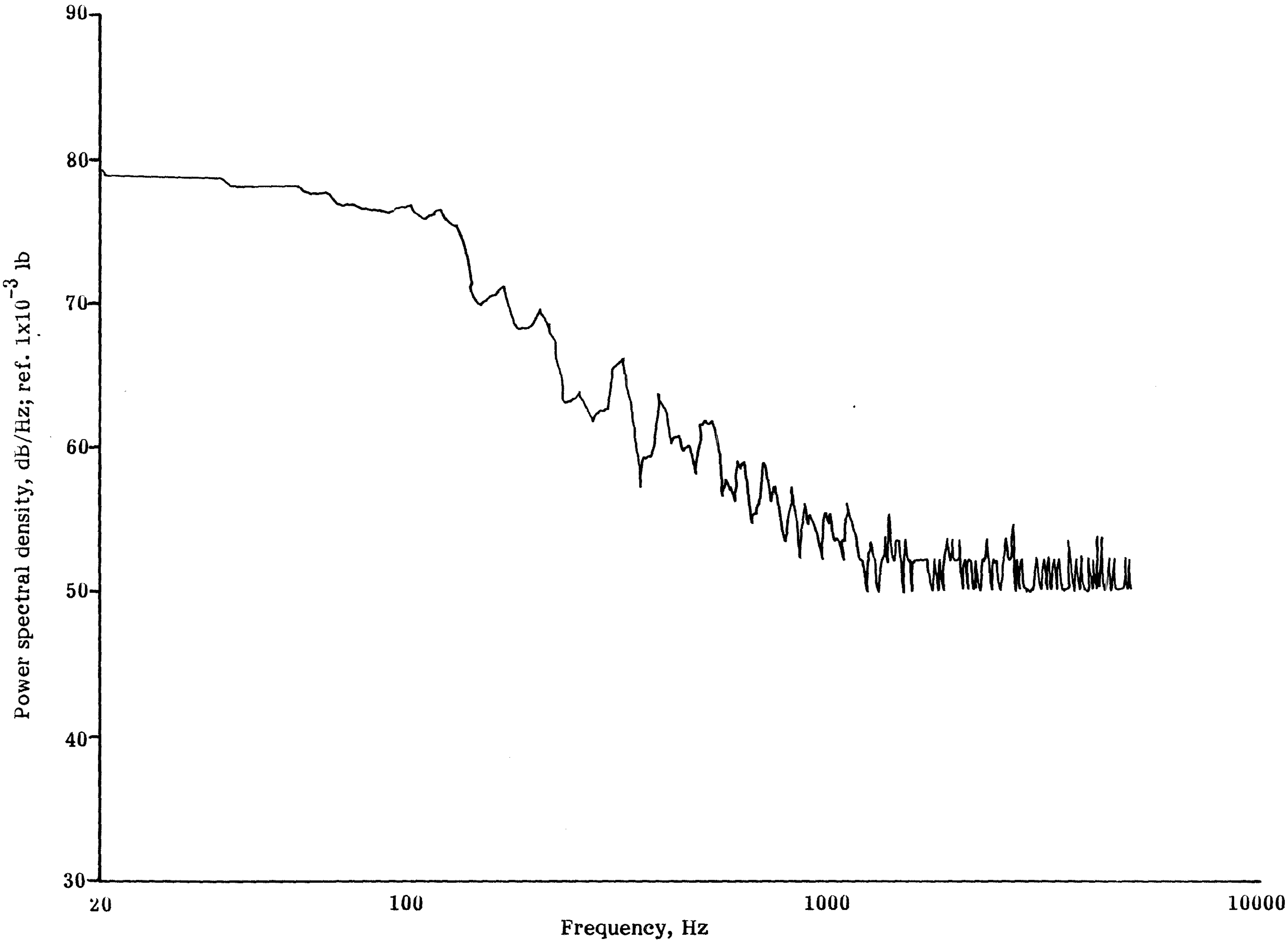


FIGURE 36. Force PSD of single bit cutting coal at 60 ips

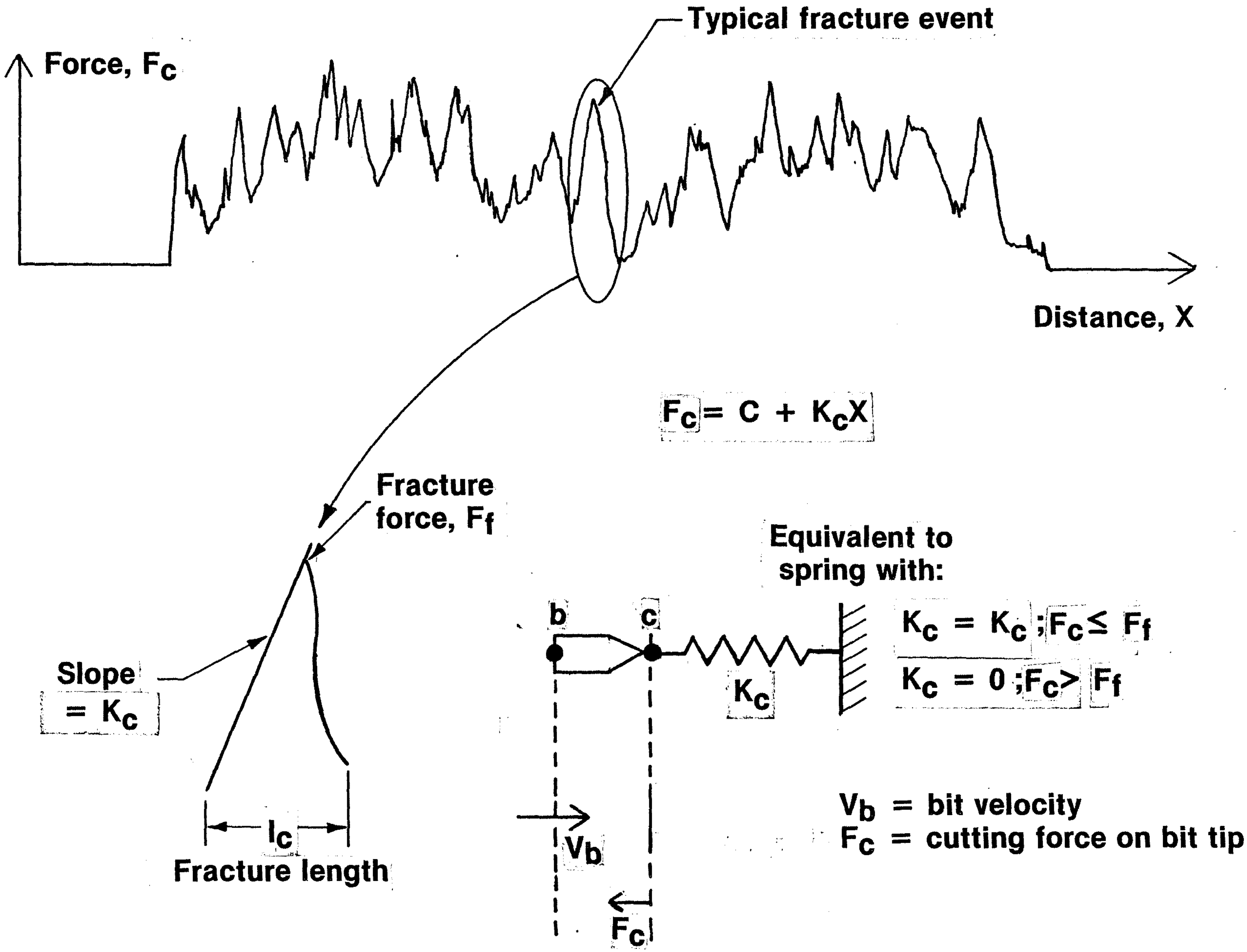
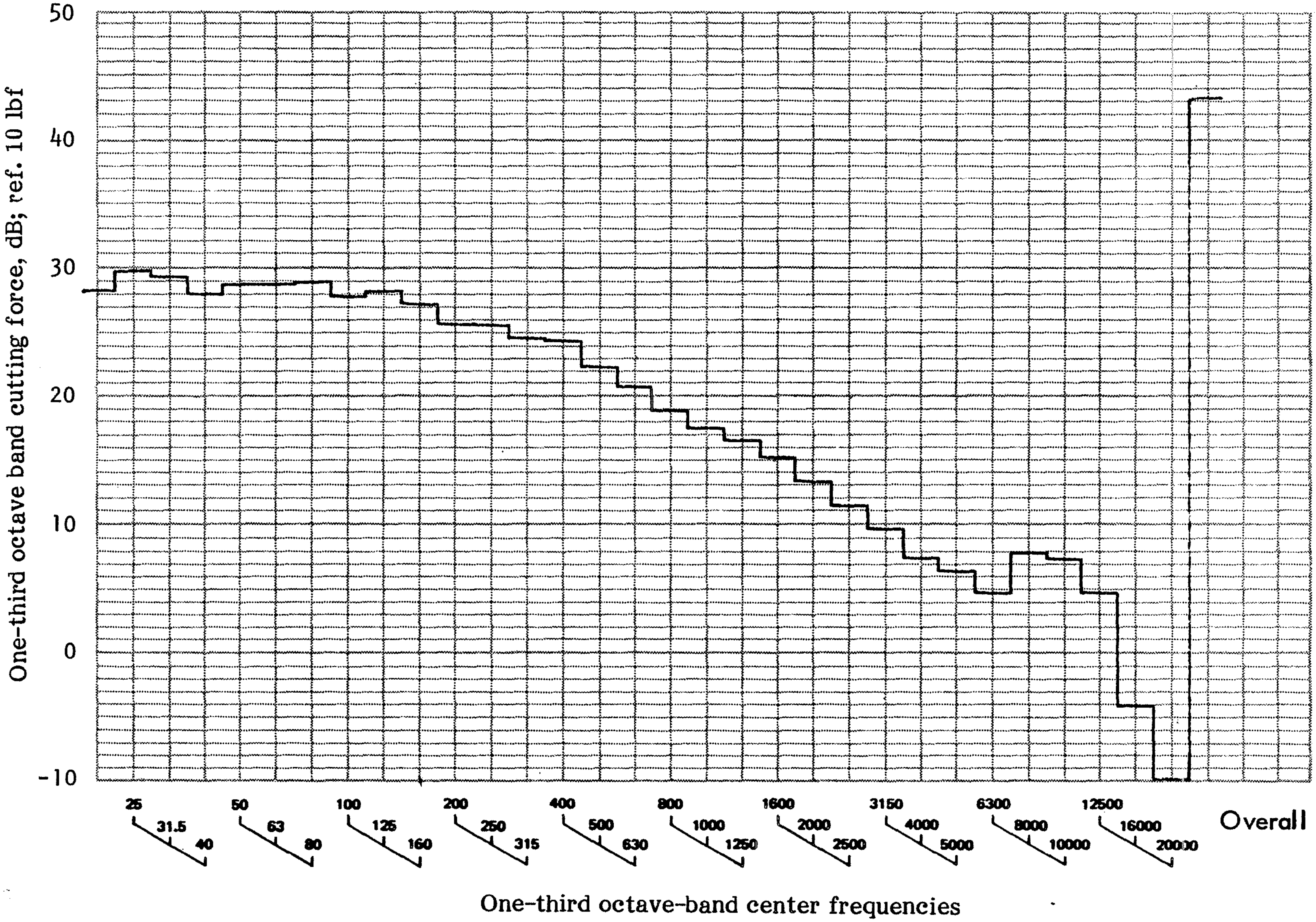


FIGURE 37. Coal cutting force model



Note: Cutting force in pounds = $10^{(dB/20)}$.

FIGURE 38. Typical One-Third Octave-Band Coal Cutting Force Spectrum

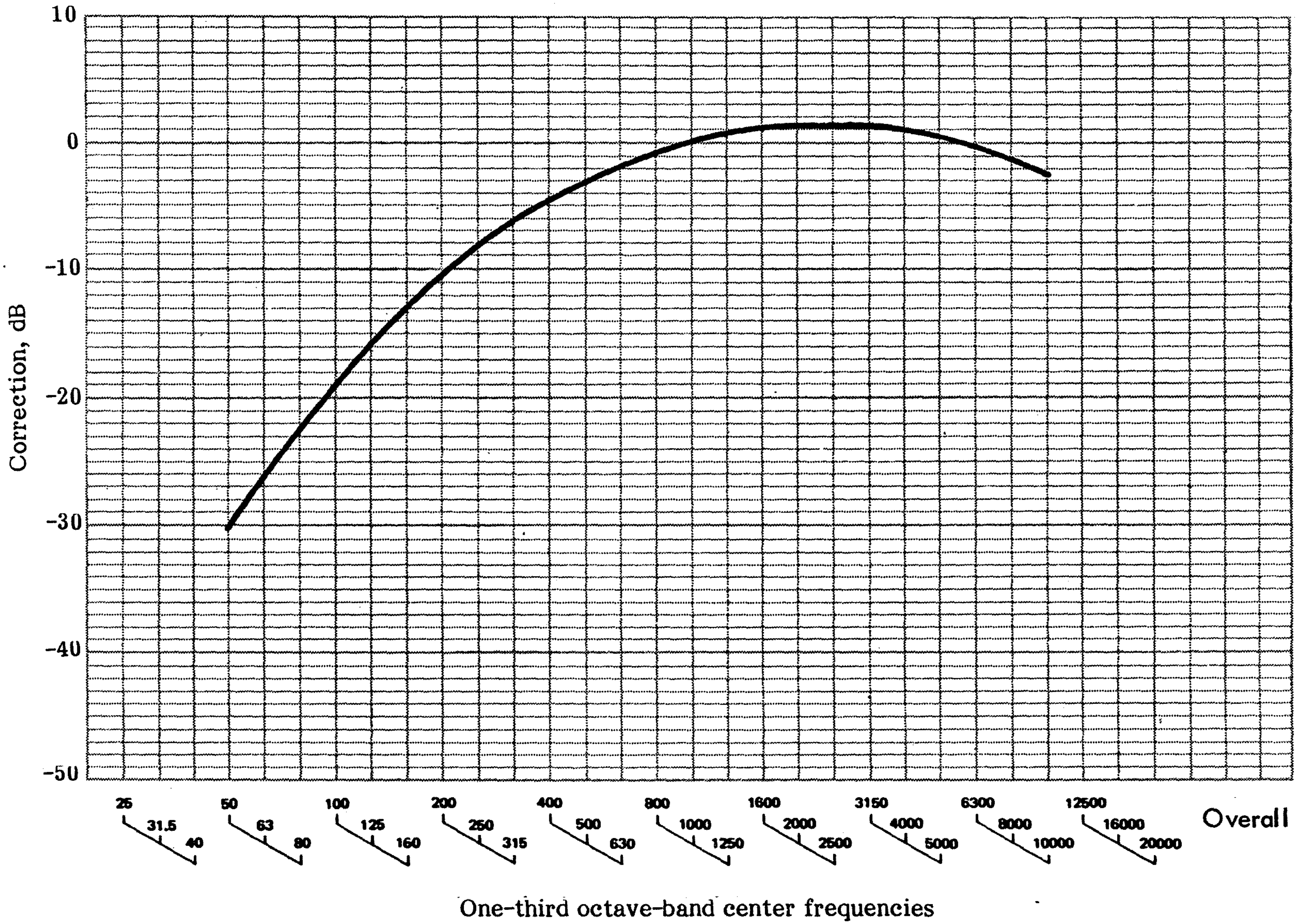


FIGURE 39. A-weighting filter response

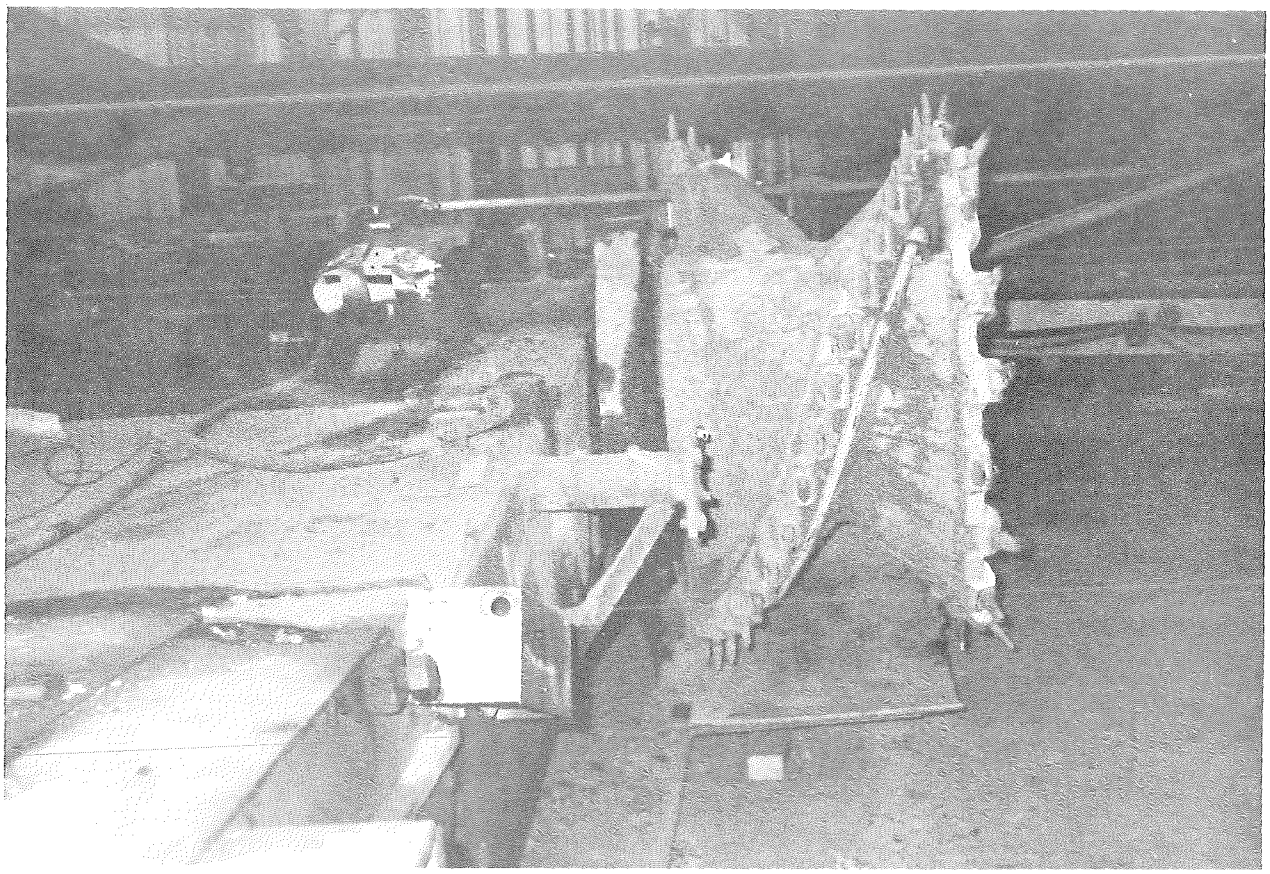


FIGURE 44. Stiffened and damped cutter drum on the USBM longwall shearer

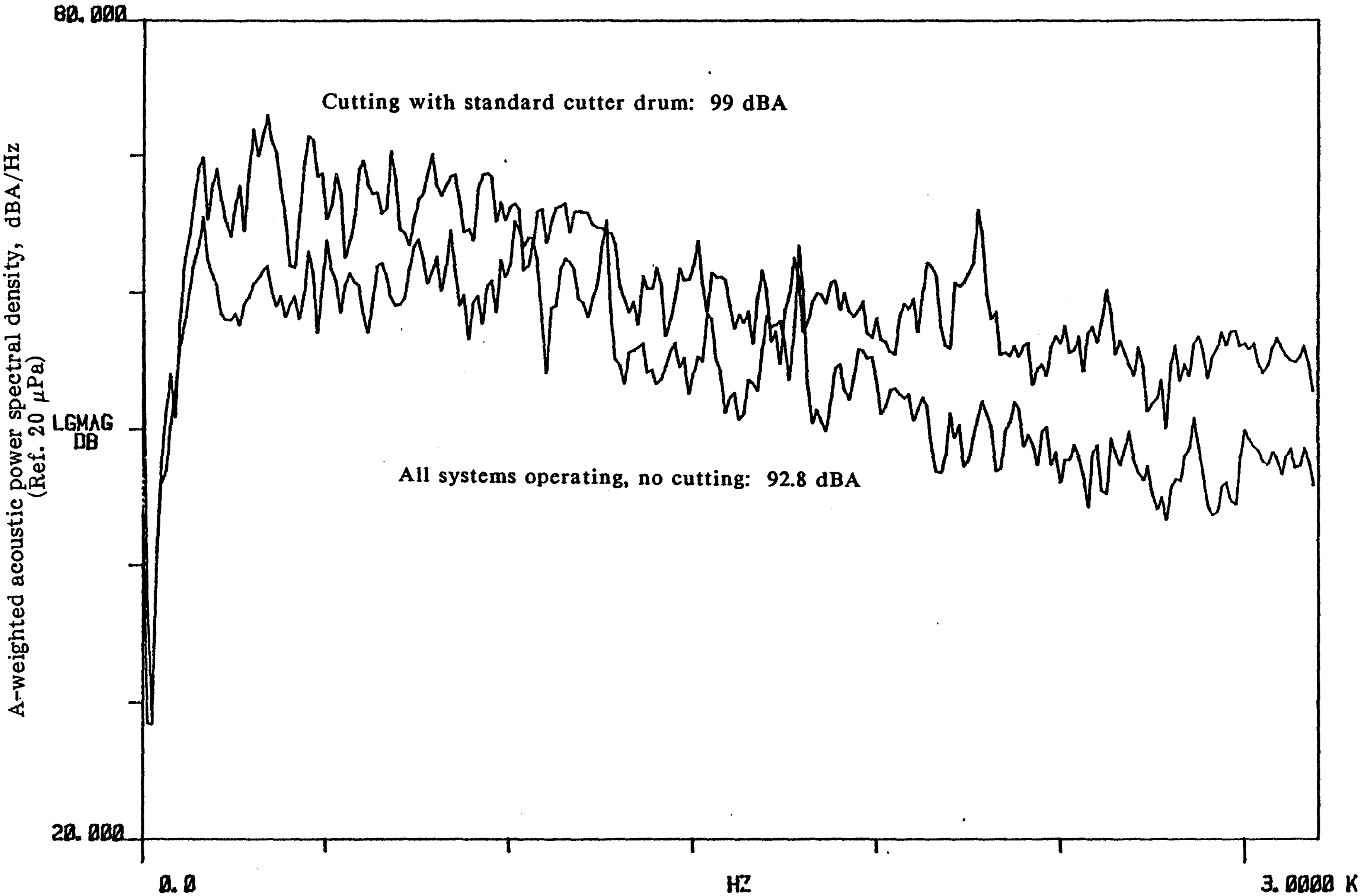


FIGURE 45. Noise from USBM longwall test

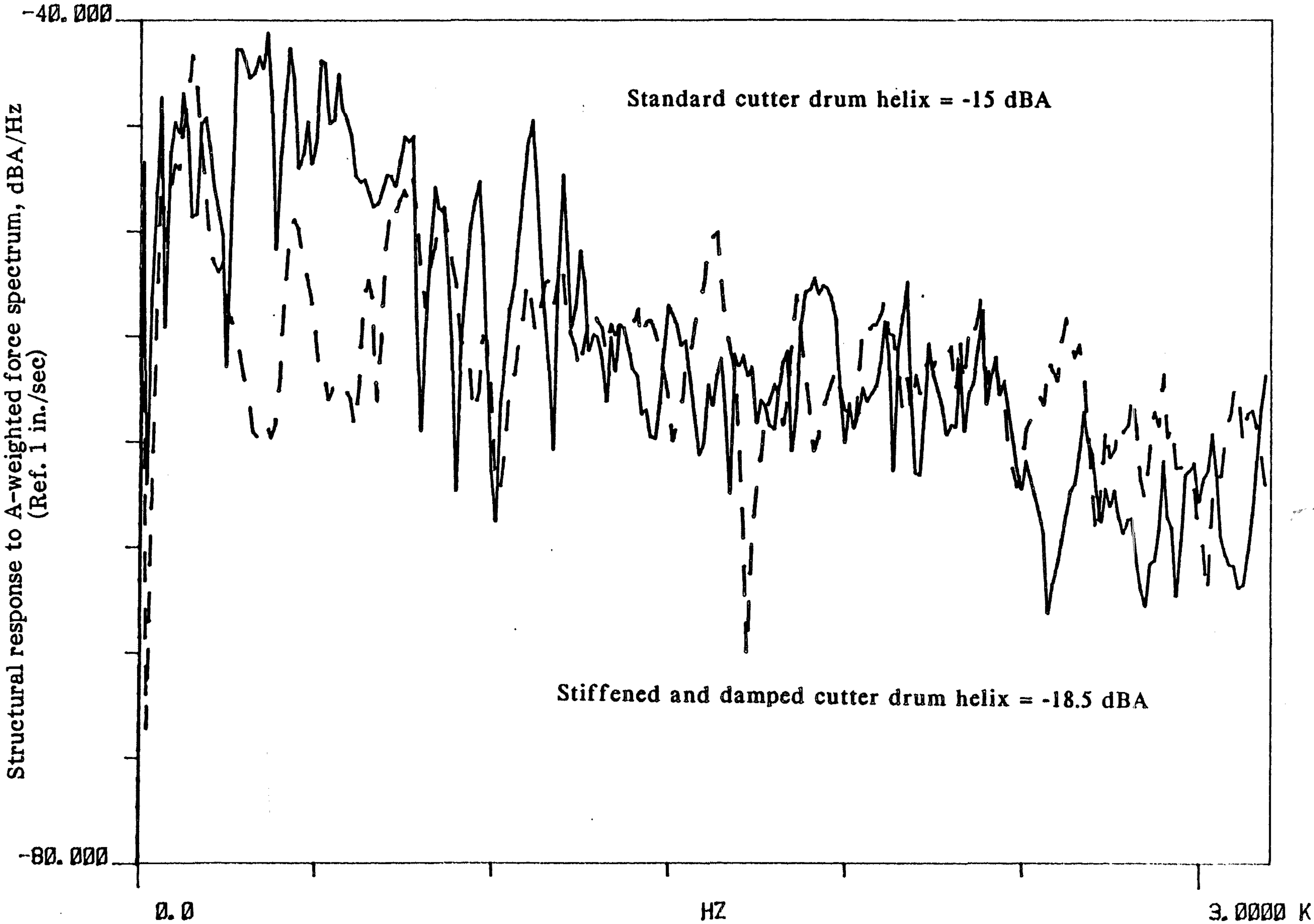


FIGURE 46. Structural response to A-weighted force spectrum

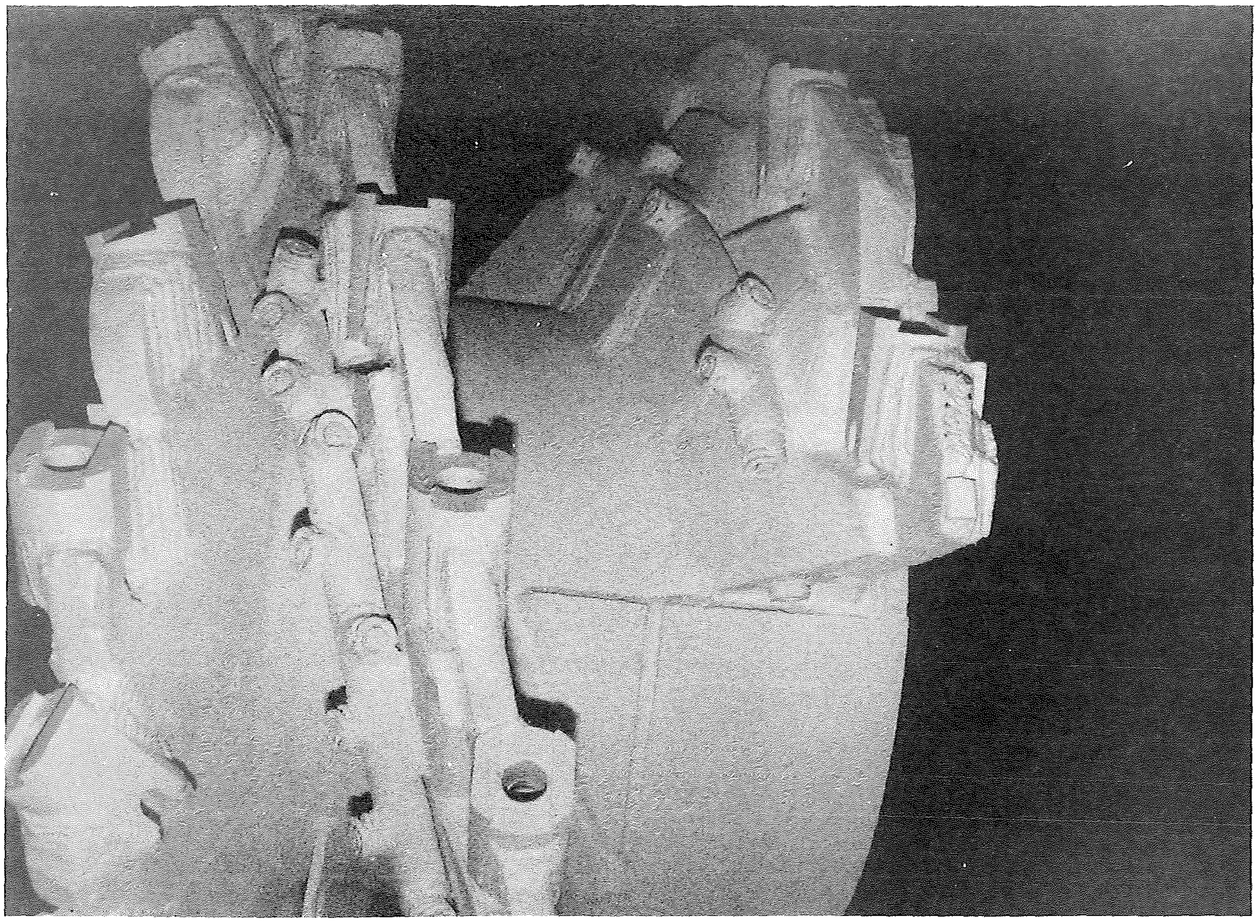


FIGURE 47. Stiffened and damped cutter drum

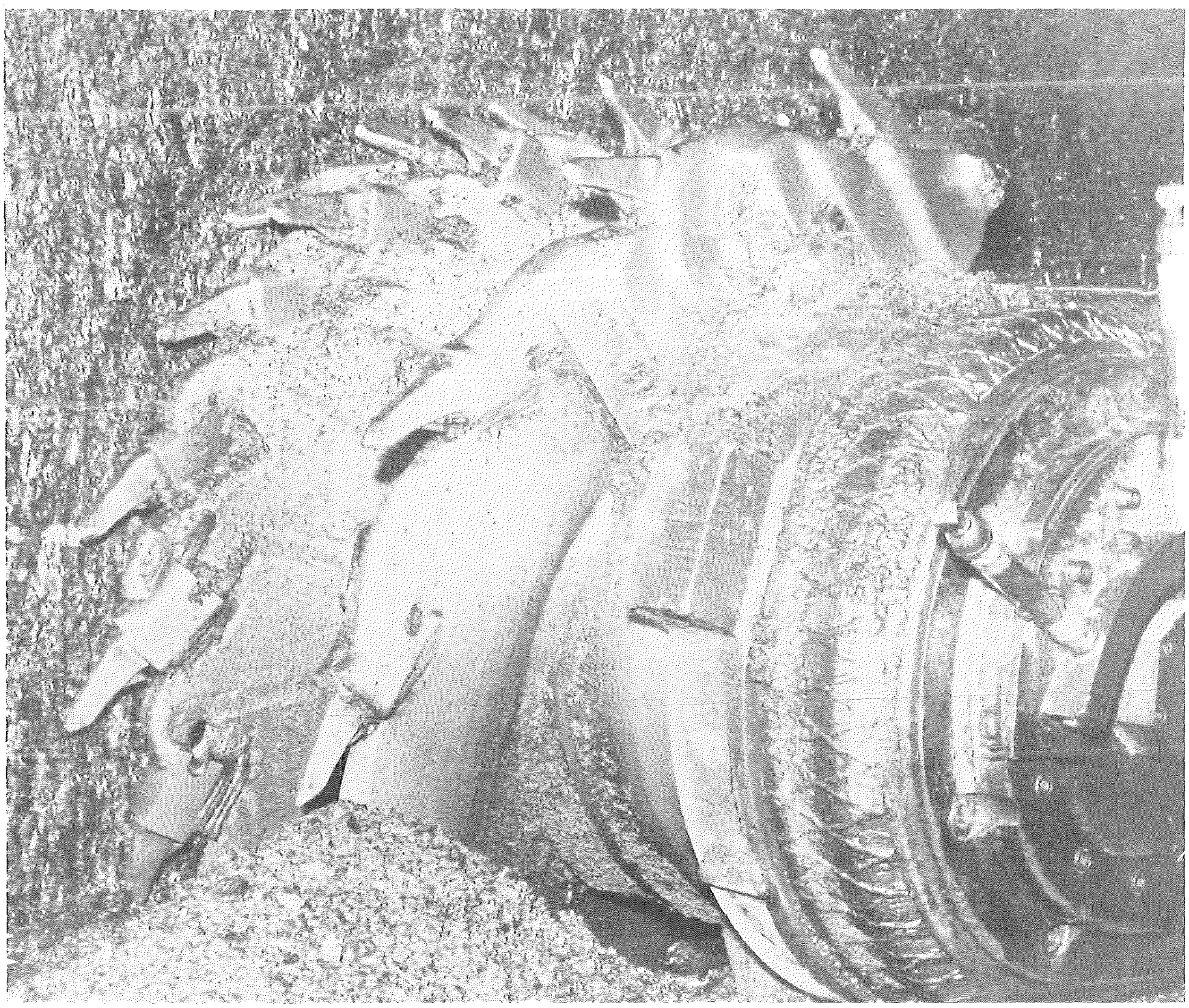


FIGURE 48. Stiffened and damped cutter drum at Consol Mine #95

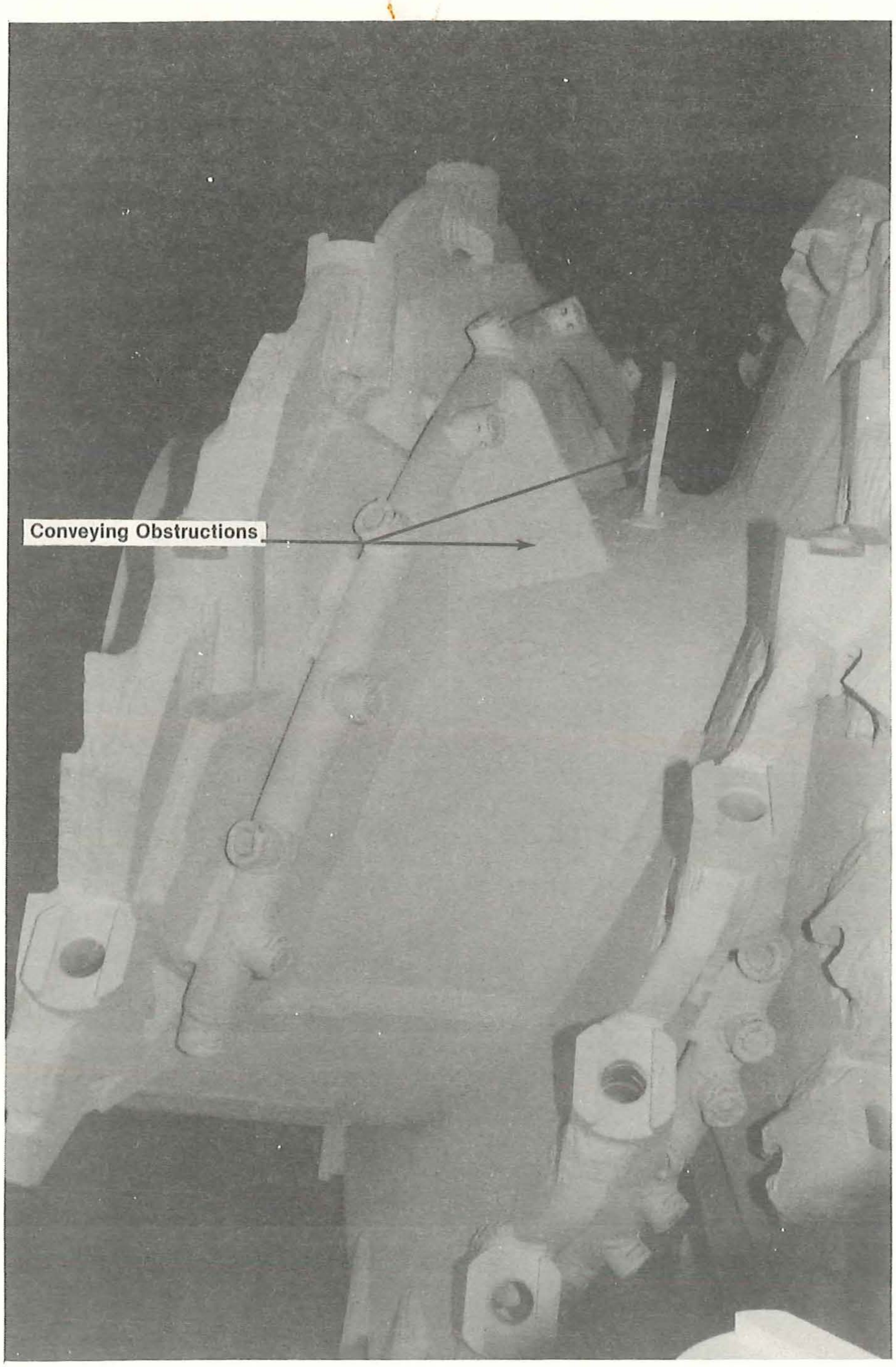
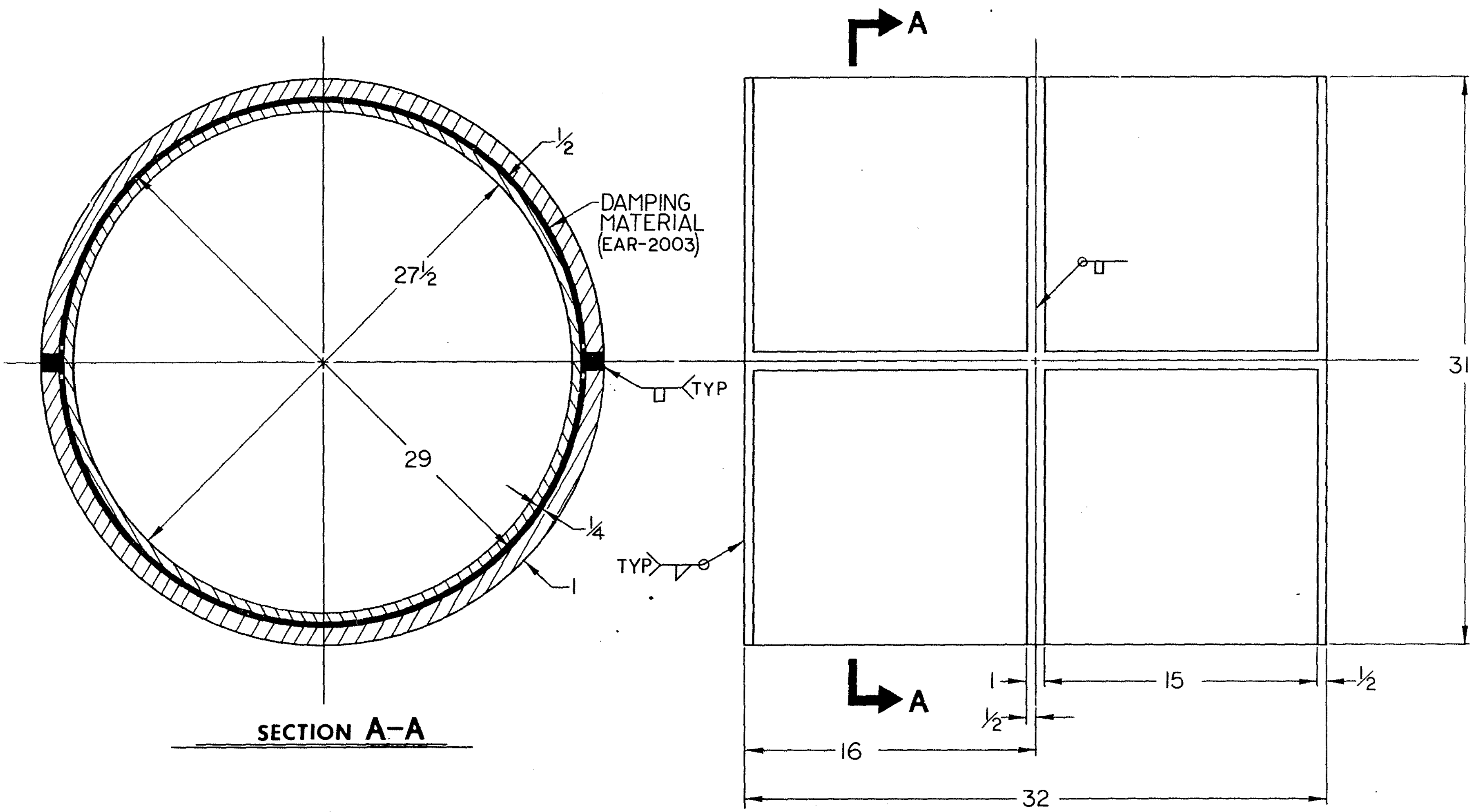


FIGURE 49. Stiffened and damped cutter drum conveying obstructions



Note: All dimensions in inches.

FIGURE 50. Stiffened and damped cutter drum core

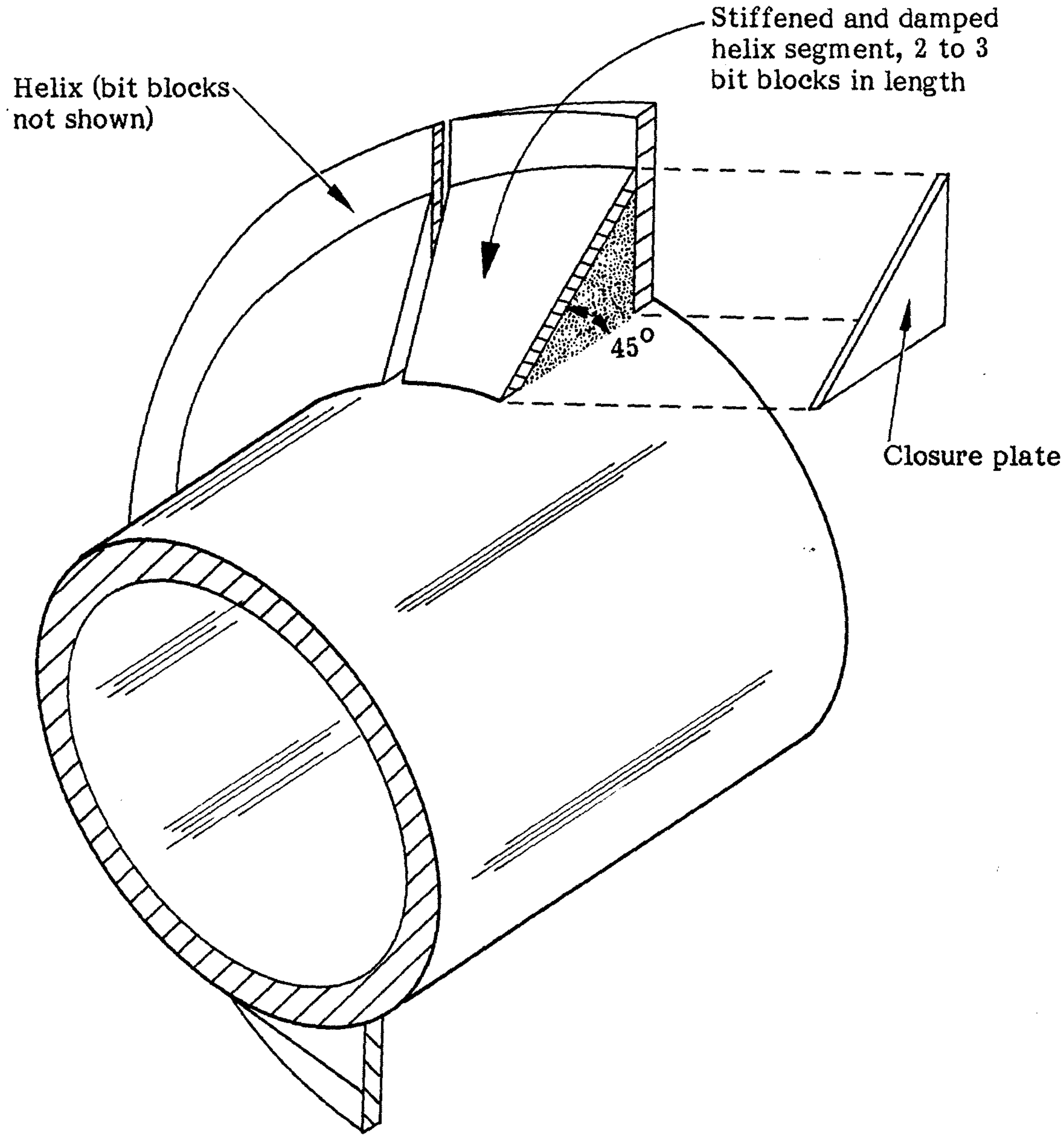


FIGURE 51. Segmented, stiffened, and damped helix configuration

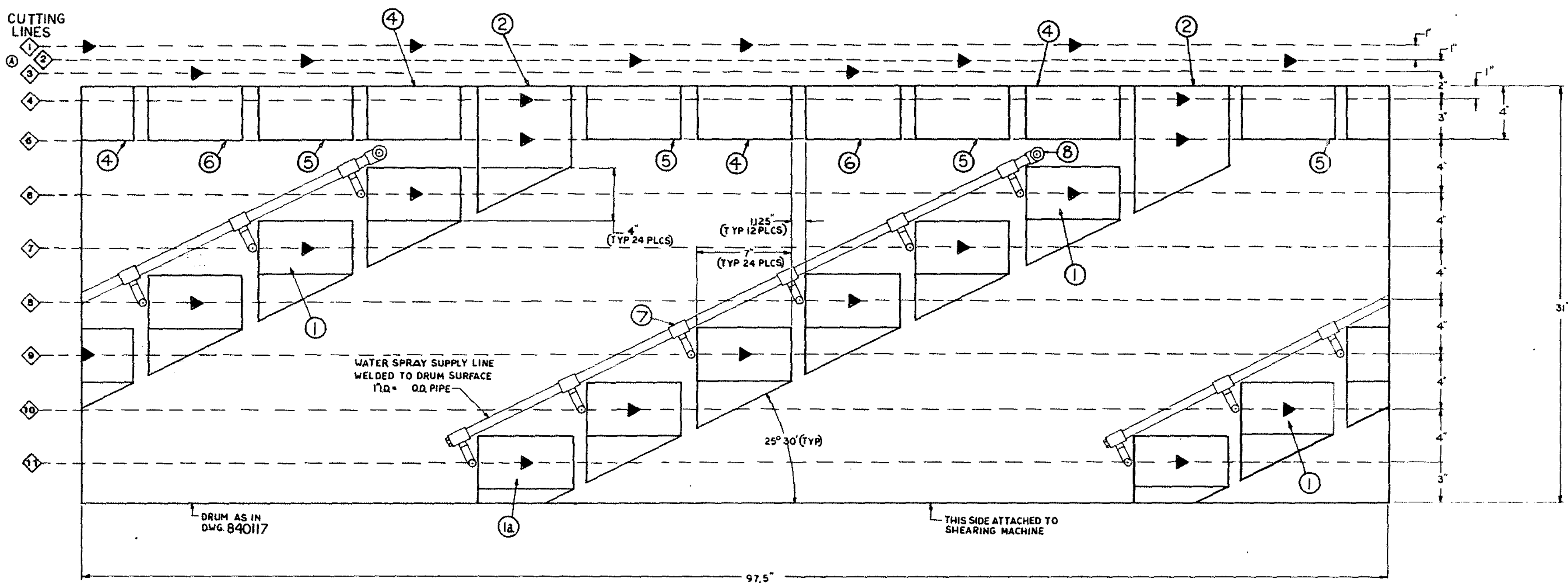


FIGURE 52. Stage-laced shearer cutter drum

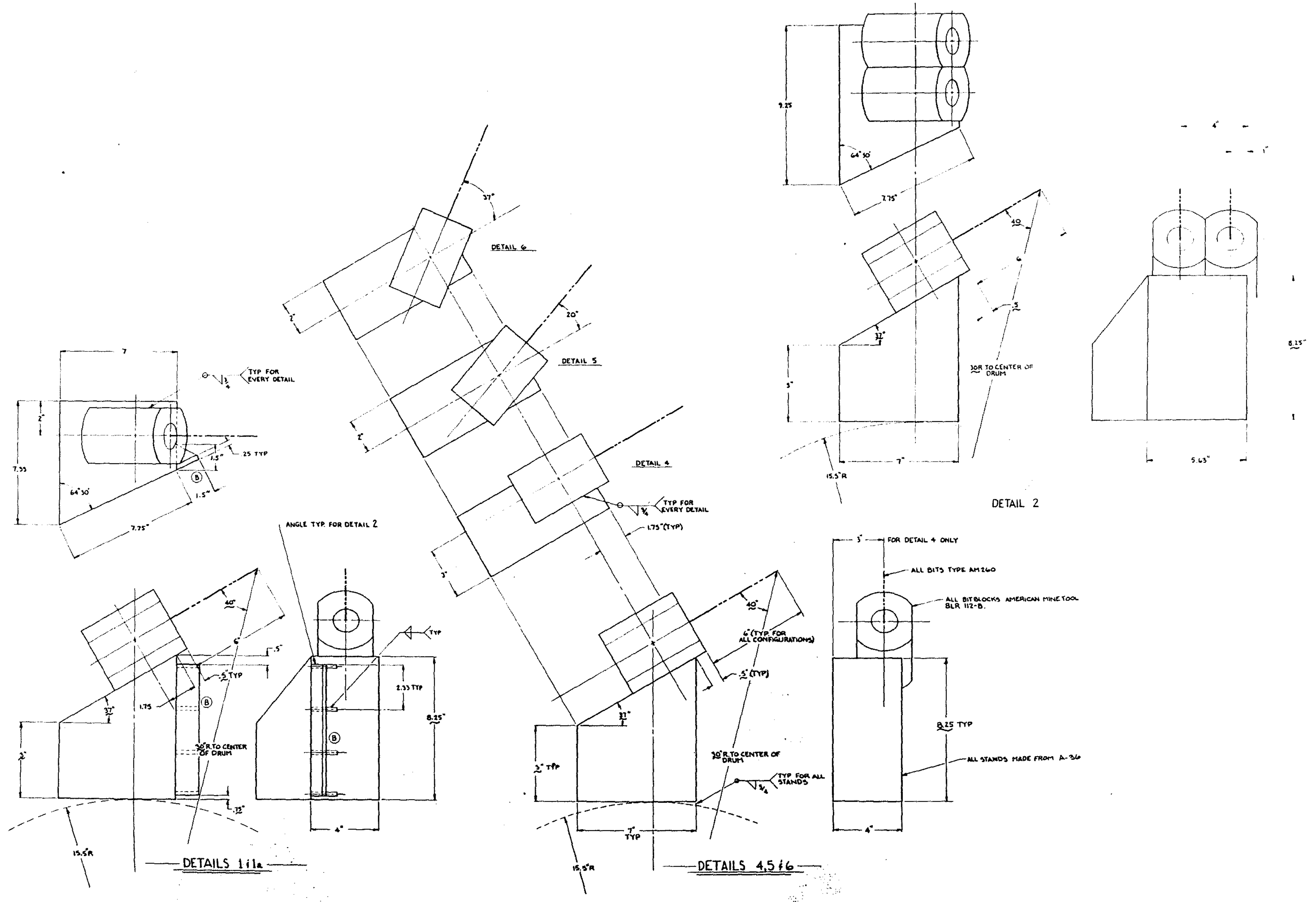


FIGURE 53. Stage-laced shearer cutter drum

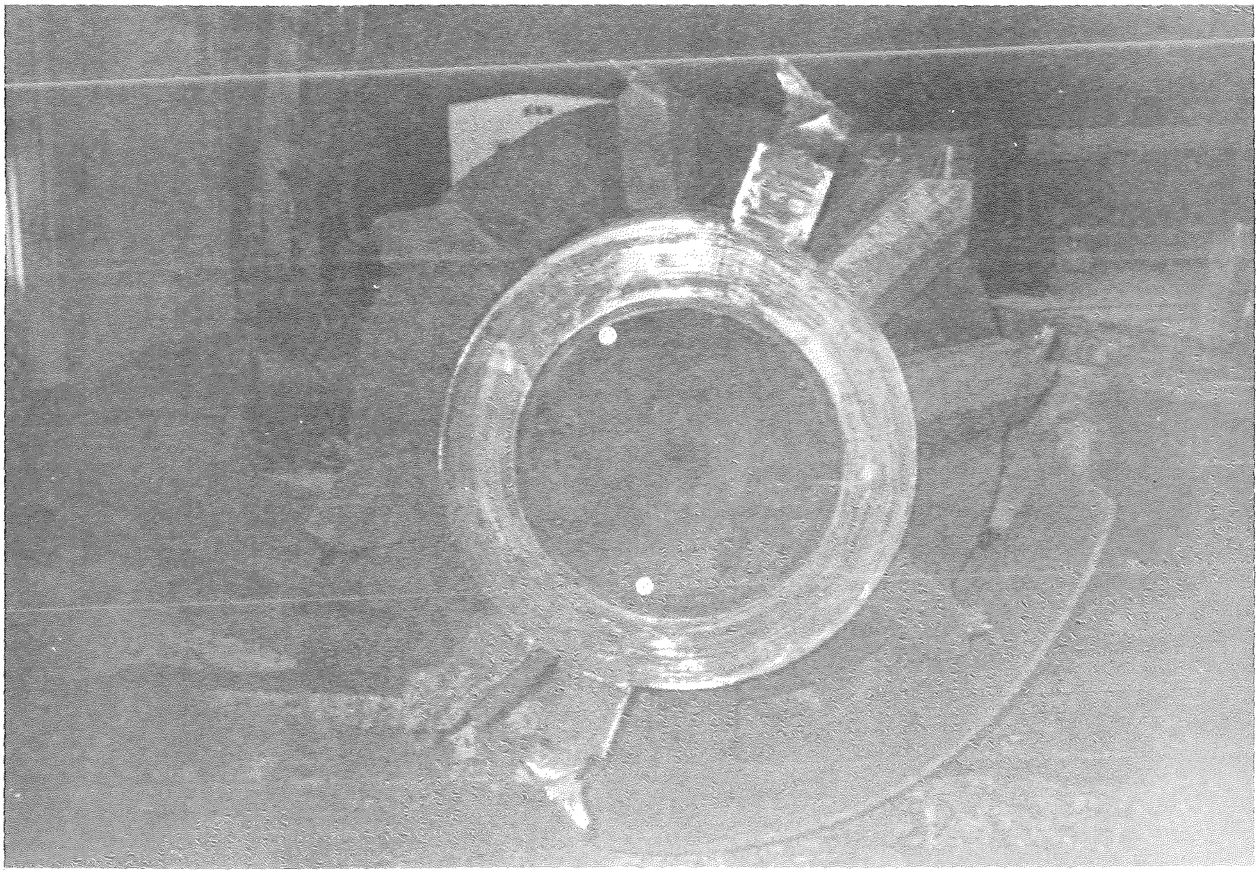


FIGURE 54. Stage-laced longwall shearer cutter drum

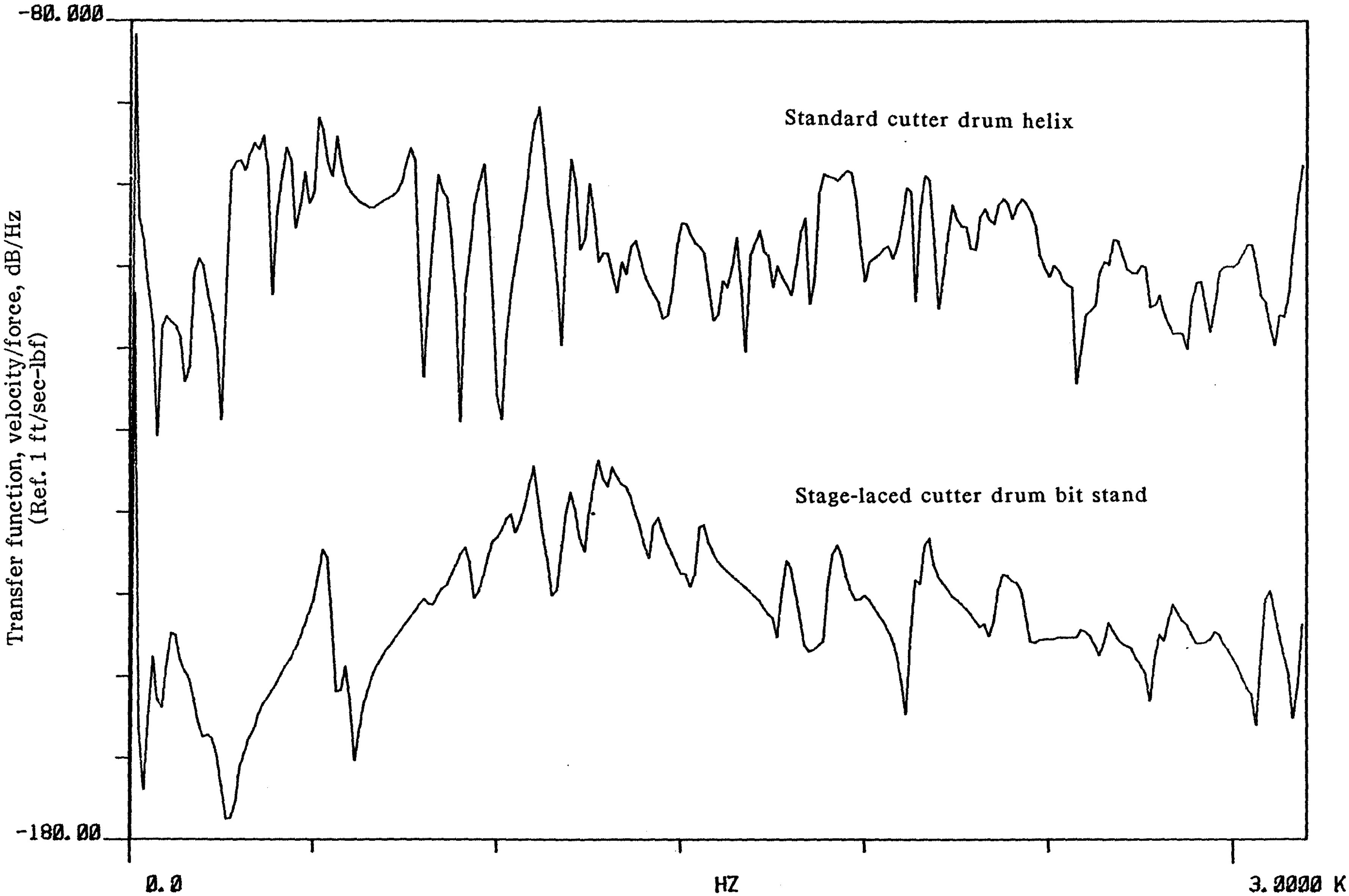


FIGURE 55. Velocity/force transfer functions

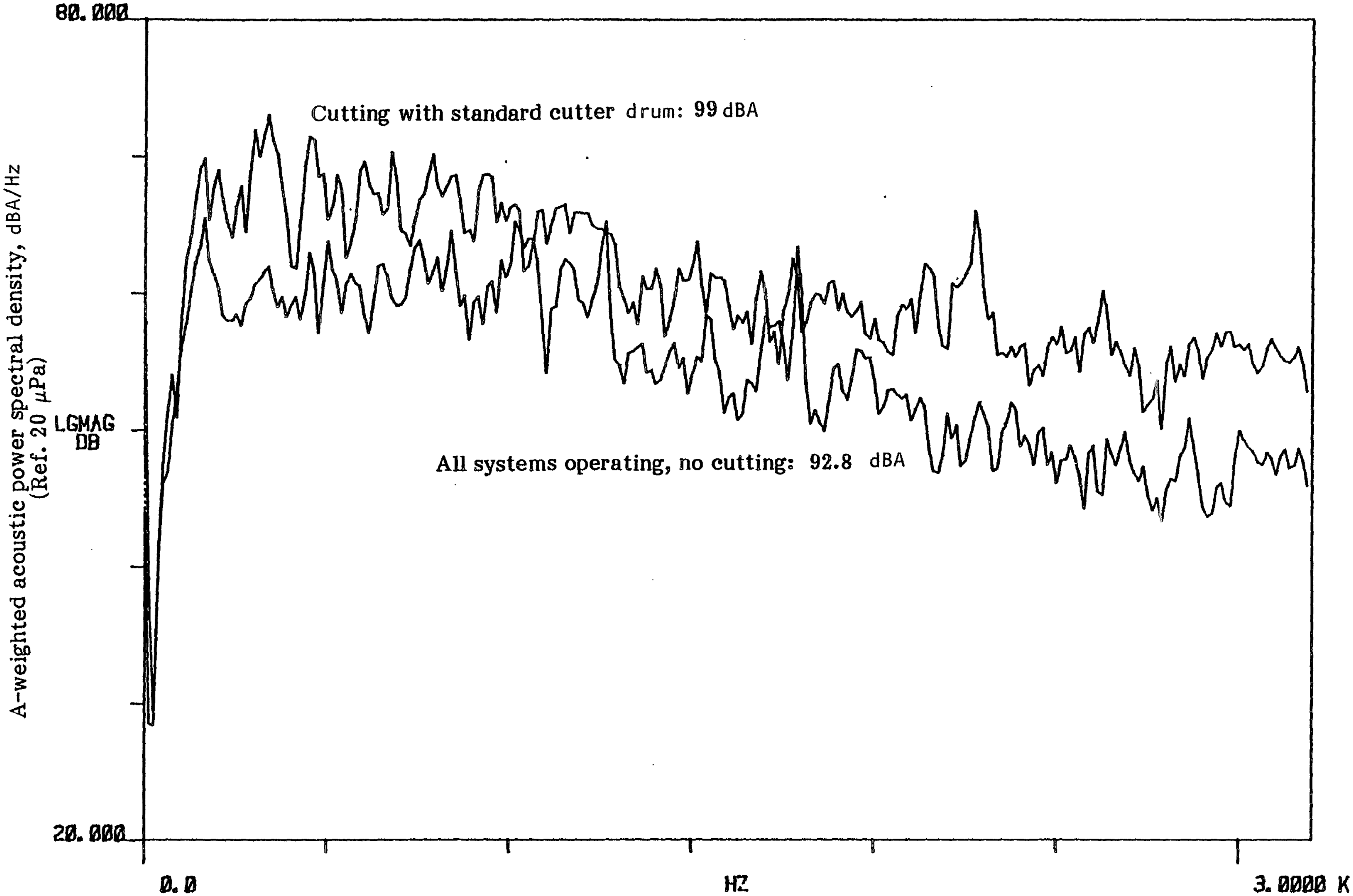


FIGURE 56. Noise from USBM longwall test

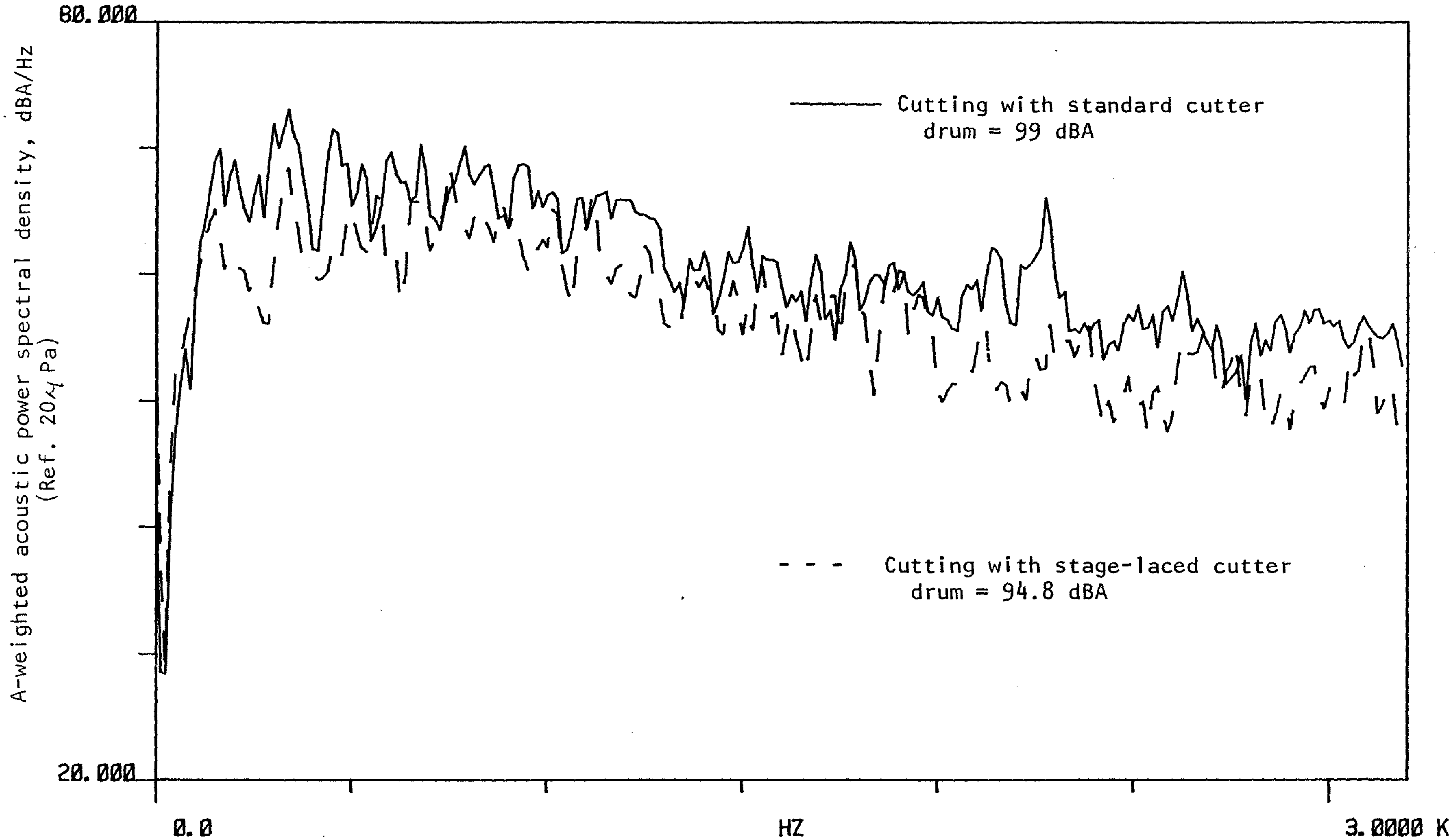


Figure 57. Noise from USBM Longwall Test

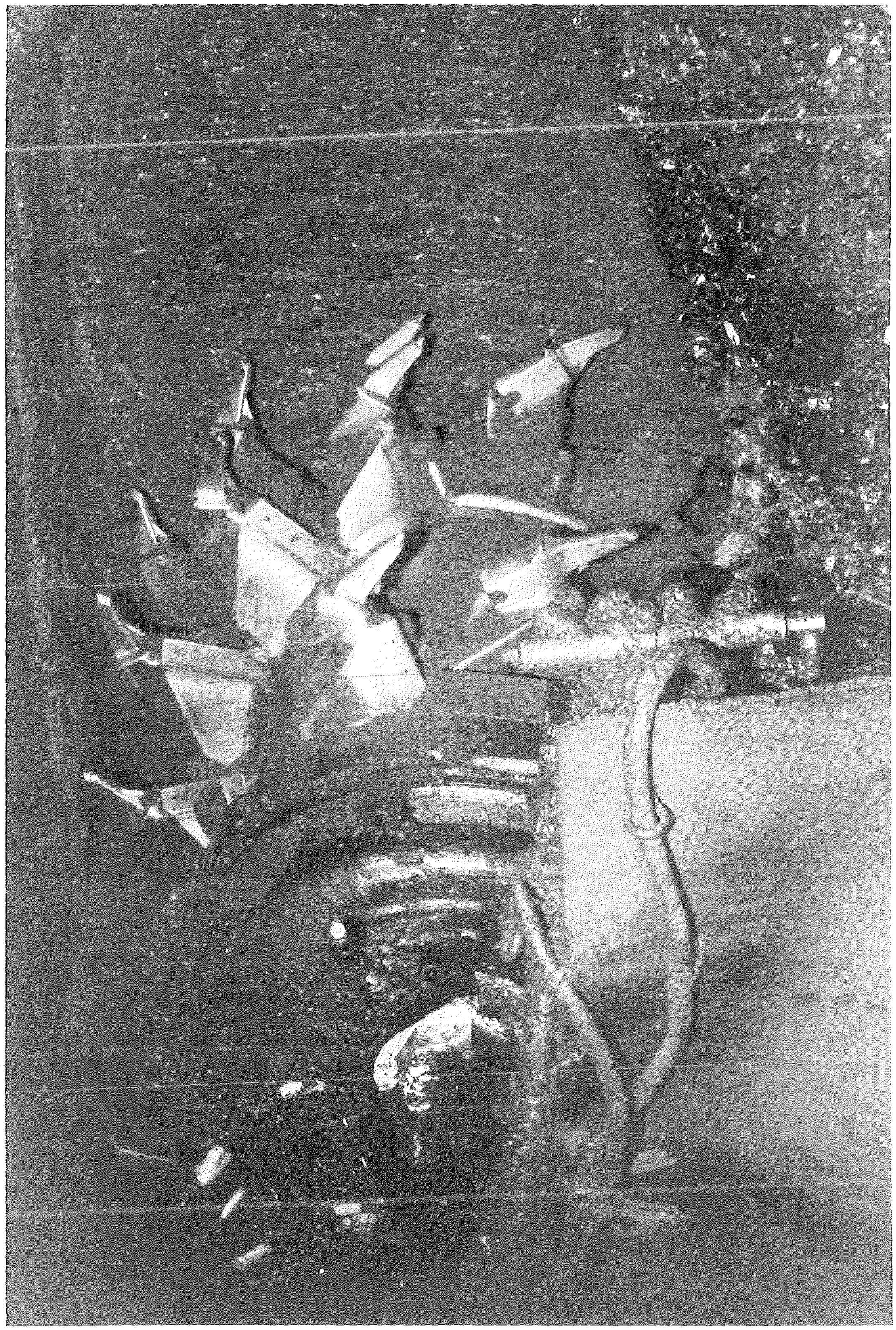


FIGURE 58. Stage-laced cutter drum

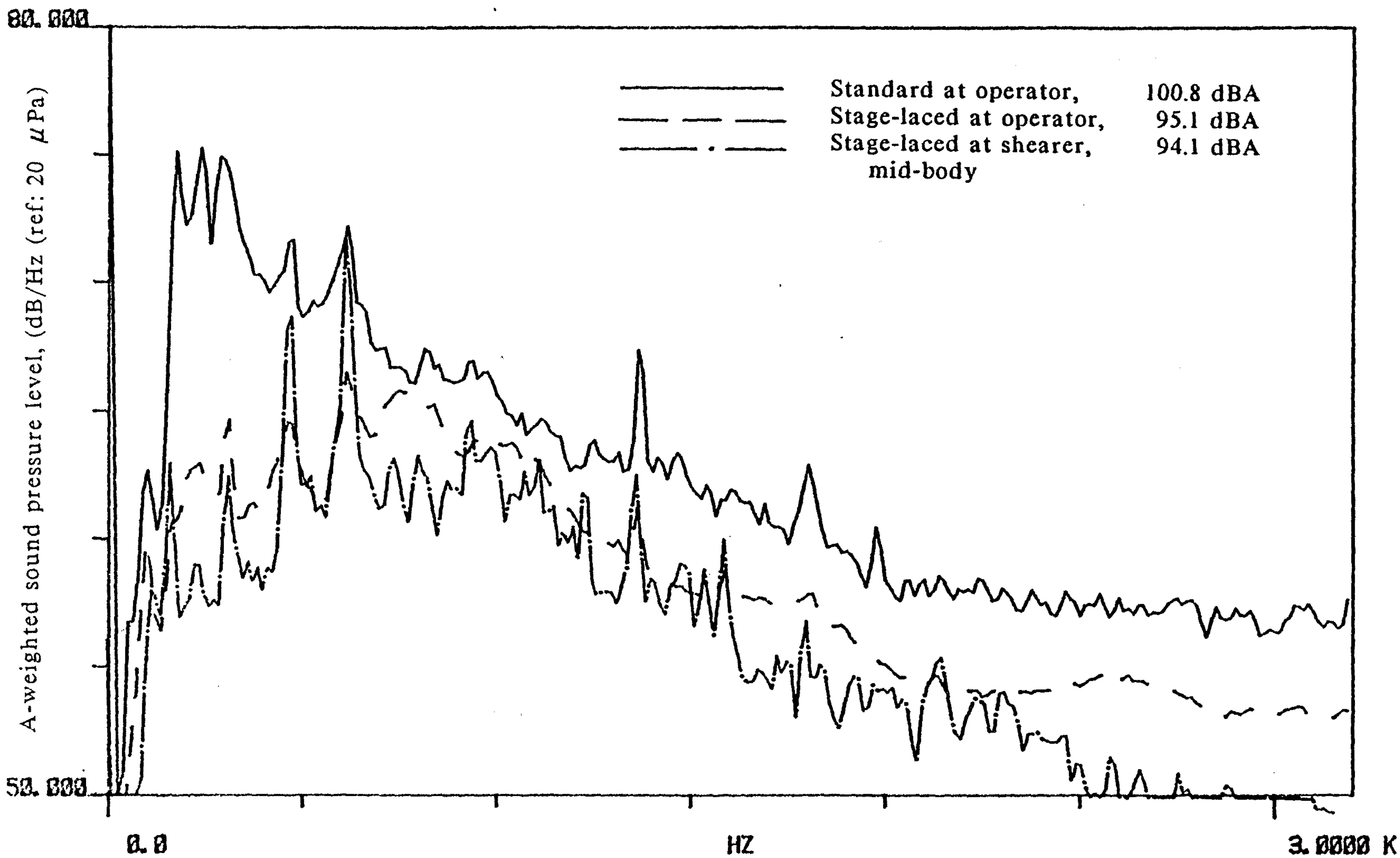
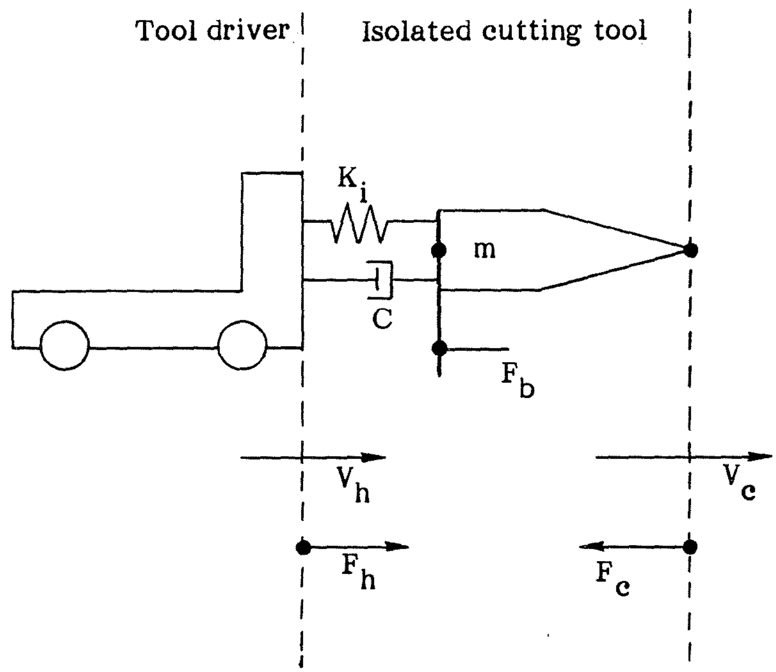


FIGURE 59. Longwall shearer noise



where K_i	- isolator spring stiffener	m	- mass of isolated cutting tool
C	- isolator damping coefficient	V_c	- velocity of cutting tool tip
V_h	- velocity of tool driver	F_b	- force on the cutting tool
F_h	- force at tool driver	F_c	- force at the cutting tool/coal interface

FIGURE 60. Single-degree-of-freedom isolated cutting tool

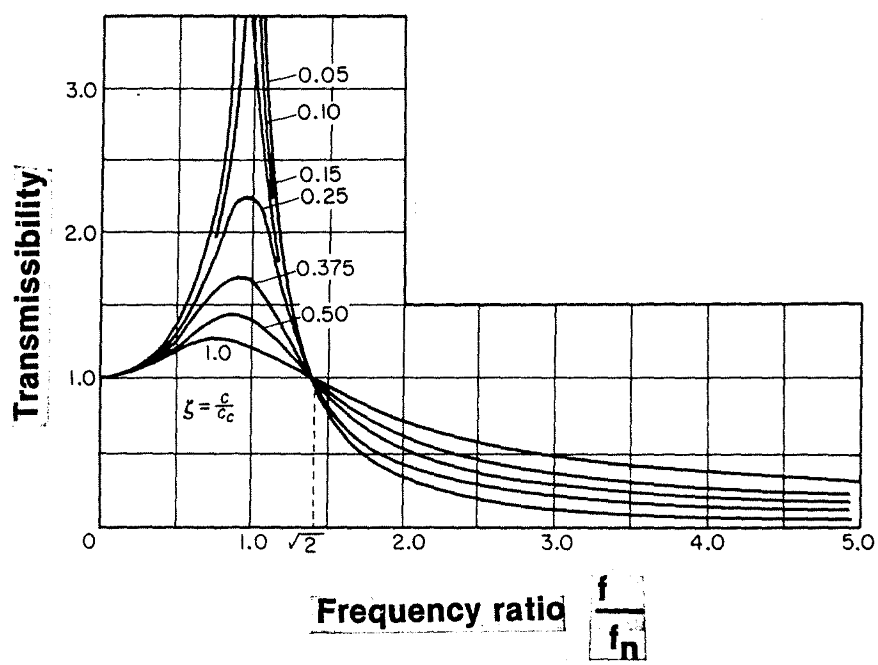


FIGURE 61. Transmissibility

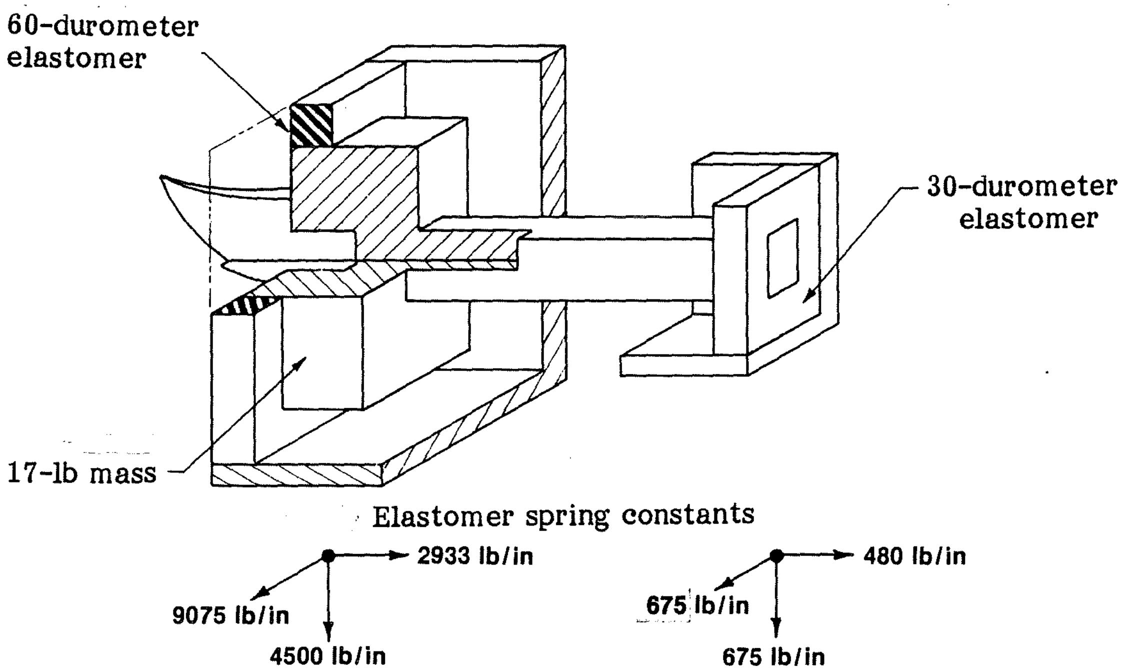


FIGURE 62. Isolated cutting tool for LCA tests

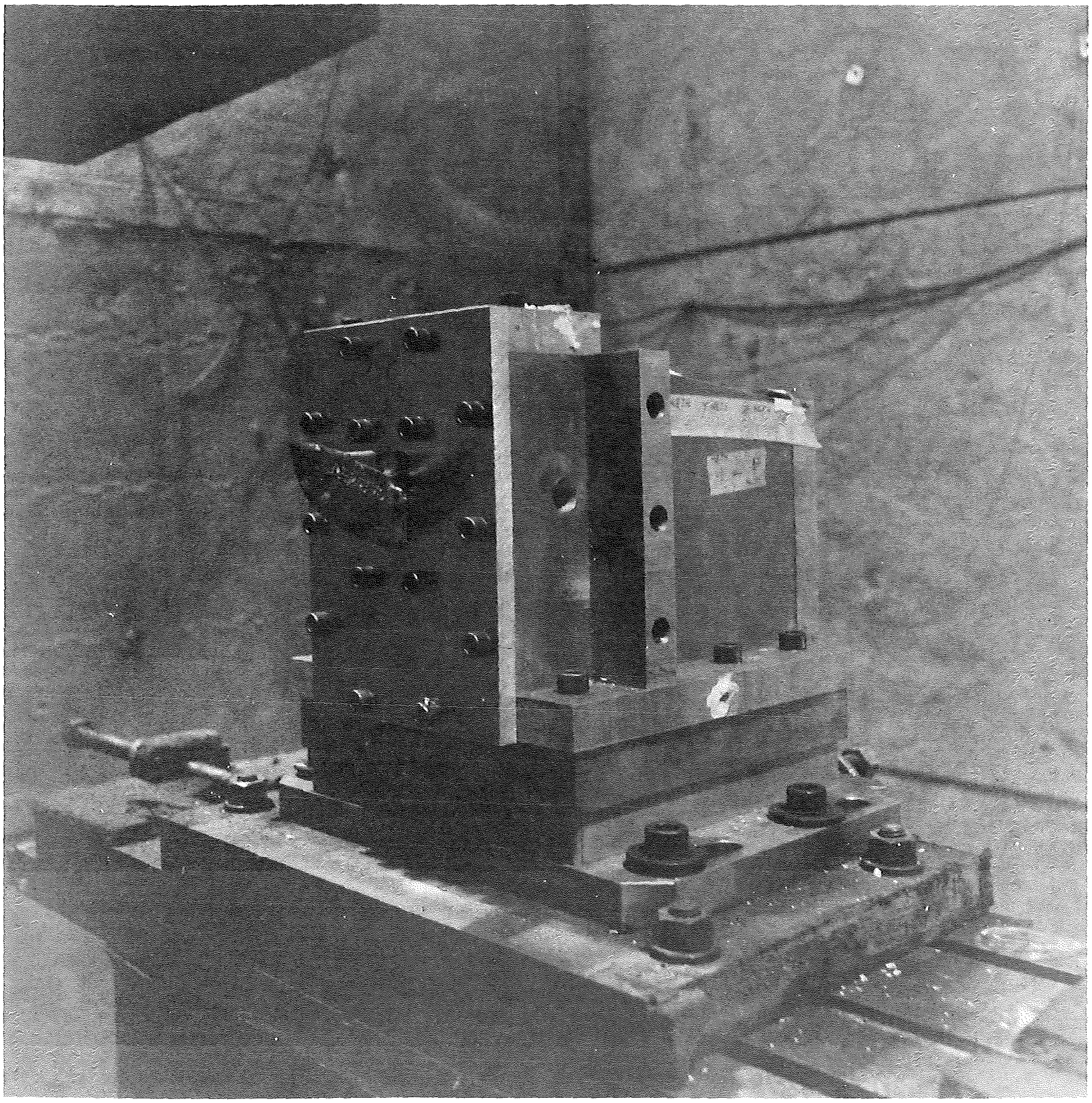


Figure 63. Rigid cutting tool for LCA tests

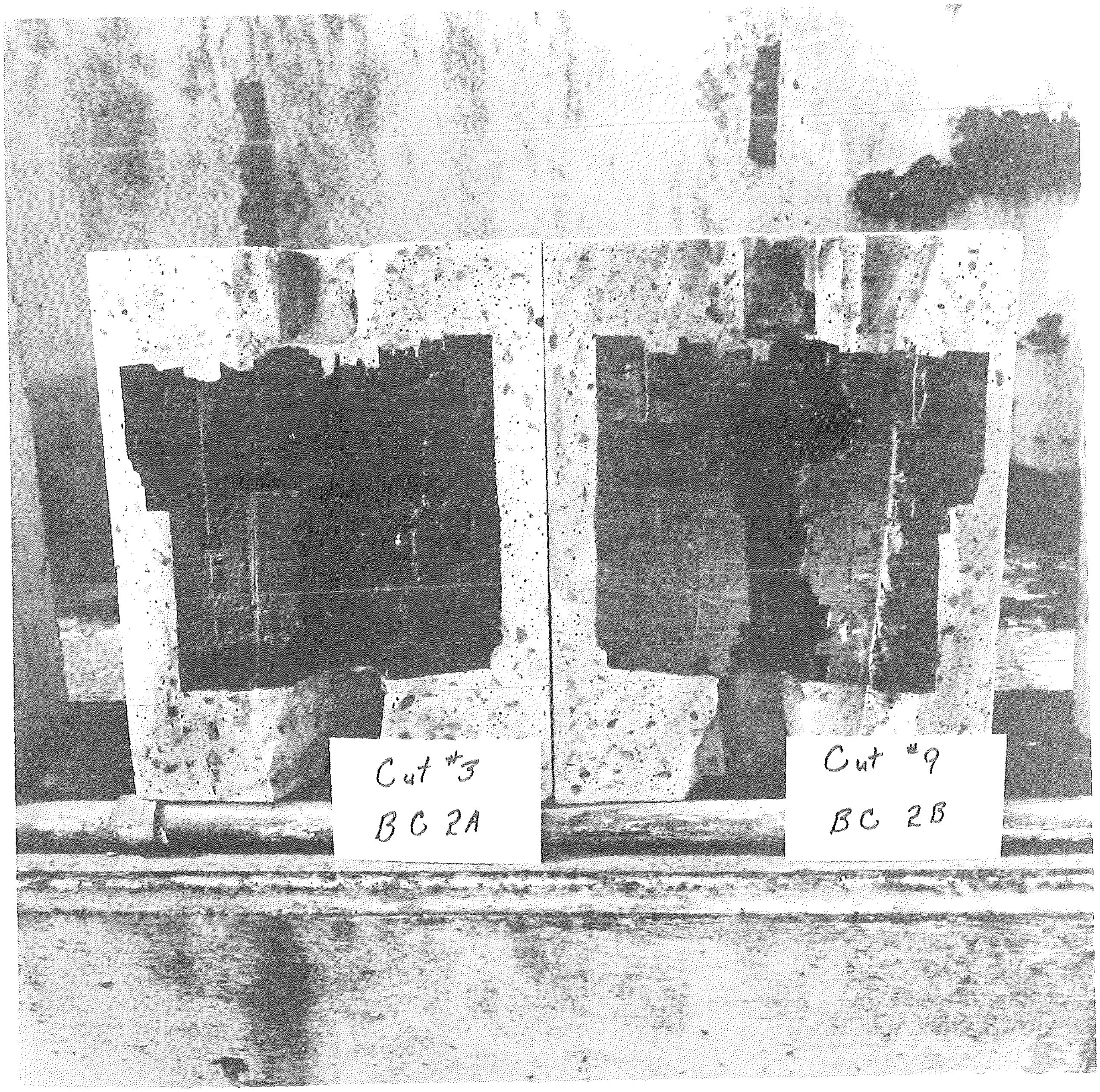


Figure 64. Typical coal samples for LCA tests

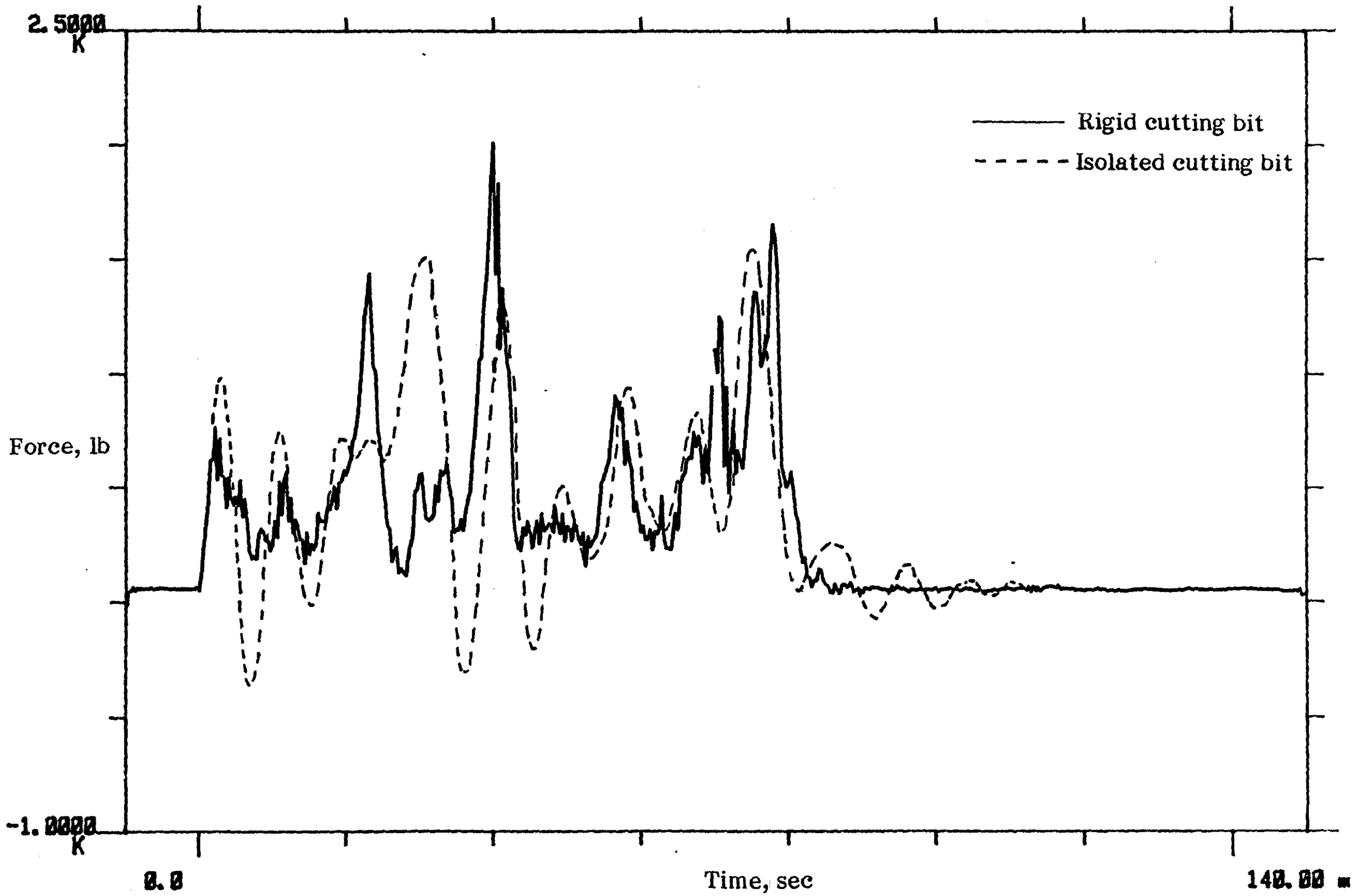


FIGURE 65. Force time histories: LCA tests

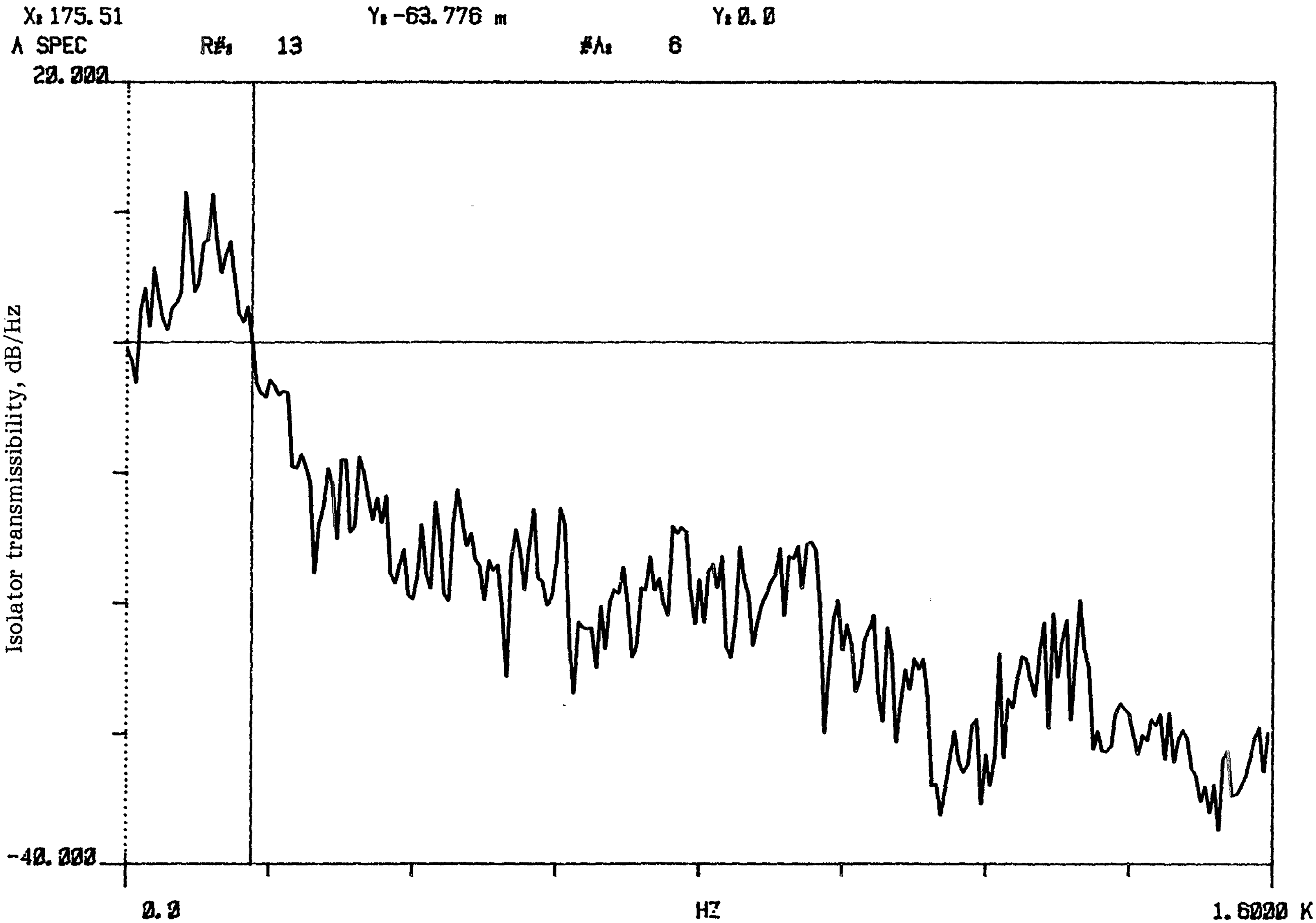


FIGURE 66. Isolator transmissibility: LCA tests

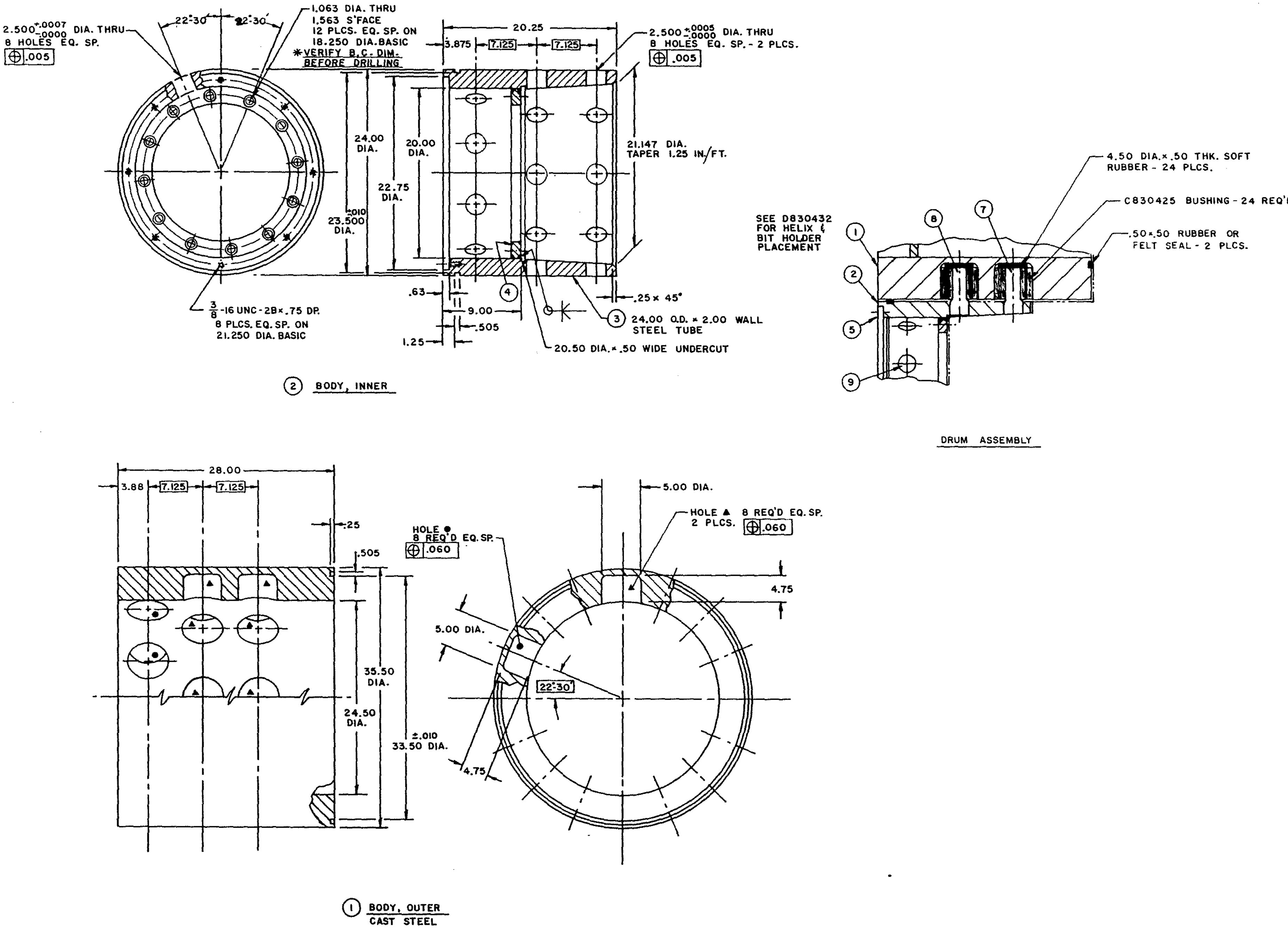


Figure 67. Stage-Laced longwall shearer cutter head with whole head isolation

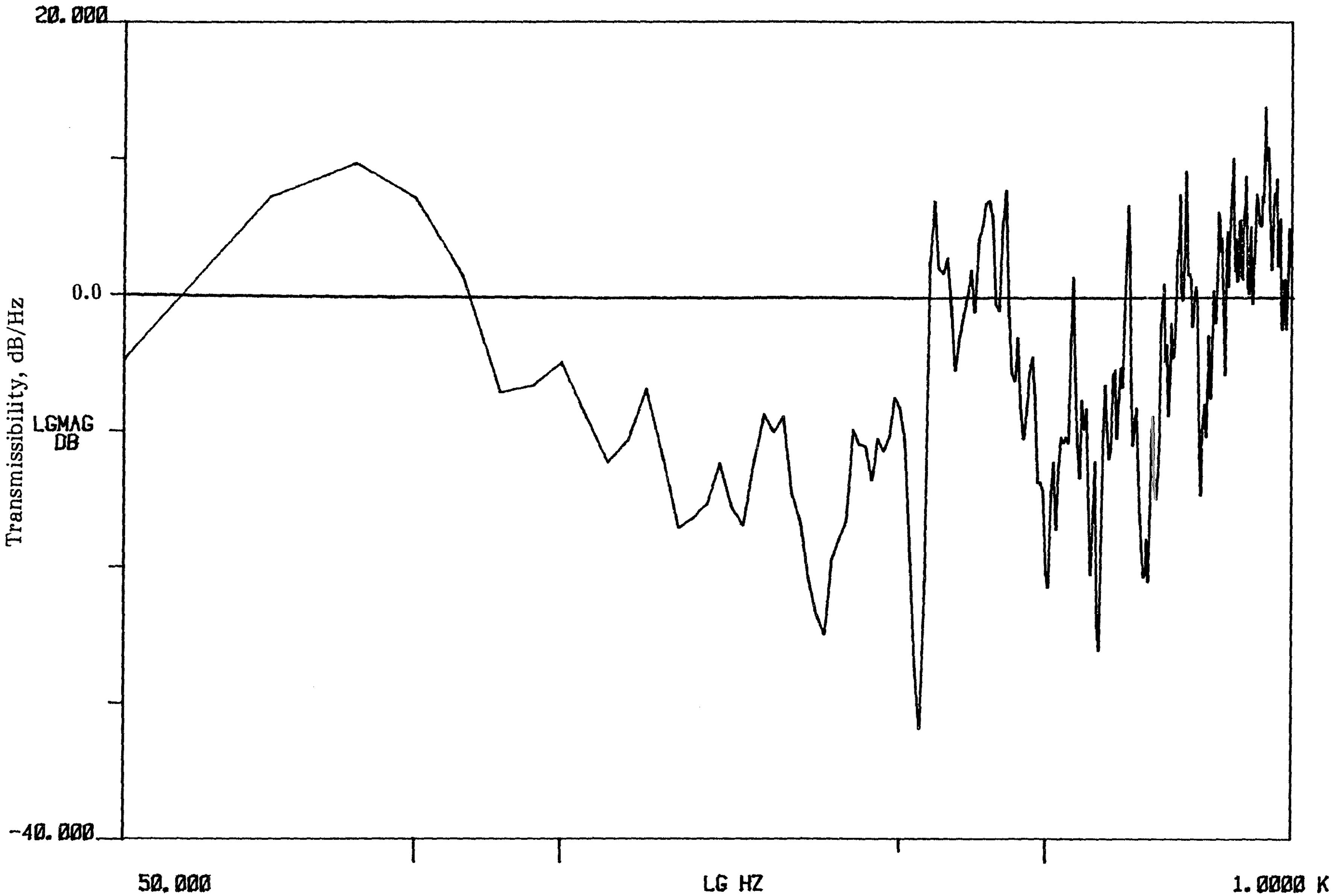


FIGURE 68. Transmissibility of isolated cutter drum from USBM longwall test

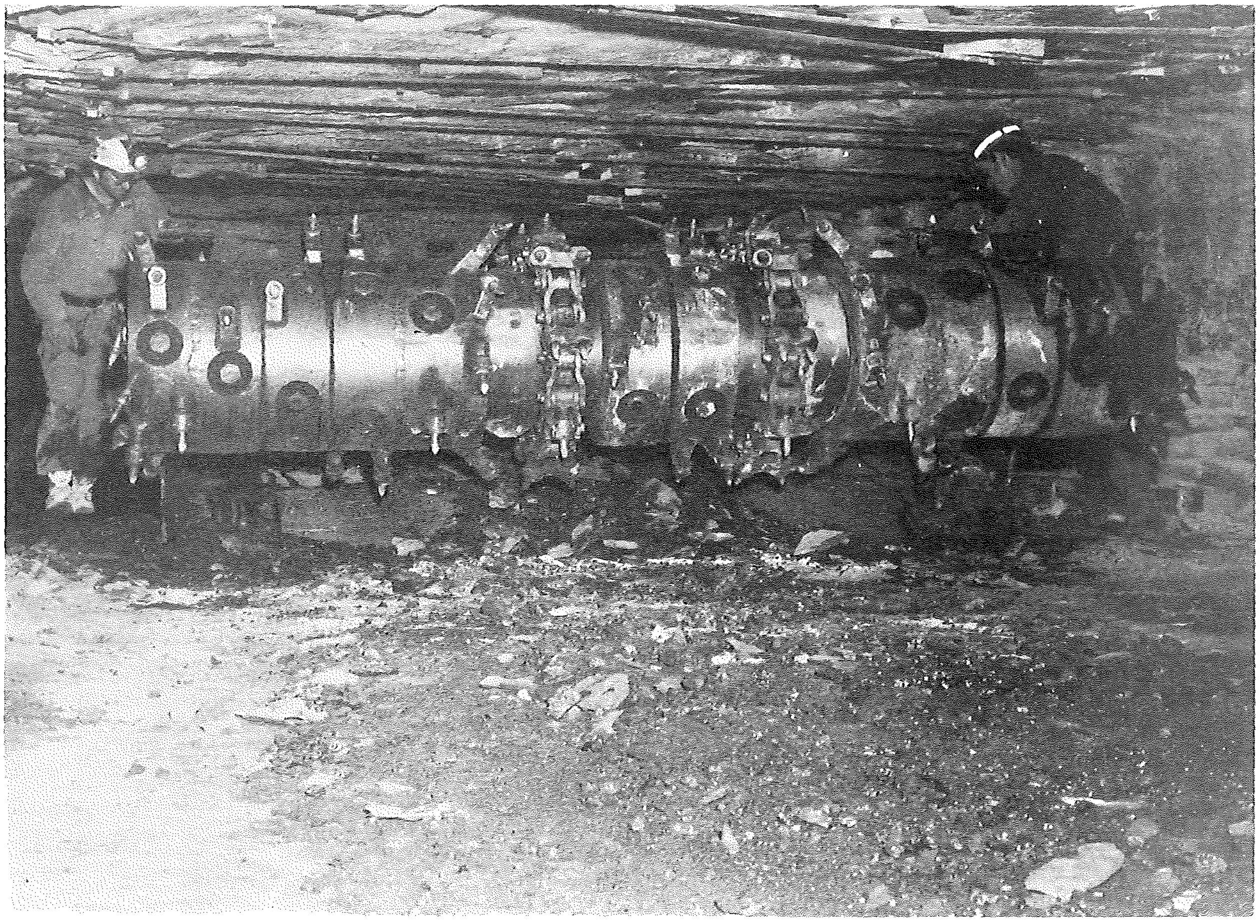


FIGURE 69. Continuous miner isolated cutter head

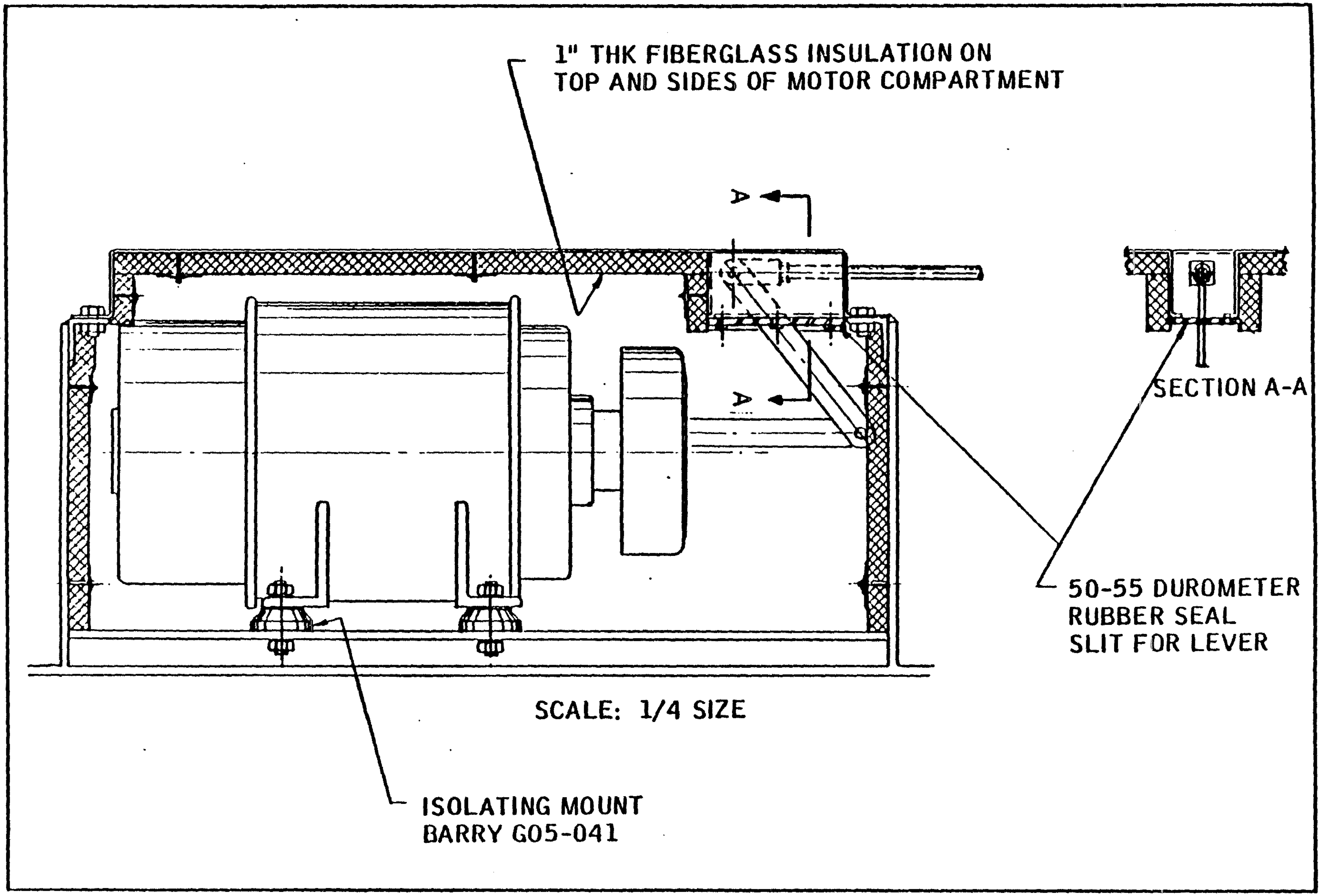


FIGURE 70. Electric motor enclosure

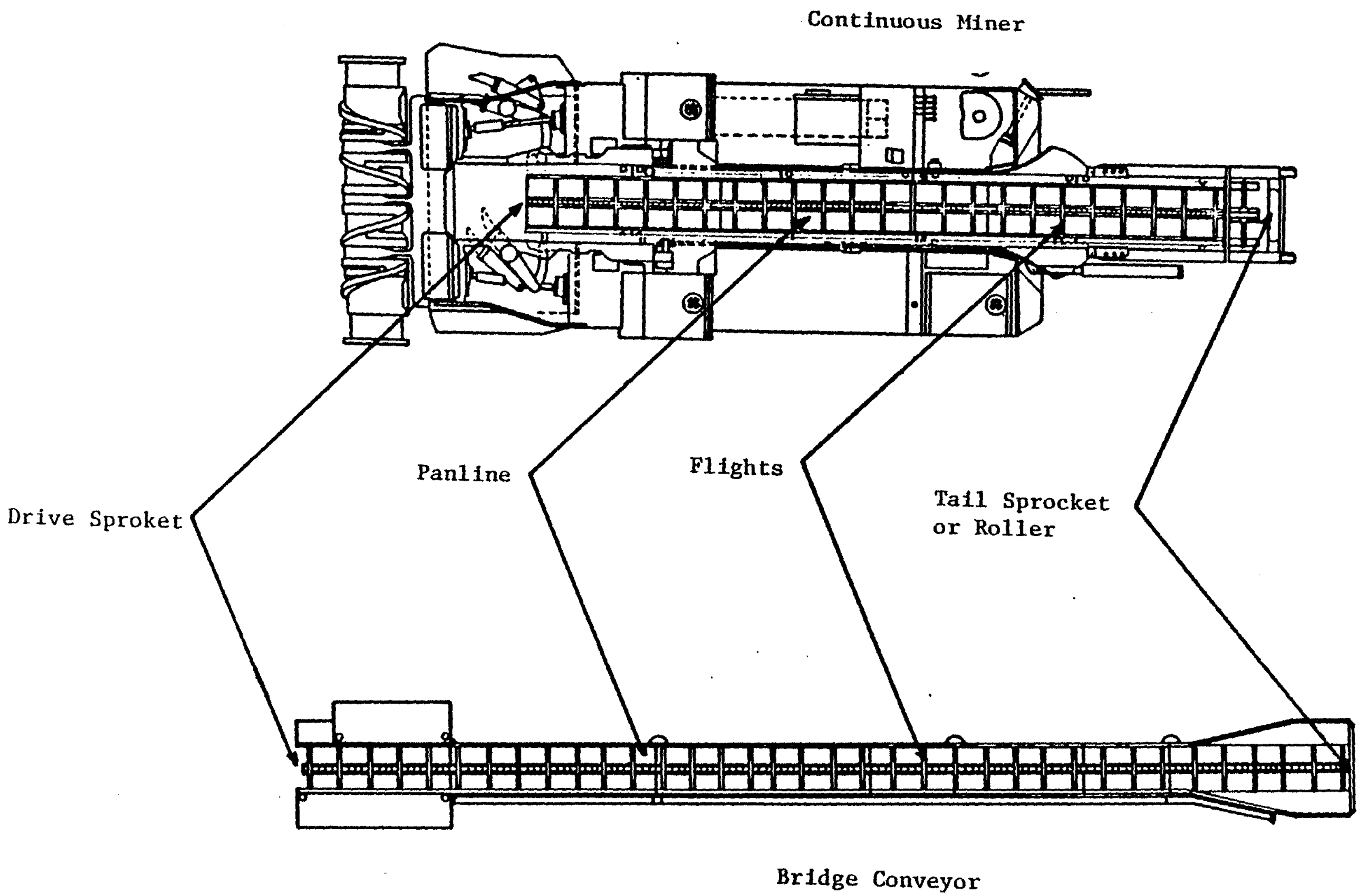


FIGURE 71. Continuous miner and bridge conveyor

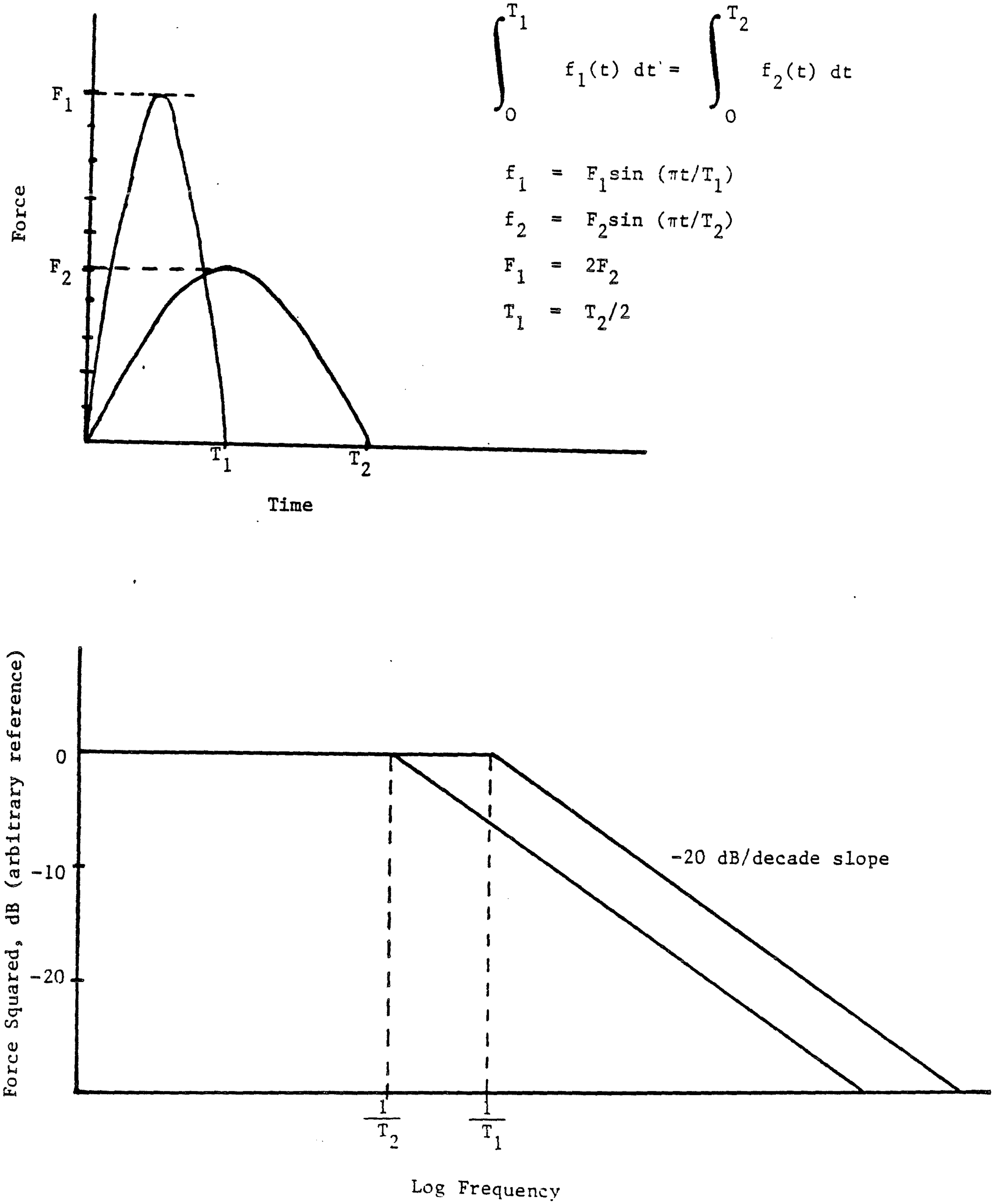


FIGURE 72. Influence of object compliance on impact force dynamics

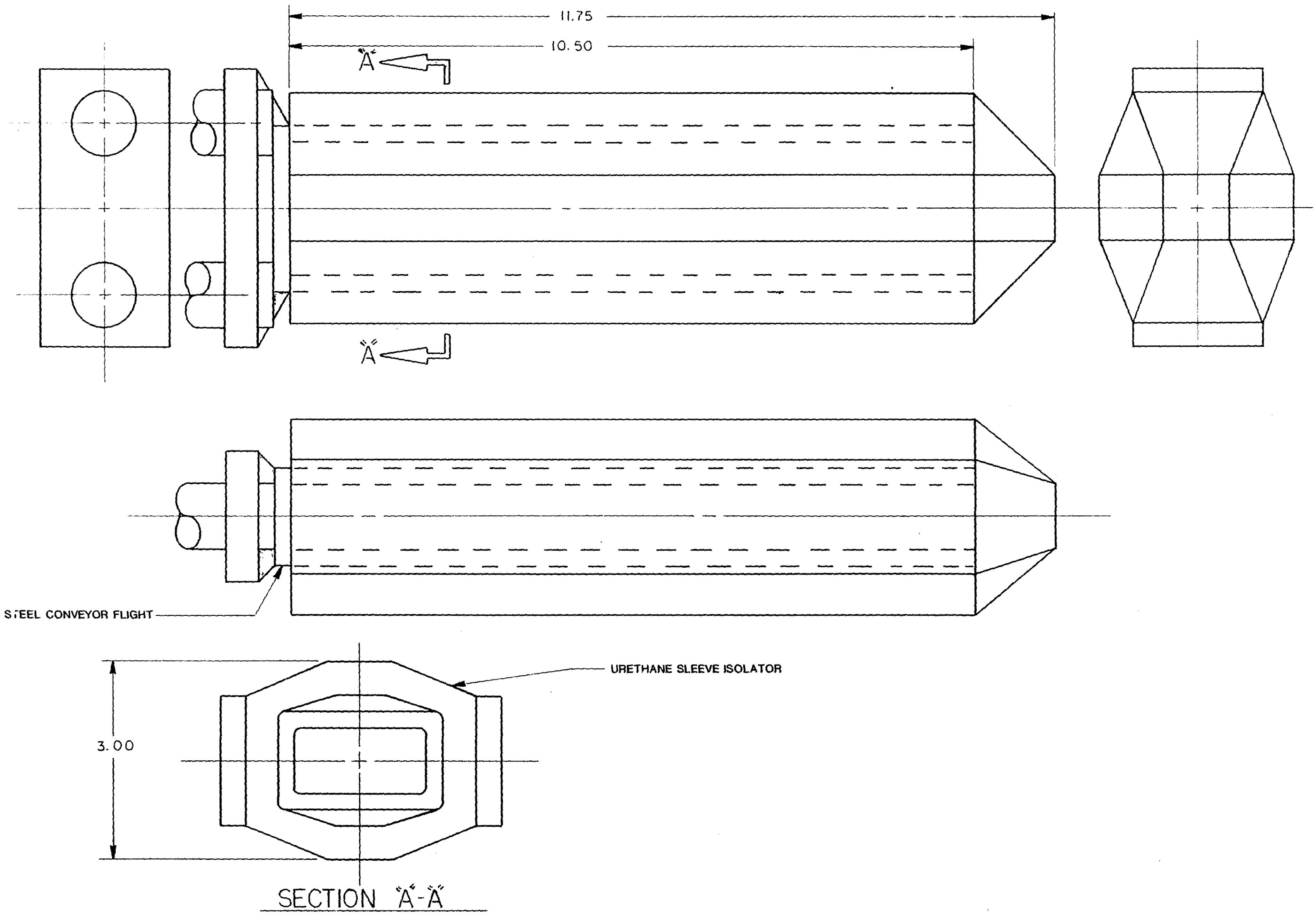


FIGURE 73. Isolated conveyor flight

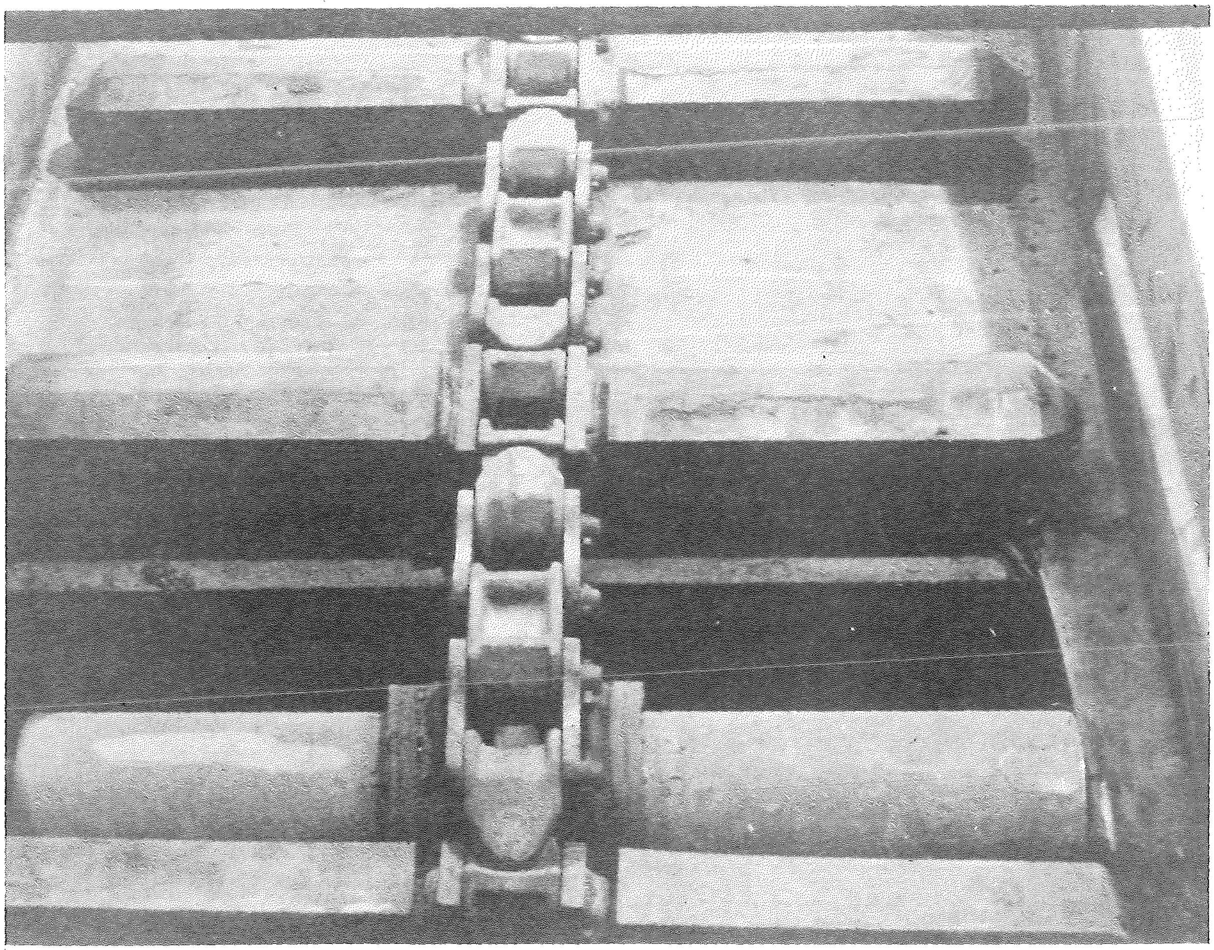


Figure 74. Urethane sleeve isolated conveyor flight

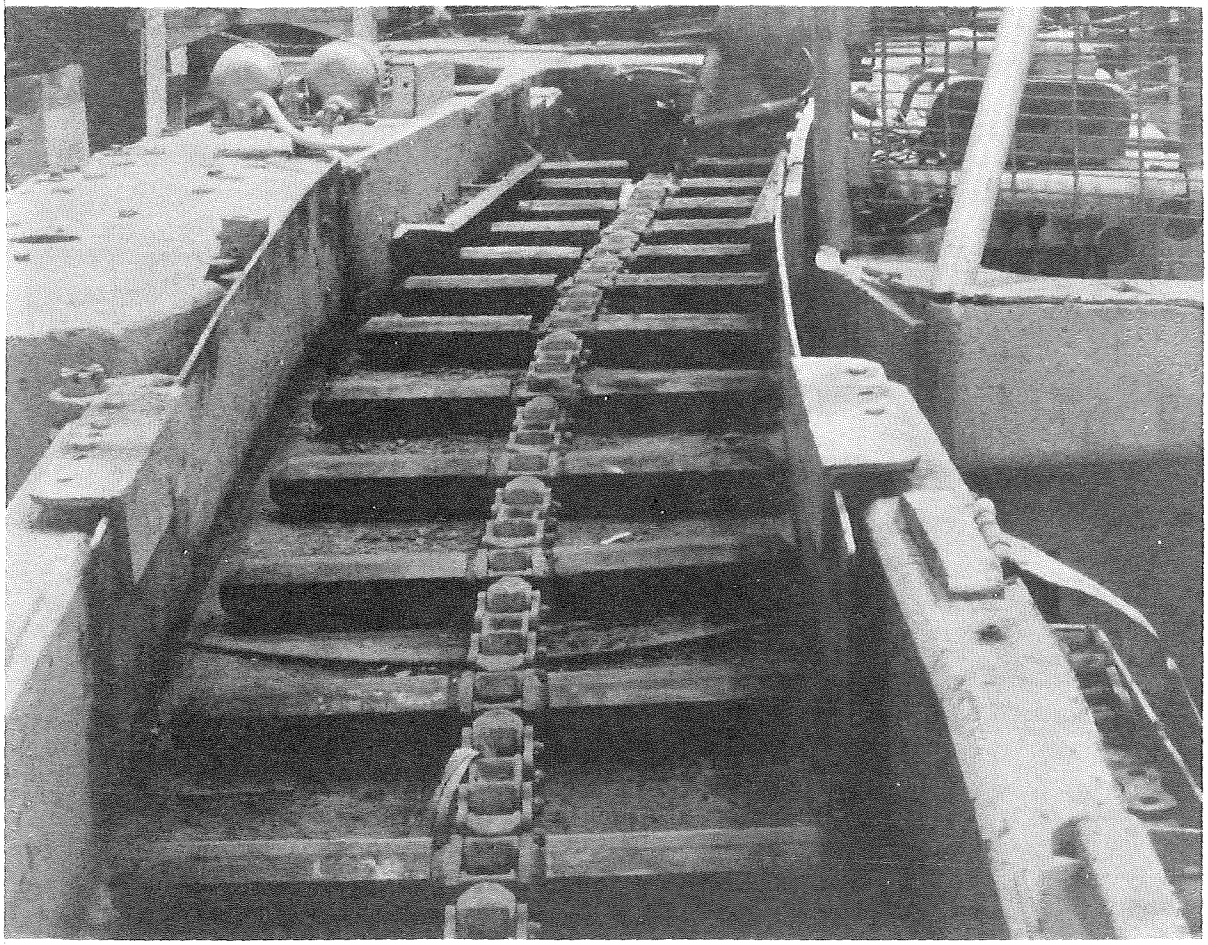


Figure 75. Urethane sleeve isolated conveyor flight

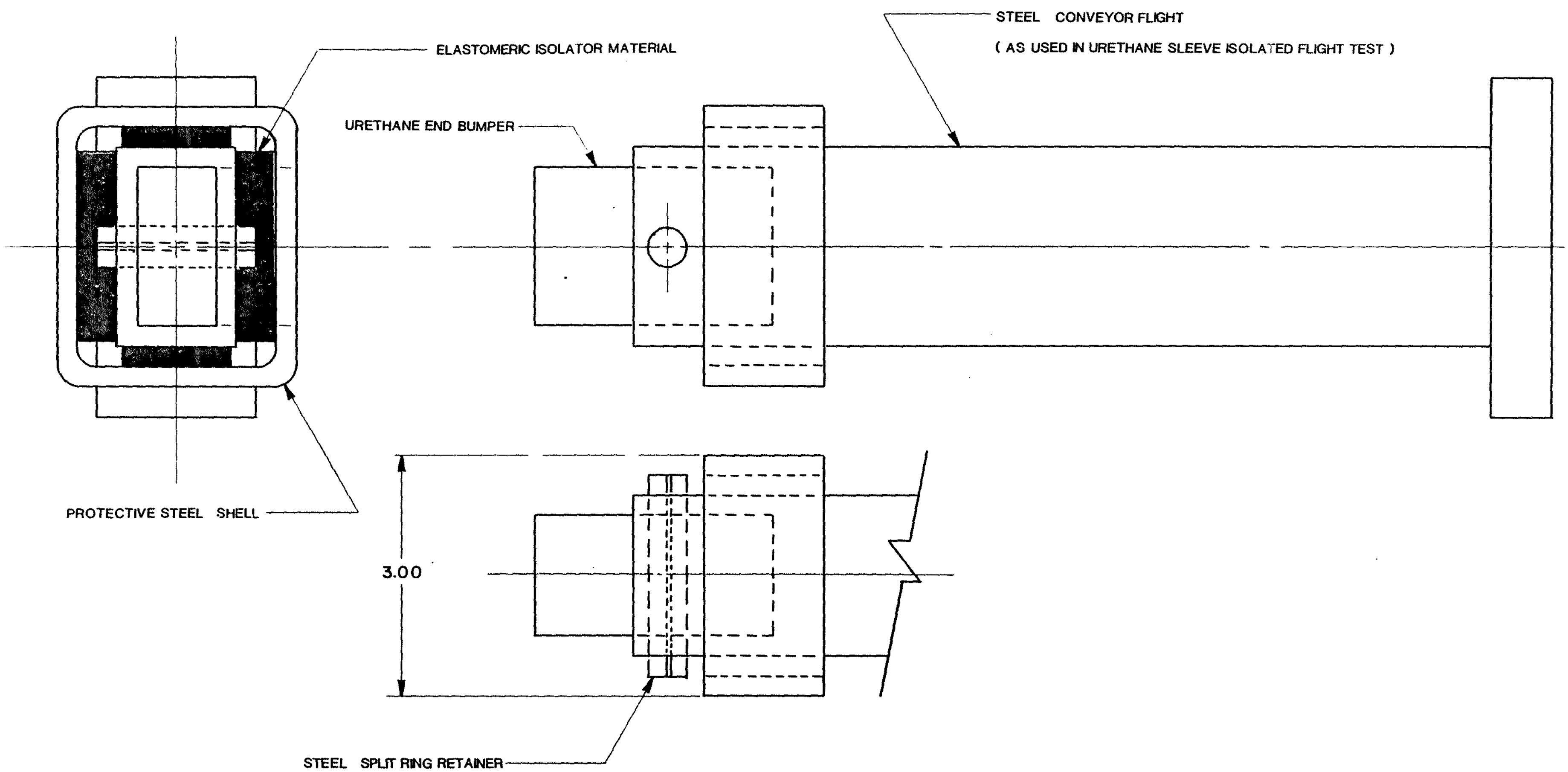


FIGURE 76. Steel Shell Isolated Flight

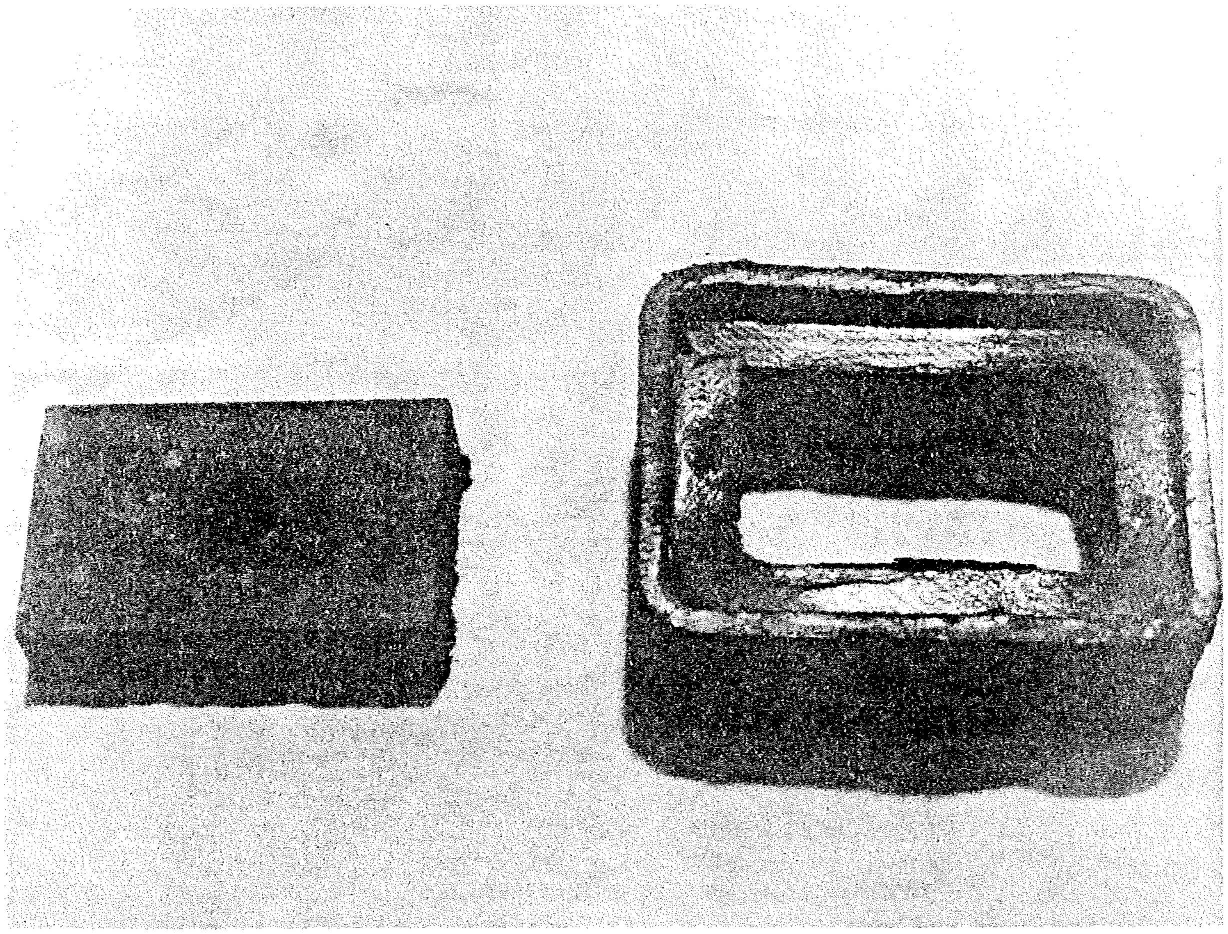


Figure 77. Steel shell isolation ring and end bumper prepared for installation on steel conveyor flight

Appendix A

REPORT ON U. S. MINE VISITS

I. Mine A

INTRODUCTION

The Mine A had been identified as a state-of-the-art longwall system.[1] This was the only low-seam plow operation identified in the initial study.[2] The mine visit enabled a two-man measurement team to record noise and vibration data on an operating longwall system. The mine was visited on April 10, 1979.

BACKGROUND

The mine produces nearly 1.2 million tons of coal annually and supplies a nearby generating station. Daily production of 6,000 tons of steam coal is performed on a three-shift basis. The union mine, which employs approximately 450 men, uses both continuous and longwall methods.

GEOLOGY

The Upper Freeport Seam, which reaches a thickness of 78 inches at Mine A is mined. The overburden is 650 feet thick. This seam of low sulfur coal contains some narrow rock bands one foot from the roof. A seam analysis is shown in figure A1.

MINE LAYOUT

Mine A uses a standard retreating system with a right-handed face. Panel length is normally 3,400 feet; however, this panel, one left off fifteen east north, was only 1600 feet long. A layout of the panel is shown in figure A1. Ventilation on the face is

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[1] Hoop, T., and R. M. Slone, Jr. Noise Study of Longwall Mining Systems, Phase I Report. Wyle Laboratories Technical Memorandum TM 79-4, Mar. 1979, pp. 21-22.

[2] Hoop and Slone; see footnote 1, p. 23.

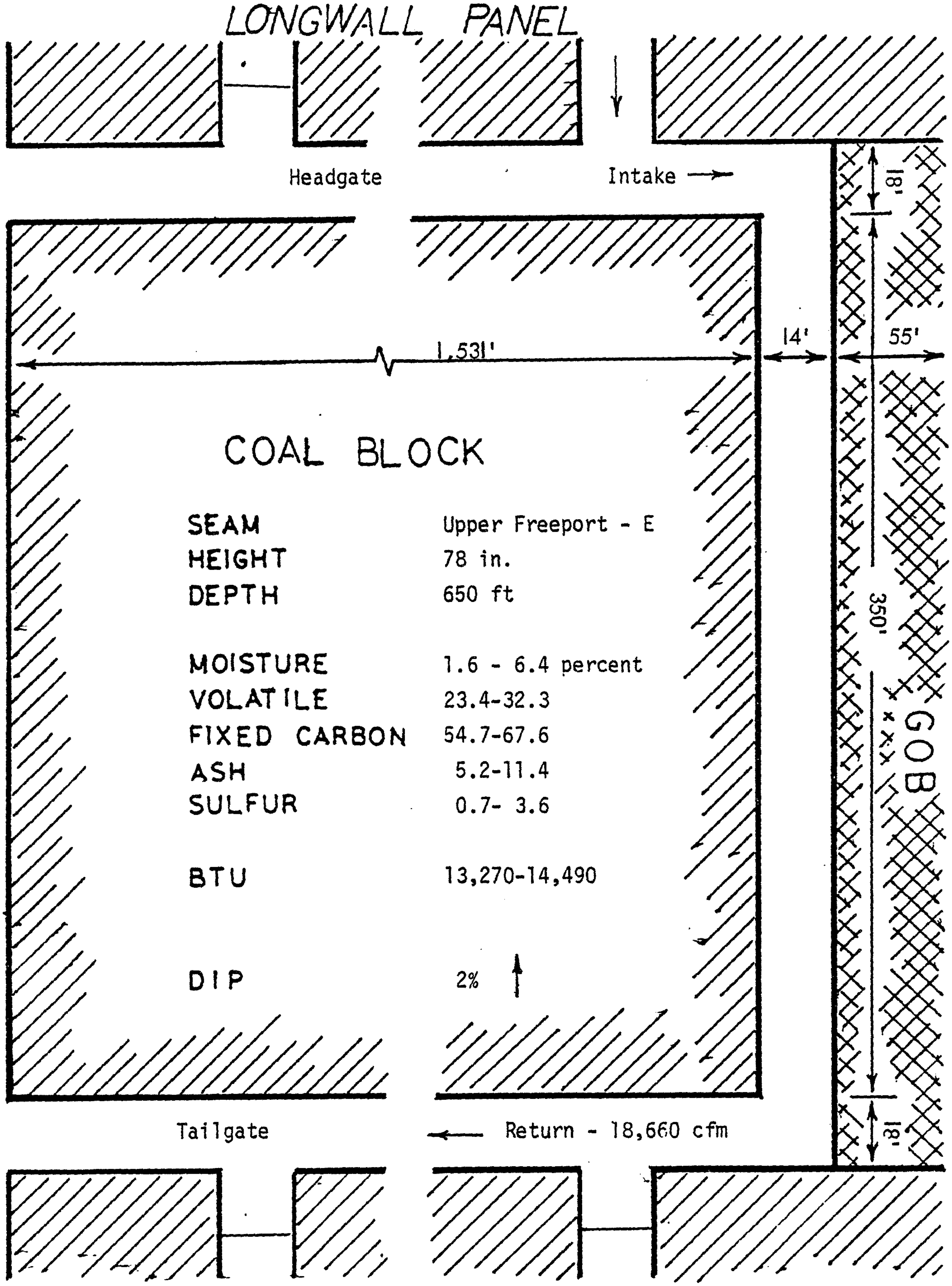


FIGURE A1. Mine A seam analysis and longwall panel layout brought in at the last open crosscut in the headgate entry and is exhausted out the tailgate. The quantity of air measured at the tailgate was 18,660 cfm. A diagram of the headgate is shown in figure A2, and the tailgate and face plans are shown in figure A3.

At the headgate, a 30-inch face conveyor dumps onto a high-speed, 28-inch stage loader. This 100-foot conveyor in turn dumps onto a bridge conveyor of matched capacity which then feeds a 36-inch panel belt. This "S" configuration allows the belt to be centered in the entry with extra room along the face conveyor for the plow to cut out. A master control panel is mounted along the panline anchoring jack. From this position, the headgate operator controls the plow, the conveyors, and the pump units, which are on the right-hand side of the belt by the tail piece. Roof clearance over all headgate machinery is normally in the two-foot range.

Unlike most systems, the mine uses 100% hydraulic oil for the support fluid. Each of the three pumps run at a pressure selected for its job function. Ram feed is 700 psi, frame-down pressure is 1,100 psi, and up pressure is set at 2,200 psi.

The 350-foot, 30-inch face conveyor is protected by 75 double frame supports. Each of five support operators is assigned a section of 25 frames. The Westfalia 7-26 gleithobel plow is powered by two 125-hp motors and is hauled by a 26x92-mm trapped chain. No sumping procedure is required, and the plow is equipped with water sprays for dust suppression. Machinery used in this mine is summarized in table A1.

MINING CYCLE

1. The plow takes a 2¼-inch cut from the headgate to the tailgate.
2. The face conveyor is automatically snaked by continuous pressure rams as the face is cut.
3. The plow is stopped at the tailgate, direction is reversed, and a 2-inch cut is taken toward the headgate.
4. Face supports are advanced according to roof conditions, but they must be advanced at least once after every six cycles.
5. No sumping is necessary since the plow cuts out into both entries.

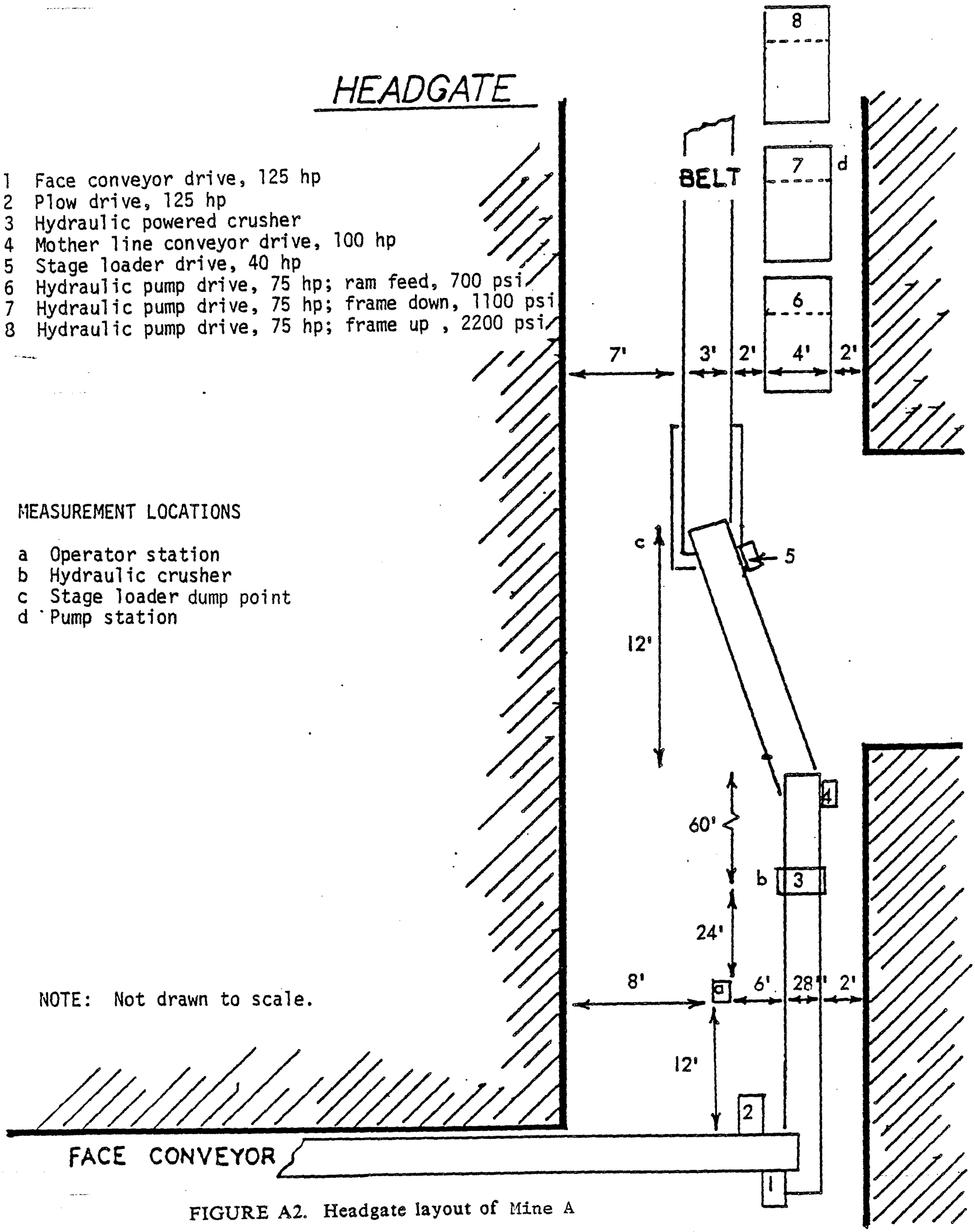
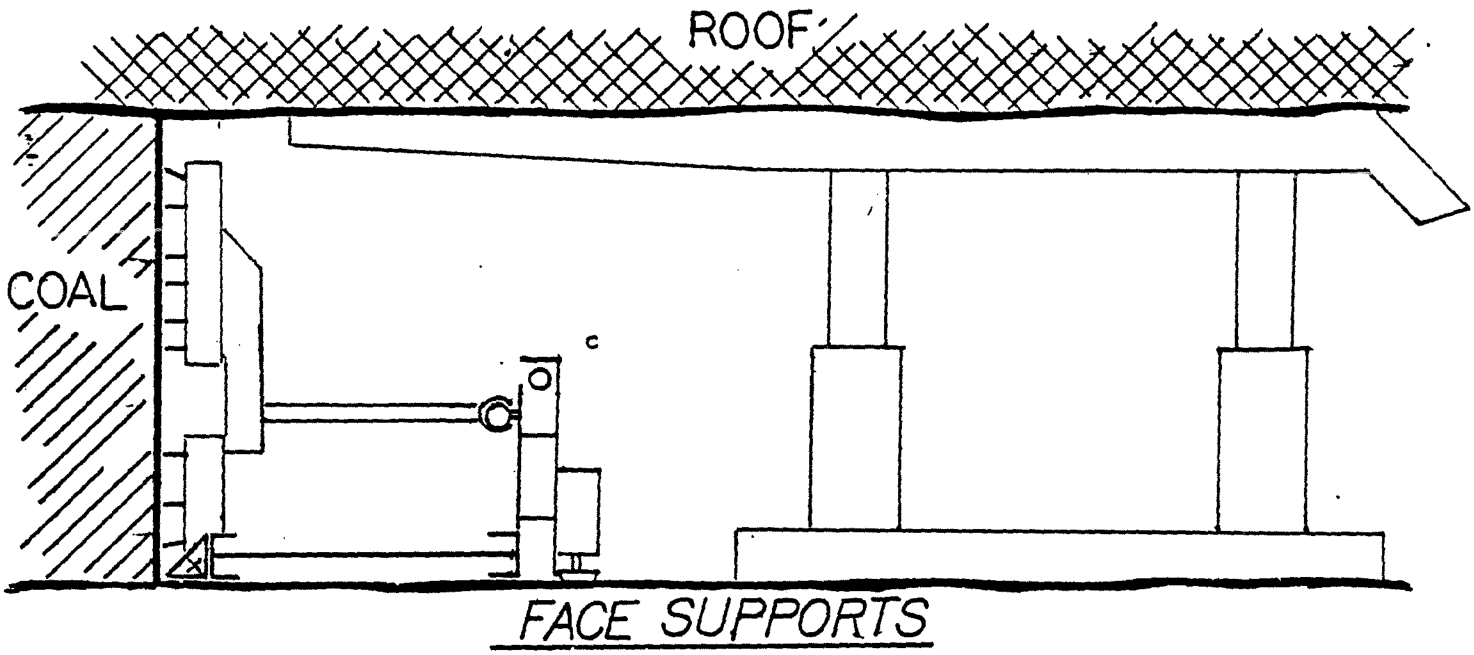


FIGURE A2. Headgate layout of Mine A



Panzer Kl.1, double frame, 4x175-ton

Measurement Locations

a Operator station
b Tail drive
c Chock line position

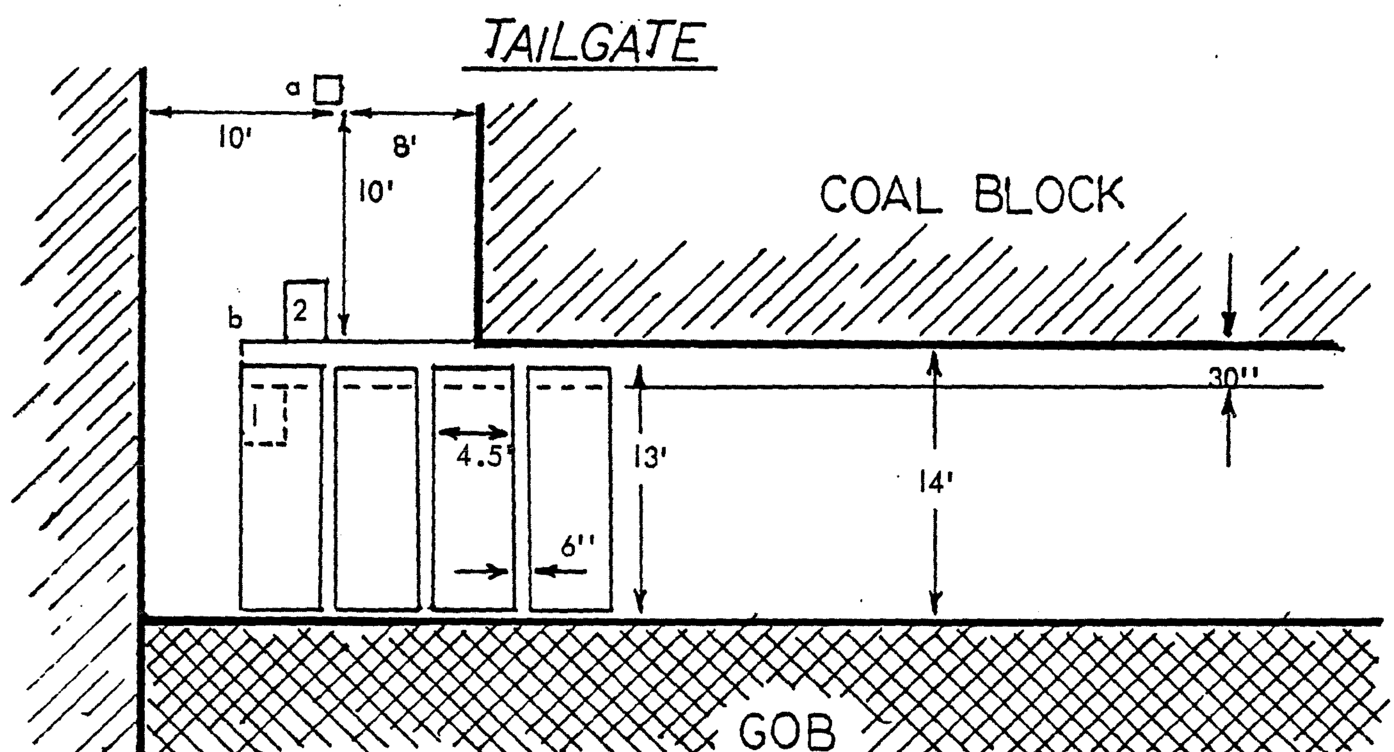


FIGURE A3. Mine A tailgate layout and face plans

TABLE A1. Machinery used in Mine A

Machinery, manufacturer, model, and type	Power	Speed	Comments
Winning machine, Westfalia 7-26 gleithobel plow	2x125 hp	135 fpm	Trapped haulage chain 26x92 mm, 2-in. depth of cut.
Face conveyor, Westfalia PF 111-700, 30-in. conveyor	2x125 hp	150 fpm	Double inboard 22-mm chain, 1-m between flights.
Headgate conveyors:			
Westfalia, Mother Line, 30-in. conveyor	100 hp	180 fpm	3/4-in. triple chain.
Westfalia stage loader		180 fpm	
Face supports, Westfalia K1.1, 4-leg frame	175 ton/leg		Yield load, 7200 psi; working range, 44 in. to 68 in.
Hydraulic pumps, Reliance drive motor, Westfalia system	75 hp	160 gpm at 1800 rpm	3 units. Up pressure = 2,200 psi, down pressure = 1,100 psi, ram pressure = 700 psi.
Crusher	hydraulic 700-psi feed		5 rings with 4 bits.

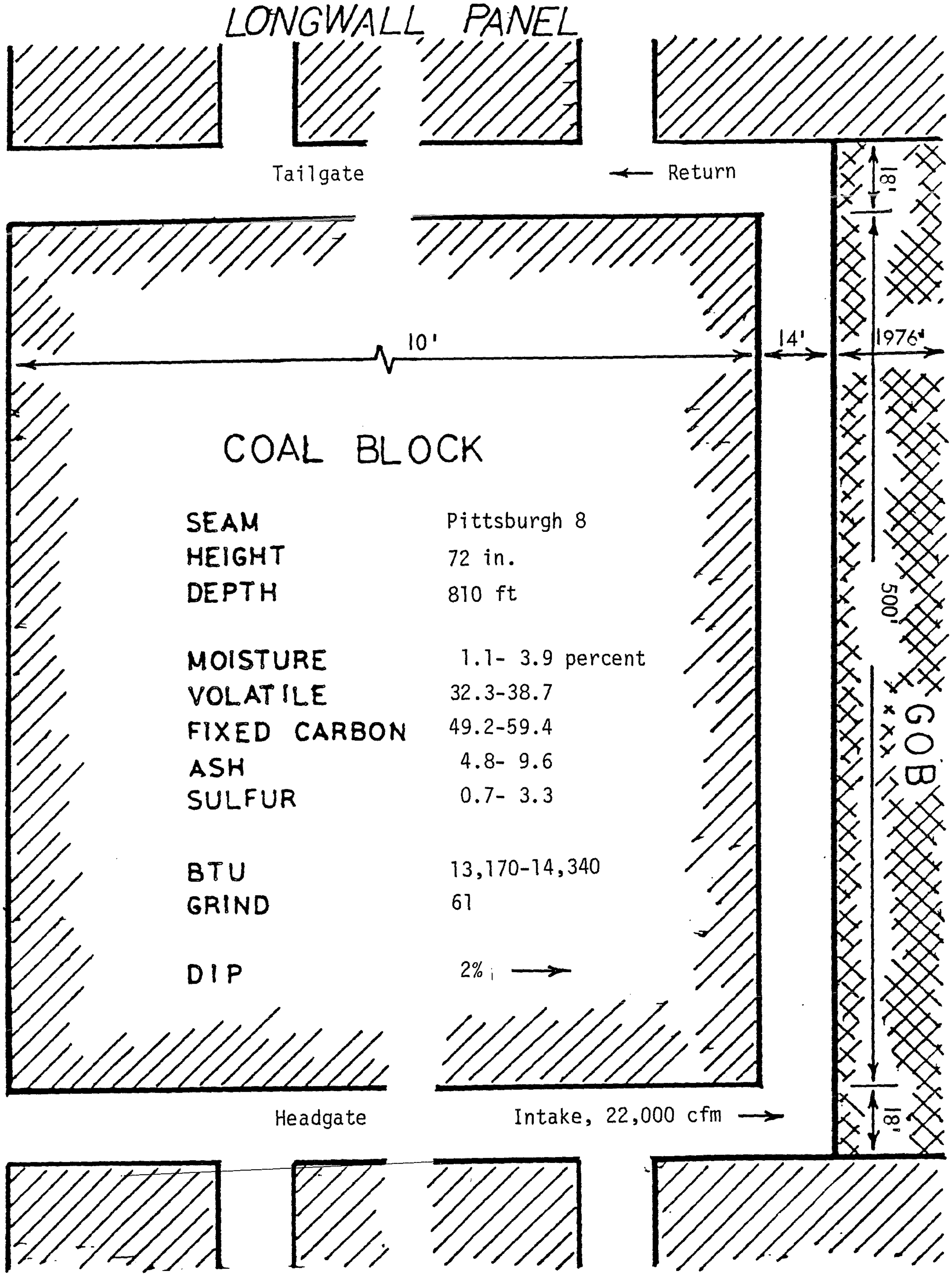


FIGURE A4. Mine B: seam analysis and longwall panel layout creating a drier working area at the headgate. Roof clearance over the headgate machinery is normally three feet. Figure A5 shows the headgate layout and figure A6 the tailgate and face plans.

The 500-foot face is protected by 99 Hemscheidt 2-leg shields, and an Eickhoff EDW-300L shearer is used to mine the coal. The single-strand coal chain is driven by dual 200-hp motors at a speed of 200 feet/minute. Dust suppression is by water spray. Dust problems are further reduced by mining only in the direction of the air flow. Machinery is summarized in table A3.

MINING CYCLE

1. The shearer cuts to the tailgate with the leading drum cutting the top 70 percent of the seam.
2. The leading drum is lowered and the bottom stump is cut as the shearer moves toward the headgate.
3. The shearer is trammed to the headgate without cutting.
4. The face conveyor is advanced starting at the tailgate.
5. Supports are advanced when the face conveyor is positioned.
6. The shearer is angled into the coal face starting at support 20 and increasing toward support 1.
7. The drums are repositioned, ready to begin the next cut.

MINE CREW

The crew is made up of 10 men, nine UMWA workers and one company foreman. Two men operate the shearer while four additional men move the supports. Coal is cut in only one direction, thus the entire length of the panline is advanced at the same time. The mechanic and section foreman perform safety and preventive maintenance until a breakdown occurs. A supply man is used periodically. Workers and face duties are listed in table A4.

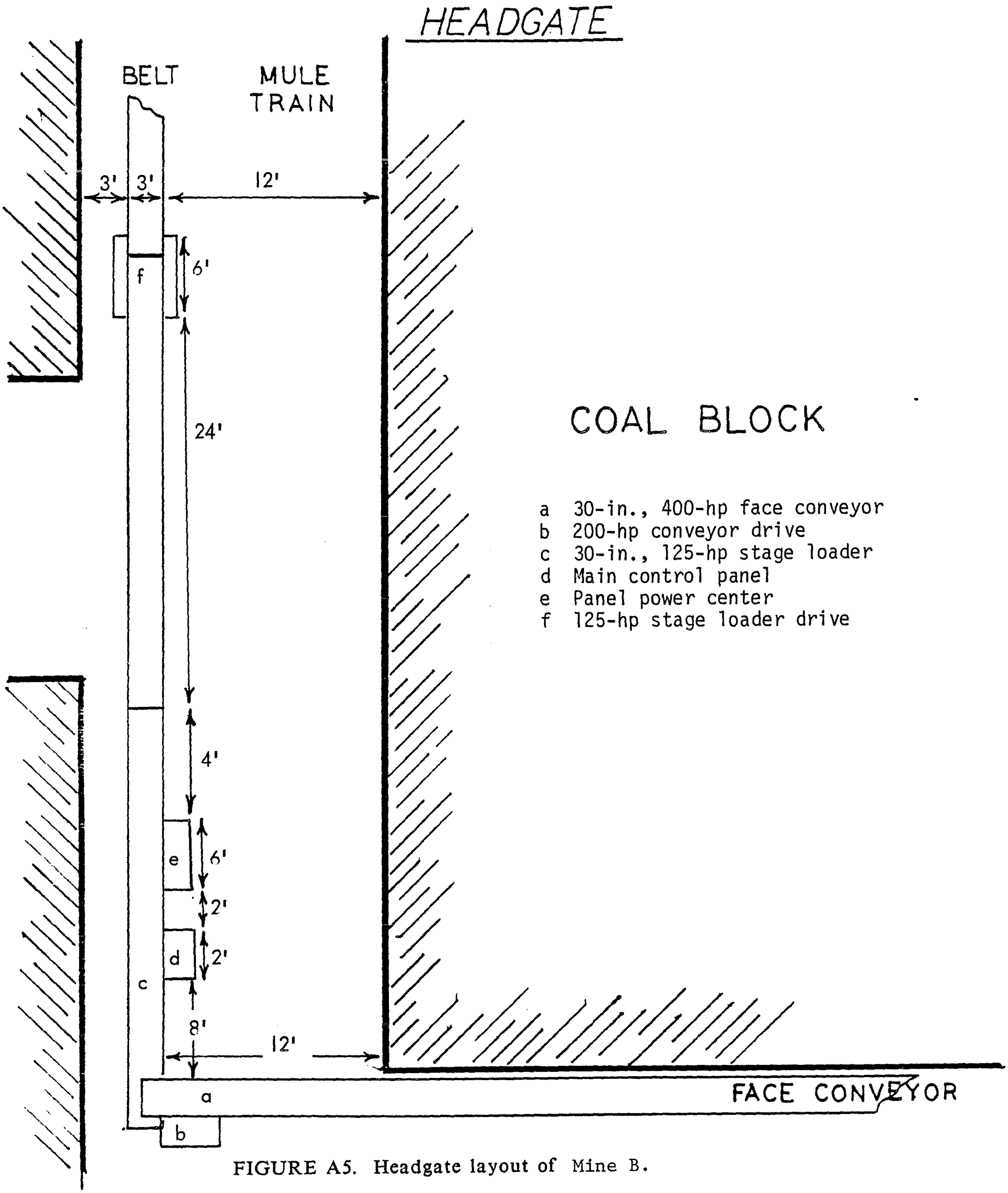


FIGURE A5. Headgate layout of Mine B.

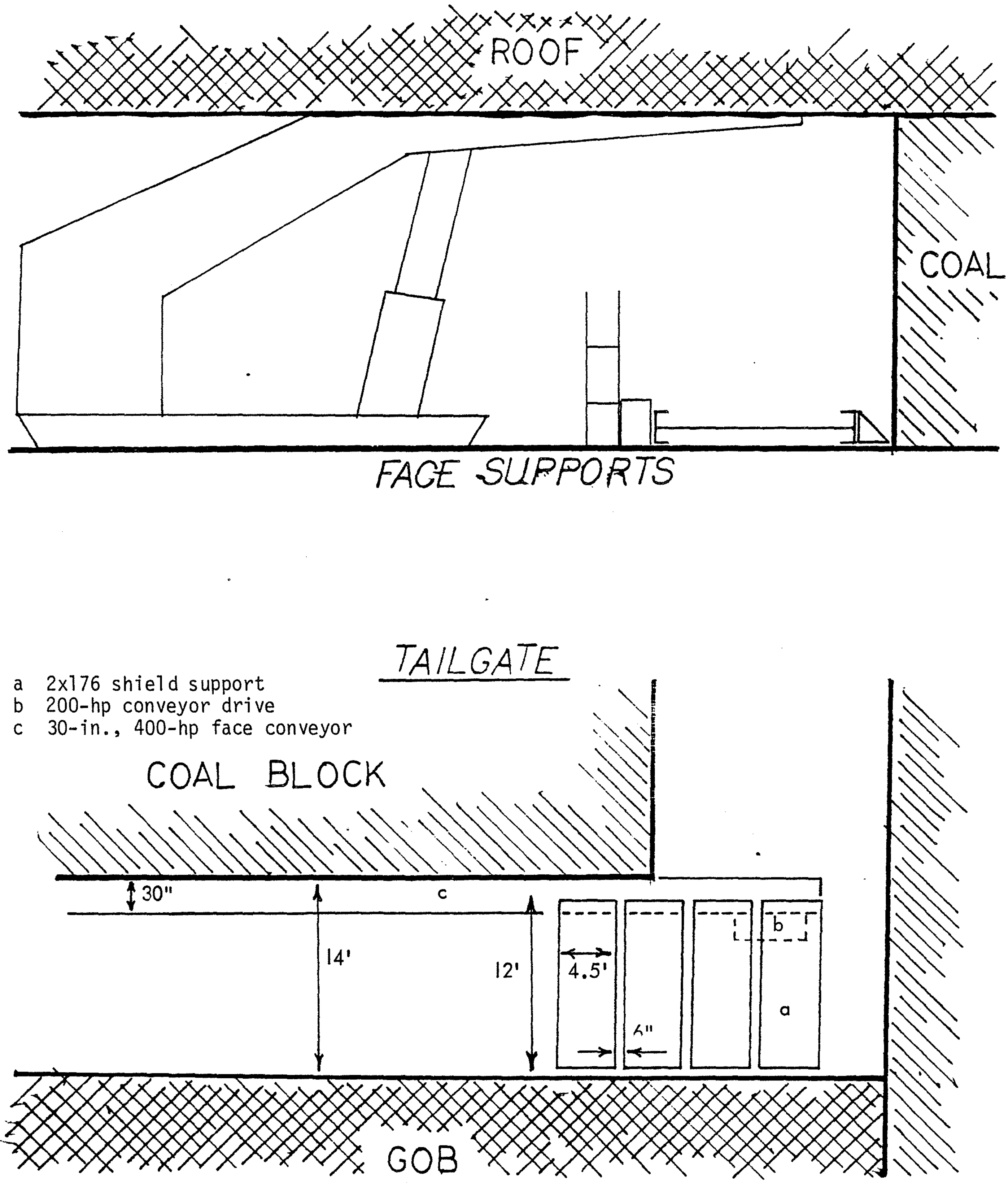


FIGURE A6. Mine B: tailgate layout and face plans

TABLE A3. Machinery used in Mine B

Machinery, manufacturer, model, and type	Power	Speed	Comments
Winning machine, Eickhoff, EDW-300-L, double drum	300 kw	20 fpm; 52 rpm, drums	30x60-in. drums, Eickotrack haulage.
Face conveyor, Eickhoff, EKF, single center	2x200 hp	220 fpm	850-ton/hr capacity, 30x92-mm chain, 30-in. pan width.
Headgate conveyor, Eickhoff	125 hp	300 fpm	26x92-mm center chain.
Face supports, Hemscheidt, 320, 2-leg shield	176 ton/leg at 300 bar		Yield load = 405 bar; working range = 42-90 in.
Hydraulic pumps	75 hp	22 gal/min at 300 bar	5% hydraulic oil solution, not located in headgate.

TABLE A4. Task summary of Mine B face workers

Crew member	Quantity	Location		Task variety		Job description
		Fixed	Varied	None	Some	
Foreman	1		X		X	Direction of work force and general health and safety.
Headgate operator	1	X		X		Stationed at main control panel; responsible for all stop/start controls.
Shearer operator	2	X		X		Travels along with shearer; controls speed and cut horizon.
Support operator	4	X		X		Advances panline after empty cut; advances supports.
Mechanic	1		X		X	Preventive maintenance and minor repairs while producing coal.
Supply	1		X		X	Transports materials to face area; clean up; supports face crew as needed.

III. Mine C

INTRODUCTION

Mine C which had been designated a state-of-the-art longwall system,[3] is a low-seam mine operating a single-drum shearer. A two-man field measurement team visited the mine and recorded noise and vibration data on April 11, 1979. On the day of the visit, roof conditions prohibited coal production.

BACKGROUND

Mine C is located west of Elderton, Pennsylvania. This mine produces 900,000 tons of coal annually and supplies the a power plant nearby. The coal is transported from the mine to the power plant on an underground conveyor system. Three-hundred and seventeen miners, who produce 2600 tons of coal daily, are employed on a three-shift-per-day schedule.

GEOLOGY

The Lower Freeport Seam, which has a maximum height of 44 inches at Mine C, is mined. The very shallow coal bed has an overburden of less than 200 feet in some areas. The seam geology is shown in figure A7.

MINE LAYOUT

Mine C uses a standard retreating system on an approximately 4,000-foot-long panel. The face visited, 4 south 1 right, is a left-handed face (figure A7). Ventilating air is brought in through the last open crosscut along the headgate and exhausted through the tailgate. On the day of the visit, 14,400 cfm of air was measured at the headgate.

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3 Hoop and Slone (footnote 1).

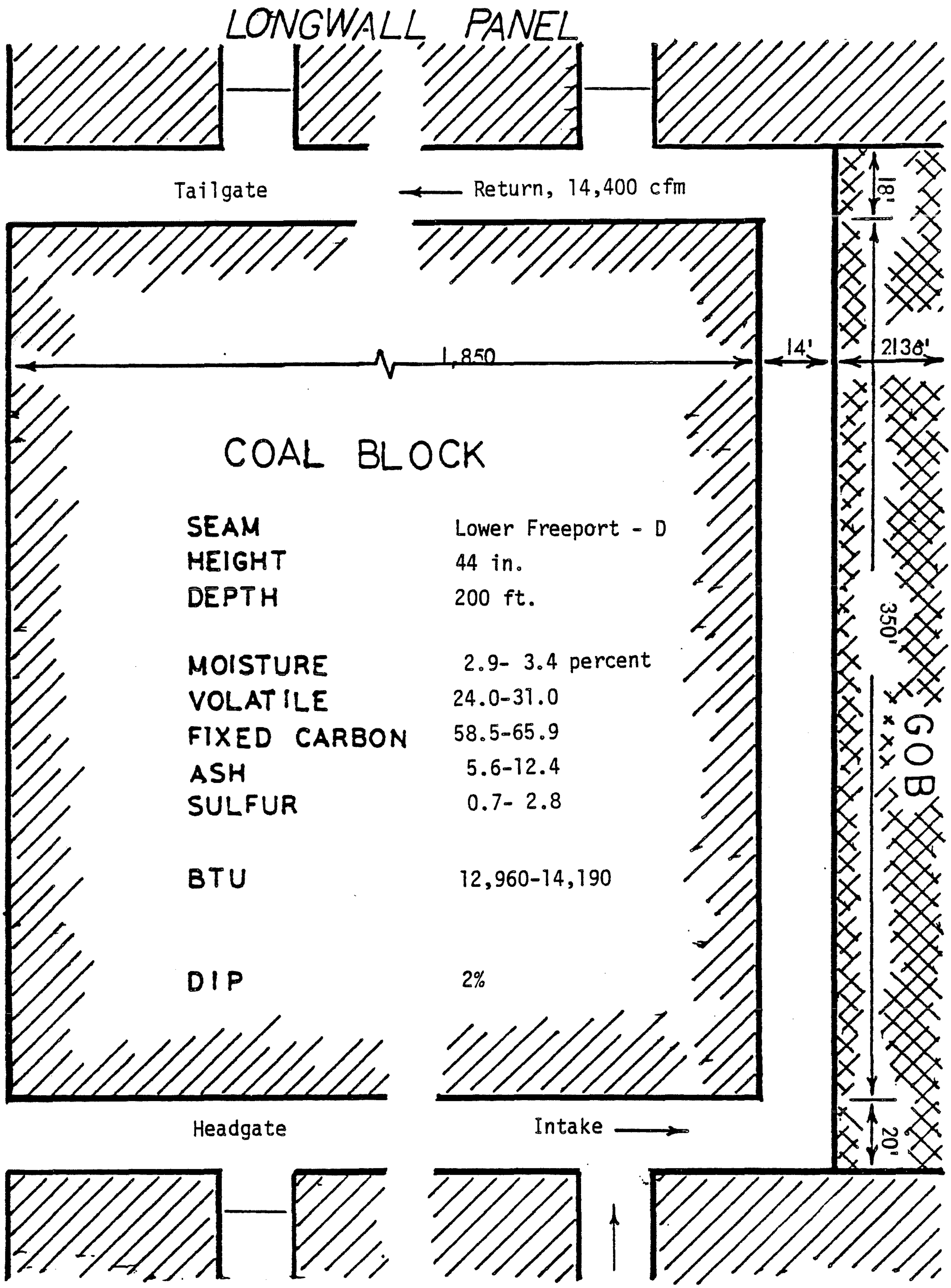


FIGURE A7. Mine C: seam analysis and longwall panel layout

At the headgate, where the roof clearance is approximately one foot, a 30-inch face conveyor dumps onto a high-speed, 30-inch piggyback conveyor. This unit feeds a 36-inch panel belt, which runs at 325 fpm. To give maximum cutout distance for the shearer, the stage loader and panel belt are placed along the left rib. A mule train is located by the belt tail where the master control unit is mounted. From his position between the hydraulic pumps and the electrical distribution units, the headgate operator controls all conveyors, shearing machine power, and the hydraulic pumps. Two 75-hp pumps, using 5% soluble oil solution, power the 75 face supports. Both pumps operate at 3,500 psi with chocks using full pressure for setting and half pressure for lowering and snaking. A layout of the headgate is shown in figure A8, and figure A9 shows the tailgate and face plans.

A 350-foot Eickhoff face conveyor with a single-center chain and powered by two 125-hp drives is used at Mine C. Coal is won with an Eickhoff single-drum ranging arm shearer with a 24x44-inch cutting drum. Twenty-foot wide entries enable the machine to cut out the coal at each end, thus simplifying sumping procedures. Dust suppression is by water sprays through the drum and auxiliary machine-mounted sprays. Machinery used in Mine C is summarized in table A5.

MINING CYCLE

1. The shearer takes a full cut of 24 inches from the headgate to the tailgate.
2. The face conveyor is advanced approximately 60 feet behind the shearer by the snaker.
3. After the shearer cuts out at the tailgate, the face conveyor advances, and the shearer begins cutting toward the headgate.

MINE CREW

Eight men, 7 UMWA workers, and one company foreman make up the work crew. The mechanic and foreman perform a variety of tasks in several locations while the face workers do repetitive tasks on the face. Crew members, job functions, and locations are summarized in table A6.

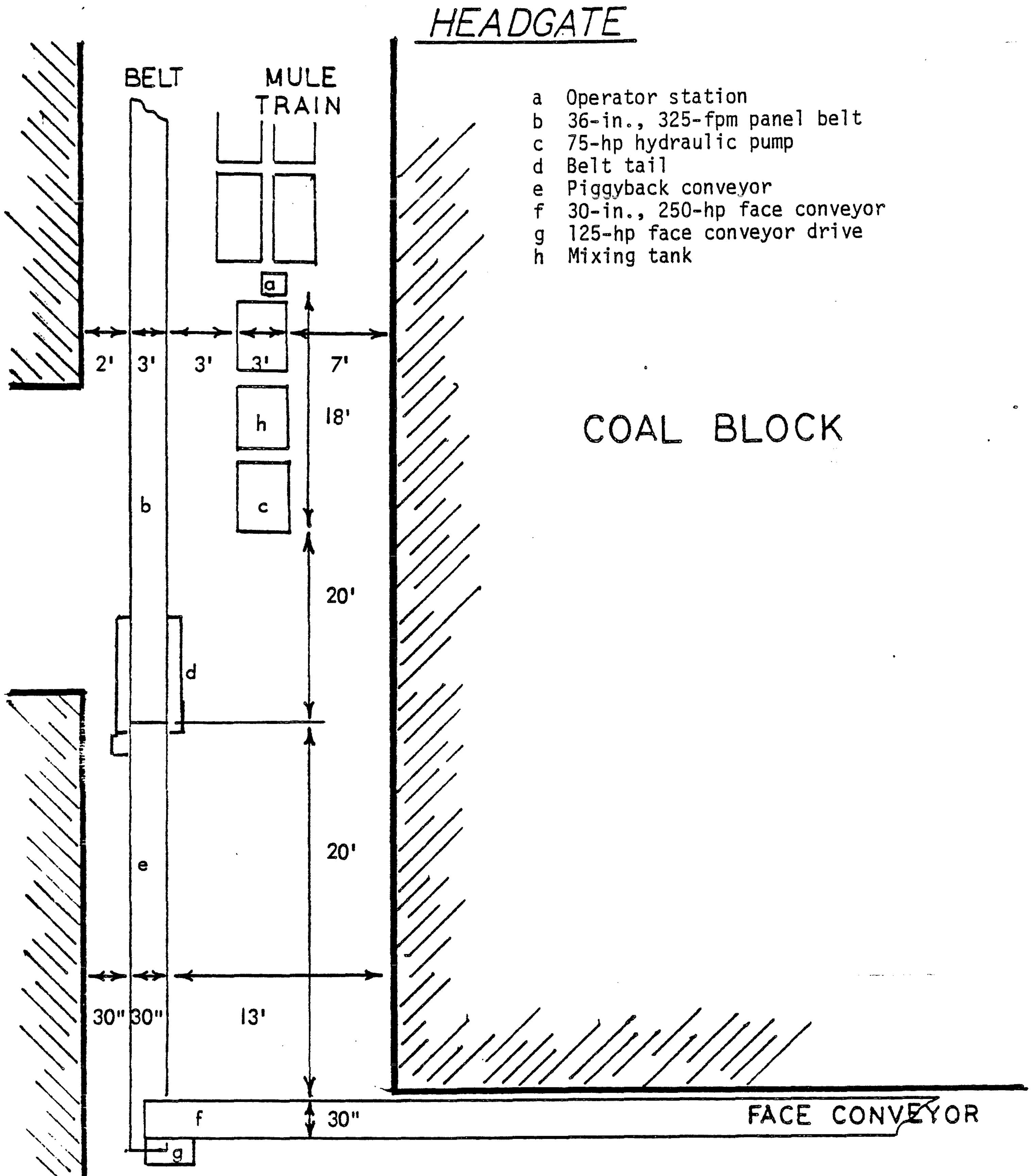
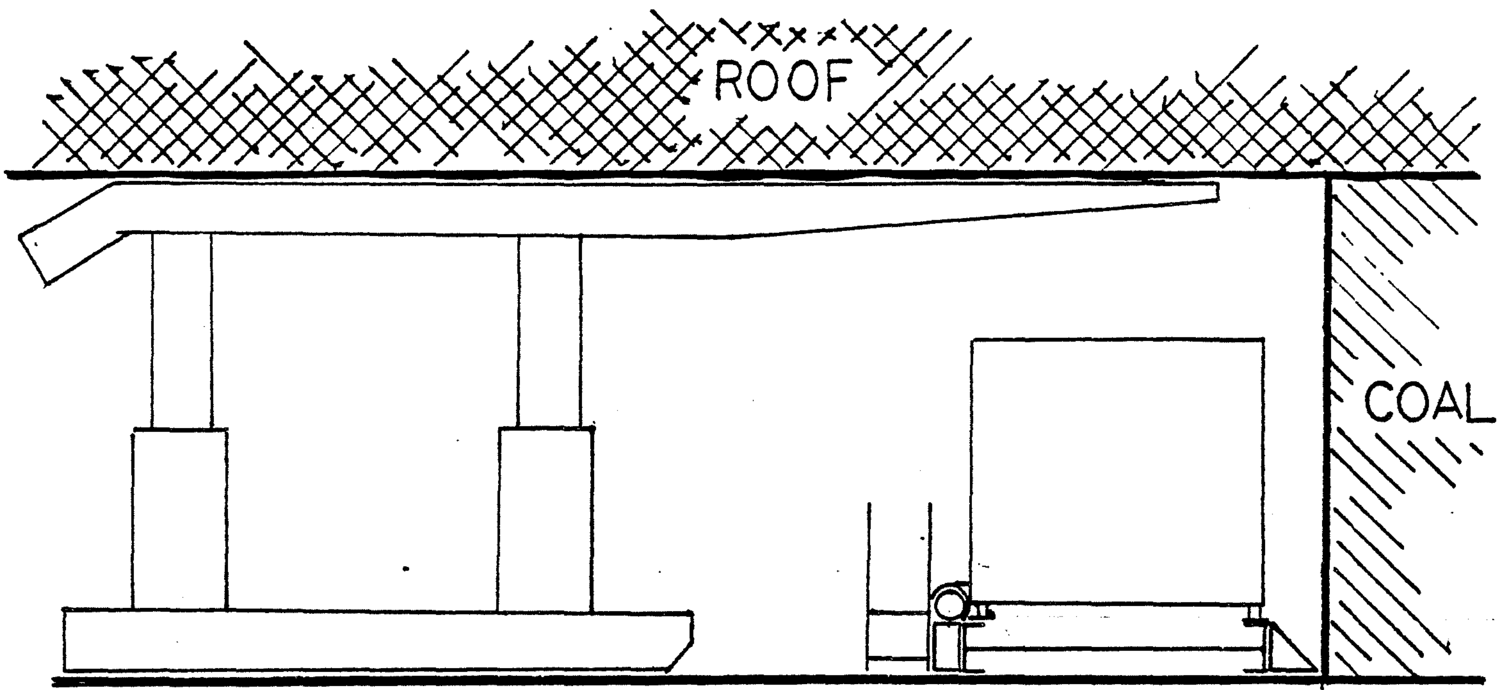


FIGURE A8. Headgate layout of Mine C.



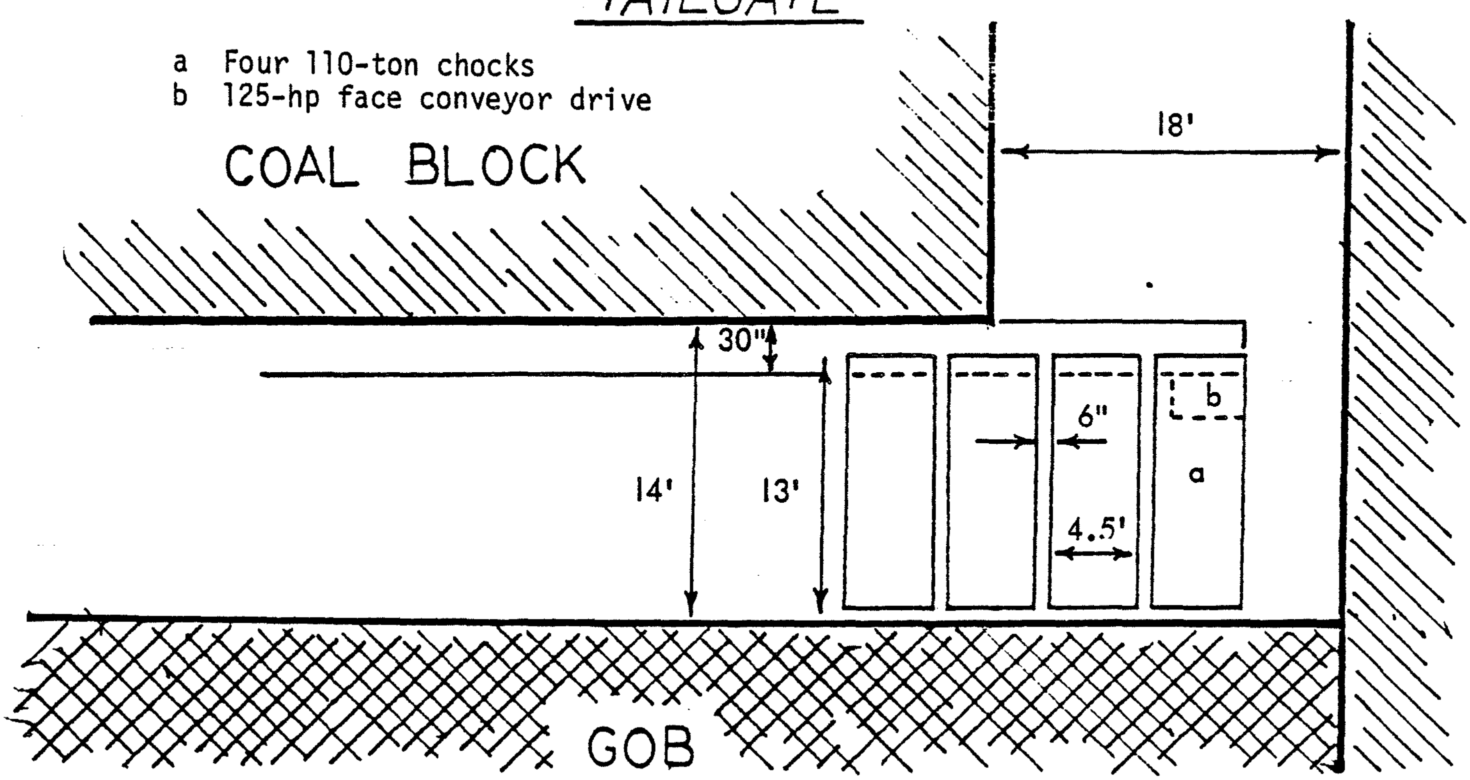


FIGURE A9. Mine C: tailgate layout and face plans

TABLE A5. Machinery used in Mine C

Machinery, manufacturer, model, and type	Power	Speed	Comments
Winning machine, Eickhoff, EW-150L, single drum	230 hp	30 fpm, 48 rpm	Exposed haulage chain; 24x44 in. drum.
Face conveyor, Eickhoff, EKF 3, 30-in. conveyor	2x125 hp	180 fpm	Single center chain, 26-mm; 1-m between flights.
Headgate conveyors, Long Airdox, 1033, 30-in. piggyback	40 hp	210 fpm	31 ft long, 18 in. between flights.
Face supports, Kloeckner-Ferromatik, 3859, 4-leg chock	110 ton/leg at 3500 psi		Yield load = 420 kg/sq cm, working range = 36 to 72 in.
Hydraulic pumps, Reliance	75 hp	1600 gpm at 1800 rpm	Two units each at 3500 psi, 5% oil solution.
Crusher	NA	NA	None.

NA - Not available.

TABLE A6. Task summary of Mine C face workers

Crew member	Quantity	Location		Task variety		Job description
		Fixed	Varied	None	Some	
Foreman	1		X		X	Direction of work force and general health and safety.
Headgate operator	1	X		X		Stationed at main control panel; responsible for all stop/start controls and pump and belt maintenance.
Shearer operator	1	X		X		Travels along with shearer; controls speed and cutting path.
Snaker	1	X		X		Travels approximately 30 ft behind the shearer, advancing the AFC as coal is removed.
Support operator	3	X		X		Travels with the shearer, advancing the roof supports as AFC advances.
Mechanic	1		X		X	General maintenance throughout the section.

IV. Mine D

INTRODUCTION

Mine D which has been identified as a state-of-the-art longwall system employing a double-drum shearer,[4] was visited on May 3, 1979.

BACKGROUND

Mine D is located in Bald Knob, West Virginia. This Beckley area mine produces 1.2 million tons of steam coal annually and employs 600 union workers on a three-shift-per-day schedule.

GEOLOGY

The coal mined at Mine D is multiple-bedded coal from the Eagle Seam and ranges from 2 to 10 feet thick. This seam, also known as the Middle War Eagle, lies at a depth of 500 feet. An analysis of the seam is given in figure A10.

MINE LAYOUT

Mine D uses a standard retreating system. Most panels are approximately 3,500 feet long and 500 feet wide (figure A10). The panel visited, 1 left off 2 east, had 110 feet remaining to be cut. Ventilation on this right-handed face is in through the headgate and out the tailgate. Air quantity is 17,000 cfm.

A 30-inch face conveyor at the headgate dumps coal onto an identical stage conveyor. From there the coal is transferred to a 36-inch panel belt and finally into rail cars. Since the systems use a double-drum shearer, entry width and cutout distances are not critical. A master control panel is mounted along side the stage conveyor, eight feet from the face conveyor. From there, the headgate operator controls the on/off action of the shearer, conveyors, and pumps. Panel distribution boxes and support system hydraulic pumps are located by the tail piece, between the left-hand rib and the belt.

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[4] Hoop and Slone (footnote 1).

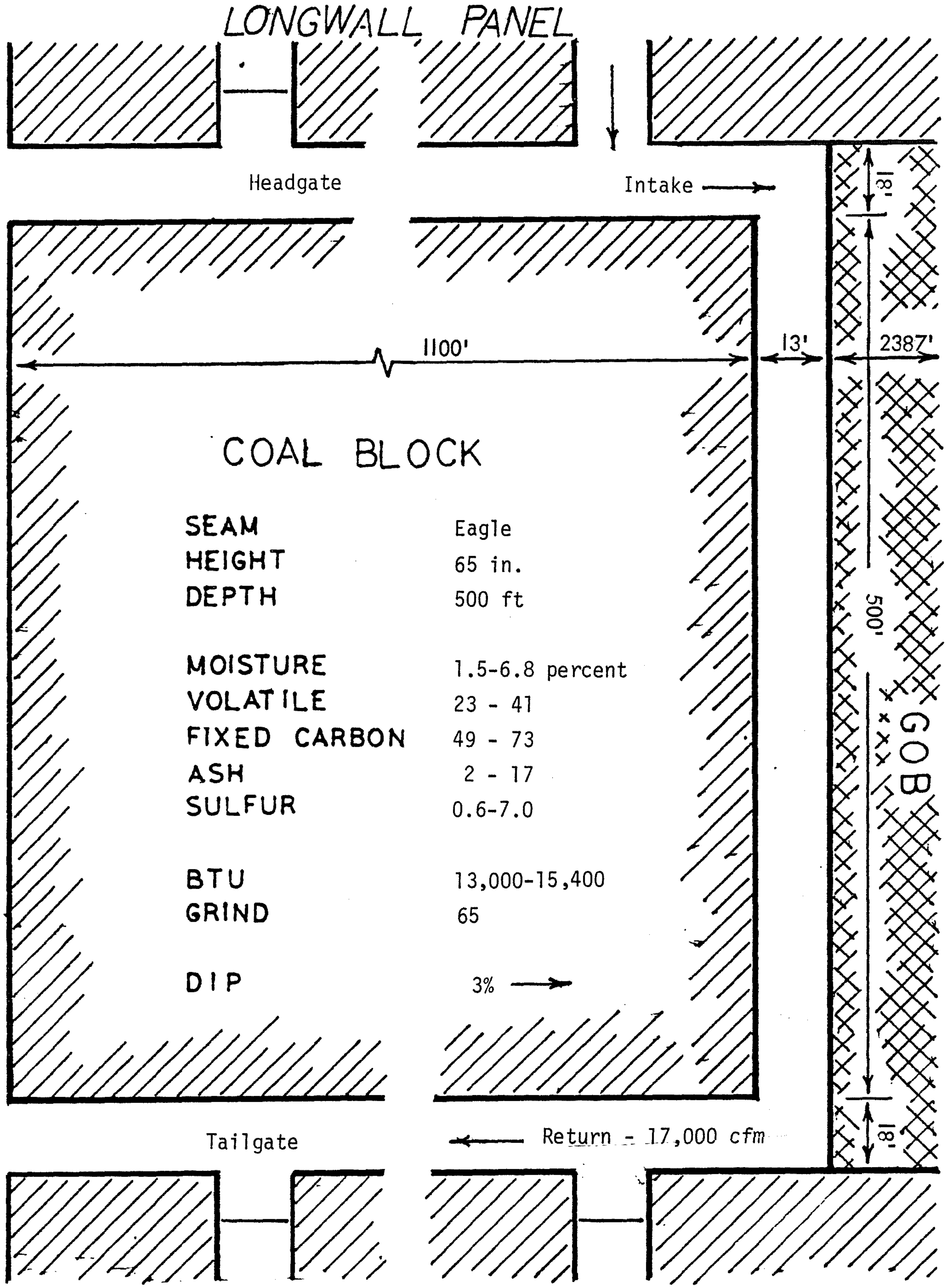


FIGURE A10. Mine D: seam analysis and longwall panel layout

Both pumps, which run at 3,500 psi, use a 5% hydraulic oil solution mixed in a common tank. Roof clearance over the headgate machinery ranges from one to three feet.

The 500-foot, 30-inch face conveyor is protected by 110 four-leg shield supports. The Sagem double-drum shearer is propelled by a Perard Peratrak system. Coal is moved along the face by a 26x92-mm double inboard chain. Two 175-hp motors move the chain at 226 fpm. Dust suppression is accomplished by water sprays mounted on the shearer. Figure A11 shows the headgate layout, and figure A12 shows the tailgate layout and face plans. Machinery used in this mine is summarized in table A7.

MINING CYCLE

1. The shearer cuts to the tailgate with the leading drum cutting the top 70 percent of the seam.
2. The snaker follows appproximately 30 feet behind the shearer, advancing the face conveyor 30 inches.
3. The shield operators advance the supports when a section of the face conveyor is fully advanced.
4. The shearer cuts out at the tailgate. Then it reverses direction and the drums.
5. The shearer makes an angle cut for approximately 45 feet.
6. The shearer reverses direction and drums and makes an angle cut to the tailgate, completing the sumping procedure.
7. The shearer cuts from the tailgate to the headgate.
8. The shearer makes an S-type sump.

MINING CREW

The crew is made up of ten men, nine UMWA workers and one company foreman. Since the shearing machine has double drums, two operators are required. Here, as in most setups, the mechanics and foreman move about the section performing safety and preventive-type duties unless a breakdown or problem occurs. A utility worker is used to do all the "dead" work. A separate crew is located at the transfer point where coal is dumped from the belt into rail cars. Table A8 lists face workers according to location and task.

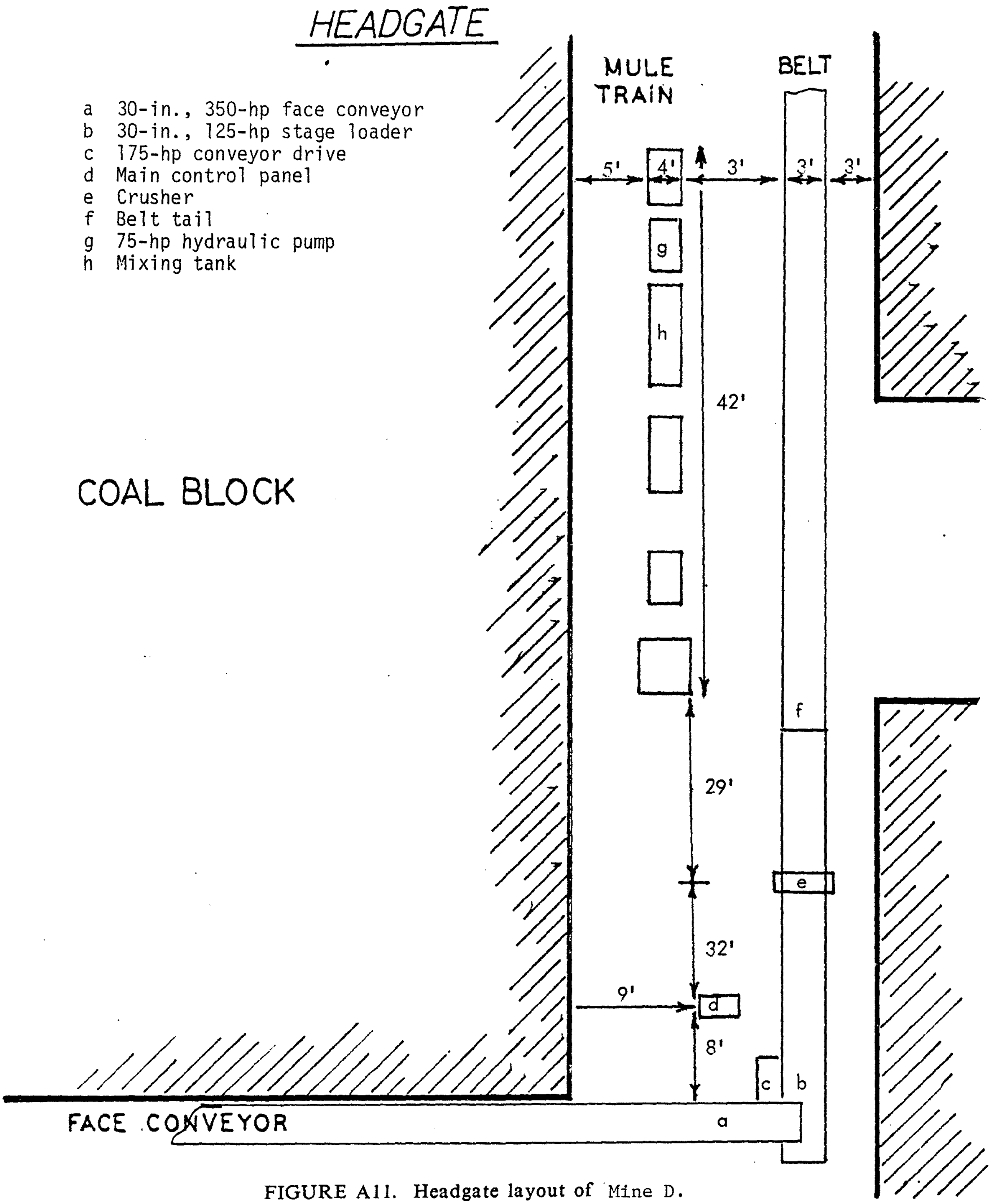
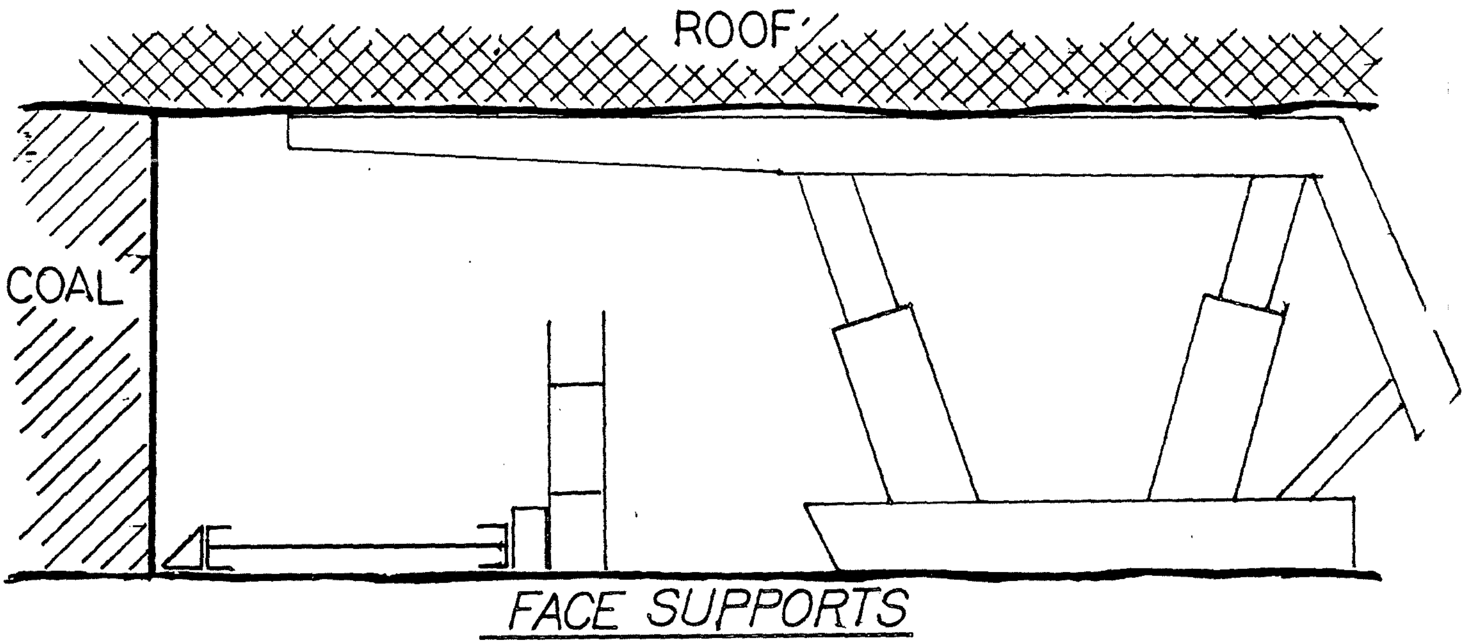


FIGURE A11. Headgate layout of Mine D.



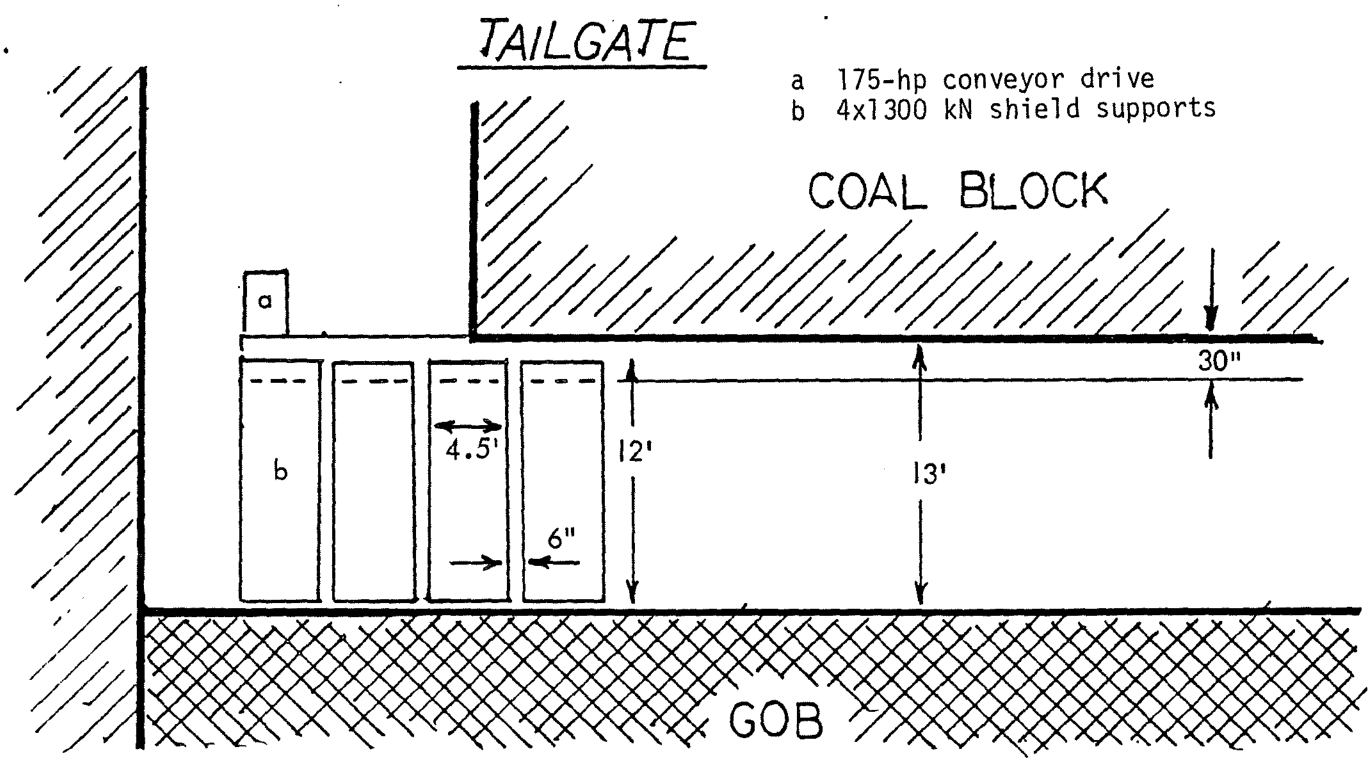


FIGURE A12. Mine D: **tailgate layout and face plans**

TABLE A7. Machinery used in Mine D

Machinery, manufacturer, model, and type	Power	Speed	Comments
Winning machine, Sagem Sirrus, 400, double drum	400 hp	30 fpm; drum, 46 rpm	30x54-in. drums, Perard Peratrak chainless haulage.
Face conveyor, Perard, PF II/600, double inboard	2x175 hp	226 fpm	400-ton/hr capacity, 600-mm width, 26x92-mm chains.
Headgate conveyor, Perard, M IV V/600, double inboard	125 hp	226 fpm	500 ton/hr; 26x92-mm chains.
Face supports, Westfalia, BS 2.1, 4-leg shield	1300 kN/leg at 3500 psi		yield load, 520 bar; working range, 42-90 in.
Hydraulic pumps, Reliance motor, WOMA pumps	75 hp	100 l/min at 315 bar	5% hydraulic oil solution.
Crusher, rotary type	75 hp	100 l/min at 315 bar	4 rows with 4 bits.

TABLE A8. Task summary of Mine D face workers

Crew member	Quantity	Location		Task variety		Job description
		Fixed	Varied	None	Some	
Foreman	1		X		X	Direction of work force and general health and safety
Headgate operator	1	X		X		Stationed at main control panel; responsible for all stop/start controls.
Shearer operator	2	X		X		Travels along with shearer; controls speed and cutting horizon.
Snaker	1	X		X		Travels behind the shearer, advancing the face conveyor.
Support operator	3	X		X		Travels behind the snaker, advancing the roof supports.
Mechanic	1		X		X	Preventive maintenance and minor repairs while producing coal.

V. Mine E

INTRODUCTION

The Mine E had been identified as a candidate mine for study because it uses an in-web shearer.[5] It was to be visited on October 3, 1979.

BACKGROUND

The Mine E is located in West Virginia. This mine, which is in the Beckley area, produces one million tons of metallurgical coal annually. Mine employment is 400.

GEOLOGY

The Pocahontas No. 3 Seam, recognized as one of the best low-volatile metallurgical coals, is mined at Mine E. The No. 3 seam is part of the Pocahontas formation, which is found in the southeastern region of the Appalachian Plateau. The formation is primarily sandstone and ranges from a thickness of 200 feet to 700 feet. A seam analysis is presented in figure A13.

MINE LAYOUT

The Mine E uses a standard retreating system, and normal panel size is 480 feet by 5,300 feet (figure A13). The field measurement team visited the panel and, at that time, there was 1,200 feet remaining to be cut. Air is circulated at 25,000 cfm on this right-handed face from the headgate to the tailgate.

At the headgate, coal is transferred from the face conveyor to a stage loader. Next, the coal is run on a 36-inch panel conveyor belt. Finally, the coal is loaded into rail cars for transportation outside. Since the in-web shearer has a drum at each end and travels ahead of the panline, cutting to the ends of the face conveyor is easy. The main

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[5] Hoop and Slone (footnote 1).

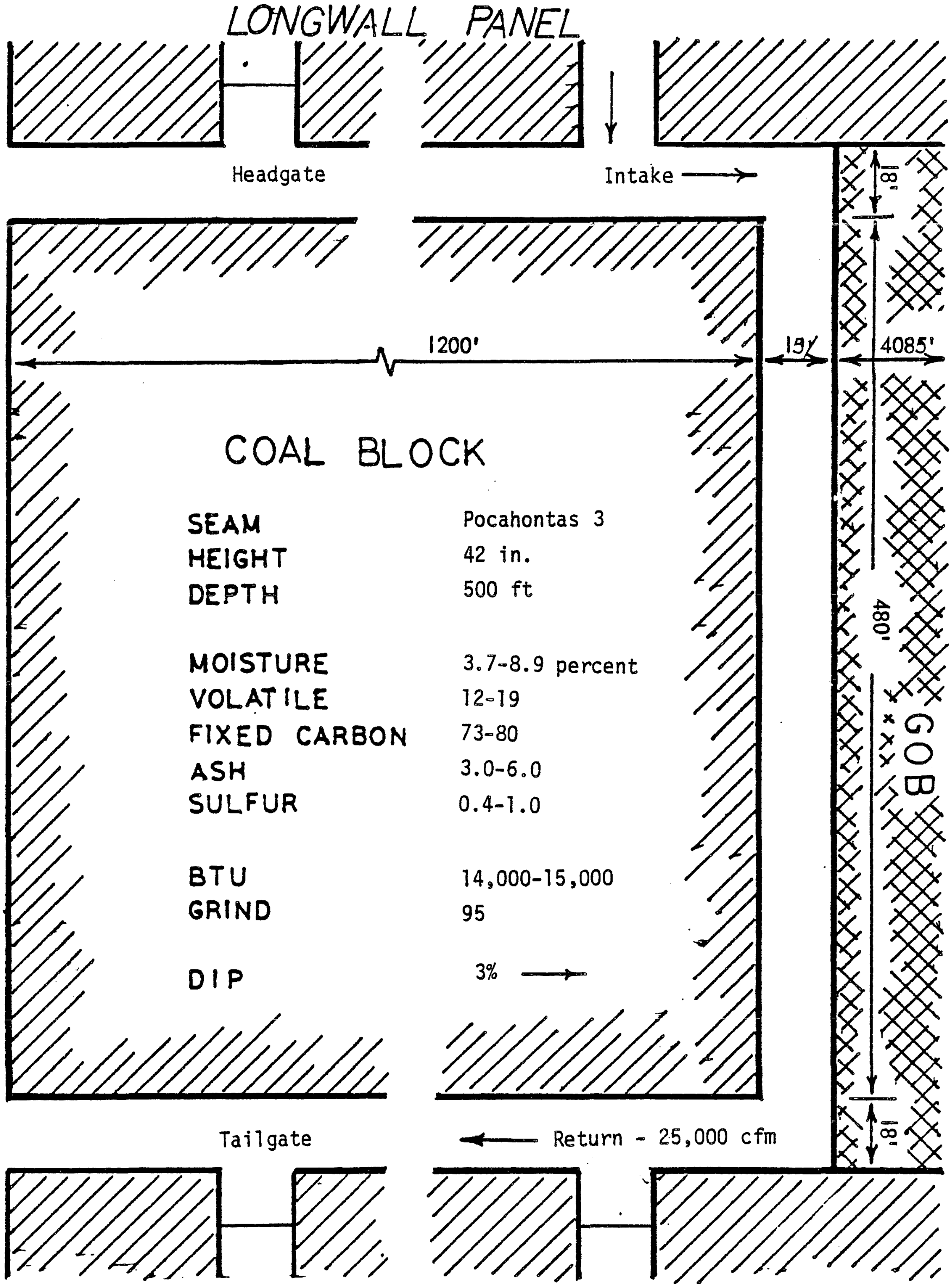

FIGURE A13. Mine E: seam analysis and longwall panel layout control panel is skid-mounted and pulled by the mule train. The mule train consists of a hiker, two panel power boxes, two hydraulic pumps, and a mixing tank. Hydraulic fluid is a mixture of 95% water and 5% soluble oil. Clearance over the headgate machinery is generally 20 inches. Figure A14 shows the headgate layout.

The face is supported by 100 Dowty 4-leg chocks. The shearer is trammed across the face by a mechanical haulage system. Tramming is accomplished by a driving sprocket engaging a stationary 22-mm chain stretched along the face. Coal haulage is achieved by a single 26-mm chain driven by two 150-hp motors. Table A9 gives a summary of the machinery used at Itmann.

MINING CYCLE

1. The shearer cuts to the tailgate with the leading drum cutting the entire seam height.
2. The face conveyor is advanced 30 feet behind the shearer.
3. The roof supports are advanced when the face conveyor is positioned.
4. The shearer cuts toward the headgate, increasing the depth of cut from 0 to 30 inches during the first 100 feet.
5. The tailgate conveyor and supports are advanced.
6. The shearer cuts to the tailgate, then to the headgate, and the conveyor and supports are advanced 30 feet behind the shearer.
7. The shearer is sumped at the headgate by the same procedure used at the tailgate.

MINE CREW

The crew is made up of 12 men, 11 UMWA workers and 1 company foreman. Two operators are required for the double-drum shearer. The mechanics and foreman move about the section performing safety and preventive-type duties unless a breakdown occurs. Because of the seam height, face workers must crawl along the panline on their hands and knees. Table A10 summarizes face workers according to task and location.

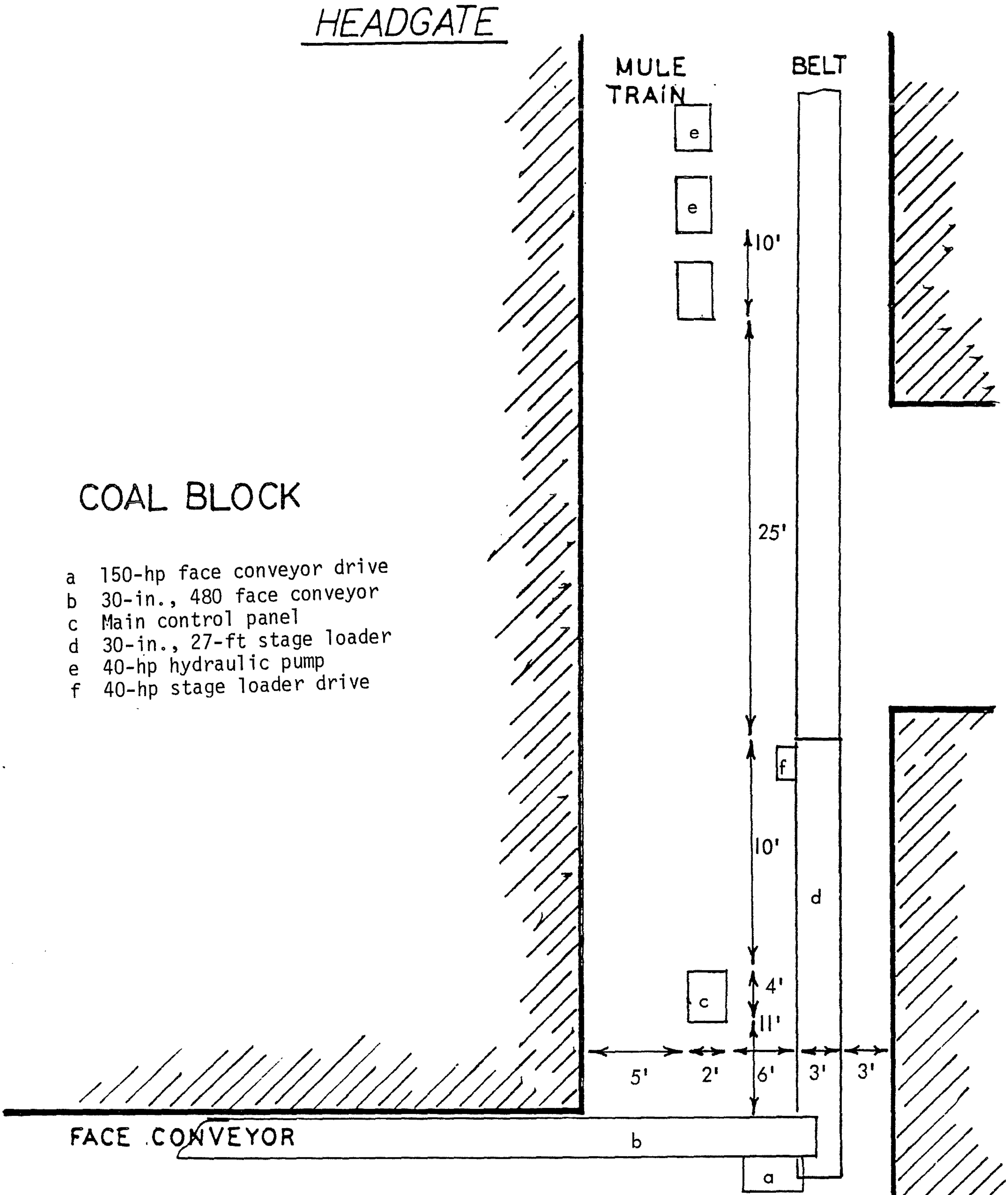

FIGURE A14. Headgate layout of Mine E.