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DEVELOPMENT AND EVALUATION OF NOISE CONTROL TECHNIQUES FOR TACONITE PROCESSING EQUIPMENT

Contract USBM J0377014
Bolt Beranek and Newman Inc.
10 Moulton Street
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16. Abstract (Limit: 200 words)

The objectives of this research were to assess worker noise exposure to taconite processing plants; to identify the major noise-generating equipment, and to develop, install, and evaluate noise control treatments for those sources. Three areas of taconite processing equipment, grinding mills, and fine and crusher screens were investigated. Extensive noise and vibration measurements were made on grinding mills in which the process parameters affecting mill noise were identified and the level of shell impact noise around the mill was defined. A multiplexing telemetry system was used to obtain shell vibration measurements from a rotating mill. Mill noise control methods were investigated and a partial mill barrier was built and tested on a semiautogeneous mill. A noise reduction of approximately 3 to 6 dBA was achieved on the falling side of the mill. The use of rubber mill liners on autogeneous mills was evaluated. Noise sources on fine screens were quantified and noise control treatments were developed. Noise reductions of 11 to 13 dBA were achieved at positions around the fine screen and some in-service testing was done. Noise measurements of crusher screens were obtained and noise control through resilient screen decks appears to hold the most promise. Additional benefits might be lower life-cycle cost because the service life of resilient decks is claimed to be longer than that of steel.

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FOREWORD

This report was prepared by Bolt Beranek and Newman Inc., Cambridge, MA under USBM Contract No. JO377014. This contract was initiated under the Metal and Non-Metallic Mine Health and Safety Program. It was administered under the technical direction of the Pittsburgh Research Center with Tom Bobick acting as Technical Project Officer. Alan Bolton, Jr. was the Contract Administrator for the Bureau of Mines. This report is a summary of the work completed during the period April 1979 to December 1982, and was submitted by the authors in February 1983.

The authors wish to thank the following member producers of the American Iron Ore Association (AIOA) for their invaluable assistance:

The Cleveland-Cliffs Iron Company
The Hanna Mining Company
Pickands Mather and Company

In particular we wish to thank Mr. Bob Neil of the AIOA and Cleveland-Cliffs Iron Company, Mr. Wally Choquette of Hanna Mining Company, Mr. Bob Buescher of National Steel Pellet Company, and Bob Garrett of Sherman Mine for their cooperation on this project.

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1. INTRODUCTION

1.1 Background

The permissible noise exposures of miners in the metal and non-metal mining industries are specified in the Federal Mine Safety and Health Act of 1977 (see Appendix A). Since many of these miners are subject to noise exposures that exceed those allowed by the Act, the United States Bureau of Mines has sponsored research into techniques to abate noise in mining operations. Of particular interest to the Bureau in the metal and non-metal industries is noise in the taconite processing plants [1]. This process involves crushing, grinding, screening, and agglomeration of a very hard and abrasive ore. The size of the machinery and the requirements for power (estimated to be 100 kW per ton of taconite ore [2]) are quite large and, consequently, high levels of noise are generated in these plants. Typical plant capacities are in the range of 2 to 5 million tons per year. Thus, taconite processing operations are conducted on a vast scale.

Taconite processing was selected for the USBM-sponsored research, as opposed to other types of ore processing, because the extreme hardness of taconite ore results in high noise levels and a considerable amount of equipment maintenance. Both factors increase the difficulty of implementing a successful noise control program, and it is believed that results from the treatment of taconite plants should be applicable to other types of ore-processing plants.

Due to significant plant expansions in the 1970's, production capacity in the United States rose to approximately 90 million tons of pellets per year. The total U.S. market for pellets is on the order of 100 million tons per year; therefore, major expansion of production capacity is unlikely, especially given the global downturn in the iron and steel markets in the last two years. Figure 1.1 shows the level of annual shipments from U.S.

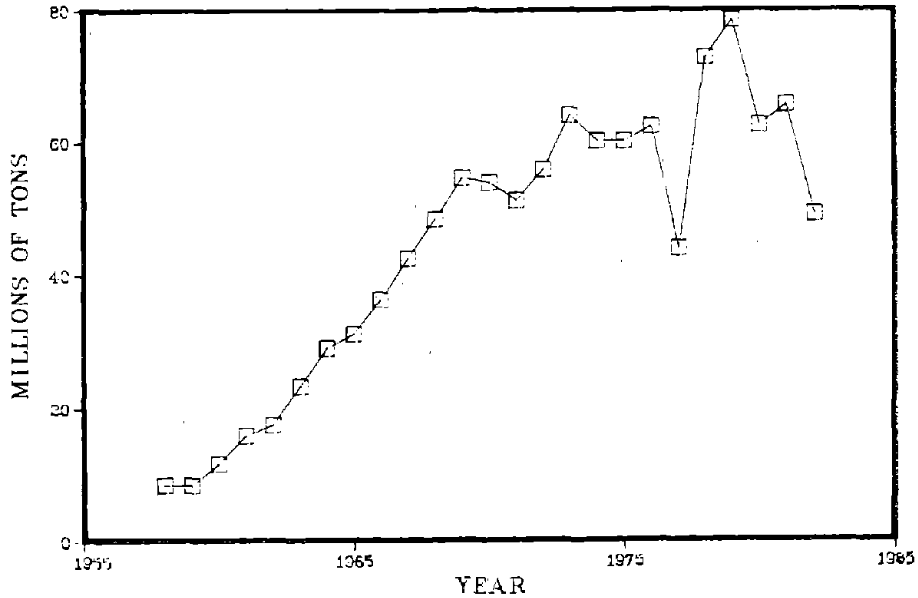


FIG. 1.1. ANNUAL SHIPMENTS OF IRON ORE PELLETS FROM U.S. PLANTS.

iron ore pellet plants for the period 1958 to 1982 [3]. In 1982, shipments were sharply reduced compared to 1981, down to less than 50 million tons. It should be noted from Fig. 1.1 that the record shipment occurred in 1979 and the years since have shown significantly reduced demand. On account of this supply/demand situation, the USBM has decided to concentrate on retrofit noise control efforts, rather than new plant noise control, as the best way to have maximum impact on reducing the noise exposure of workers in taconite plants.

1.2 Overall Project Objectives

The overall USBM research effort in taconite plants has been divided into three phases. Phase I, performed under USBM Contract J0366055, defined a noise control strategy for taconite processing plants; the objectives were:

- To assess worker exposure in processing plants
- To define preliminary noise abatement goals
- To identify the equipment causing the most noise.

Further details of this effort are described in the Phase I final report [4]. The primary objectives of Phase II, discussed in Ref. 5, were:

- To diagnose the specific noise sources on the equipment identified in Phase I
- To develop conceptual noise control techniques for these sources.

The work was divided into the following steps:

- In cooperation with the Phase I contractor and the U.S. Bureau of Mines, specific machines were selected for detailed study.
- In cooperation with the member taconite producers of the American Iron Ore Association, six processing plants were visited for the study of the identified machines, and noise and vibration measurements were made on the machines to determine the noise generating mechanisms.
- Conceptual noise control treatments were formulated for each machine.

The objectives of Phase III (the project detailed in this final report) were:

- To develop specific prototype noise control treatments for the equipment identified in Phases I and II.
- To install the treatments on machines in service and monitor their performance, mechanical durability, and cost.

1.3 Equipment Selected for Noise Control Development

There are three distinct areas in a taconite processing plant: (1) the fines crushing plant, (2) the concentrator, and (3) the pellet plant. The Phase I study identified the area noise reduction goals shown in Table 1.1. In addition, the Phase

I study identified the noise sources in four taconite plants and provided information that helped establish criteria for selecting equipment for detailed study in Phase II. This equipment included:

- Crusher sizing screens
- Mills (rod and semi-autogenous)
- Fine screens with pneumatic rappers
- Vacuum disc filters.

Crushers and crusher sizing screens in fines crushing plants, mills and fine screens in concentrators, and vacuum disc filters in pellet plants were identified as the most important noise sources unique to metal and non-metal ore processing. Noise control of crushers has been demonstrated as part of a separate Bureau of Mines research program [6]. Also, the measurements of both noise level and typical exposure times made during Phase II indicated that disc filters were not as serious a noise problem as previously estimated. Therefore, they were not included in Phase III.

TABLE 1.1. NOISE REDUCTION GOALS BY AREA OF PLANT.

Area	Goal
Fines crushing	4 - 9 dBA
Concentrators	2 - 5 dBA
Pellet Plants	4 - 5 dBA

Although Phase III was restricted to noise control design and evaluation for crusher sizing screens, mills, and fine screens with pneumatic rappers, one must remember that in most plants other equipment not unique to metal and non-metal ore processing also needs noise control treatments in order to comply with present noise regulations. This other equipment includes slurry and tailings pumps, motors, gears, vacuum pumps, and Roots blowers, for example.

In cooperation with the American Iron Ore Association (AIOA), visits were made to three taconite plants on the Iron Range to view the three pieces of equipment, make initial measurements, and select candidate sites for the noise control demonstration on each equipment item. With additional assistance from the AIOA, meetings were scheduled with individual mining companies to request their cooperation with Bolt Beranek and Newman Inc. (BBN) on this project. Requirements for mine labor and equipment downtime were explained. Following these meetings, agreements were signed with the cooperating companies to allow BBN to acquire acoustic measurements, analyze plant operations, and develop, install, and evaluate noise control treatments that were installed on the particular equipment utilized at that plant. The decision was made at the outset of the work to concentrate the early project effort on grinding mills and fine screens with pneumatic rappers. These equipment items - Allis-Chalmers mills and Dorr-Oliver Rapifine DSM screens - were both located at the same plant.

The project schedule was impacted rather heavily by market conditions in the taconite industry. As the U.S. steel industry fell into a recession after mid-1981, the taconite plants (which are mainly captive subsidiaries of U.S. steel companies) were shut down for extended periods. In fact, 1980 was a relatively poor year in the steel industry, and a large midyear drop in capability utilization occurred (see Fig. 1.2). This figure shows the percent of capability utilization in the steel industry over the term of this project. Until the second quarter of 1981, it was above 80% with the exception of the previously mentioned 1980 midyear dip. From the third quarter of 1981 to the present

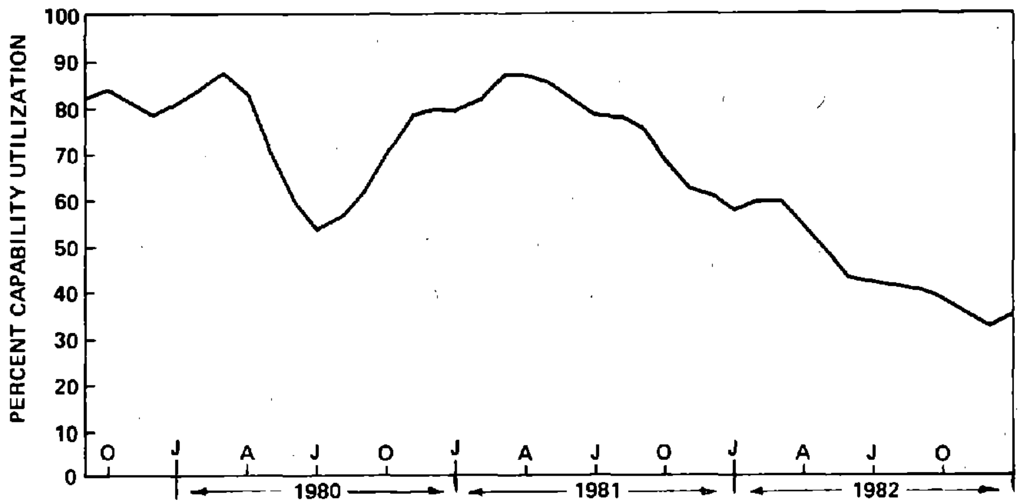


FIG. 1.2 PERCENT OF CAPABILITY UTILIZATION IN U.S. STEEL INDUSTRY DURING TERM OF THIS PROJECT.

time, there has been a steady decline to levels in the 30 to 35 percent range.

Working around the shutdowns of the plant, noise control treatments were demonstrated on the mills and fine screens. Unfortunately, the plant at which a cooperative agreement had been reached for work on crusher sizing screens shut down in January 1981 for an indefinite period. This was before work had commenced at the plant. The plant remained shut down until the fourth quarter of 1982, when its owner closed it permanently. Due to the finite resources of the project, the effort that would have been required to obtain a new crusher screens cooperator, and the uncertainty governing the prospects for full-time plant operation, it was decided to revise the focus of the work on crusher screens, by reviewing and reporting work performed by BBN and others in the taconite and allied mineral industries.

1.4 Report Organization

In Sec. 2 of this report, the noise control demonstration on grinding mills is described, together with a discussion of how various grinding parameters affect mill noise. The fine screens noise control demonstration is discussed in Sec. 3 and the review of the crusher screens is presented in Sec. 4. A discussion of potential benefits from administrative noise controls in taconite processing plants is given in Sec. 5. The overall project conclusions are presented in Sec. 6.

2. MILLS

Results from the Phase II study [5] identified grinding mills as the primary noise source in the concentrator plant. An overall noise reduction goal of 2 to 5 dBA was established for these areas. In this section, we present the results and analysis of an extensive noise control program aimed at achieving this goal. Section 2.1 provides an overview of the types of grinding mills and the associated noise levels. In Sec. 2.2, specific sources of mill noise are described and the influence of various parameters on the noise level identified. The various noise control options are discussed in Sec. 2.3. The two most promising noise control methods, partial noise barriers and rubber liners, have been tested and are discussed in detail in Secs. 2.4 and 2.5.

2.1 Overview

2.1.1 Grinding techniques

There are three types of primary grinding mills used in the taconite industry. These are rod mills, autogenous mills, and semi-autogenous mills.

Rod mills, see Fig. 2.1, are primary grinding mills that reduce product size by line contact between rods extending the full inside length of the mill. Such contact (attrition grinding) results in selective grinding of the large particles first, giving a uniform range of product size distribution. Rod mills can accept feed up to approximately 2 in. from secondary and tertiary crushers, and the product (4 to 35 mesh) is passed to ball mills for further grinding. The typical size of a large 2000 hp primary rod mill is 20 ft long by 15 ft in diameter. The steel rods added to the charge are usually 3 to 5 in. in diameter when new.

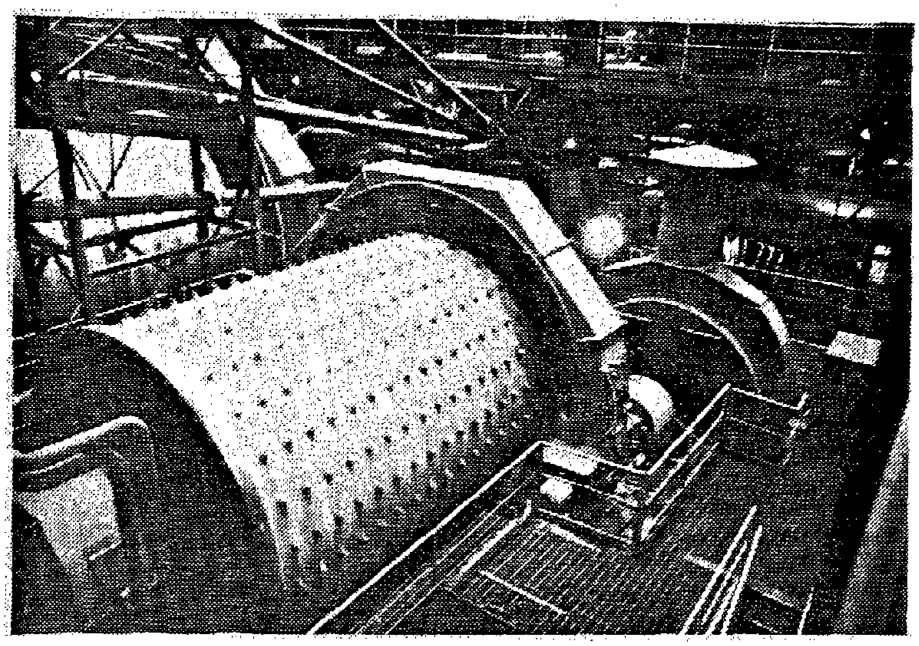


FIG. 2.1. ROD MILLS.

In primary autogenous grinding, a single large mill is used to reduce mine ore to a finely crushed product. The ore itself acts as the grinding medium. These mills, therefore, replace both secondary and tertiary crushers and rod mills. Unlike rod mills, the diameter of autogenous or semi-autogenous mills (see Fig. 2.2) generally exceeds its length. Mill diameters are typically 24 to 36 ft, and lengths are usually 10 to 20 ft. In autogenous grinding the main reduction mechanism is abrasion, where frictional forces cause fragments to be torn loose. Impact and attrition grinding usually account for only 30 to 40% of the work.

In the semi-autogenous process, steel balls are introduced into the charge. This increases the amount of impact grinding in the mill and speeds the throughput of material. The steel balls used in the grinding media are 3 to 5 in. in diameter and usually account for 3 to 9% of mill volume.

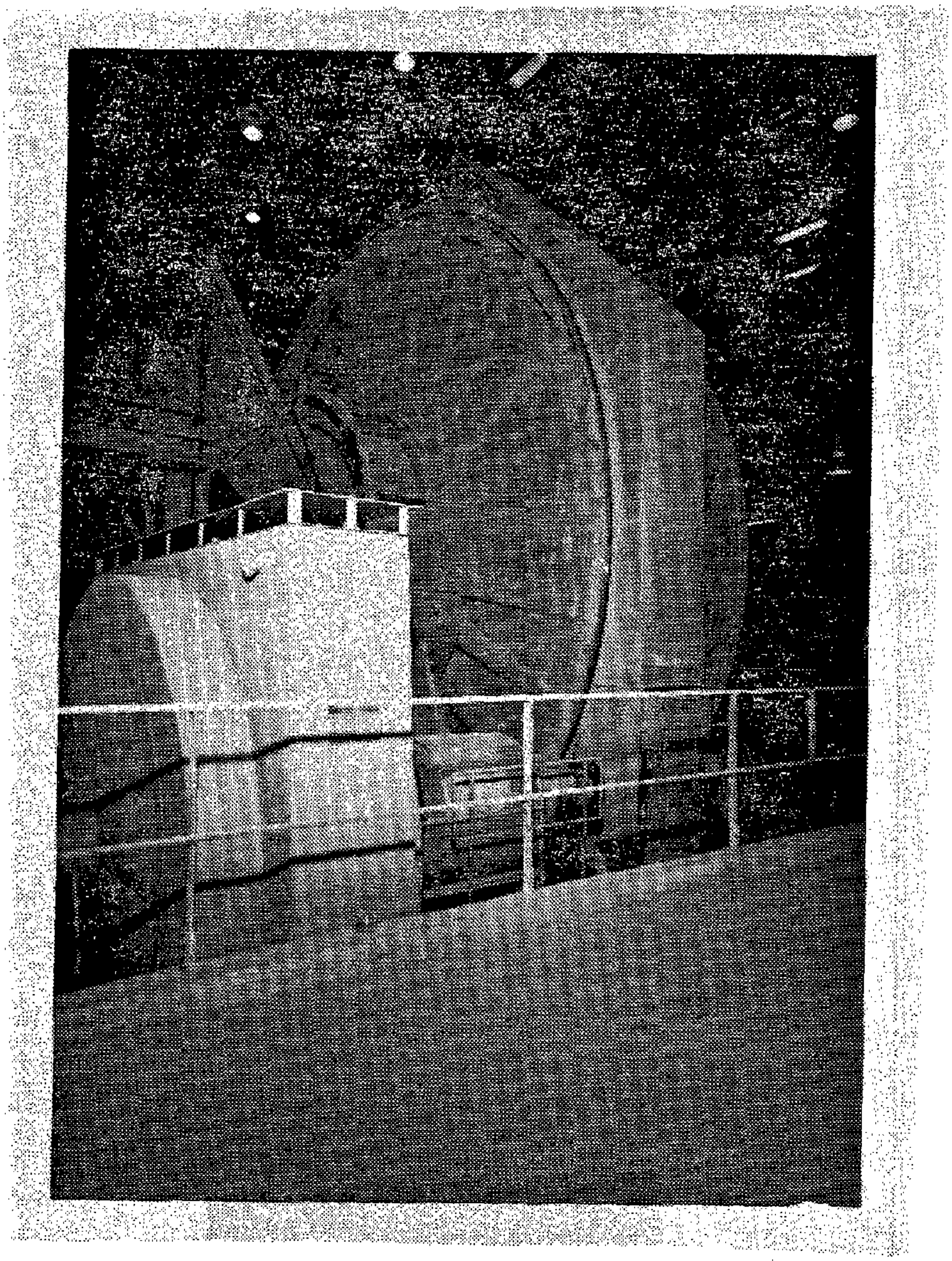


FIG. 2.2. SEMI-AUTOGENOUS MILL - DISCHARGE END.

Most mills rotate in the range of 65 to 76% of critical speed* (typically 10 to 12 rpm). The trajectory path of a particle carried up by a shell liner in a mill rotating at 75% of critical speed is plotted in Fig. 2.3. The kidney-shaped region is the area that might be occupied by the mill "charge" when it is operating. The charge typically occupies 40% of the mill volume. The media (rocks, balls, or rods) are carried upward by the liners on the "rising side," and will eventually fall onto other material underneath rather than on the shell liners. In addition, media constantly roll and tumble down the slope of the

*Critical speed is the lowest rpm that will cause a particle on the shell liner to centrifuge. At less than critical speed the particle falls off the shell liner before it reaches the top of the circular path of the liner.

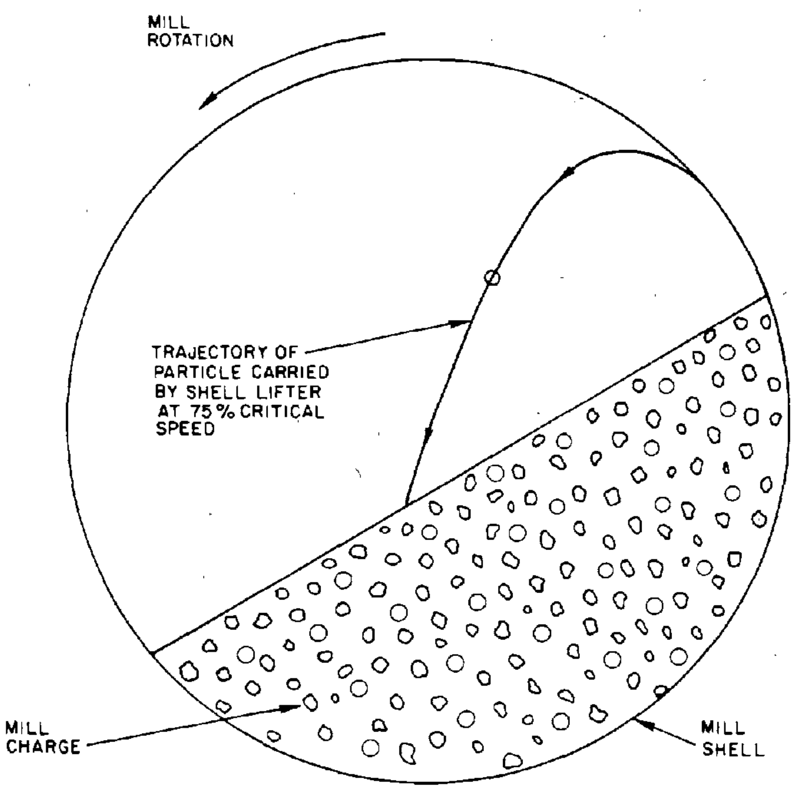


FIG. 2.3. PARTICLE TRAJECTORY AND CHARGE POSITION INSIDE TACONITE MILL (FALLING SIDE AND RISING SIDE).

charge onto the shell liners in the lower quadrant or "falling side" of the mill.

2.1.2 Typical noise levels

In most concentrator plants, the primary grinding mills are the dominant noise source within the area they occupy. Typical noise levels measured around a rod mill and a semi-autogenous mill are shown in Table 2.1. In general, there is little variation in the sound level on the mill floor around a large primary grinding mill. Such relatively constant noise levels are characteristic of the reverberant noise environment in a taconite plant, and make source diagnosis using standard measurement techniques difficult.

TABLE 2.1. PRIMARY MILL NOISE LEVELS.

Mill Type	Size	No. of Mills in Plant	Measurement Location	Sound Level (dBA)
Rod	17-1/2-ft × 11-ft diameter	3	1 1/2 ft from shell, falling side	98
			1 1/2 ft from shell, rising side	95
			10 ft from trommel	93
			On balcony, 15 ft above trommel	92
			On balcony, feed end	92-95
			On Mill floor, 10 ft from feed	96-98
Semi-Autogenous	18-ft × 27-ft diameter	10	1 1/2 ft from shell, falling side	105
			1 1/2 ft from shell, rising inside	98
			15 ft from shell, falling side	99-100
			On catwalk 20 ft from mill	95
			Below mill	101-105
			In nearby welding bay	93

Noise measurements for all three types of primary grinding mills were made in more than ten different concentrator plants. Figure 2.4 shows the noise level distributions for each type of mill. The typical measurement location was 15 ft from the falling side of the mill.

The average overall noise level for rod mills was 95 dBA with a standard deviation of ±2.5 dB. The number and size of the mills varied considerably in the rod mill plants visited. For

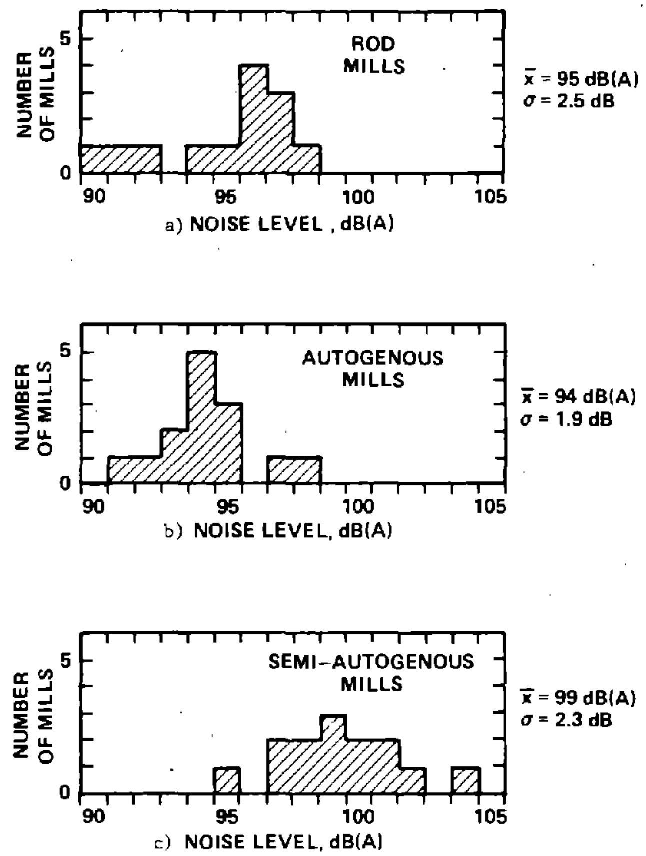


FIG. 2.4. DISTRIBUTION OF GRINDING MILL NOISE LEVELS FOR THREE DIFFERENT TYPES OF TACONITE CONCENTRATOR PLANTS. (MICROPHONE POSITION APPROXIMATELY 15 FT FROM FALLING SIDE OF THE MILL SHELL.)

example, one plant had a single 40-ft-diameter rod mill, while another had thirty 12-ft-diameter mills.

The overall average noise level for autogenous mills is 94 dBA ±2 dB. The lower standard deviation is due in part to the fact that autogenous mill sizes are more uniform than rod mills, with diameters usually 30 to 36 ft. Also, the feed input to the autogenous mill is carefully controlled and is kept relatively constant between mills. This is particularly true for plants that use froth flotation cells in the removal of silicates, since the pH concentration of the mill has to be carefully controlled.

Semi-autogenous mills have the highest noise levels, with an average of 99 dBA ±2.5 dB. The addition of the steel balls to the charge can significantly increase the noise level by 2 to 6 dBA over a similarly sized autogenous mill. In addition, the distribution of data shown in Fig. 2.4 is relatively wide for semi-autogenous mills. This is due to the variety of mill operating conditions, feed inputs, and charge makeup that are typical of semi-autogenous operations. These parameters are discussed in more detail in the next section.

2.1.3 General parameters in mill noise

There are a large number of parameters that affect the noise level of a primary grinding mill. However, because of the size of the equipment and complexity of the grinding circuit, it is often very difficult, if not impossible, to isolate and determine the direct effect of any one of these variables.

The parameters can be divided into three groups, which are shown in Table 2.2. As will be apparent, these groups are not necessarily mutually independent of one another, and are separated only for convenience of the present discussion.

In group I, the method or type of grinding will affect the noise level produced by a mill. The steel rods added to the charge in a rod mill not only assist the grinding process, but can impact with the mill liners/shell causing noise. In autogenous grinding, only ore will impact with the liners/shell, while with semi-autogenous grinding, the addition of steel balls to the charge will increase the noise level.

Most taconite plants currently use wet grinding circuits. Dry grinding typically requires 25% more power input than a wet mill, which may mean larger and possibly noisier auxiliary equipment (motor drives). The hardness of the ore is another factor that may influence the amount of impact noise, and can vary from mine to mine.

In group II, the physical parameters of the mill typically affect the grinding mechanism and efficiency, but they may also

TABLE 2.2 GENERAL FACTORS THAT INFLUENCE MILL NOISE.

I.	**Grinding Process**
	Method of Grinding
	Wet vs Dry Grinding
	Ore Hardness
II.	**Physical Parameters**
	Mill Size (diameter and aspect ratio)
	Grinding Media
	Liners
	Mill Speed
	Power Consumption (drive equipment)
III.	**Mill Specific**
	Wear Rate (liners)
	Number of Mills Running
	Amount of Auxiliary Equipment
	Floor Openings
	Spacing between Mills
	Damping of Shell
	Acoustic Environment of Plant
	Microphone Location

increase the noise level. For example, mill diameter, aspect ratio, and mill speed are chosen for maximum grinding efficiency. Sufficient data do not exist to quantify the effects of these parameters on the noise level, however, it is likely they are significant factors.

The amount and size of the grinding media added to the charge have a direct effect on the noise level, and this is discussed further in Sec. 2.2. The power consumption of the mill determines the size of the motor and gear drives. The significance of the noise from auxiliary equipment will depend, in part, on the level of grinding mill noise.

Group III includes factors that are specific to individual mills, and to the acoustic effects in the measurement of the noise level. The amount of liner wear appears to affect the noise, particularly on the falling side of the mill. The number of other mills and amount of auxiliary equipment operating at any one time is also a significant factor. If there are large openings or steel grates around the sides of the mill, noise from underneath the mill can be reflected by the lower levels. The plant reverberation time and the extent of the acoustic near field are additional acoustical parameters that affect the overall noise level.

Many of the factors discussed here are dependent on one or more of the other parameters. For instance, ore hardness will affect the quantity of grinding media added, the power consumption, and the liner wear. Consequently, with the present program, it has been possible to identify the effects of only a few of these parameters. These parameters and the associated noise sources are discussed in full in the next section.

2.2 Mill Noise Sources: Measurement and Analysis

This section presents the results of noise and vibration measurements made on specific mills in order to quantify the noise sources identified in Phase I [5] of this program. In addition, the effects of the various grinding parameters and mill equipment on the overall plant noise level are investigated.

2.2.1 Mill shell radiation

It was apparent from previous measurements [5] that the largest mill noise source was radiation from the falling side and the lower half of the mill shell (see Fig. 2.5). The first step in the measurement program was, therefore, to determine how much vibration was transmitted from the mill liners to the mill shell, and finally radiated as noise. The outer shell of a large 27-ft-diameter semi-autogenous mill was instrumented with a series of accelerometers, and the vibration levels were monitored over a period of time under normal operating conditions.

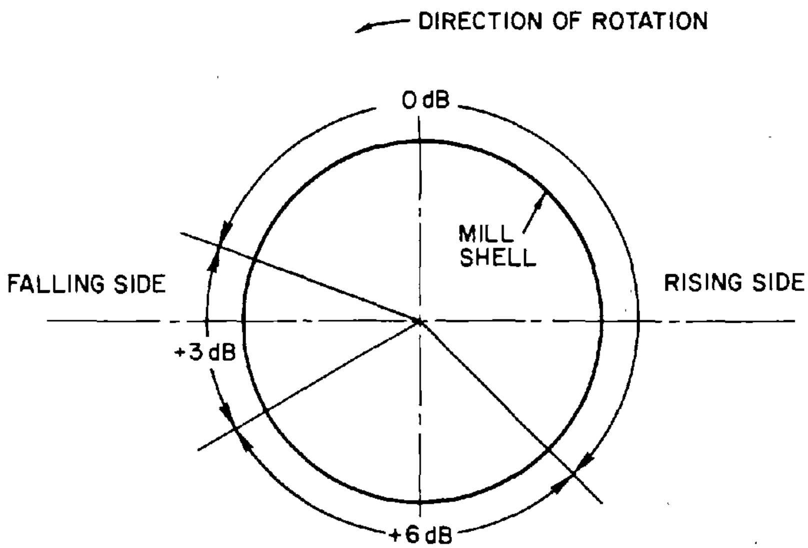


FIG. 2.5 RELATIVE NOISE RADIATION FROM DIFFERENT PORTIONS OF ROD MILL SHELL.

Instrumentation System

The instrumentation system consisted of up to 10 accelerometers mounted on the mill shell and end caps, connected to a 16-channel multiplexor and a telemetry system (see Fig. 2.6). The telemetry system then transmitted the vibration data to various signal conditioning and tape recorder equipment. A more detailed description of the multiplexor and telemetry system is given in Appendix A. The accelerometers were mounted to the mill shell via 1-1/2-in.-diameter steel pads, which were previously welded to the mill shell. The accelerometers were located on an axial line around the shell, as shown in Fig. 2.7. Position Nos. 2, 7, and 8 were attached to the feed and discharge end caps, while channel #1 was used as a dummy signal to check instrumentation noise. Cable leads were supported by special brackets welded to the shell and were fed back to the multiplexor and telemetry system.

The multiplexor was mounted to the edge of the feed end cap. A base plate was welded between two liner bolts, and the

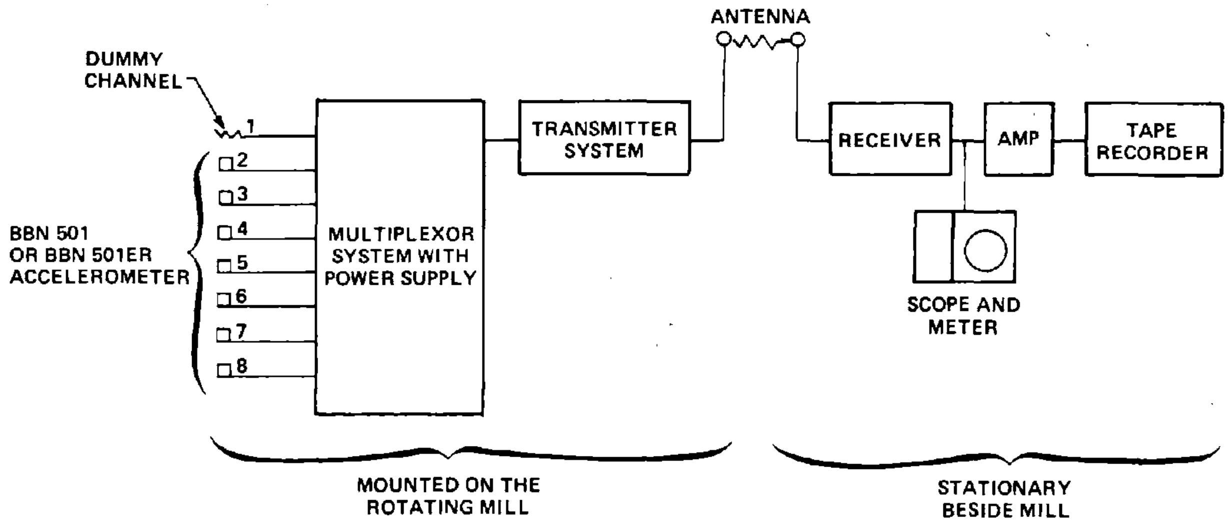


FIG. 2.6. INSTRUMENTATION SYSTEM FOR VIBRATION MEASUREMENTS ON ROTATING GRINDING MILLS.

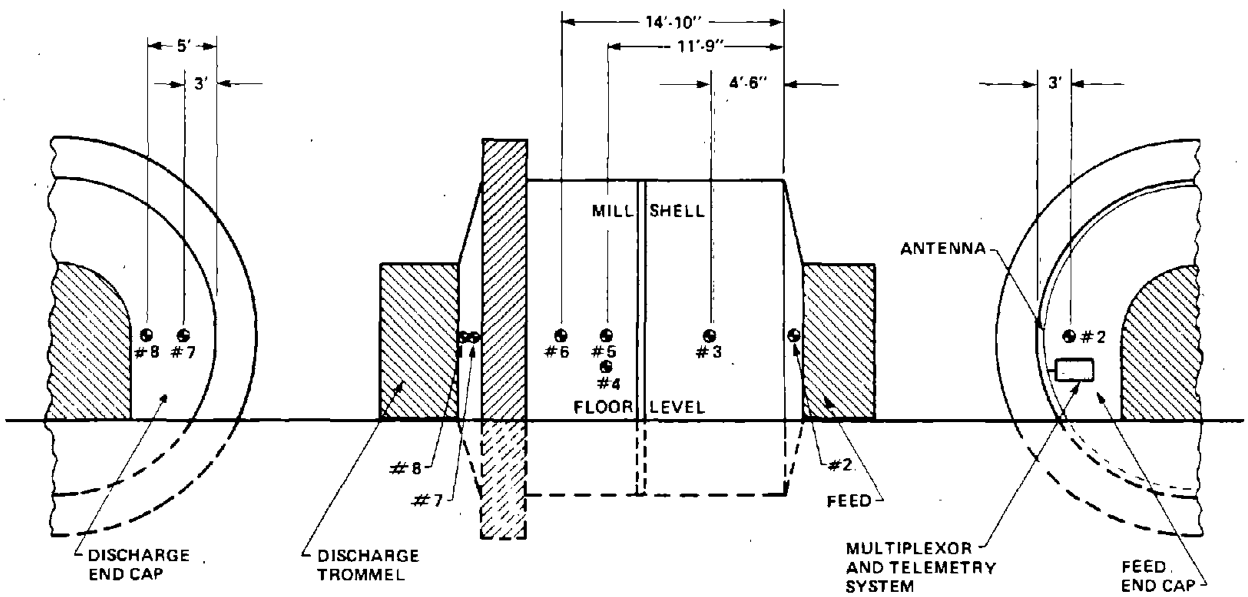


FIG. 2.7. ACCELEROMETER LOCATIONS ON THE SHELL AND END CAPS OF A 27-FT-DIAMETER SEMI-AUTOGENOUS GRINDING MILL. NUMBERS REFER TO CHANNEL IDENTIFICATION.

outer case of the multiplexor attached to this mounting. Sponge foam and rubber were used to vibration isolate the system from the base plate and the mill shell (see Fig. 2.8). An FM trans-

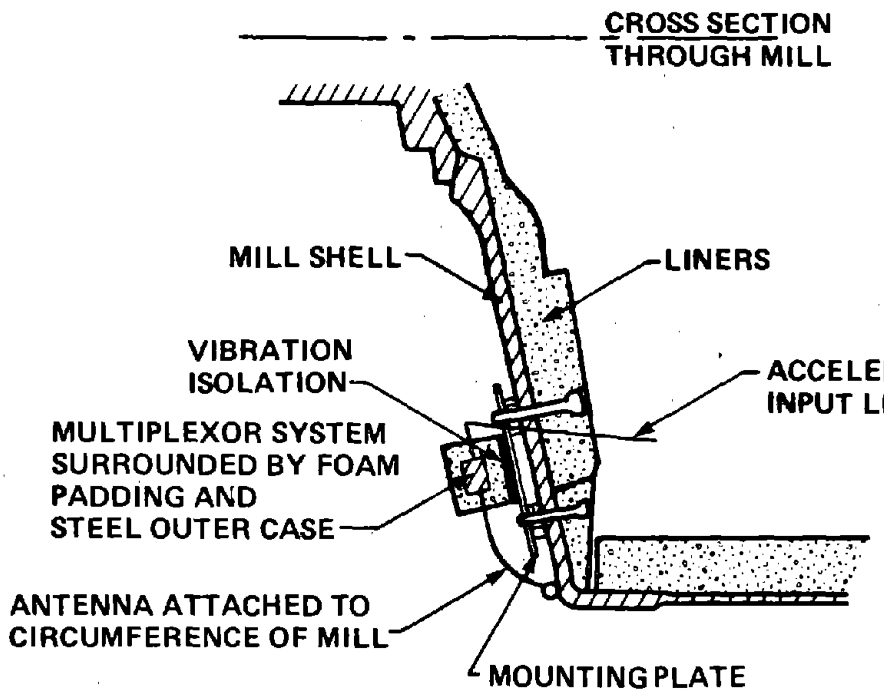


FIG. 2.8. MOUNTING ARRANGEMENT OF THE MULTIPLEXOR AND TELEMETRY SYSTEM TO THE GRINDING MILL.

mitter antenna was attached to the circumference of the feed end cap. Figure 2.9 shows the antenna mounted to the mill, providing continuous antenna coverage during each rotation of the mill. Steel brackets with rubber isolator grommets were attached to the end cap and used to support the antenna wire.

Measurement Results

The mill was run with the measurement system installed and vibration data were obtained at regular intervals for several days. The average throughput of taconite during this time was 281 long tons/hr, with an average power consumption of 5446 kWh. Typical acceleration signals from four complete mill rotations are shown in Fig. 2.10.

The vibration levels have a large dynamic range with most of the energy concentrated in only a small part of the mill rotation cycle. This indicates that only a small portion of the mill

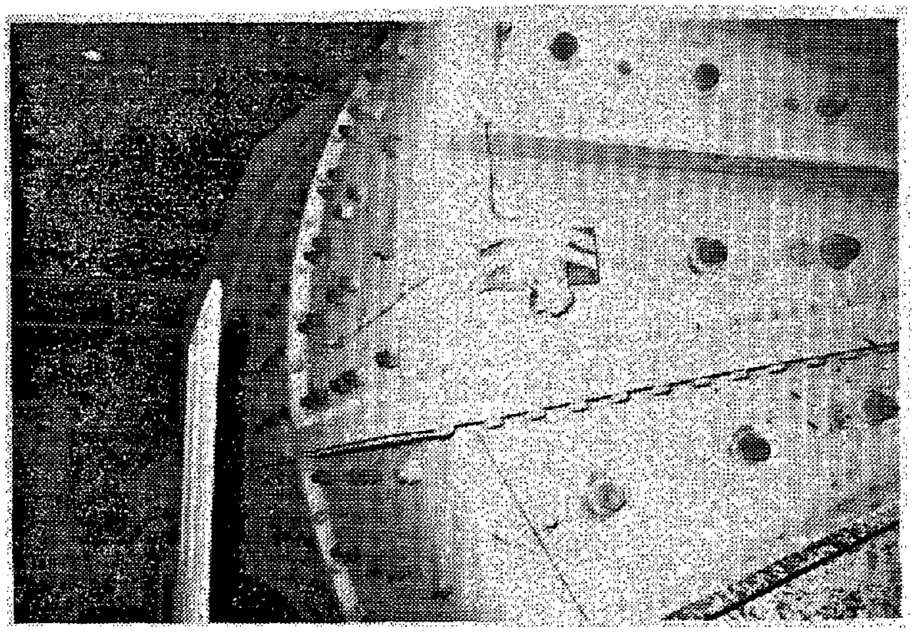


FIG. 2.9. CLOSEUP OF CIRCUMFERENTIAL ANTENNA ATTACHED TO THE MULTIPLEXOR SYSTEM. NOTE ACCELEROMETER LEADS RUNNING ALONG THE LENGTH OF THE MILL.

shell is subjected to high acceleration levels that radiate sound energy. The reference line in Fig. 2.10 indicates the position of the accelerometers moving past the mill floor level on the falling side of the mill. The peak acceleration levels, therefore, occur on the falling side of the mill, from a few feet above the mill floor to the lower half of the mill. Note that with a mill speed of 11.46 rpm, the approximate perimeter speed of the accelerometers is 16.2 ft/sec, and the duration of the peak acceleration is approximately 1.25 secs, which corresponds to 20.3 ft or almost exactly a quarter of the mill shell perimeter.

Figure 2.10 also shows the acceleration level from channel #4, located on the end of a liner bolt. The peak acceleration levels are higher than on the mill shell and the peak duration is shorter. One might expect this since a liner bolt is much lighter in comparison to a section of the mill shell, and also since it is connected directly with the inside of the mill.

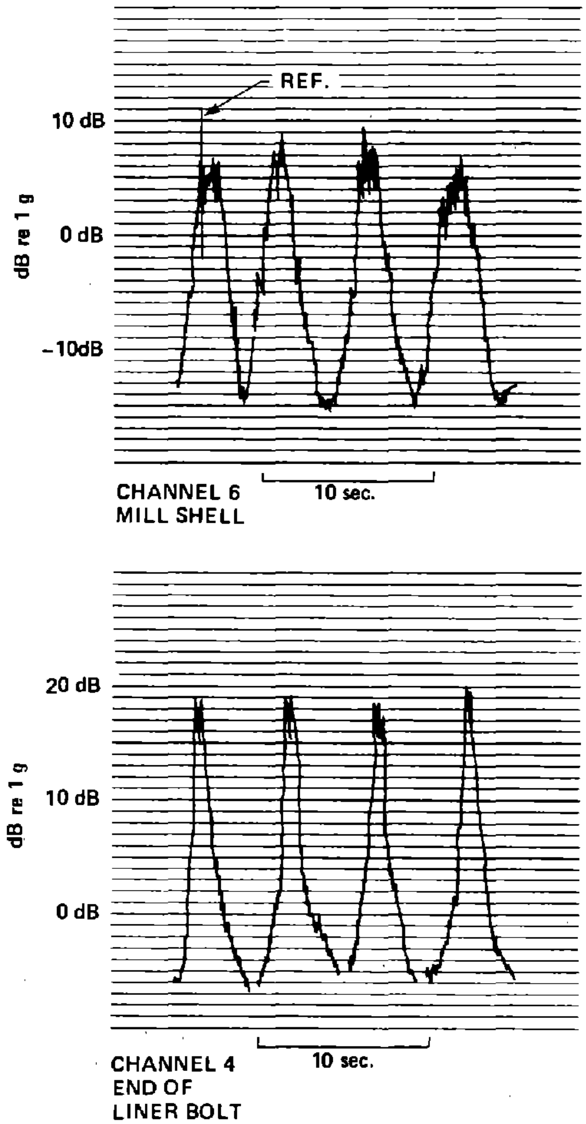


FIG. 2.10. ACCELERATION LEVELS ON THE SHELL OF A ROTATING SEMI-AUTOGENOUS MILL. FOUR MILL ROTATIONS.

Typical traces from other positions on the shell and end caps are shown in Fig. 2.11. Data from the mill shell (channels #3 and #5) appear very similar to channel #6 in Fig. 2.10, while the vibration on the end caps (channels #7 and #8) shows slightly different waveforms. Impacts toward the outside perimeter of the discharge end cap (channel #7) appear for only a short period of time, while closer to the center of the discharge (channel #8) multiple impacts occur.

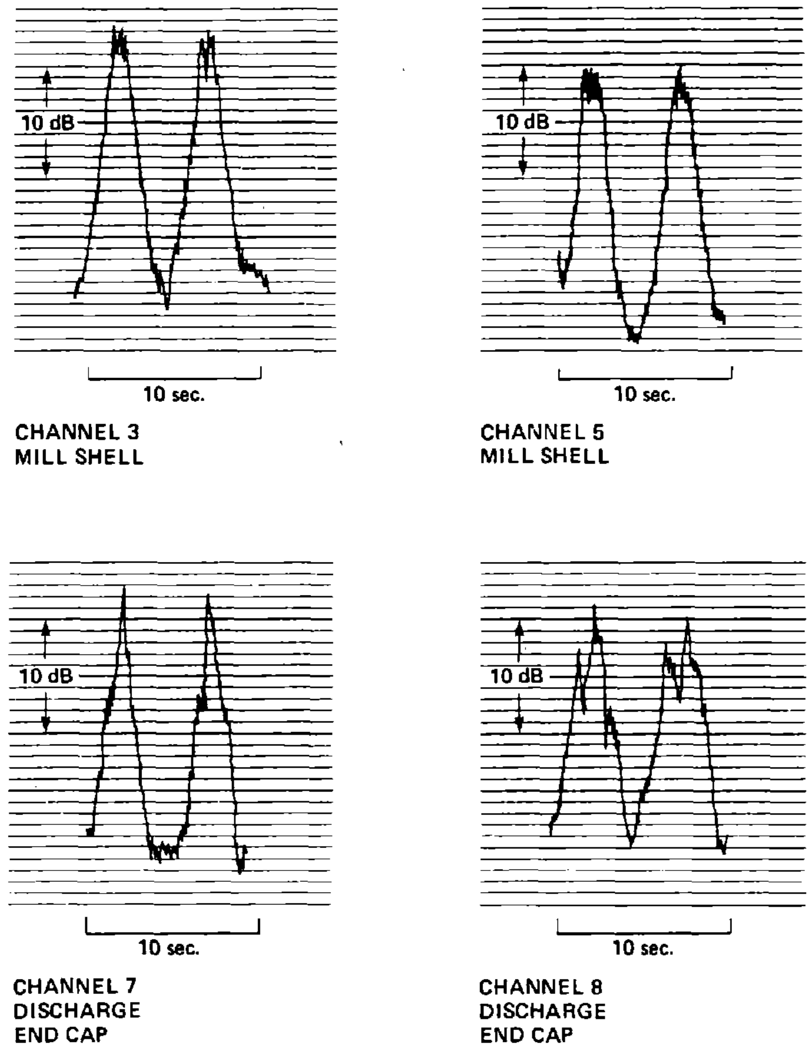


FIG. 2.11. TYPICAL ACCELERATION LEVELS ON THE SHELL AND END CAPS OF SEMI-AUTOGENOUS MILL.

One-third octave band acceleration spectra are shown in Fig. 2.12 for three accelerometer channels in which the signals were analyzed using a 1-sec integration time over the signal peak. The spectra show levels dominated by energy in the frequency range 500 Hz to 5 kHz. The instantaneous levels on the discharge end cap (channel #7) are shown to be significant.

Previous noise measurements made close to the mill shell indicated that the noise spectrum peaked in the same frequency

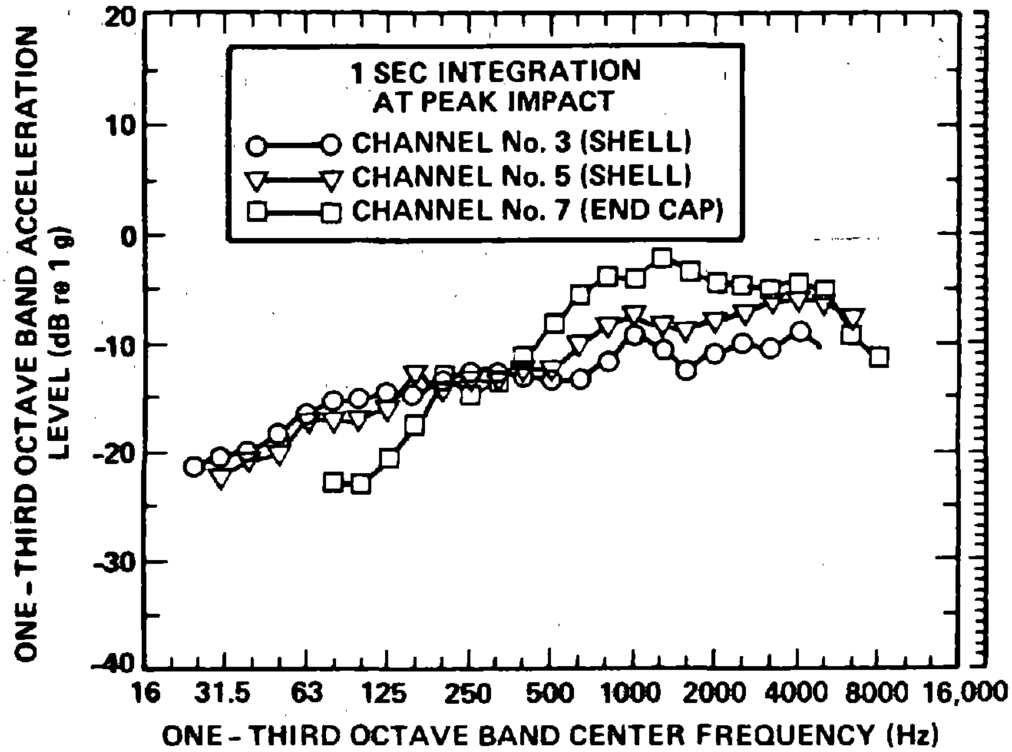


FIG. 2.12. ONE-THIRD OCTAVE BAND ACCELERATION LEVELS MEASURED ON THE SHELL OF SEMI-AUTOGENOUS MILL.

range as these acceleration measurements. The acceleration time histories filtered in the 500-Hz, 1-kHz, and 2-kHz octave bands are shown in Fig. 2.13. The filtered vibration levels from the mill shell (channels #5 and #6) are very similar to the overall time histories, and the peak impacts occur soon after the mill floor level (marked REF on Fig. 2.13) is passed. For channel #7, on the discharge end cap, the impacts are much lower at this reference point, although the peak impact acceleration occurs at the same position for all channels.

From the results above, it appears that impacts from the steel balls and contact with the charge on the falling side of the mill are responsible for most of the mill shell vibration levels. This vibrational energy is contained in the frequency range 500 Hz to 5 kHz; there is a very quick falloff in these frequencies after the peak level occurs. A reduction or elimina-

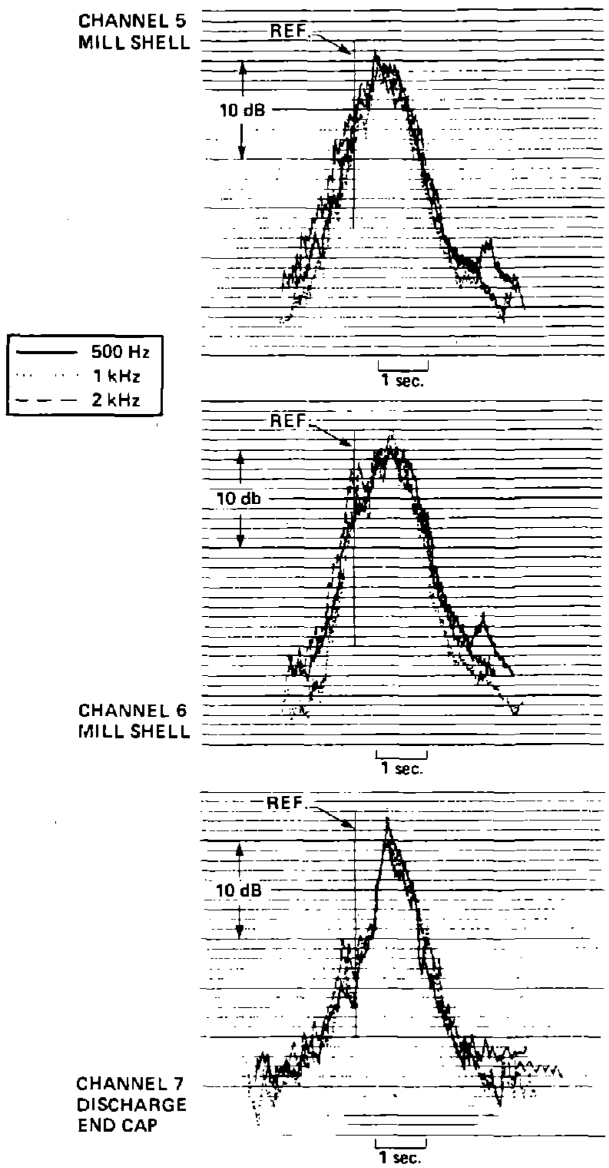


FIG. 2.13. ACCELERATION LEVELS FROM SHELL OF GRINDING MILL FILTERED INTO OCTAVE BAND COMPONENTS.

tion of these acceleration levels would provide a considerable reduction in the overall level. Alternatively, if the noise level around that portion of the mill where the peak acceleration levels occur was eliminated, there would be potential for noise reduction. From the speed of mill rotation, one can estimate the shell area where the peak acceleration occurs. As an example, Fig. 2.13 shows that the -10 dB values occur approximately 4 ft above the reference point and 20 ft below it. Also, the results show that the vibration levels on the end caps are significant, and that these areas will also have to be treated.

2.2.2 Directional microphone measurements

A series of directional microphone noise measurements were made around the shell of a semi-autogenous mill. The purpose of these measurements was to use the results from the acceleration data, discussed above, and to map out the relative acoustical source strengths around the mill shell. A directional microphone was used in these tests to reduce the contributions of background noise levels from other mills and from the reverberant environment in the concentration plant.

Measurements

A directional microphone was built and tested at BBN. When the microphone was rotated 90° relative to the source direction, the response was more than 10 dB below the reference level*. At angles greater than 90°, the response was better than 15 dB.

After initial test and calibration, the microphone was used to measure noise from the shell of the 27-ft-diameter semi-autogenous grinding mill. Measurements were made around the complete perimeter of the mill, as shown in Fig. 2.14. At each position the directional microphone was held approximately 16 in. from the rotating shell, and four source directions† were measured. Results with the microphone pointing into the mill are shown in Fig. 2.15 for seven positions around the shell. The data clearly illustrate that the highest noise levels are around

*Reference level was defined as the level measured with the microphone pointing directly at the source.

†Directions are: Into the mill, out from the mill, upwards and downwards.

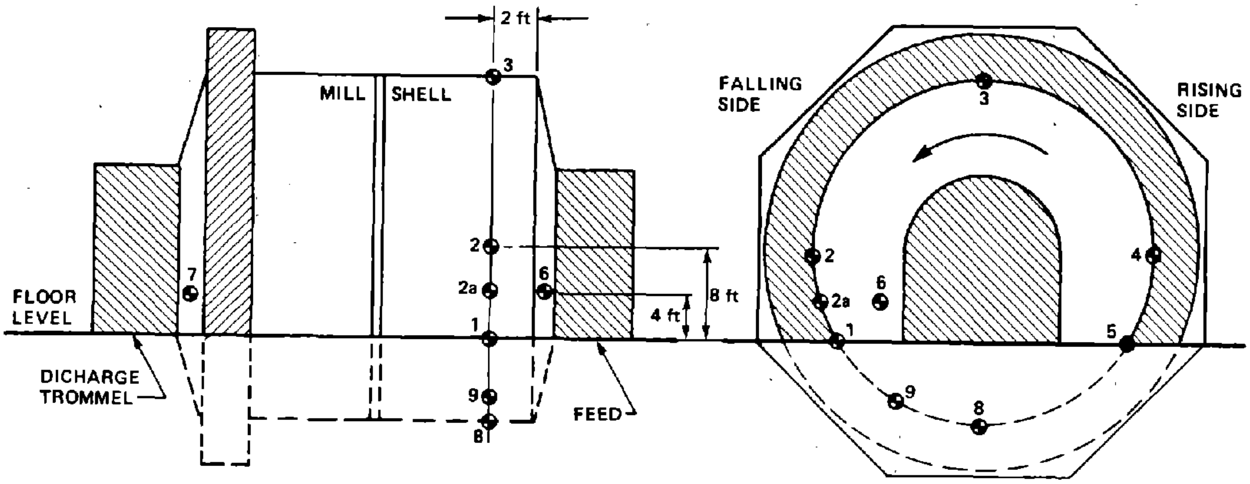


FIG. 2.14. DIRECTIONAL MICROPHONE POSITIONS ON SEMI-AUTOGENOUS MILL.

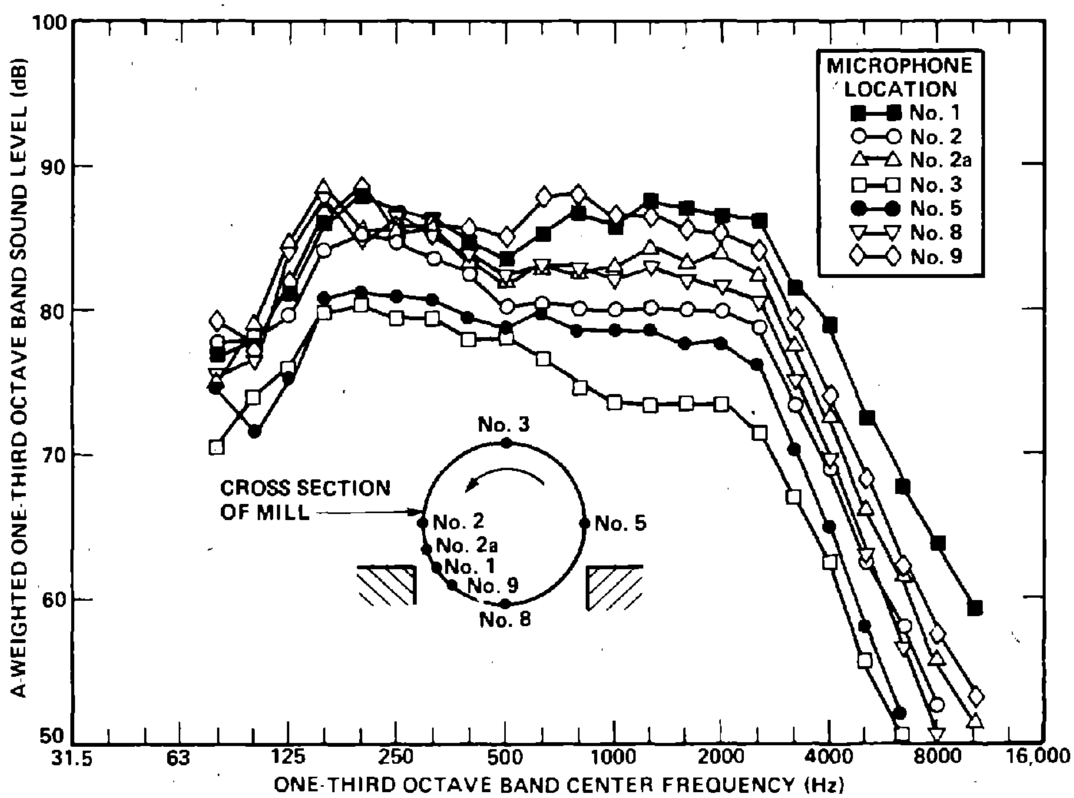


FIG. 2.15. ONE-THIRD OCTAVE BAND SPECTRA FOR SEVEN LOCATIONS ON GRINDING MILL. DIRECTIONAL MICROPHONE.

a narrow range on the falling side of the shell, and that levels on top of the mill and on the rising side are significantly lower. When the directional microphone is rotated relative to the mill shell, one obtains spectra typical of that shown in Fig. 2.16. At position #1, close to the mill impact and charge grinding area, the noise is coming directly from the shell and from below the mill. Note that the mill is mounted between two floor levels, and below the mill there is a built-in concrete floor and wall area. It is highly reverberant below the mill and noise is reflected from the floor to the upper levels. Measurements made with the microphone pointed upwards or outwards from the shell are 5 dB lower than with the microphone pointed into the mill.

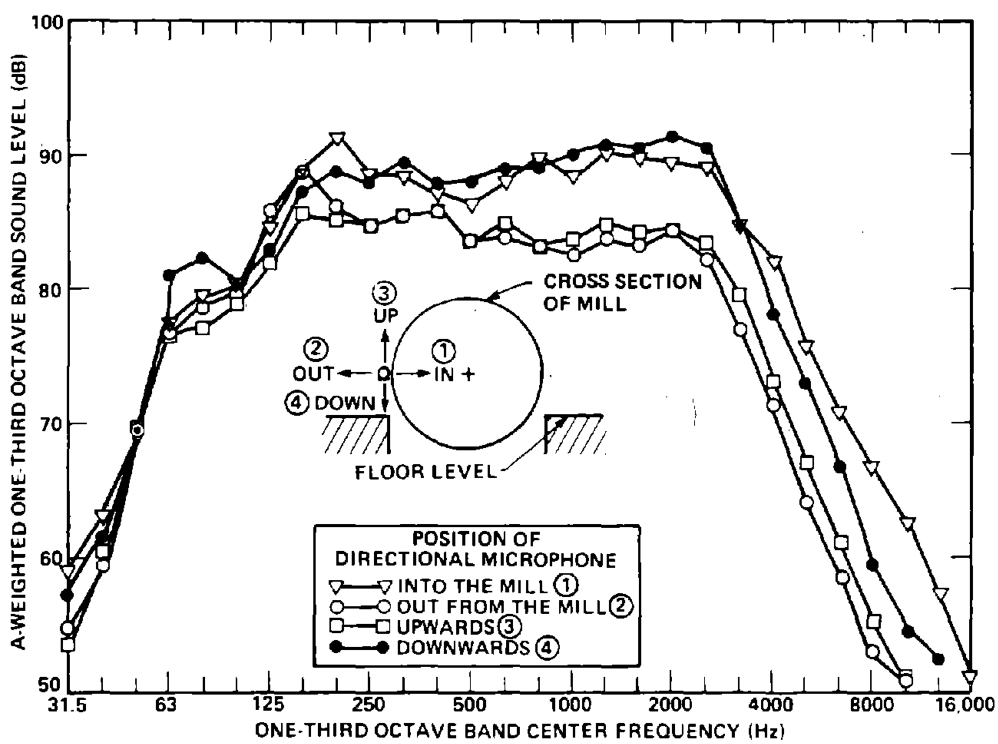


FIG. 2.16. ONE-THIRD OCTAVE BAND SPECTRA OBTAINED WITH DIRECTIONAL MICROPHONE AT POSTION #1.

At the midpoint of the falling side (position #2), the dominant noise is coming from the downward direction, rather than directly into the shell (see Fig. 2.17).

Analysis of Source Strengths

Using these measurements, and the results from Sec. 2.2.1, the relative source strength for different areas around the mill shell was calculated. These areas are illustrated in Fig. 2.18, where the levels are referenced to 0 dB for the falling side of the mill. If one assumes that these areas have relatively uniform nearfield sound pressure levels, one can calculate an equiv-

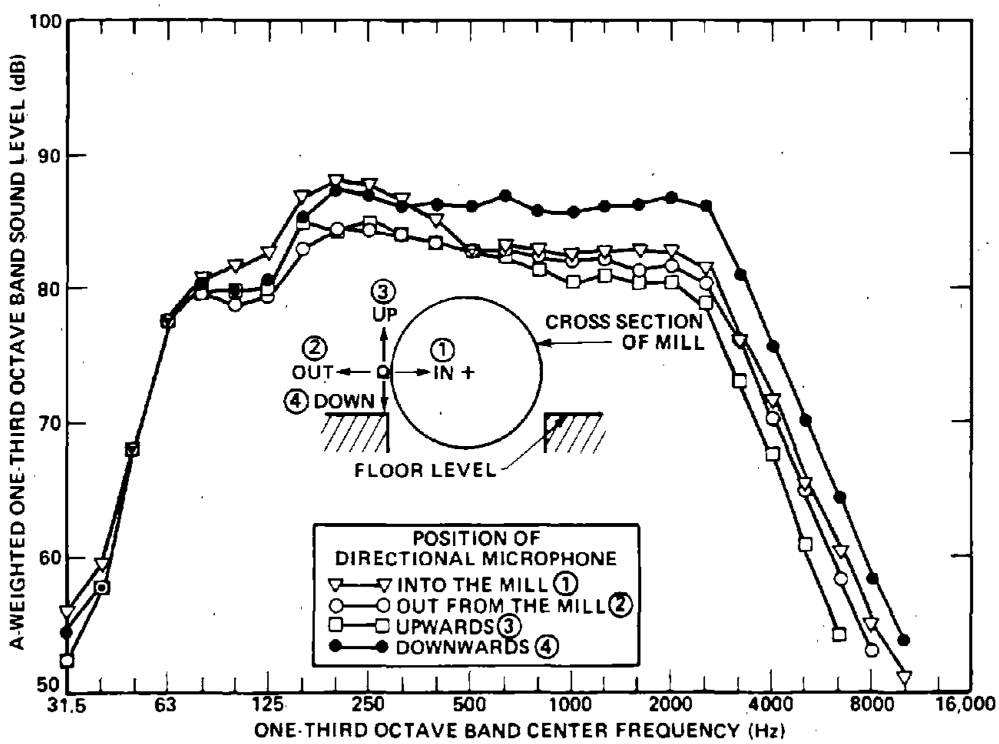


FIG. 2.17. ONE-THIRD OCTAVE BAND SPECTRA OBTAINED WITH DIRECTIONAL MICROPHONE AT POSITION #2.

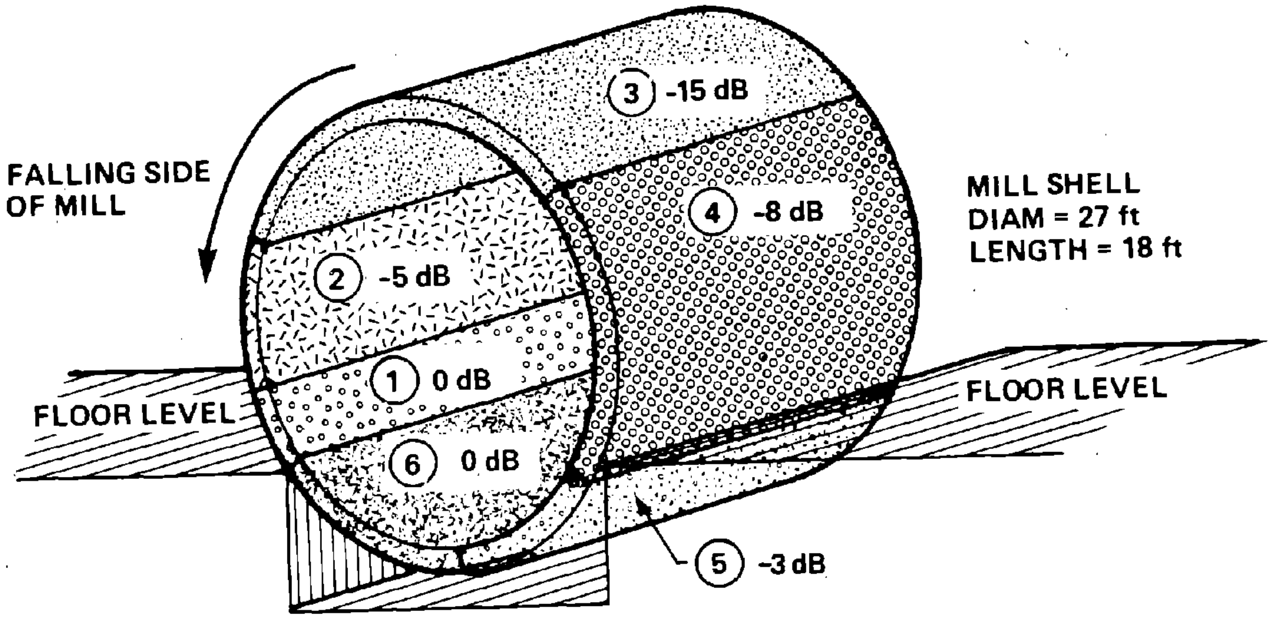


FIG. 2.18. MILL SHELL DIVIDED INTO RELATIVE POWER LEVELS FOR SPECIFIED AREAS OF MILL.

alent radiating area, and hence an equivalent sound power level. These calculations are shown in Appendix B for the complete mill shell, including both feed and discharge ends. On the basis of these results, the potential noise reduction from a noise barrier for the mill shell and/or the end caps was calculated. The reductions are summarized in Table 2.3 with detailed calculations given in Appendix B.

TABLE 2.3. POTENTIAL MILL NOISE REDUCTION WITH PARTIAL BARRIERS.

Portion of Semi-Autogenous Mill Shell Enclosed	Noise Reduction (dB)
Falling side of shell only	1.4
Falling side of shell and adjacent end caps	3.0
Lower third of shell and both end caps	5.2

A partial barrier over the falling side of the mill, as well as on the feed and discharge end caps, will be required to obtain at least a 3 dB noise reduction. Covering the shell alone will not provide a significant reduction. The discharge end, in particular, is a noisy area and requires treatment on the falling side. The areas underneath the mill are also important sources; when these areas, plus the adjacent end cap areas, are covered, a 5 dB noise reduction can be obtained.

These calculated results were tested when a partial mill barrier was designed and constructed around an operating mill. The results from the partial barrier are described in Sec. 2.4.

2.2.3 Noise level as a function of grinding media and liner wear

Some of the parameters that affect the noise level of a grinding mill were outlined in Sec. 2.1.3. In this section, two of the physical parameters, grinding media and liner wear, are described in terms of measurements made on specific grinding mills.

Media Design Parameters

In semi-autogenous mills and rod mills, the size of the grinding media is one of the principal factors that affect the efficiency and capacity of the grinding process. For any given mill application, the sizing of the media involves two major opposing factors.

1. As the diameter of the media increases, the pressure between the surfaces increases, allowing larger particles to be broken up.
2. For small diameter media, the surface area available for grinding increases, resulting in increased grinding capacity.

The sizes used in rod mills vary from 1-1/2-in. to 5-in.-diameter rods, and in semi-autogenous mills, 3/4-in. to 5-in.-diameter balls.

The volume of media added is calculated as a percentage of the total mill volume (usually 3 to 9%), and is dependent on the required grinding efficiency and on economic factors. A high percentage of grinding balls will provide an efficient grind, but will require a larger power consumption, as well as higher metal wear of media and mill liners. Figure 2.19 shows the power consumption per long ton of output for increasing amounts of balls added to a typical semi-autogenous mill.

The consumption of media in wet grinding results from two processes. The first is surface abrasion by contact with the charge and the liners. The second is corrosion of the freshly exposed media surface. Typical consumption in wet grinding mills varies from 0.5 to 2.0 lb of media/ton of charge.

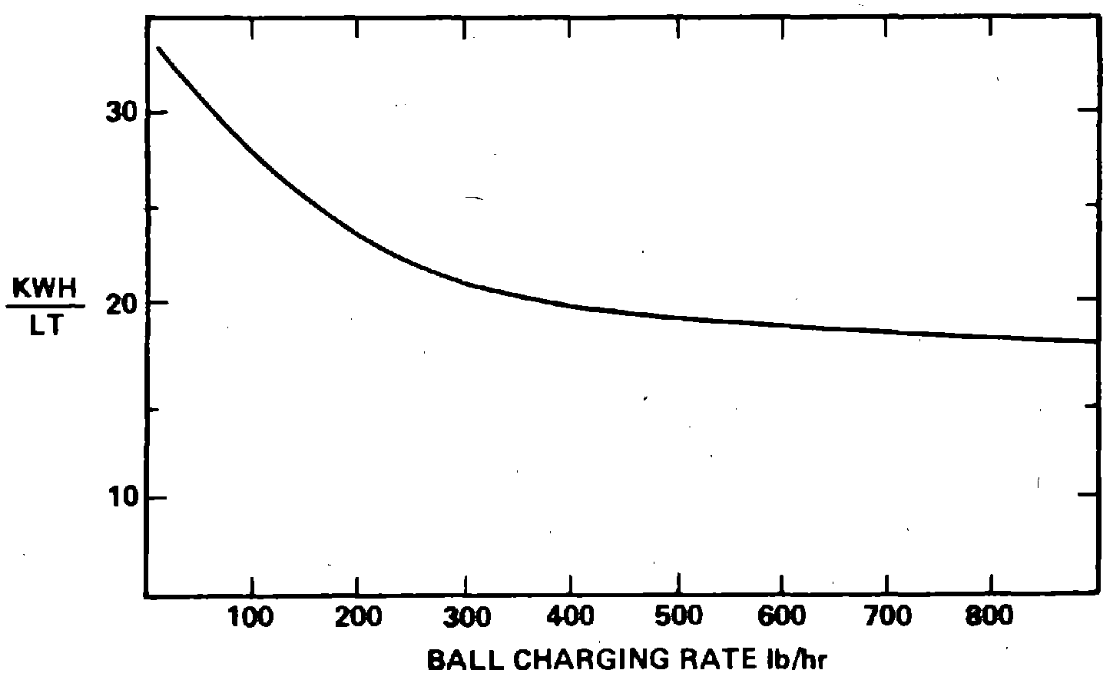


FIG. 2.19. TYPICAL POWER CONSUMPTION IN KWH/LONG TON VERSUS BALL CHARGING RATE FOR A LARGE SEMI-AUTOGENOUS MILL.

Variation of Noise Level with Grinding media

The noise levels of semi-autogenous mills operating with different quantities of grinding media (4-in.-diameter balls) were measured. These measurements had to be obtained during production conditions, since it was not possible to run a single mill under carefully controlled tests. Consequently, the results given in the next two sections should be regarded as general trends rather than absolute levels for all mills.

Figure 2.20 shows the variation in noise level vs the amount of grinding media added to the charge of a 27-ft-diameter semi-autogenous mill. The microphone was located 2 ft from the falling side of the mill, which ensured its being in the near field of the mill being tested. The increase in level due to the addition of grinding media appears to be approximately 4 to 5 dBA

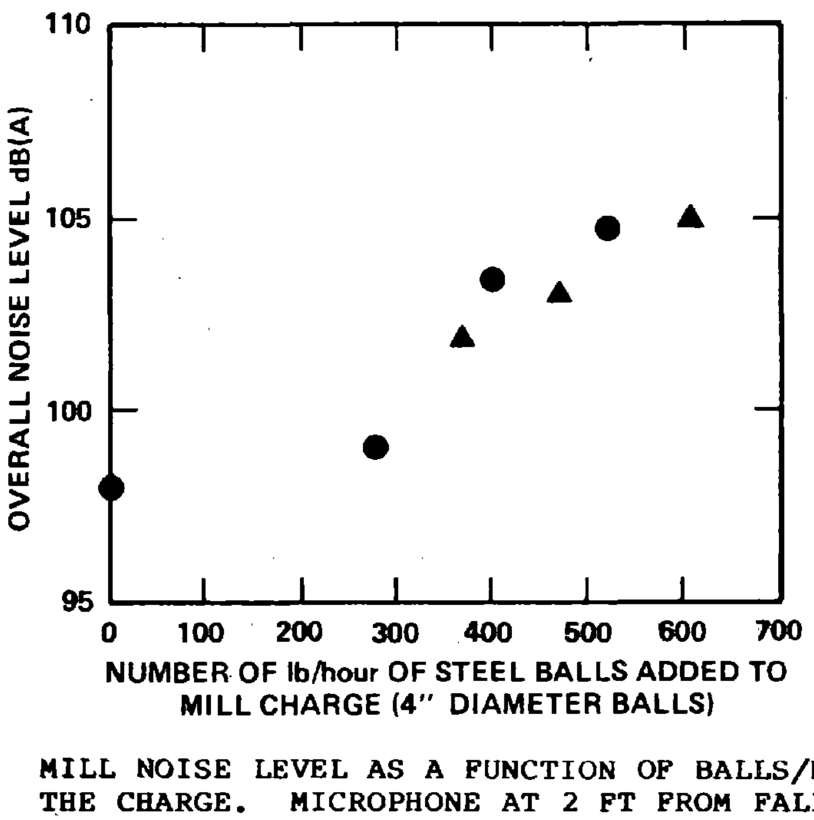


FIG. 2.20. MILL NOISE LEVEL AS A FUNCTION OF BALLS/HOUR ADDED TO THE CHARGE. MICROPHONE AT 2 FT FROM FALLING SIDE OF SHELL. DATA TAKEN WITH APPROXIMATELY 1900 HR OF LINER WEAR.

per doubling of media weight added. Note that the data point with 0 lb/hr of media was for an autogenous mill. The corresponding one-third octave band spectra are shown in Fig. 2.21 for three different amounts of media addition. The increase in noise level is only at frequencies above 500 Hz where the shell impact noise is the dominant source. At lower frequencies, the motor and auxiliary drive equipment noise levels are basically the same. Note that these spectra were obtained from three mills that had approximately equal amounts of liner wear (1500 hr). This is important, since the next section shows that noise level is also a function of liner wear.

Variation of Noise Level with Liner Wear

The parameters that affect liner wear are complicated and include many of the same factors that were discussed in the previous section. In addition, the type of liner material, the

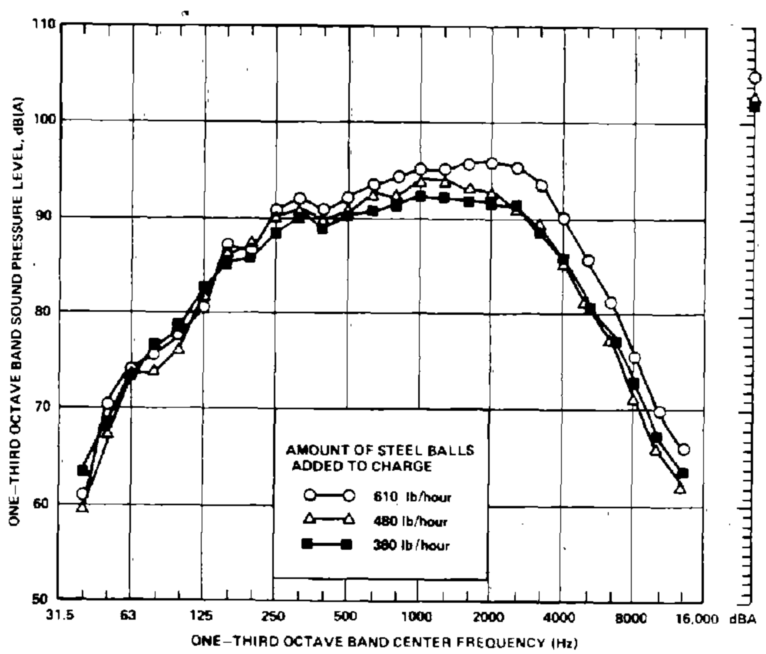


FIG. 2.21. ONE-THIRD OCTAVE BAND NOISE LEVEL FROM SEMI-AUTOGENOUS MILL FOR DIFFERENT AMOUNTS OF BALLS ADDED TO THE CHARGE. (MICROPHONE AT 2 FT FROM FALLING SIDE OF MILL SHELL.)

shape and size of liner, and the ore hardness are all factors. A typical liner configuration for an autogenous or semi-autogenous grate discharge mill is shown in Fig. 2.22. The main shell liners or rails (Nos. 7, 8, and 9) have lifters that are 6 in. high, and these areas (especially at the feed end) usually have the highest wear in the mill. Table 2.4 lists the estimated life expectancy for these particular liners on a semi-autogenous mill. Item numbers refer to the liners indicated in Fig. 2.22. For this mill, the rails have a life expectancy of 2800 hr, or approximately 5 to 7 months, depending on plant use.

TABLE 2.4. LIST OF MILL LINERS ON ALLIS-CHALMERS 18-FT x 27-FT DIAMETER SEMI-AUTOGENOUS GRINDING MILL.

Item No.	No. Required Per Mill	Description	Estimated Weight (lb)	Life Expectancy (Hr)
1	14	Feed Head Throat Liner	800	40,000
2	14	Feed Head Inner Liner	1040	40,000
3	28	Middle Feed Head Liner	720	7,200
4	28	Feed Head Liner	2610	1,800
5	14	Outer Ring Liner	1350	3,600
6	28	Feed Head Corner Block	310	20,000
7	56	Shell Liner - Feed End	1775	2,800
8	56	Shell Liner - Center	1945	2,800
9	56	Shell Liner - Discharge End	1860	2,800
10	28	Grate - 3/4 in. Radial Slots	440	5,000
26	28	Discharge Head Filler Ring	600	20,000
27	28	Outer Discharger	750	20,000
28	28	Middle Outer Discharger	615	30,000
29	28	Middle Discharger	560	30,000
30	28	Middle Inner Discharger	460	30,000
31	14	Inner Discharger	360	30,000
32	7	Discharge Cone - Right	2170	20,000
33	7	Discharge Cone - Left	2170	20,000
34	28	Middle Wear Plate	1290	10,000
35	28	Inner Plate	285	30,000
36	28	Lifter Bar Outer	375	2,500
37	28	Lifter Bar Middle	765	2,500
38	14	Lifter Bar (Long)	410	18,000
39	14	Lifter Bar (Short)	320	18,000

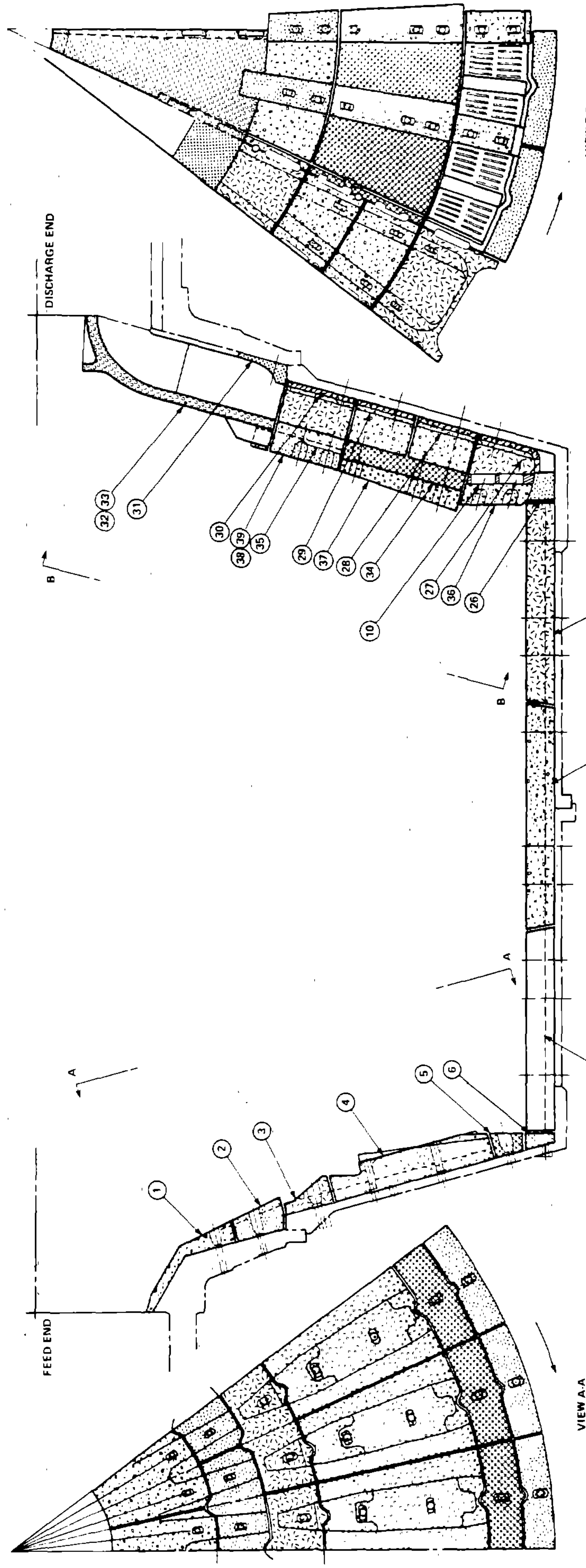


FIG. 2.22. TYPICAL ARRANGEMENT OF LINERS ON ALLIS-CHALMERS SEMI-AUTOGENOUS MILL. NUMBERS REFER TO ITEMS LISTED ON TABLE 2.4.

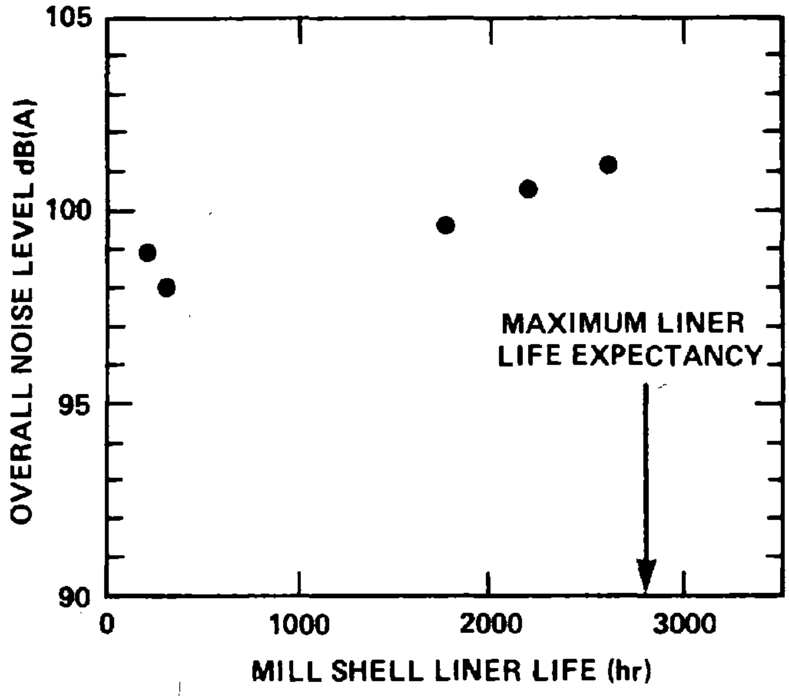


FIG. 2.23. MILL NOISE LEVEL VERSUS LINER WEAR IN HOURS. MICROPHONE AT 2 FT FROM FALLING SIDE OF MILL SHELL.

For one particular mill, the noise level was monitored at different stages during the liner wear cycle. Figure 2.23 shows the results from these measurements, which indicate an increase in noise level with increasing liner life. The microphone was located 2 ft from the falling side of the mill shell and the rate of grinding media addition was kept constant.

This increase in noise level is not entirely expected. One might predict that as the lifters wear down, the charge will not rise as high, and the impact energy of the charge hitting the shell will be reduced. However, a number of other factors must be taken into account. At the beginning of a wear cycle, the liners will have had new backing material installed, and the liner bolts will have been correctly torqued down. After some wear, the bolts may become loose, and the backings may wear out, resulting in conditions for higher impact noise on the shell.

Additional measurements on other types of mills over complete liner wear cycles are necessary to define this effect further.

2.2.4 Noise from a single mill

One of the problems associated with noise measurements in concentrator plants is background noise contribution. The highly reverberant environment and the large number of machines running simultaneously make it almost impossible to measure the noise from a single mill. Usually one has to rely on nearfield measurements and assume little contribution from other sources.

However, during one off-peak production period, a complete concentrator plant was shut down except for a single mill line.* From measurements of the noise from this one source, it was possible to determine exactly the contributions from all other plant equipment, and also to compare the single mill noise data with vibration data from Sec. 2.2.1.

Measurement

The noise from a single primary mill was measured and then monitored as additional mills were switched back into operation. The results from these measurements are summarized in Fig. 2.24, which shows the one-third octave band spectra measured 15 ft from the falling side of mill #2.

As mill #2 was started up, no feed was applied initially, and then, after a few minutes, full feed was applied. As feed was applied, the overall level increased by 3 dBA to 95.5 dBA, and the increase was in frequencies above 250 Hz where grinding noise dominates. As expected, noise below 250 Hz, which is mainly motor and gear drive noise, remained the same. When an

*A mill line or circuit consisted of a semi-autogenous primary mill, a secondary ball mill, and one section of separators, screens, and pumps.

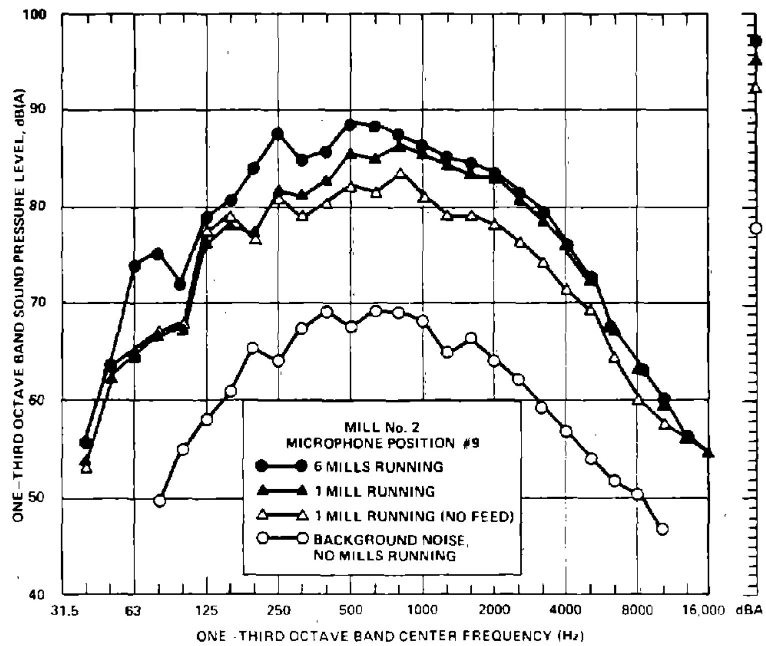


FIG. 2.24. ONE-THIRD OCTAVE BAND NOISE LEVEL MEASURED AT MILL #2 WHEN THE MILL IS OPERATING SINGLY AND WHEN A TOTAL OF FIVE ADDITIONAL MILLS ARE RUNNING.

additional five mills were switched on, the increase in noise at the position by mill #2 increased by 2 dBA. In this case, most of the increase in noise was in the low frequencies, because of the additional motors, gears, and secondary mills, that were activated. Plant background noise (when all mills were shut down), is the bottom curve shown in Fig. 2.24, and the overall noise level was 78 dBA.

The increase in noise level between one and six mills for 17 different positions around mill #2 is shown in Fig. 2.25. The location of these numbered positions are referenced in Fig. 2.26. Close to the mill shell and on the falling side of mill #2 (positions 6, 7, 8), the increase in noise from the additional mills is only 1 to 1-1/2 dBA. In the far field, and particularly on the rising side of the mill (positions 15, 16, 17), the increase is 3 to 4 dBA when all mills are running. These measurements indicate that the noise level at a location between the mills is

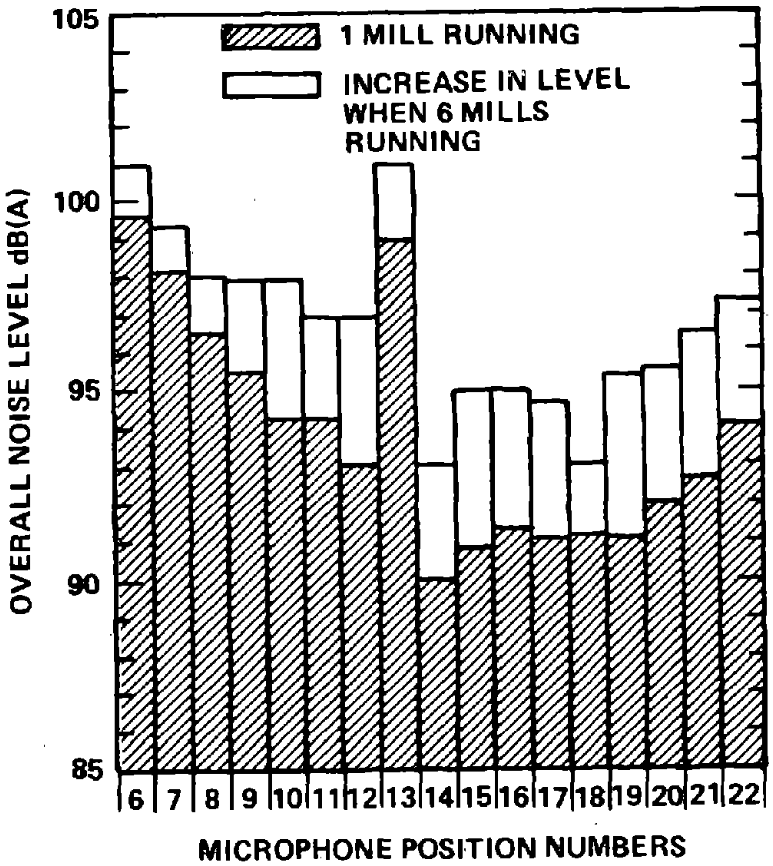


FIG. 2.25. COMPARISON OF THE NOISE MEASURED AROUND MILL #2 WHEN MILL #2 IS RUNNING SINGLY AND WHEN A TOTAL OF SIX MILLS ARE OPERATING. (MICROPHONE POSITIONS ARE REFERENCED IN FIG. 2.26.)

dominated by noise from the falling side of the closest mill, and that it is approximately 5 to 6 dBA higher than the noise from the rest of the plant equipment.

Comparison of Vibration and Noise Levels from a Single Mill

In this section, the noise measured from the single operating mill is compared with mill shell vibration data discussed in Sec. 2.2.1. The acceleration levels from the shell vibration can be used to compute a calculated sound pressure level close to the shell. There are significant problems with such a calculation because of the unknown radiation efficiency and vibrational

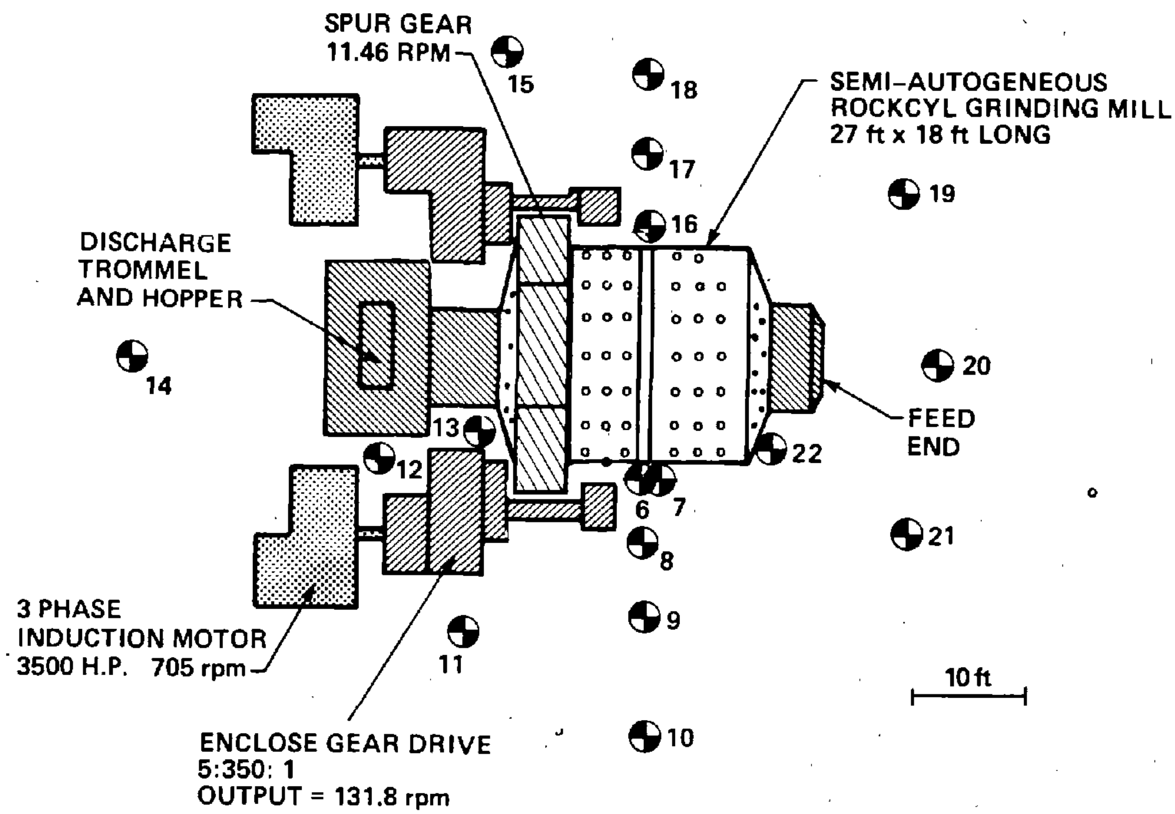


FIG. 2.26. MICROPHONE LOCATIONS AROUND SEMI-AUTOGENOUS MILL.

characteristics of the mill shell, as well as the difficulty in calculating the noise in the near field of such a large structure.

However, by making a number of simplifying assumptions, an estimate of this sound pressure level can be made. One approach to this calculation is to assume the nearfield sound pressure level (p) is related to the shell particle velocity (υ) by the expression $p = \rho c \upsilon$. Another standard method is to calculate the sound power level, π, as $\pi = \sigma \rho c A \langle \upsilon \rangle^2$, assuming a radiation efficiency of $\sigma = 1$, where A is some area of the vibrating shell.

Both of these calculations have been made and are shown in Fig. 2.27 as the shaded region. The measured sound pressure level is indicated by the data points and compares reasonably well with calculated levels. In spite of this very simplified

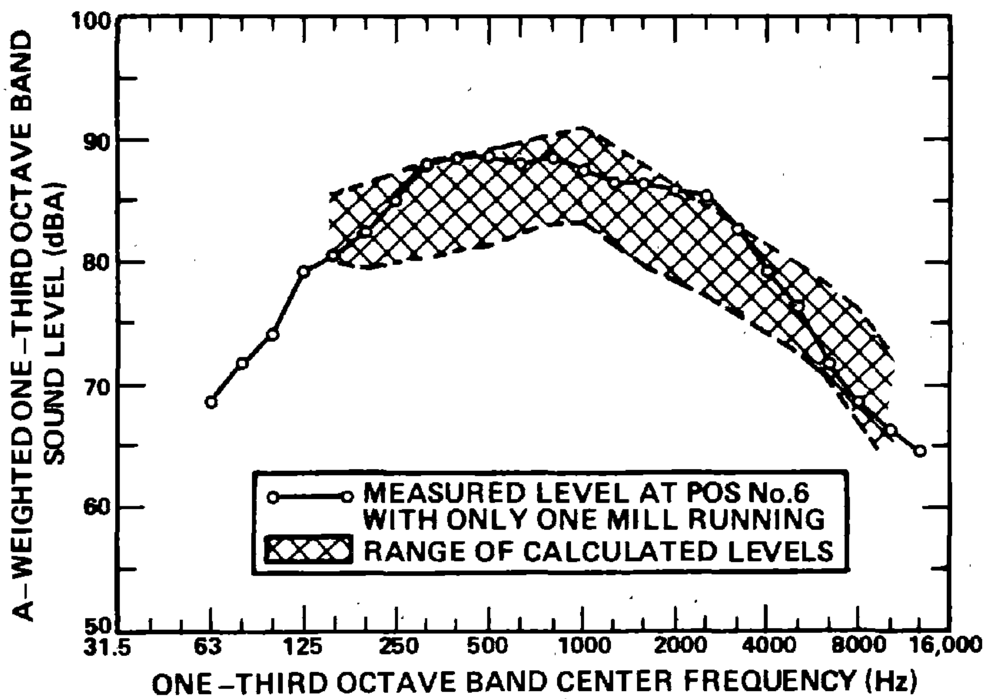


FIG. 2.27. COMPARISON OF THE RADIATED MILL NOISE CALCULATED FROM MILL SHELL VIBRATION MEASUREMENTS WITH THE MEASURED NEARFIELD.

analysis, these results compare sufficiently well to confirm that the dominant noise source is radiation from the mill shell.

2.2.5 Mill drive auxiliary equipment

In most taconite concentrator plants, the drive equipment is not the dominant noise source, and may in fact be completely masked by the mill shell noise. The objective of the present program was to investigate and control the dominant shell noise. However, a few comments regarding noise from the drive systems are included here because tones from the drive equipment can be important at certain frequencies.

A typical drive arrangement for a large primary mill is shown in Fig. 2.28. A dual pinion drive is used with a large spur gear mounted on the perimeter of the mill. The potential noise sources are the large (typically 3500 hp) motor drives and the single-stage-reduction gear boxes.

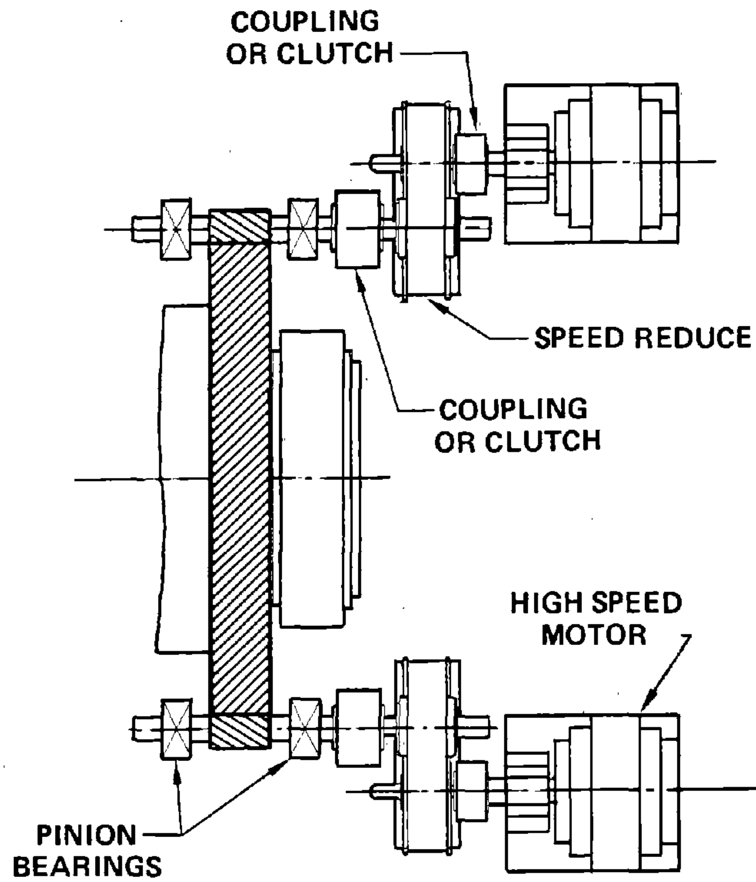


FIG. 2.28. TYPICAL MOTOR AND GEAR DRIVE ARRANGEMENT FOR PRIMARY GRINDING MILL.

Gear Noise

The one-third octave band noise level measured in an autogenous concentrator plant is shown in Fig. 2.29. A strong tone in the 315-Hz band is apparent throughout this plant. This tone was identified as coming from the Falk reduction gear boxes used on six of the grinding mills. A narrow band analysis of the noise measured 2 ft from the gear box is shown in Fig. 2.30. The spectrum is a classic gear noise shape with tones at low frequencies (<50 Hz) and at mid frequencies (>300 Hz). Table 2.5 lists the name plate specifications for the gear drive. From these, one can identify the low frequency tones as harmonics of

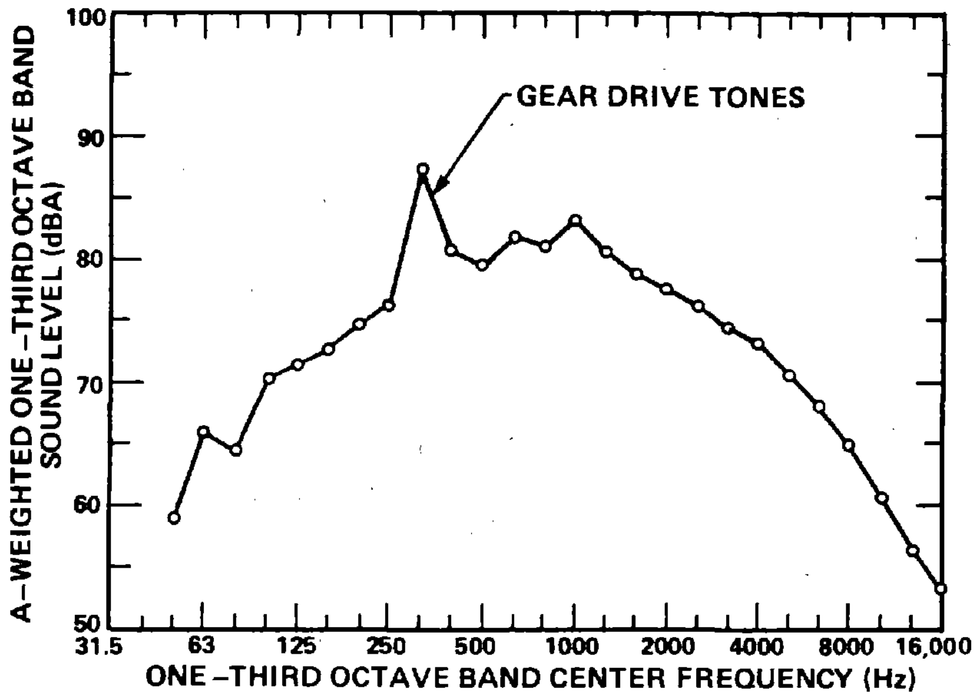


FIG. 2.29. ONE-THIRD OCTAVE BAND LEVEL OF NOISE MEASURED IN WALKWAY OF AUTOGENOUS GRINDING PLANT.

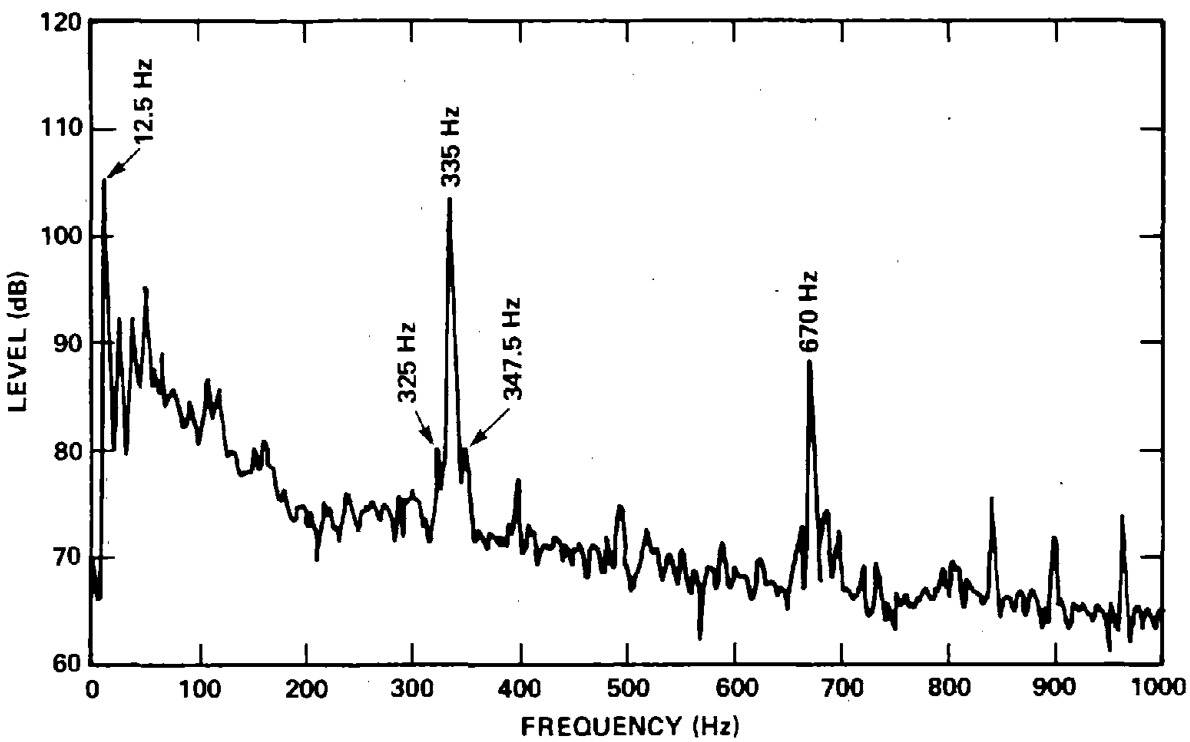


FIG. 2.30. NARROW BAND ANALYSIS OF NOISE MEASURED 2 FT FROM GEAR DRIVE OF AUTOGENOUS MILL.

TABLE 2.5. NAME PLATE SPECIFICATIONS FOR MILL REDUCTION GEAR BOX.

Herringbone, Double Involute Gear Drive		
Service, hp	=	4725 hp
Input	=	720 rpm
Output	=	140 rpm
Gear ratio	=	5.143:1
Pinion	=	28 teeth
Gear	=	144 teeth

the input shaft rotation rate (12 Hz). The tone at 335 Hz is the tooth-meshing frequency with side bands at 322.5 Hz and 347.5 Hz. Harmonics of the tooth-meshing frequency and the higher-order side bands are also present.

Similar measurements were made on a gear drive for a semi-autogenous mill. The narrow band analysis shown in Fig. 2.31a shows the tooth-meshing frequency at 237 Hz, but the spectrum includes other noise sources. In fact, the one-third octave band spectrum measured in the aisle between two mills (see Fig. 2.31b) shows the gear noise masked by the mill shell noise.

Motor Drive Noise

The other major source of auxiliary noise is from the large synchronous drive motors. These motors usually produce strong tones caused by rotor-stator interactions or the blade-passing frequency of the motor cooling fans. Narrow band data given in Fig. 2.32 show multiple tones between 1.5 and 2 kHz and harmonics at 3 to 4 kHz. In addition, there is much low frequency energy at 60 Hz and 120 Hz due to electromagnetic sources.

In most cases, the noise from these motors is masked by the mill noise. One exception to this was a semi-autogenous plant where the noise measured in the vicinity of the motors produced

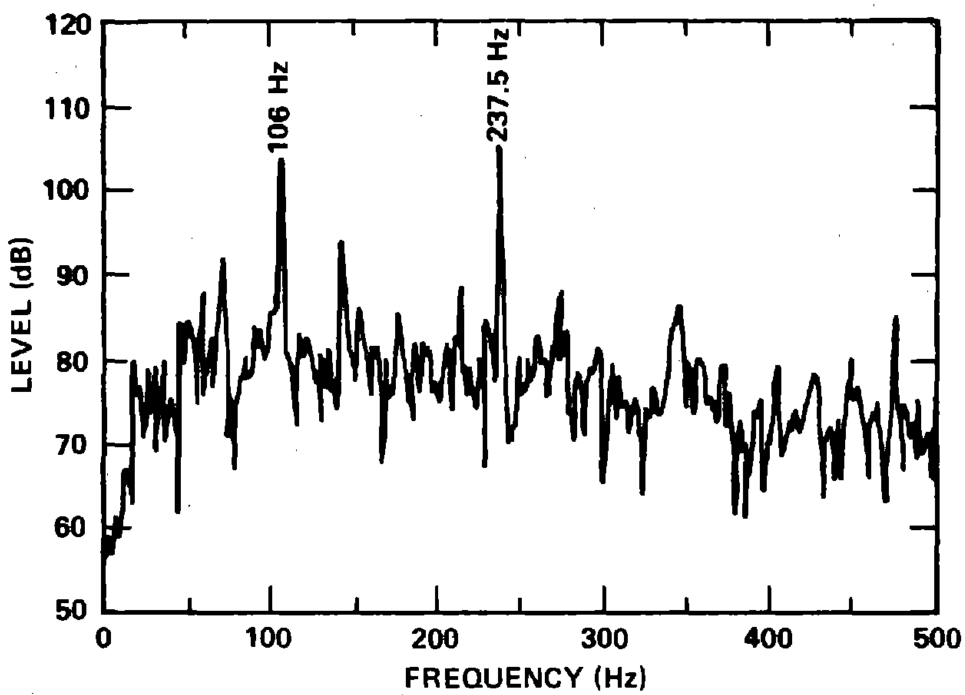


FIG. 2.31a. NARROW BAND ANALYSIS OF NOISE MEASURED 2 FT FROM GEAR DRIVE OF SEMI-AUTOGENOUS MILL.

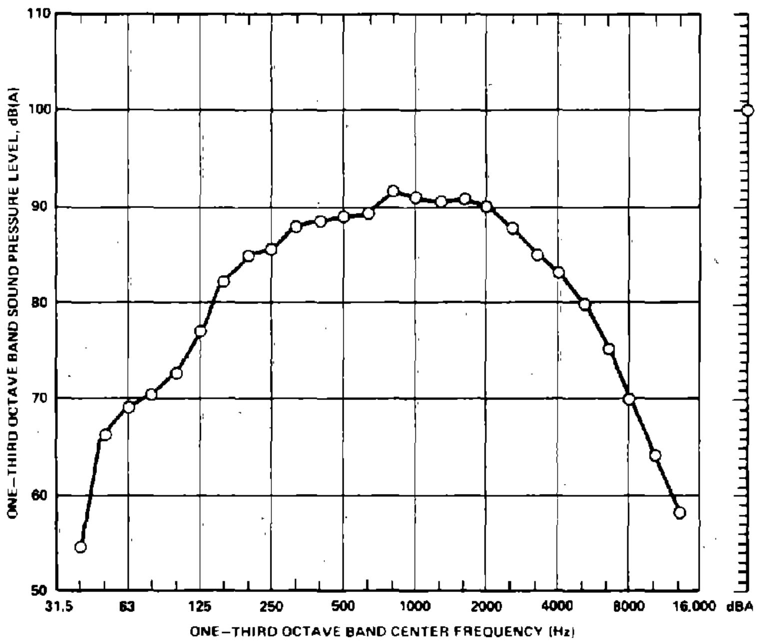


FIG. 2.31b. ONE-THIRD OCTAVE BAND NOISE LEVEL IN SEMI-AUTOGENOUS CONCENTRATION PLANT. NOTE THAT THE TONES FROM THE GEAR DRIVE SHOWN IN (b) ARE MASKED BY GENERAL MILL NOISE.

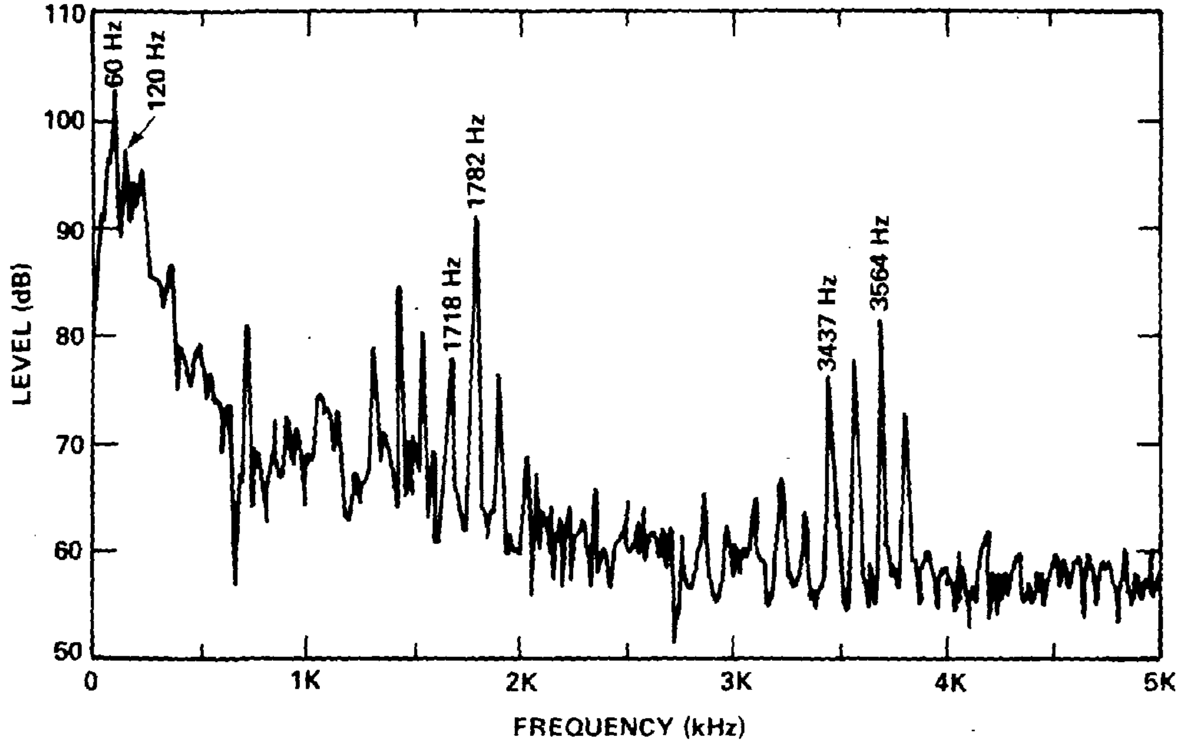


FIG. 2.32 NARROW BAND ANALYSIS OF NOISE MEASURED CLOSE TO MOTOR DRIVE OF SEMI-AUTOGENOUS MILL.

tones that clearly dominated the overall spectrum level, as shown in Fig. 2.33.

Noise Control of Auxiliary Drive Equipment

Mill drive equipment is not usually the dominant noise source, but it may become significant once the mill shell noise is treated.

An estimate of the different source contributions is shown in Fig. 2.34. The contribution of the drive equipment noise, measured in the near field, has been calculated for a position in the walkway aisle. The overall plant noise at the same position is compared with these levels. For this particular equipment, the drive equipment tones and harmonics become significant noise sources, once the shell noise has been reduced by 5 dBA.

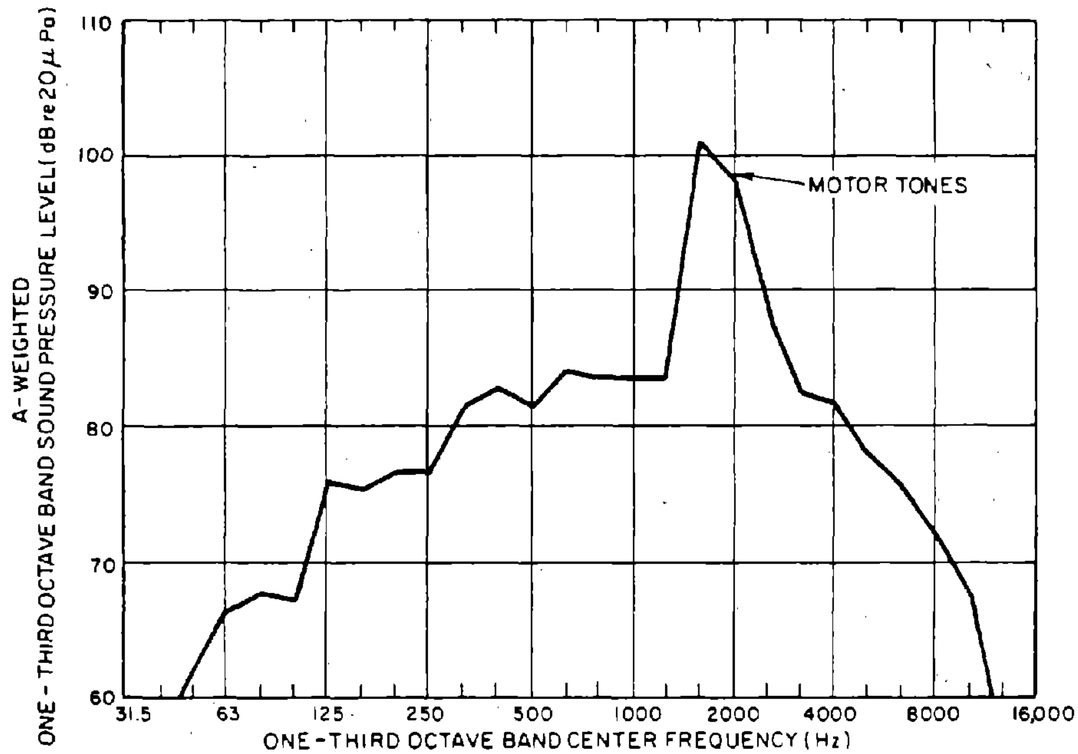


FIG. 2.33. ONE-THIRD OCTAVE BAND SPECTRUM OF NOISE 3 FT FROM AN AEROFALL MILL MOTOR.

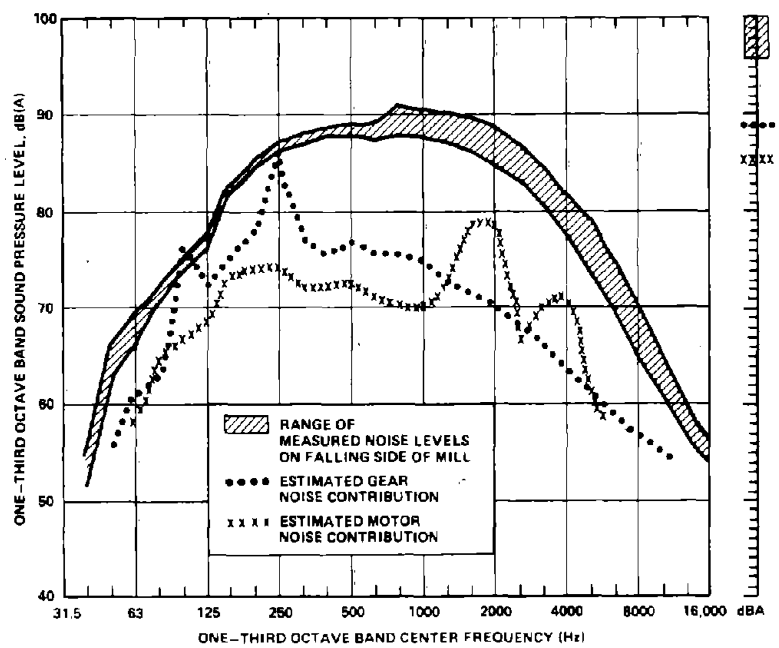


FIG. 2.34. ESTIMATED CONTRIBUTION OF MOTOR AND GEAR DRIVE NOISE TO THE OVERALL GRINDING MILL NOISE LEVEL.

For the autogenous plant shown in Fig. 2.29, the gear box noise already dominates in the 315 Hz one-third octave band. Control of this tone could be achieved by a number of methods. The small amount of energy in the first side bands, shown in Fig. 2.30, indicates that the gears might need a geometry redesign. Modification of the tooth face might reduce the tooth-meshing tone. A less expensive method of noise control would be an enclosure around the gear box, although adequate provision for cooling would have to be required.

It is worth noting that half of the mills at the same plant use a Philadelphia gear drive, and the noise produced by these does not contain the tone measured with the Falk gear drive (see Fig. 2.35).

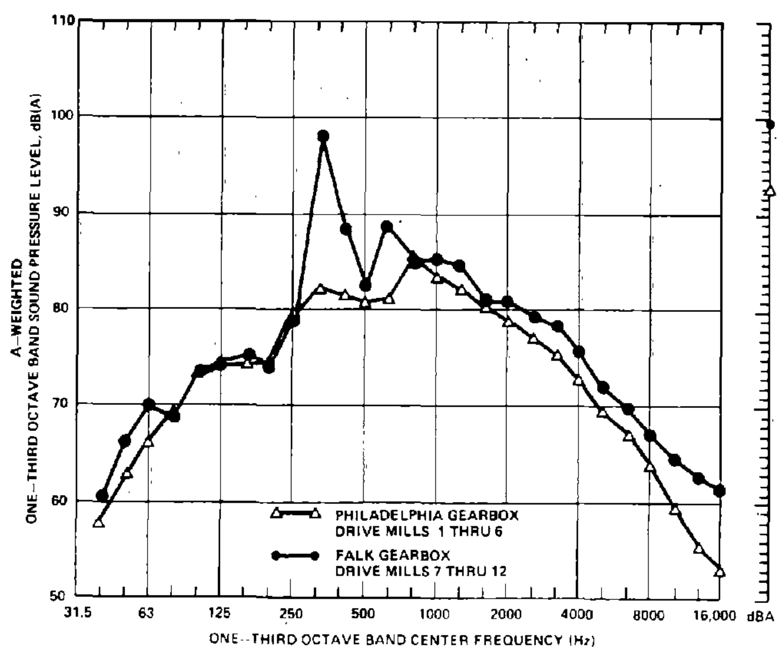


FIG. 2.35. ONE-THIRD OCTAVE BAND NOISE LEVELS MEASURED CLOSE TO TWO DIFFERENT GEAR DRIVES IN THE SAME AUTOGENOUS GRINDING PLANT.

The control of motor noise would require further investigation, although modification of the cooling fan, an inlet muffler, or an enclosure are possible solutions.

2.3 Noise Control Options

In Phase II of this program, a number of noise control options were considered for rod mills and autogenous/semi-autogenous mills. In light of extensive measurements made in the present program, and after several discussions with the major mill manufacturers and taconite plant operators, a list of possible noise control options was drawn up. Some of the original ideas from Phase II were reevaluated, additional methods were considered, and new information from Canada and Europe was obtained. The two most promising options were analyzed in detail and then installed. The noise reductions achieved, the treatment durability, and the mine acceptability are discussed in Secs. 2.4 and 2.5. Table 2.6 lists the noise control options considered at the start of this program.

TABLE 2.6. NOISE CONTROL OPTIONS FOR PRIMARY GRINDING MILLS.

Noise Control Treatments	Rod Mills	Autogenous and Semi-Autogenous Mills	Comments
Shell Damping	✓	✓	Needs to be proven, maintenance problems.
Full Enclosures/Lead Curtains	✓	-†	Very expensive, maintenance problems.
Partial Enclosure	✓	✓	Potential needs to be proven.
Trommel Enclosure	✓	-	Reduces trommel noise. Doesn't affect shell noise.
Process Control	✓	✓	Affects grinding performance and secondary circuits. Benefits may be hard to control.
Liner Backings	✓	✓	Potential for only small noise reduction.
Liner/Shell Vibration Isolation	✓	✓	Good noise reduction. Durability and economic problems.
Rubber Liners	-	*	Wear and economic factors need to be proven.

†Too expensive for large mills.

*Autogenous mills only.

2.3.1 Shell damping

Having identified the vibration of the mill shell as the dominant noise source, we considered increasing the damping of the mill shell. This could be achieved by either attaching a layer of viscoelastic material to the outside of the mill shell or by using a constrained layer construction. However, early in the program, it was determined that damping treatment was not a feasible noise control option. The reasons for this are discussed below.

On most large primary grinding mills the shell is approximately 2 to 4 in. thick, which would require almost an equal thickness of viscoelastic material to increase the damping significantly. Even if the material were available in these sizes, the costs would be prohibitive.

In addition, a series of loss factor measurements was made on a rod mill and a semi-autogenous mill to determine the existing shell damping. The measured loss factors on both the outside of the mill shell and on the shell liner (inside the mill) are shown in Fig. 2.36. These results indicate that the mill shell and liner assembly are already heavily damped, and that it would be difficult to increase this damping significantly.

2.3.2 Mill enclosures

The concept of an enclosure around the grinding mill has been tried at Intalco, WA, where two 13-ft-diameter Aerofall mills were enclosed in separated hardboard houses. No acoustic data were available for this installation. In Canada [7,9,11] a 16-ft × 11.5-ft-diameter rod mill was covered using a 30-ft × 35-ft × 20-ft-enclosure. Noise levels inside the enclosure were 112 to 115 dBA, while the levels outside were 92 to 98 dBA.

However, this application was significantly easier because the mill was situated in an existing bay that was enclosed on

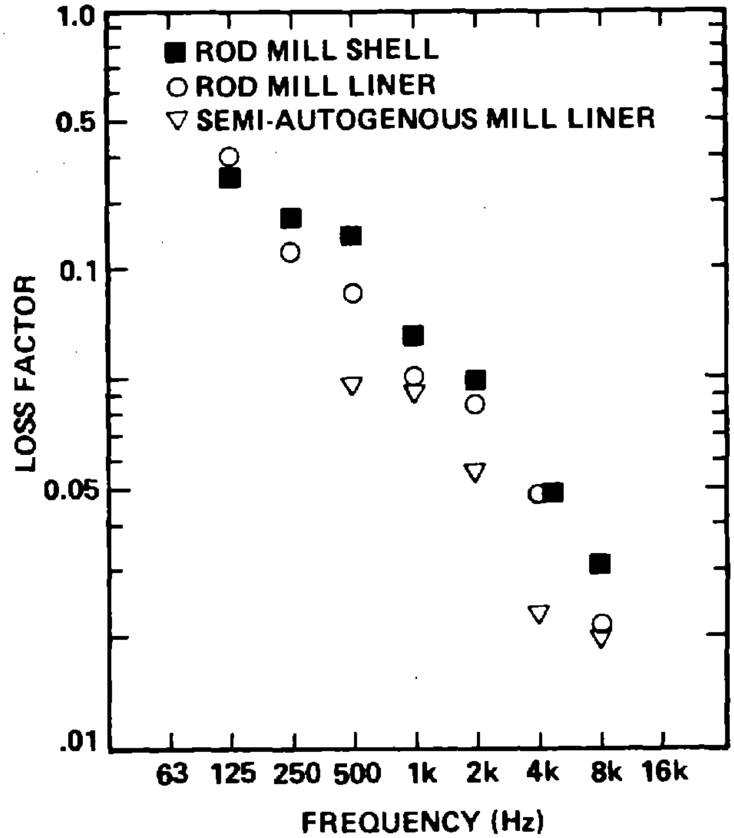


FIG. 2.36. TYPICAL MILL SHELL AND LINER LOSS FACTORS.

three sides. The concept of a full mill enclosure for taconite plants that contain many mills in one large open area is neither realistic nor feasible. The costs for even a small rod mill are very high, and are prohibitive for a large autogenous mill. Other problems are that of mill maintenance and operating requirements. A full enclosure would severely limit access to the liner bolts for changing or tightening the mill liners. Also, visual access to the mill shell is required to check for leakage or loose liners. Interference with the perpetual cleanup necessary in a taconite plant, including small-loader operation and hosing down, would be a significant problem.

The use of enclosures around the trommel on rod mills has been tried by several mine operators. Figure 2.37 shows an

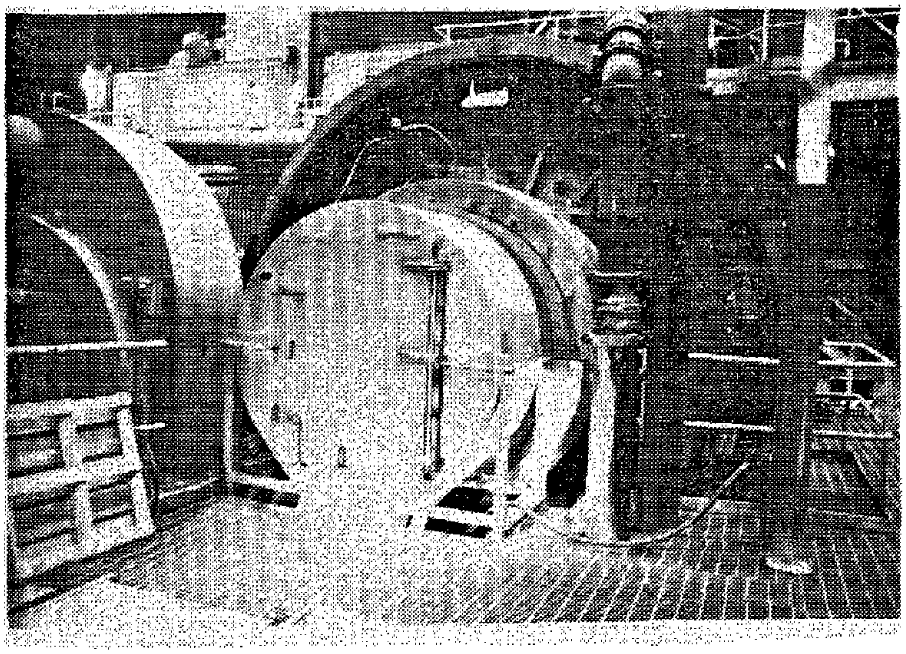


FIG. 2.37. ROD MILL TROMMEL ENCLOSURE.

example of a trommel enclosure fitted to a 16-ft × 11-ft-diameter rod mill. If the enclosure is tight fitting, it will significantly reduce the noise coming from the discharge end of the mill. However, the noise radiated directly from the shell will still be a problem. Also, trommel enclosures are usually only beneficial on rod mills, which have high discharge end noise due to their open trommel arrangement.

From the noise measurements and analysis discussed in Sec. 2.2, it was apparent that a partial barrier would potentially reduce the shell noise. A partial barrier would cover the lower half of the mill where the high shell noise levels were generated. The advantage of a partial barrier over the full enclosure is that it can be designed to be easily removable for maintenance, and it will not hinder visual inspection of the mill. Also, some plants already use steel guards in front of the mill and it would be relatively simple to incorporate the barrier into this guard. Obviously, the associated noise reduction with a partial barrier will be much less than for a full enclosure.

Calculations made in Sec. 2.2 estimated a 5 dBA reduction when the falling side of the shell and both end caps are covered. The performance of a partial barrier had to be verified on an actual grinding mill, and these results are presented in Sec. 2.4.

2.3.3 Process control

The results presented in Sec. 2.2 indicate that mill noise is dependent on the number of grinding balls added to the charge, the charge condition, and the liner life. Therefore, one noise control option might be to optimize these parameters to obtain the quietest running condition. (In fact, for several years, one mill manufacturer has marketed an acoustic feed control system [8], whereby the mill noise level was monitored and used to adjust the feed input to the mill.)

However, after discussion with the plant operators, it appears that it is not possible to take advantage of these effects without seriously affecting the grinding efficiency. For example, reducing the charging rate of balls added to the mill from 500 lb/hr to 250 lb/hr might provide a 2 to 3 dB reduction in semi-autogenous mill noise. However, it would also result in approximately a 12% increase (see Fig. 2.19) in power consumption, as well as upsetting throughput and balance with the secondary circuits.

In another example, it was determined from measurements made on rod mills that the noise level was dependent on the condition of the mill charge. There was approximately a 3 dBA variation in noise level at the falling side of the mill for different slurry densities. However, making changes in the slurry density also provided problems in mill throughput and secondary circuits.

Another problem associated with process noise control has been that mills in different plants operate at different conditions, and factors that affect one mill may not affect another. For example, noise measurements made in a plant that had autogenous mills indicated that noise level did not vary with changes in the amount of charge added to each mill. In order to generalize these factors, a carefully controlled model or full-size mill study, in which each parameter could be varied individually, would have to be made.

2.3.4 Liner backings/vibration isolation

Measurements were made on a number of different rod mills that used either rubber or epoxy-backed materials between the mill shell and the liners. Taking into account the variations between different mill operating conditions, the results seemed to indicate that rod mills with rubber backings were approximately 2 to 3 dB quieter than those with epoxy backings [5]. The purpose of the backing material is to prevent water and slurry leakage and, hence, shell corrosion. Consequently, the recent trend in the industry has been to use the rubber-backed material rather than the epoxy materials. None of the rod and semi-autogenous mills seen in this phase of the program were using the epoxy backings.

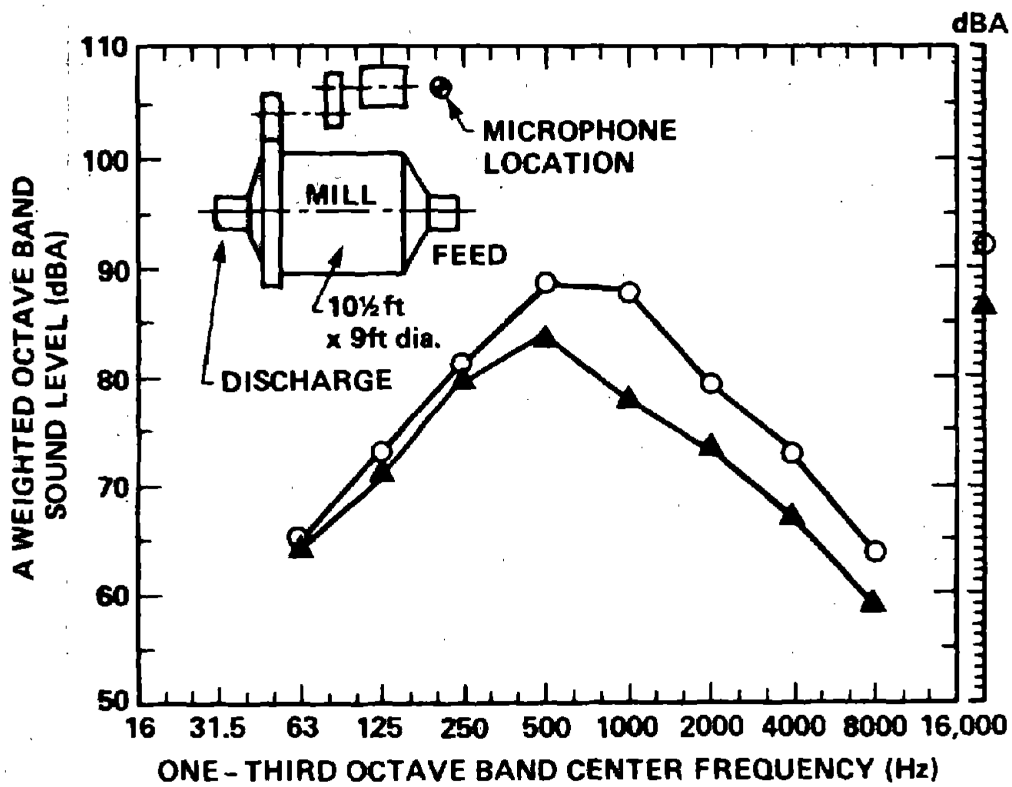
A more efficient method of using the rubber backing would be to incorporate vibration isolation between the mill shell and the liners. This would involve using rubber washers on the liner bolts to prevent the transmission of vibration from the liners to the shell. In its simplest form, this type of treatment would involve a rigid or waffled (to increase free surface area and reduce the stiffness) rubber sheet as a liner backing and a rubber washer under the nut on the outside of the shell.

Several different designs of shell vibration isolation were presented to the two major mill manufacturers and to plant operators. Agreement could not be reached on the designs because of the costs involved in testing the concept, the uncertainty of the durability, and the manufacturers' premise that a rigid liner fixture was essential. Consequently, this method has not been tested on an operating mill. However, one method of resolving the durability problems might be to test the vibration isolation on a scale model, such as that used by mill manufacturers to test liner designs. Unfortunately, such a scale model was not available for the present program.

2.3.5 Rubber liners

The use of rubber in the taconite industry has been increasing during the last 10 to 15 years. Initially, rubber was used on chute linings and on screen decks to increase wear resistance, as well as to reduce noise levels. More recently, rubber liners were used in secondary ball mills. The primary benefits of the rubber liners were to reduce wear, to improve grinding performance, and to reduce downtime during liner replacement. A side benefit was the reduction in noise level. As a noise reduction option, rubber liners potentially offer a large reduction, since they effectively isolate the falling charge from the mill shell.

A small, 10-1/2-ft × 9-ft-diameter pebble mill in Scandinavia (where most of the rubber development work has been done) was lined with rubber [9]; the reduction in noise level over the steel liners is shown in Fig. 2.38. A 6 dBA noise reduction was achieved at a position towards the feed end. Until recently, rubber liners were only used on secondary taconite grinding mills. For the large primary mills, the wear performance of the rubber was considered to be uneconomical. Within the last three years, owners of a mine in Canada and one in Northern Michigan have been evaluating the use of rubber in primary autogenous mills. Both these sites have been visited, and noise measurements and linear wear information are discussed in Sec. 2.5. However, at the present time, there are no primary semi-autogenous or rod mills using rubber liners in the U.S. taconite industry. The balls and rods added to the charge greatly increase the amount of impact and abrasion in the mill. Signi-



o--o steel liners
▲--▲ rubber liners

FIG. 2.38. NOISE REDUCTION ACHIEVED [9] ON SMALL PEBBLE MILL WITH RUBBER LINERS.

ficant improvements in the material wear characteristics have to be made before rubber liners can gain widespread acceptance in the industry.

2.4 Partial Mill Barrier

Analysis discussed in Sec. 2.2 indicated that a partial mill noise barrier would provide a 5 dBA reduction in noise level at the mill falling side. In order to test this noise control option, a prototype mock-up barrier was installed on a 27-ft-diameter semi-autogenous mill.

2.4.1 Design

The design of the prototype is shown in Fig. 2.39. A close-fitting barrier was installed on the falling side and end caps of the mill. The barrier fit inside the guard fence, approximately 10 in. from the mill shell. In production designs, the barrier could be incorporated into the guard fence, and easily removed when relining was necessary. The floor area around the edges of the mill had a 2-ft-wide section of open steel grating that was sealed to prevent mill noise reflecting up from the basement.

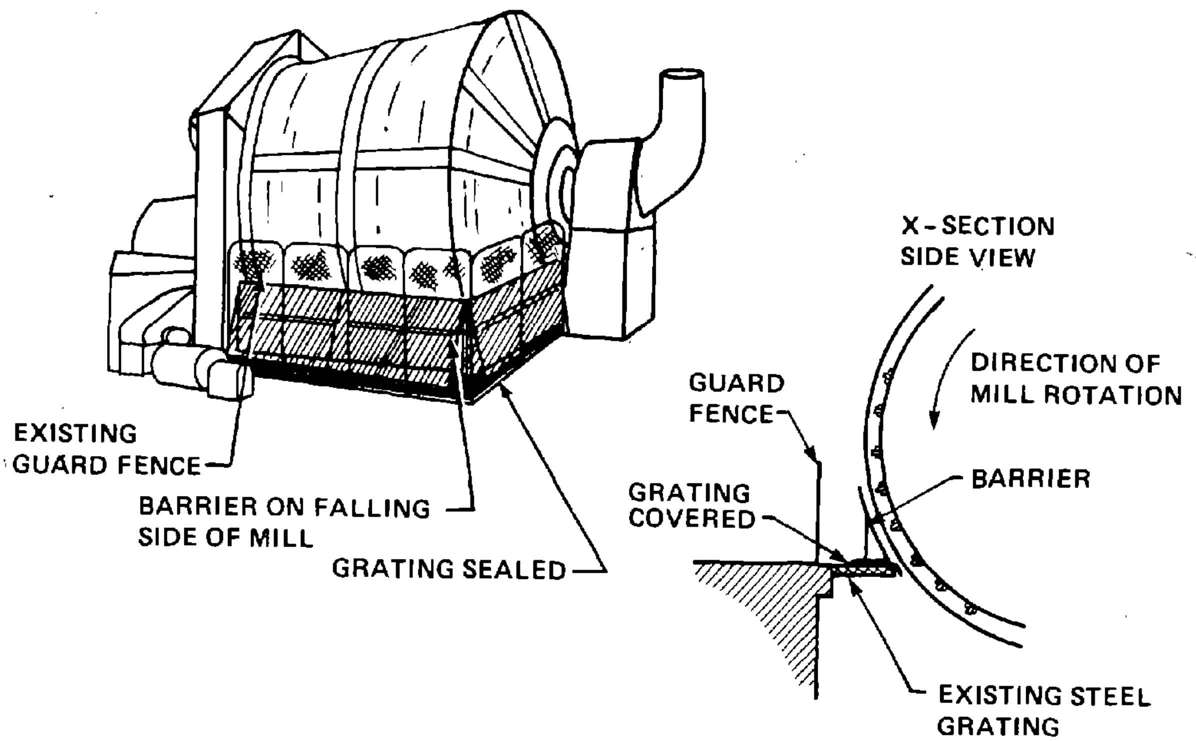


FIG. 2.39. PROTOTYPE DESIGN FOR PARTIAL MILL NOISE BARRIER.

The construction of the demonstration barrier is shown in Fig. 2.40 and consists of a wooden x-frame support with a 1/4-in. plywood covering. A layer of 3/4-lb/sq ft loaded-vinyl sheeting was used to provide additional mass to the plywood. Finally, the plywood surface was covered with an absorptive layer of 4-in.-thick fiberglass which was supported by metal studs. The barrier

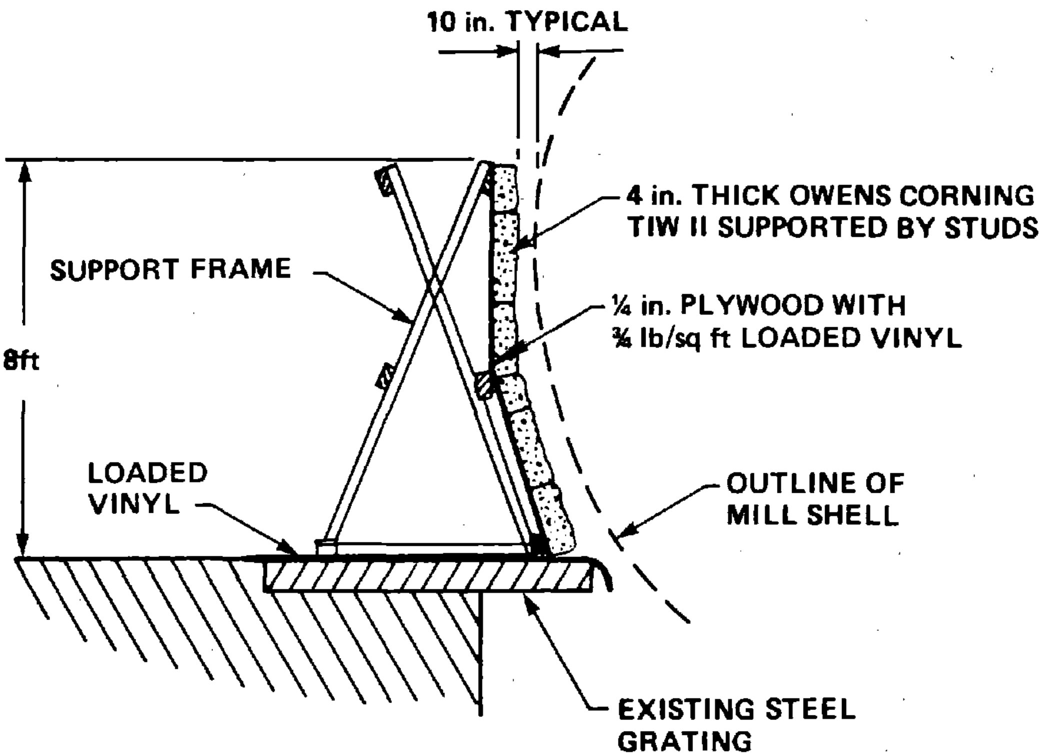


FIG. 2.40. CONSTRUCTION OF DEMONSTRATION MILL BARRIER.

was designed so that both a 4-ft-high and an 8-ft-high barrier could be tested. A gap of approximately 10 in. was left between the mill shell and the edge of the barrier to allow for clearance of the liner bolts, as well as a margin for safety. A single layer of loaded vinyl was used to cover the existing floor grating around the edge of the mill.

The acoustic design of the barrier provided at least a 10-dB transmission loss at frequencies 125 Hz to 10 kHz. The calculated transmission loss for the barrier is shown in Fig. 2.41.

2.4.2 Results

The completed barrier is shown around the mill shell and around the feed end in Fig. 2.42. Noise measurements were recorded at 18 locations around the test mill. During these measurements, it was not possible to run the test mill by itself

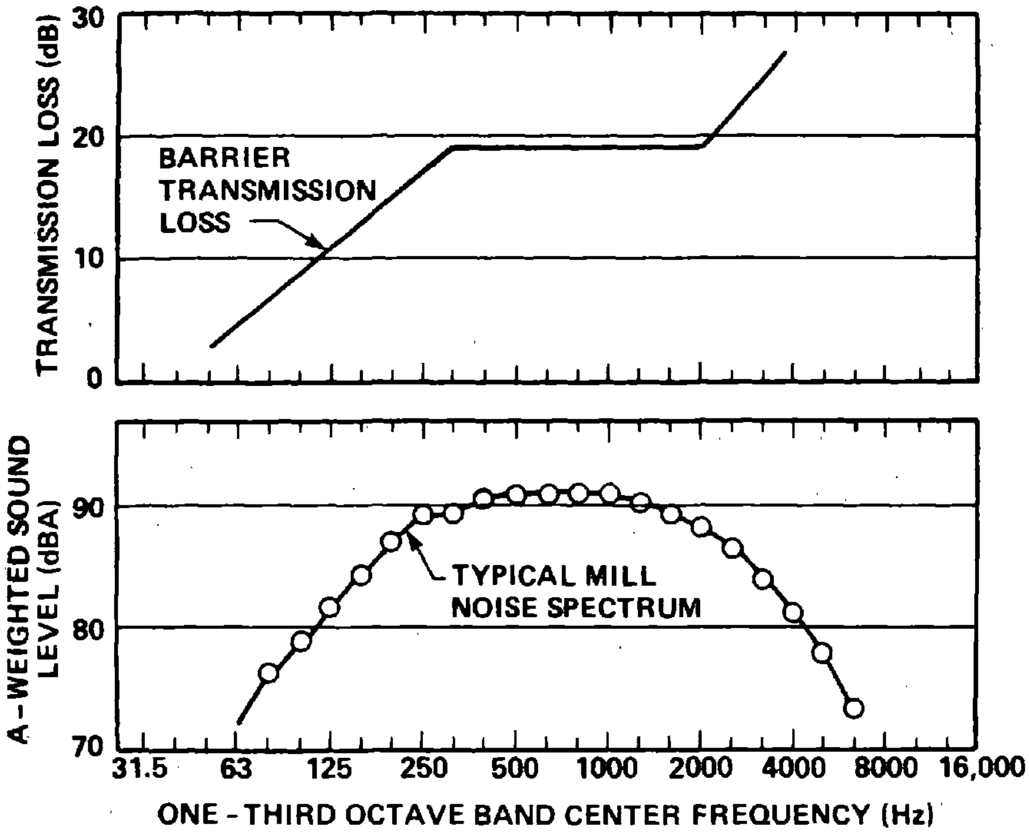
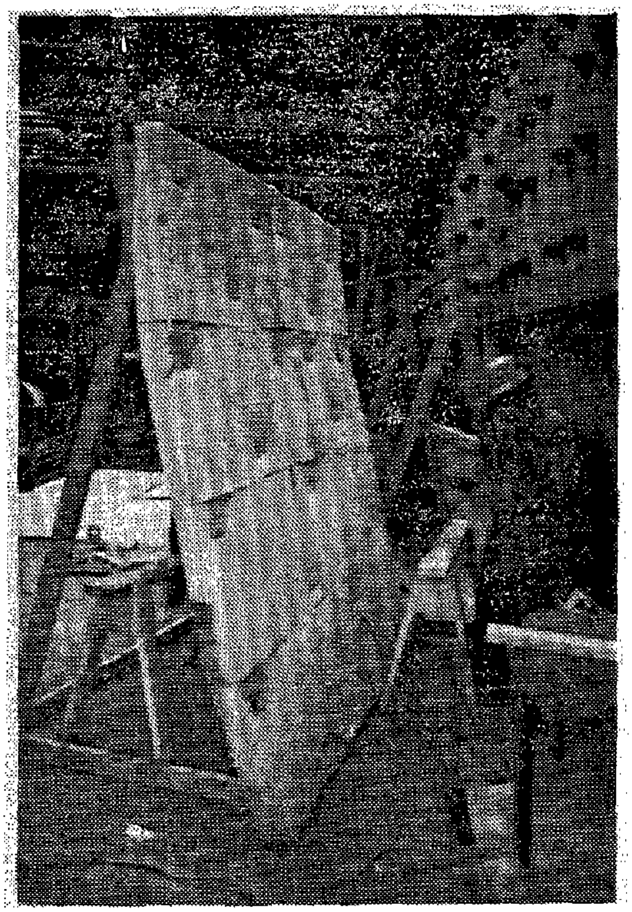


FIG. 2.41. CALCULATED TRANSMISSION LOSS OF NOISE CONTROL BARRIER COMPARED WITH TYPICAL MILL NOISE SPECTRUM.

(i.e., with all other mills and equipment shut down). However, we were able to shut down the mills adjacent to the one being tested, leaving a total of eight mills running. Noise measurements were obtained for the following test conditions:

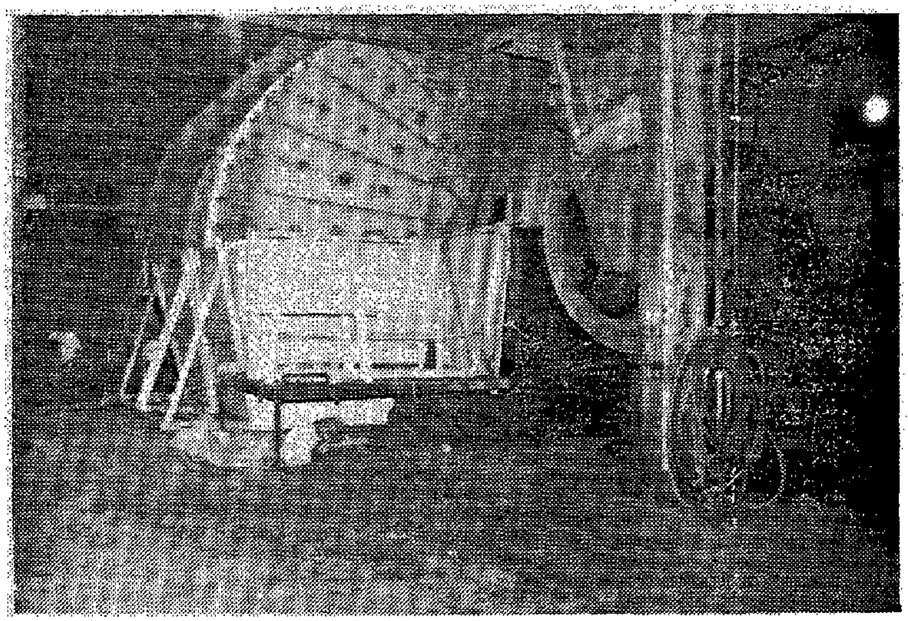
1. BASELINE - no treatments.
2. PARTIAL BARRIER - 8 ft high.
3. PARTIAL BARRIER - 4 ft high.
4. BACKGROUND NOISE - test mill and two adjacent mills shut down.

Typical one-third octave band noise levels for microphone positions on the mill floor, 15 ft from the barrier, are shown in



(a) Mill Shell

FIG. 2.42. PARTIAL MILL NOISE BARRIER INSTALLED ON SEMI-AUTOGENEOUS MILL.



(b) Feed End

FIG. 2.42. PARTIAL MILL NOISE BARRIER INSTALLED ON SEMI-AUTOGENOUS MILL.

Fig. 2.43. A 6 dBA noise reduction was achieved with the 8-ft barrier, while the 4-ft barrier provided 2 to 3 dBA reduction. Background noise from the other mills at these positions was 90 dBA, and contributed 1 to 2 dBA to the noise level measured with the 8-ft barrier. Close to the barrier, in the "shadow zone," there was a 10 dBA reduction in noise level, while at the feed end of the mill the reductions were only 2 to 6 dBA. This is because of the noise contributions from the feed chute and the other mills.

A summary of the noise reductions achieved at 17 positions (see Fig. 2.26) around the mill is shown in Fig. 2.44. The mill noise was significantly reduced in all areas adjacent to the 8-ft barrier. On the rising side of the mill (position Nos. 15 to 18), where there was no barrier, the noise levels were the same for all test conditions. The barrier reduced noise levels at all

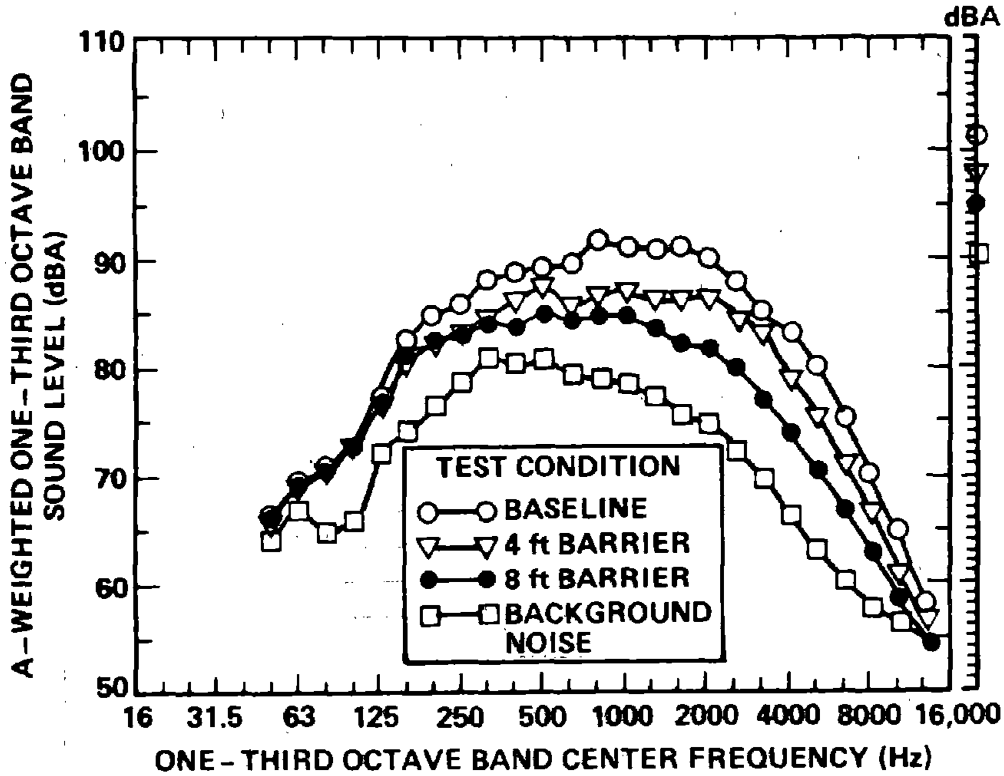


FIG. 2.43. ONE-THIRD OCTAVE BAND SPECTRA MEASURED APPROXIMATELY 15 FT FROM SEMI-AUTOGENOUS MILL, SHOWING PERFORMANCE OF PARTIAL MILL BARRIER.

positions on the falling side to 95 dBA or lower, except for two locations. The 97 dBA level at position No. 13 resulted from the gear drive noise, as well as the close proximity to the mill discharge end. The higher level at position No. 11 was due to noise underneath the mill coming up through a steel floor grating at this position. A further 2 dBA reduction, to 94.5 dBA, was achieved by covering this grating with a layer of loaded vinyl material. With this additional reduction, the barrier reduced the falling side noise levels to below the levels on the rising side of the mill.

A shortage of materials prevented the installation of a barrier on the rising side of the mill. However, measurements made close to the shell indicated that the dominant source was noise reflected up from below the mill. It was estimated that simply

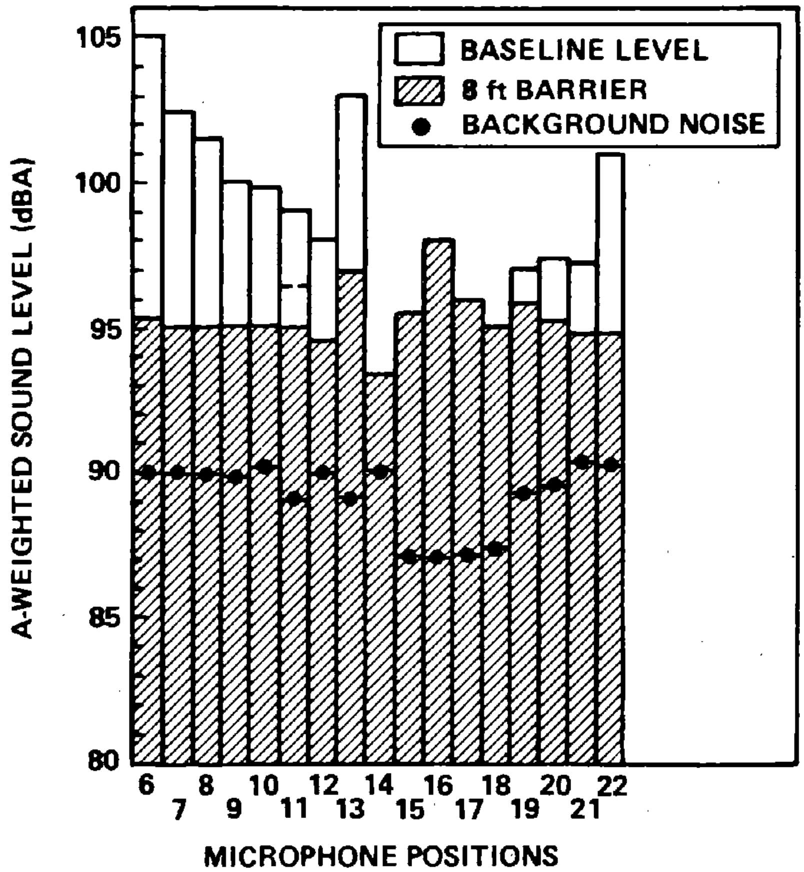


FIG. 2.44. SUMMARY OF NOISE REDUCTIONS ACHIEVED WITH PARTIAL NOISE BARRIER AROUND SEMI-AUTOGENOUS MILL. MICROPHONE POSITIONS REFERENCED IN FIG. 2.26.

covering the grating area on the rising side, without a barrier, would provide an additional 2 dBA reduction on this side.

2.4.3 Under mill noise

The partial barrier treatment provides noise reduction only for personnel working on the floor around the mills. However, noise levels underneath the mills are very high; these areas were also investigated. A floor plan of Level 1 (underneath the mills) with the associated noise levels, is shown in Fig. 2.45. The projection of one rotating mill is indicated by the dashed line; the mills are typically located 15 to 20 ft above the floor level. Noise levels within the concrete support structure

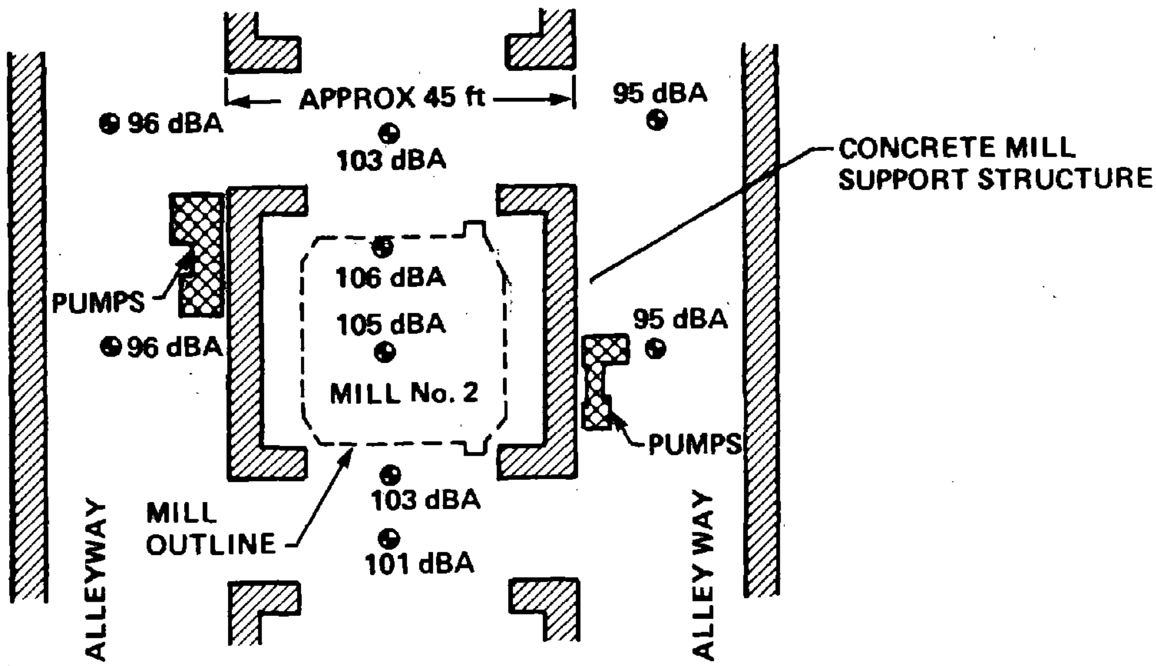


FIG. 2.45. FLOOR PLAN AND ASSOCIATED NOISE LEVELS IN BASEMENT OF SEMI-AUTOGENOUS PLANT.

directly below the mills were 101 to 106 dBA. However, most of the cleanup work that is performed by the laborer in this area is concentrated in the alleyways, where typical levels are 95 dBA. Little work is performed beneath the mills when they are running, partly due to safety reasons, and partly due to the thick taconite mud covering the floor. Consequently, although a noise curtain hung between the concrete mill support structures might be feasible, it was not considered necessary. Also, in addition to the mill noise, there are a number of large feed pumps and conveyors in the alleyways of Level 1 that contribute to the noise exposure of cleanup personnel. These pumps and conveyors would have to be treated before any reduction in noise exposure could be achieved in this area.

2.4.4 Maintenance, durability, and costs

The partial barrier tested in this installation was a prototype used to determine the noise reduction that might be achieved. The maintenance and durability of such a barrier remains to be proven. However, it is felt that a barrier could be designed that would require little maintenance, could be easily removed for relining of the mill, and would be durable. The best design would probably incorporate the acoustic barrier with the safety fence that is currently installed around each mill. Sections of the barrier could be easily removed either by hand, or with the small forklift truck that is presently used to remove the fence. Mounting slots on the mill floor would have to be laid down to locate the barrier. The open grating around the mill floor would need to be sealed off, and several small openings would be required for water drainage after cleanup operations. Absorption material on the inside of the barrier would require protection from dust, water, dirt, and mechanical damage. A fiberglass material wrapped with a thin protective coating and covered with perforated metal sheeting would accomplish this. The material costs for an industrial-grade barrier, unfinished and unpainted, for an 18-ft-long mill are estimated to be $6500. This cost allows for $16 per sq ft of material for the barrier itself, and $5 per sq ft for the floor covering. No installation costs have been estimated, since it is assumed that plant labor would be used.

Although the partial barrier appears to have worked well for this particular mill, there are certain limitations to its design. The first is that the barrier provides noise reduction on the falling side of the mill, and to a lesser extent at the ends of the mill. Therefore, the greatest benefit in reduction of noise exposure will be during relining and cleanup operations. It is not clear from the measurements made in this program how much reduction in overall plant noise will be achieved if all the grinding mills are fitted with a barrier. In addition, the largest noise reduction will be achieved with mills that have large impact noise on the mill shell, i.e., semi-autogenous mills and rod mills. The barrier may not be as effective on autogenous mills, where overall noise levels are lower.

2.5 Rubber Liners

The success of using rubber liners in Scandinavia has led to a number of taconite plants in the U.S. and Canada to refit mills with rubber liners. Secondary ball and pebble mills have been installed with rubber, and were found to be economical in terms of liner wear and overall cost.

More recently, rubber linings have been used on large primary autogenous mills. At Sherman Mine in Canada, three 10-ft × 10-ft-diameter autogenous mills have been lined with rubber. Typical liner profiles for the original steel and the new rubber liners and lifters are shown in Fig. 2.46 [10]. Only one of the 12 primary autogenous mills at the Tilden Mine, MI, has been rubber lined. Both of these plants had relatively similar mill

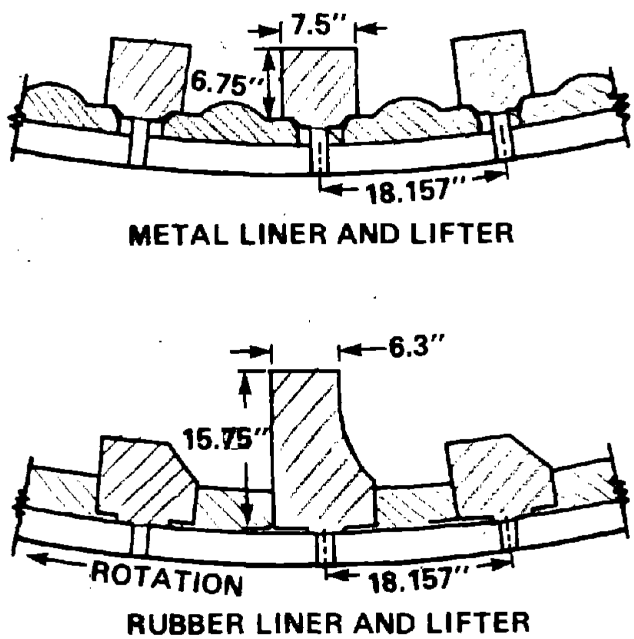


FIG. 2.46. AUTOGENOUS MILL CONFIGURATIONS FOR METAL AND RUBBER LINERS.

TABLE 2.7. PRIMARY MILL SPECIFICATIONS [10].

	Sherman	Tilden I
Manufacturer	Allis-Chalmers	Allis-Chalmers
Diameter, ft	27	27
Length, ft	10	14-1/2
Volume, ft^3	5700	8700
Mill Heads	Straight	Conical
Mill Speed		
rpm	11.2	11.1
% Critical Speed	76	75
Peripheral Speed, ft/sec	15.8	15.7
Volume Loading, %	35	35
Power, hp	3130	5354
Feed Rate, LT/hr	181	222

specifications, see Table 2.7, except for the longer length and conical mill heads at Tilden.

The primary reason these plants installed rubber liners was to improve the liner wear, reduce mill downtime, and decrease maintenance requirements. Any noise reduction was considered an additional bonus. Consequently, detailed noise measurements before and after rubber installation were not collected. However, both plants have been visited and noise measurements, as well as liner performance data, were obtained.

2.5.1 Noise measurements

The small rubber-lined Scandinavian pebble mill discussed in Sec. 2.3 showed a 6 dB reduction in noise level over steel liners. Noise measurements made at the Sherman and Tilden mines have not been as conclusive, because of the high background noise levels from adjacent mills and auxiliary equipment.

After one of the three mills at Sherman Mine had been rubber-lined, measurements [11] indicated a 1 to 2 dBA reduction in noise level around the mill, see Fig. 2.47. When all three mills were rubber lined, a small additional noise reduction was noted.

Microphone Position	Noise Level (dBA) Metal Liners	Rubber Liners (one mill only)
1	92	89
2	90	88
3	89	89
4	90	89
5	90	90
6	91	90
7	94	91

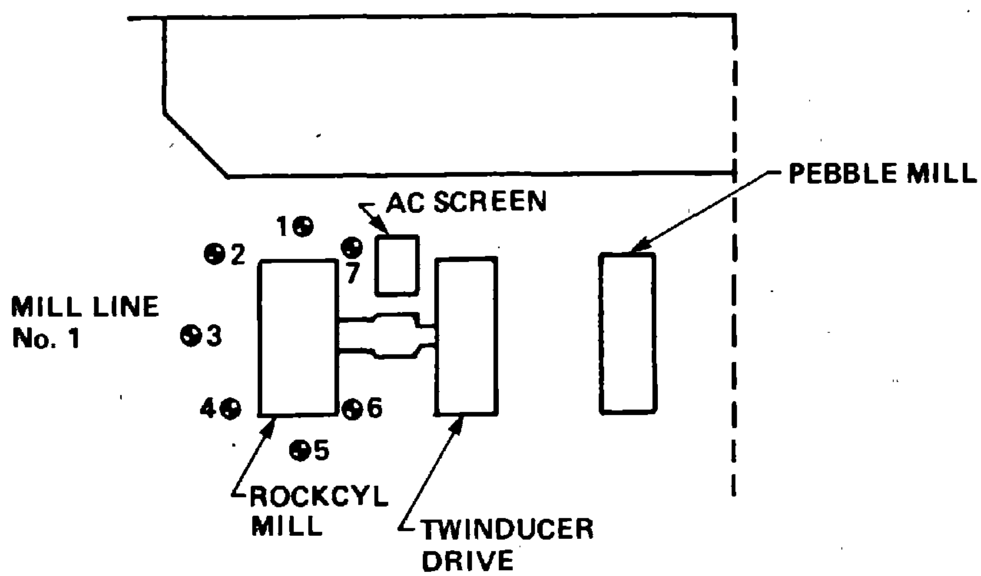


FIG. 2.47. NOISE MEASUREMENTS BEFORE AND AFTER RELINING OF ONE AUTOGENOUS MILL WITH RUBBER LINERS [11].

The main reason for these apparently small noise reductions is that the mill shell noise at the Sherman plant appears to be relatively lower than other taconite plants. Consequently, the noise from auxiliary equipment around the mills contributes significantly to the overall plant noise. The open decked A-C screens at the mill discharge are particularly noisy and are one of the dominant sources in the area. The plant is currently working to reduce the noise from these screens, and once this is done, the full benefit of the rubber liners should be realized.

At the Tilden Mine, similar noise measurements were obtained from the one rubber-lined mill. The layout of the mill, with eight different microphone positions, is shown in Fig. 2.48. The ore at Tilden is hematite, and this grinding circuit requires careful control of the material feed. Thus, it was not possible to turn off any mills adjacent to the one under test. Consequently, the background plant noise had to be considered in all the measurements. In order to improve the signal-to-noise ratio, microphone positions were chosen close to the mill shell.

Because the mill at Tilden had already been rubber-lined, it was not possible to compare rubber vs steel liners on the same mill. Instead, the noise levels around the rubber-lined mill were compared with similar measurements taken near one of the steel-lined mills. The comparison mill was selected after a noise survey of the 11 other mills. The one-third octave band

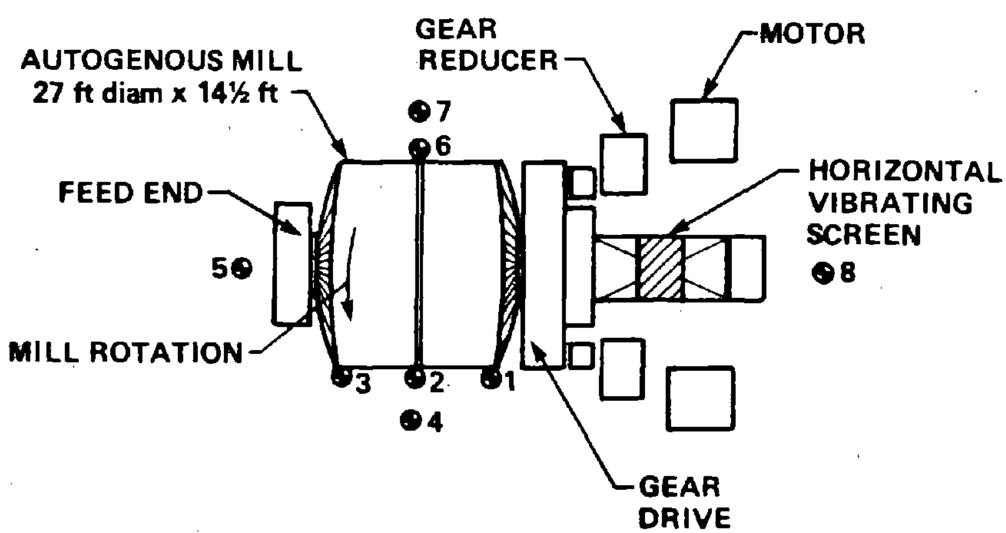


FIG. 2.48. MEASUREMENT POSITIONS AROUND AUTOGENOUS MILL.

noise levels for the rubber- and steel-lined mills are shown in Fig. 2.49. The rubber-lined mill has a 4 dBA lower noise level than the steel-lined mill. The noise reduction is only at frequencies 250 Hz to 2 kHz where impact noise from the charge dominates. At frequencies below 250 Hz, the motor and gear drive noise is the same for both mills. The plant noise level is shown in Fig. 2.49 for the same measurement position, but with the test mill shut down. The plant noise spectrum is clearly comparable to the level measured with the rubber-lined mill. Consequently, it is not possible to determine the full noise reduction achieved by the rubber liners in just one mill. In spite of this, one can still see a 2 to 3 dBA noise reduction at positions away from the mill. This is shown in Fig. 2.50, in which measurements at each position around the mill are presented. In summary, the rubber

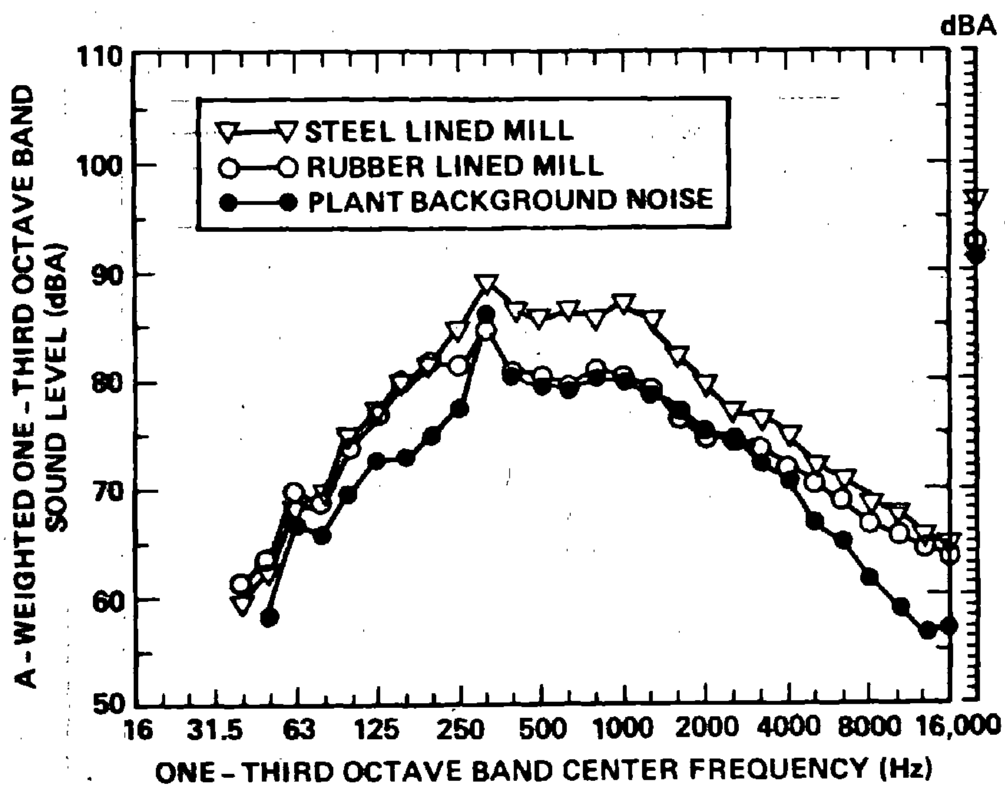


FIG. 2.49. ONE-THIRD OCTAVE BAND NOISE LEVELS FOR AUTOGENOUS MILLS WITH RUBBER AND STEEL LINERS (MICROPHONE POSITION #3).

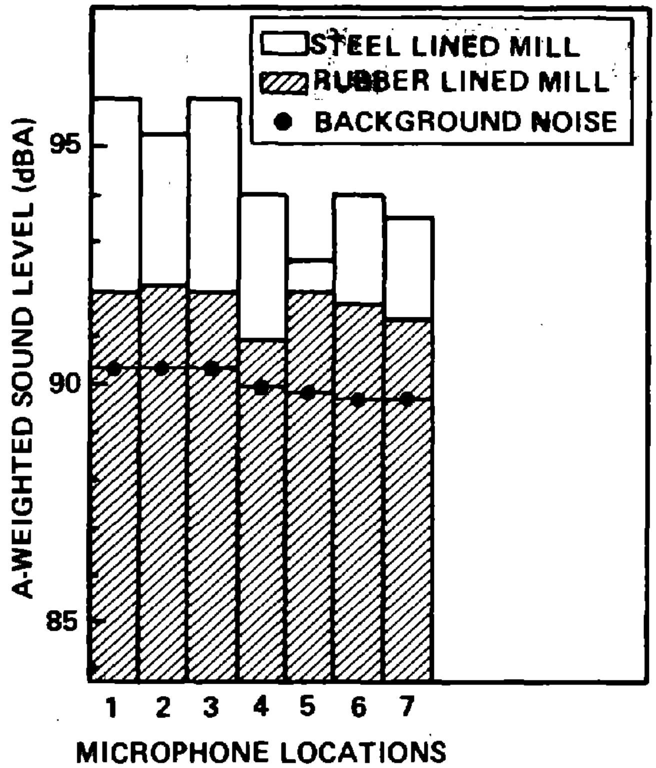


FIG. 2.50. NOISE REDUCTION FOR AUTOGENOUS MILL WITH RUBBER LINERS.

liners for this installation provided at least a 3 to 4 dB reduction in mill noise, and the actual reduction is probably higher than this.

2.5.2 Rubber liner performance

The performance of the rubber liners at both Tilden and Sherman Mines has been carefully monitored by personnel at these plants. Initial results have been published [10] and are summarized here.

The first set of rubber liners at Sherman Mine was installed in 1978, and since that time they have had experience with approximately 3 to 4 sets of liners. Since the installation at Tilden was in 1980, there has been less experience. The rubber liners installed were essentially the same at both plants except for the mill end and length variations already mentioned.

Despite these similarities, the performance results appear to be somewhat different. At Sherman Mine, the maintenance downtime was reduced and wear life was comparable to that of the steel liners, see Table 2.8. Also, the lower individual weight of the rubber liners improved material handling and reduced the potential for accidents. Personnel at the plant are pleased with these results and will be using rubber liners for future changes. At Tilden Mine, a problem developed early with the installation: the volume loading was 4 to 5% higher than with the metal liners, at the same power setting (see Table 2.9). In addition, the wear

TABLE 2.8. SHERMAN MINE - PRIMARY MILLS [10].

	3 Mills Metal Liners	3 Mills Rubber Liners
Downtime, Hr	2030	1290
Lifter Bar Wear, Months		
High Wear Areas	3.6 - 7.0	5 - 6
Low Wear Areas	5.6 - 9.0	36
Mill Lining Weight, Tons/Mill	158	59

TABLE 2.9. TILDEN MINE - PRIMARY MILLS [10].

	Metal Liners	Rubber Liners
Feed Rate, Lt/Hr	230	226
Mill HP Level	5350	5220
HP/LT Crude Ore	23.1	23.2
Volume Loading %	33 - 37	38 - 41

rate of the lifter bars was much higher than at Sherman (see Table 2.10). The rubber lifter bars had an average wear life of 3-1/2 months vs an 8-1/2 month life for the standard steel lifters. The reasons for the poor performance at Tilden are not yet apparent; additional tests are continuing.

TABLE 2.10. PRIMARY MILL LINER WEAR - AVERAGE LIFE IN MONTHS [10].

	Metal Liners	Rubber Liners
Tilden Mine		
Shell Plates	42	30
All Lifters	8-1/2	3-1/2
Sherman Mine		
Shell Plates	24	22
All Lifters	6	6

The performance of rubber liners needs additional testing before conclusive results can be drawn. In the meantime, work is being done to improve the wear and service life of the rubber. Initially, it was felt that recapping of used lifter bars might provide an additional cost benefit of rubber over metal liners. However, experience to date has not been successful, with premature separation has occurred and the recapped areas wore rapidly.

3. FINE SCREENS

3.1 General Information

The fine screens on which noise control work was performed are Dorr-Oliver 45° Rapifine DSM Model B screens with detachable, external pivot-type rappers. This type of fine screen is shown in Fig. 3.1. Fine screens are typically arranged in rows, as shown in Fig. 3.2. A two-stage operation, where a row of primary screens is located above a row of secondary screens, is commonly used. Figure 3.3 shows the general configuration of a two-stage installation. In larger facilities, rows of screens may be arranged in lines with each line servicing particular mills and magnetic separators.

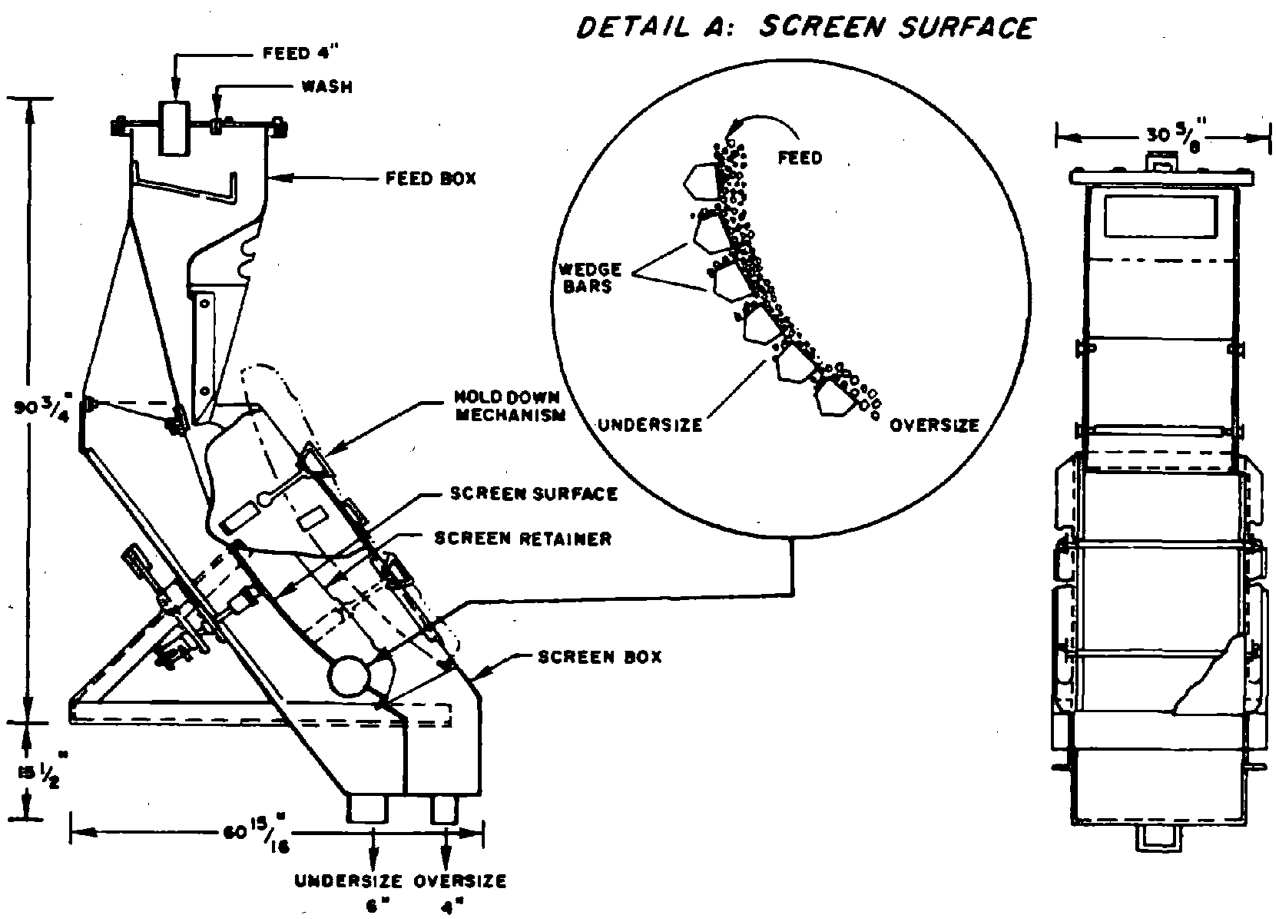


FIG. 3.1. RAPIFINE SCREENS - FRONT VIEW AND ELEVATION.

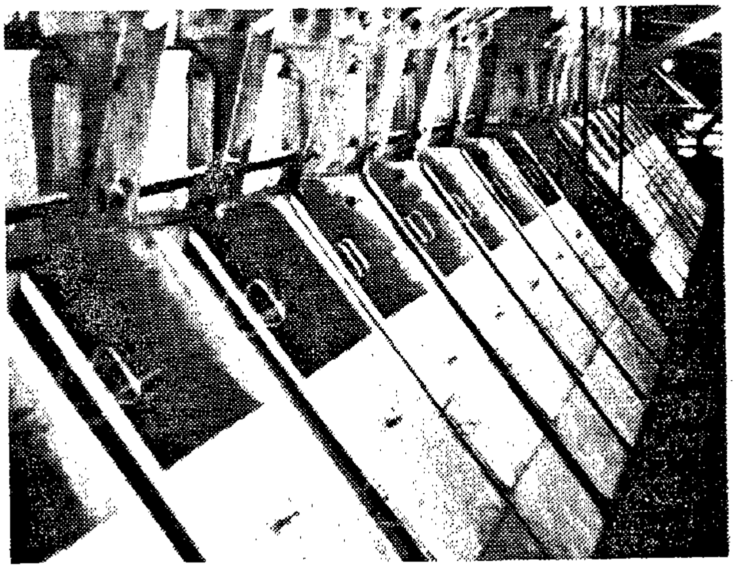


FIG. 3.2. ROW OF DORR-OLIVER RAPIFINE DSM SCREENS INSTALLED IN DEMONSTRATION PLANT.

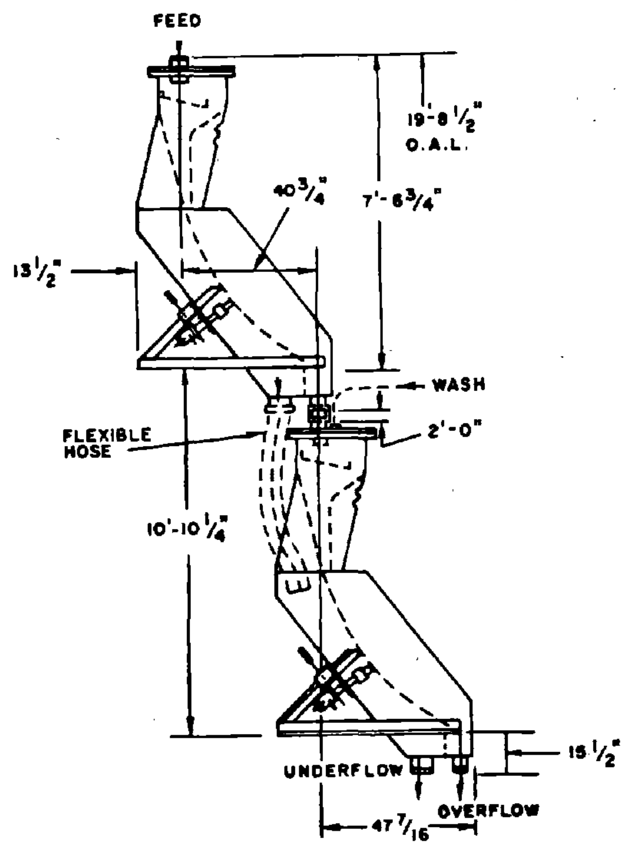


FIG. 3.3. TWO-STAGE RAPIFINE SCREEN INSTALLATION.

Fine screens are used to make the final product size separation in the concentrator, typically making the separation at 325 mesh. A slurry containing approximately 50% solids is piped to the feedbox from which it flows at low velocity over the screen surface. The screen surface is made of stainless steel wedge bars set 0.004 in. apart. The slurry follows the concave screen surface at right angles to the wedge bar openings. The undersize fraction and most of the water fall through the slots, and the dewatered oversize fraction slides down the top of the screen surface and is returned to mills for additional grinding. The screen surface has a tendency to blind, because the slurry leaves the feedbox with low velocity. This is prevented by the action of the externally mounted rapper.

The rapper supplied on the 45° Rapifine Screen is a Martin Engineering Company piston vibrator Model PV-1-1/4. This vibrator periodically raps an impact bar on the underside of the screen surface, preventing blinding. The piston vibrator is attached to one end of a counterweight arm and the counterweight arm, in turn, is attached to the rear panel of the screenbox through a pivot arrangement. The PV-1-1/4 consists of a steel piston which reciprocates inside a sleeve cylinder hitting a tool steel strike plate at one end and the rapper body end cap at the other. The reciprocating action is achieved by internal valving within the rapper body which is designed so that the piston hits the strike plate with maximum force, but strikes the end cap with only partial force. The strike plate in turn transmits the impact through the plunger rod to the impact bar on the underside of the screen surface. The counterweight arm automatically adjusts the rapper position so that there is always contact between the rapper and the plunger rod, and between the rod and the screen surface.

Maintenance of the fine screens includes reversal of the screen surface at periodic intervals (approximately every three weeks) so that the leading edge of the wedge bar does not wear excessively, out of proportion to the trailing edge. At the time of screen reversal, the slots are cleaned out by a mechanic with a wire brush. Screen life is about 6 to 12 months. Inspection of the rapper is recommended at the same time as screen reversal. Malfunction is usually caused by dirt, corrosion, and/or insufficient lubrication, according to Dorr-Oliver, and can be corrected by disassembling, cleaning, and oiling. The life of a rapper is usually many years.

Dorr-Oliver recommends a maximum operating air pressure between 20 and 30 psi and a vibration (on) time between 1/2 and 1 sec. The duty cycle is adjusted by varying the off time interval. The impact force is adjusted by changing the air pressure. Martin can supply a "silent-type vibrator" which has a 90 durometer urethane strike plate instead of the standard steel one. The company recommends the next larger size of vibrator if a "silent type" is employed. Martin also supplies sintered-bronze, adjustable flow mufflers for the exhaust port. Table 3.1 lists the sound level of the PV-1-1/4 as a function of operating air pressure.

TABLE 3.1. SOUND LEVEL OF PV-1-1/4 AS A FUNCTION OF OPERATING AIR PRESSURE.

Air Pressure (psi)	Sound Level at 6 Ft*	
	Without Muffler	With Muffler
20	95	94
40	100	100
60	104	104
80	105	103

*Tested on a small block.
Source: Martin Engineering Company.

A typical rapper cycle and the sound levels from an adjacent rapper line are shown in Fig. 3.4. Note that the sound levels are in the range of 97 to 100 dBA when the closest rapper line is on, and then drop to the order of 93 dBA when only adjacent lines are on. For a brief period during every cycle all rappers are off and the secondary mills, tail pumps, and gears set the background sound level at 91 dBA. Figure 3.5 shows a typical spectrum measured in front of a row of primary fine screens when the rapper is on. The A-weighted sound level is controlled by the high-frequency SPL's in the 1000 to 8000 Hz range. Reducing the occupational noise exposure of fine screens maintenance mechanics involves attenuating these high frequency levels.

3.2 Fine Screens Noise Parameters

3.2.1 Maintenance locations

It is the responsibility of the fine screens maintenance mechanic to clean and reverse the screen surface and lubricate

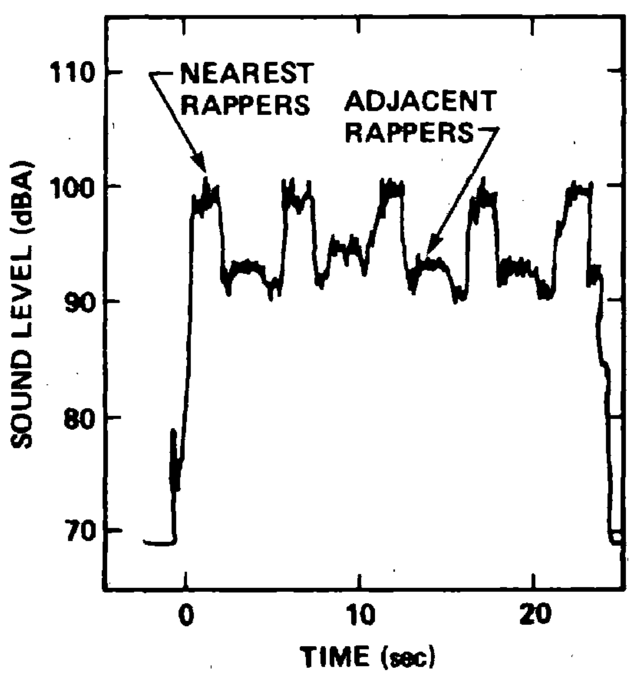


FIG. 3.4. TYPICAL RAPIFINE SCREEN NOISE. MICROPHONE LOCATED AT FRONT OF SCREEN.

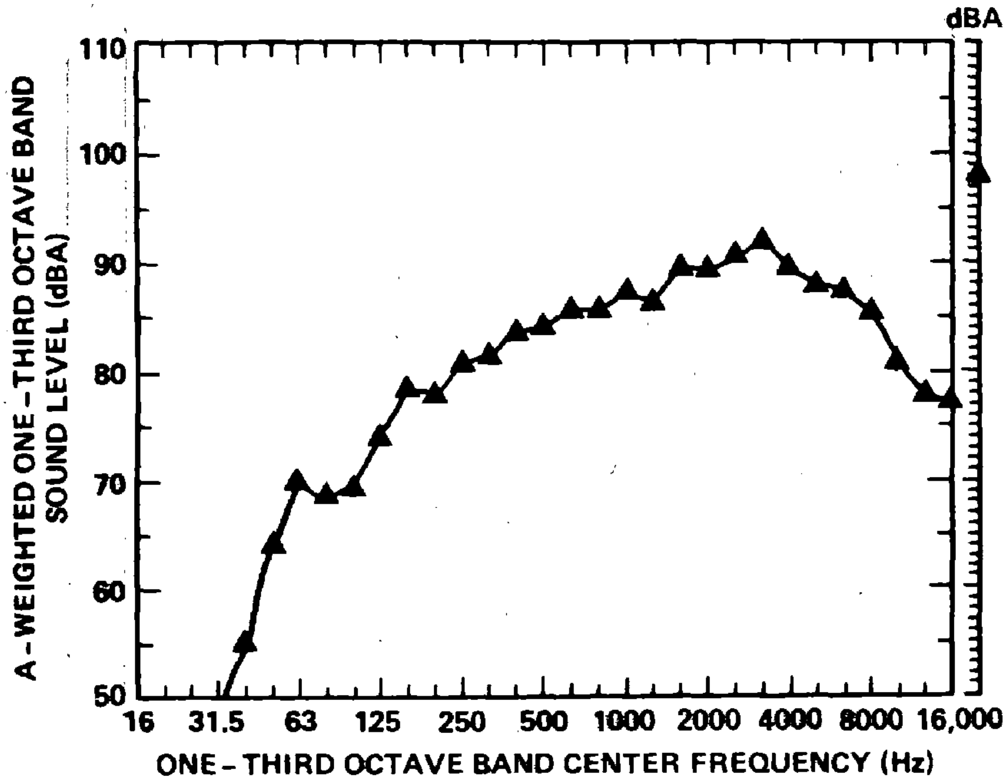


FIG. 3.5. TYPICAL SPECTRUM IN FRONT OF PRIMARY FINE SCREEN, WITH RAPPER ON.

the plunger rod on each fine screen every three weeks. Generally, the mechanic will start on a new line of sixteen primary and sixteen secondary screens at the beginning of a week and systematically work his way through the line. If time remains at the end of the week he will check out any troublespots on the rest of the lines. In performing maintenance on a fine screen, the mechanic will be in various locations around the screens. While removing and later reinstalling the screen surface, the mechanic will lean into the front of the fine screen and, although the rapper on the screen he is working on will be turned off, his ear will be closer to the noise sources on the adjacent screens than at a normal standing position in the aisle. He is also closer to the screens when greasing the plunger rod tube. The mechanic is behind the screenbox when lubricating or adjusting the counterweight arm mechanism, and he sits or kneels down on the grating when removing or installing the rapper-counterweight arm system. This position exposes him to particularly high noise levels, because he is only 2-1/2 ft from operating rappers on either side. After overhauling the screen, the mechanic will look down into the rear of the screenbox to check for good undersize flow. This requires shining a light into the screenbox and looking directly at the back of the screen surface. This is also a high noise level position. Therefore, during a screen overhaul, the fine screens mechanic is positioned at various times both in front of and behind the screens and his ears will be at normal ear height, at a lower, bent-over position, or at a kneeling/sitting height. Obviously, there is no one single location that can be defined as the operator's position.

The mechanic's exposure to different noise sources on the fine screens is further complicated when he stands between the front and back row of a two-row line. At this location he is exposed to the noise sources at the rear of the front screen and the screen surface noise on the front of the back screen. In addition, the rappers do not operate continuously - each row will be on for two seconds, off for three to four seconds, with different rows operating in programmed sequence. This all adds up to a complex spatial and temporal situation.

The Phase I study of this program, performed under USBM Contract J0366055, identified a noise reduction goal of 2 to 5 dBA for concentrators. Since the fine screens mechanic spends all of the work shift in only one area of the concentrator, more precise determination of the noise reduction required to bring him into compliance was considered necessary. To this end, the mechanic wore a personal noise dosimeter. His 8-hr time-weighted average noise level was found to be 97 dBA. This translated to a noise reduction requirement of 7 dBA.

3.2.2 Major noise sources on Rapifine screens

Three major positions were established as important locations for the fine screen mechanic's exposure: a position directly in front of the screen at a height of 57 in. and two positions behind the screen, one 57 in. high and the other 24 in. high. The front position represented screen surface reversal work, the lower rear position represented rapper-counterweight area maintenance and adjustment, and the high rear position represented the inspection process of checking the undersize fraction. The SPL spectra at all three positions, although slightly different in level and spectrum shape, were nonetheless all high-frequency dominated.

The levels of high-frequency vibration in the screen surface were found to be responsible for a major part of the high-frequency noise received at normal ear height around the screen. A previous test had determined that holding the rapper away from the plunger rod, so that the main vibration path from the rapper to the screen surface was broken (the secondary path through the counterweight arm and fine screen structure was still intact), resulted in a large drop in the high-frequency SPL's as shown in Fig. 3.6. The rapper impacts the plunger rod and screen surface at a relatively low frequency (approximately 50 to 60 Hz). Figure 3.7 presents a time domain analysis of the sound of the PV-1-1/4 rapper impacting a block in an anechoic chamber. The impact rate of once per 20 msec (or 50 Hz) is clearly visible. However, as can be seen in Fig. 3.5, the noise problem basically involves frequencies above 500 Hz.

One attractive approach considered in the initial stages of the project was to install a low-pass mechanical filter in the plunger rod. This would transmit the low frequency impact force from the rapper into the screen surface, but block the harmful high frequencies from being transmitted into the screen surface and then radiated as noise. A low-pass filter, consisting of a

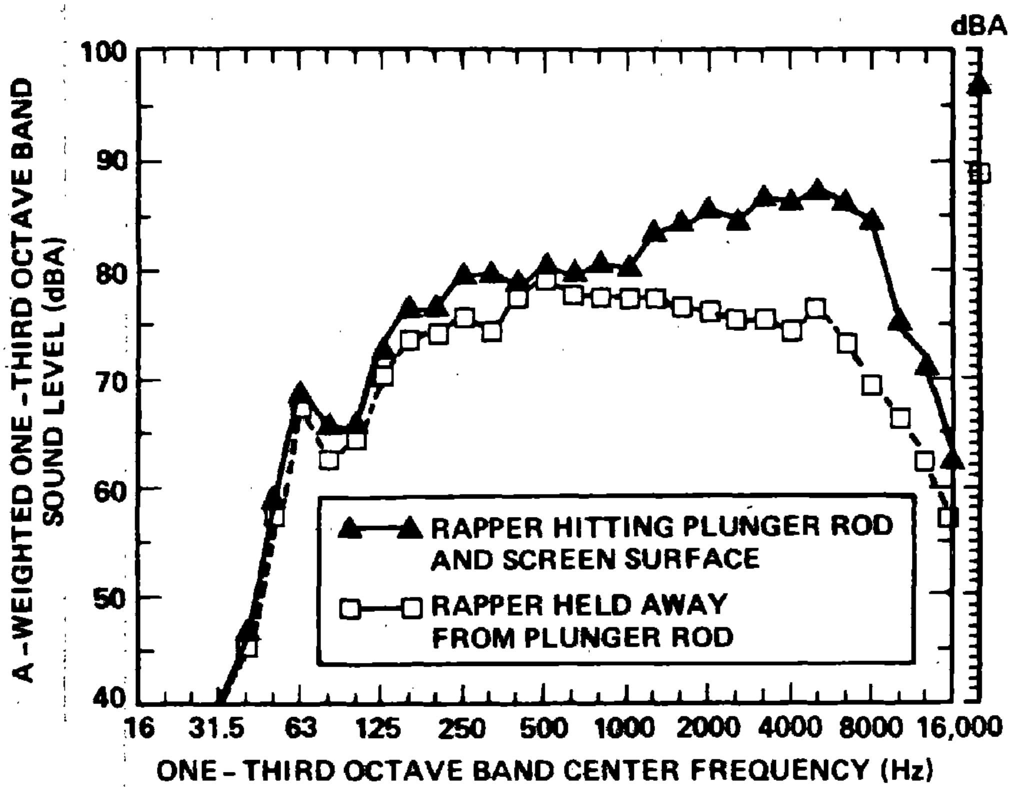


FIG. 3.6. EFFECT OF HOLDING RAPPER AWAY FROM PLUNGER ROD. MEASURED IN FRONT OF FINE SCREEN.

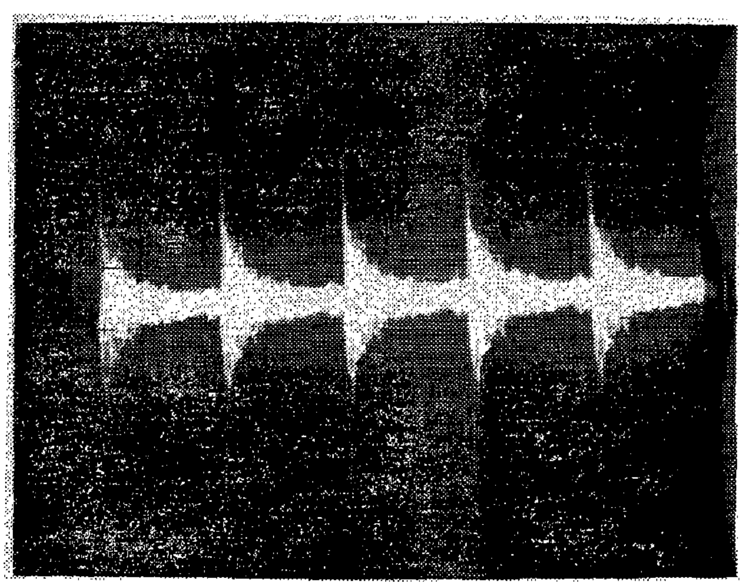


FIG. 3.7. TIME DOMAIN ANALYSIS OF RAPPER IMPACT NOISE AT 50 PSI. MEASURED IN ANECHOIC CHAMBER WITH RAPPER HITTING BLOCK.

block of 30 durometer rubber (1/4-in.-thick, 2-1/2-in. x 2-1/2 in.), was placed between the flat end of the plunger rod and the impact bar on the underside of the screen surface. This proved to be quite effective in reducing the high-frequency noise, without changing the 50 Hz rapper impact vibration level in the screen surface significantly. Figures 3.8a and 3.8b present the reduction in SPL and screen surface vibration obtained through the use of the rubber block. A noise reduction of 8 dBA was achieved. Similar behavior was observed in the vibration spectra - little change in level at the rapper impact frequency, but large reductions in the vibration levels above 500 Hz. This was obviously an effective noise reduction technique, but its effect on the operational performance of the fine screen was unknown. A long-term performance test was later organized using various durometer materials, from 30 durometer to 70 durometer rubber and 95 durometer urethane. Unfortunately, plant personnel reported that all the elastomer materials resulted in excessive blinding after two days of operation. Only the two control screens in the test row of 8 fine screens were still performing within acceptable limits. It was obvious that the high-frequency vibrations in the screen surface were a necessary part of the screening process and without them, blinding rapidly occurred. Therefore, the mechanical filter technique, which was attractive because of its small size and weight, was discarded because of operational deficiences.

Through a series of cover-and-expose tests, combined with the diagnostic tool of a cushioned plunger rod, the noise sources and noise radiating surfaces were rank ordered for each of the three selected positions. (These tests were conducted on a fine screen that was located in a quieter area of the plant where a better signal-to-noise ratio could be obtained.) Five temporary treatments were used during the rank-order testing. These were:

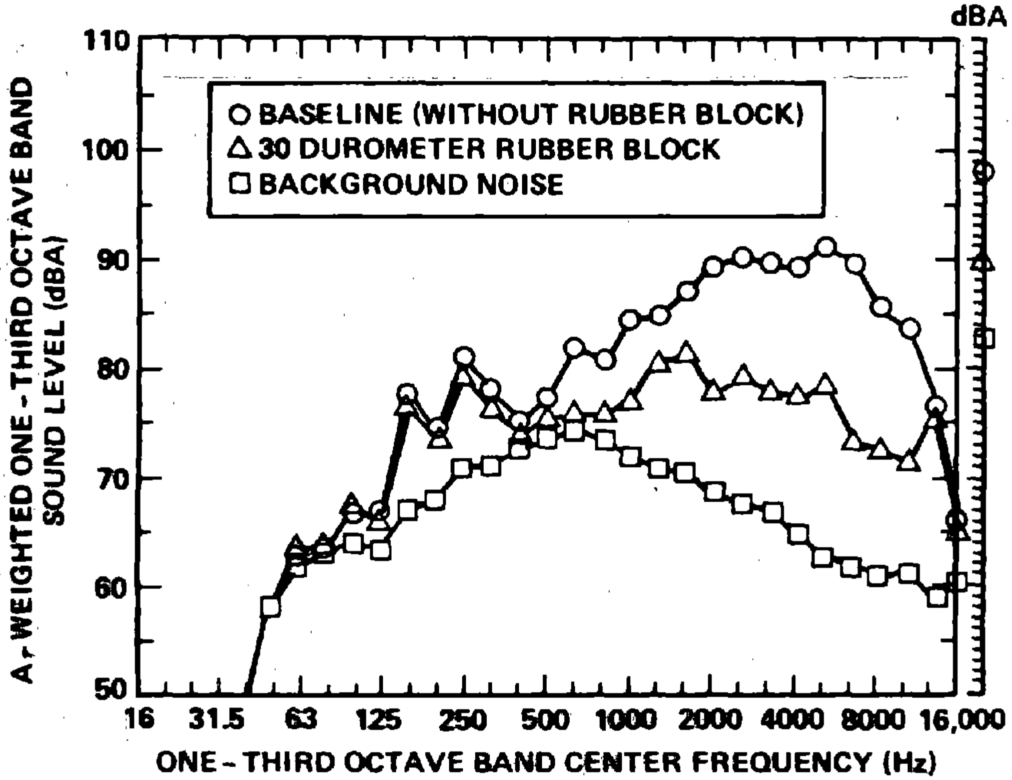


FIG. 3.8a. SPL'S MEASURED IN FRONT OF FINE SCREEN WITH AND WITHOUT RUBBER BLOCK BETWEEN PLUNGER ROD AND IMPACT BAR.

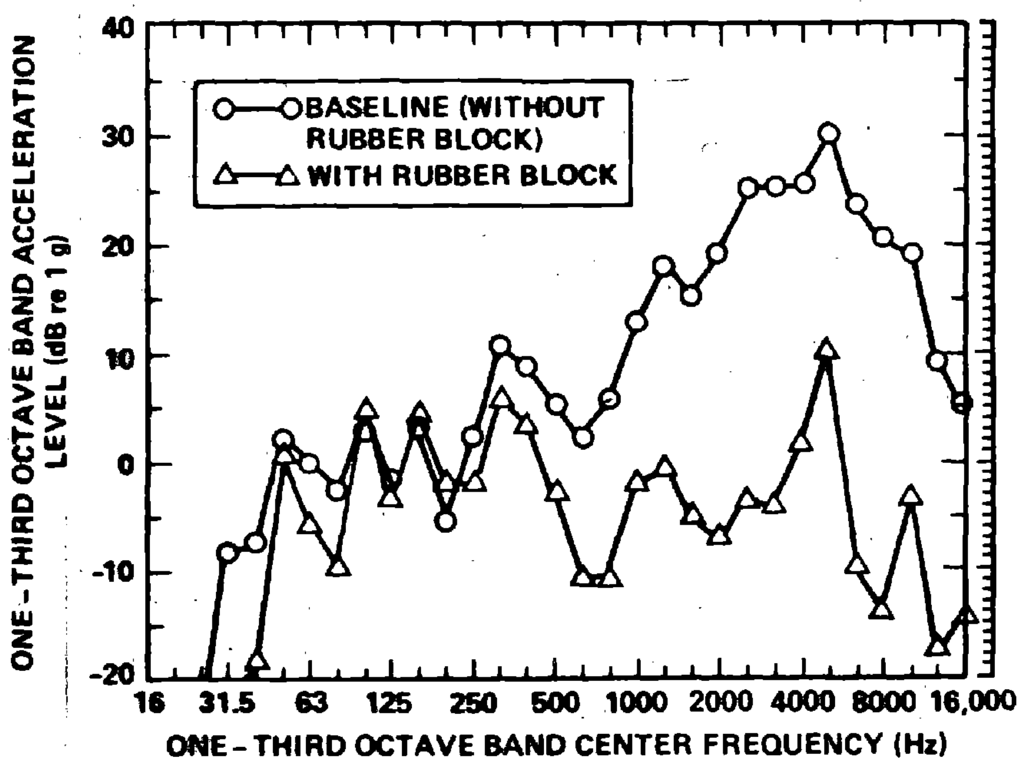


FIG. 3.8b. VIBRATION SPECTRA MEASURED ON SCREEN SURFACE WITH AND WITHOUT RUBBER BLOCK BETWEEN PLUNGER ROD AND IMPACT BAR.

1. Covering the front of the screenbox completely with lead vinyl.
2. Covering the top opening at the rear of the screenbox and the two gaps along the sides where the feedbox joins the screenbox with lead vinyl.
3. Installing a 60 durometer rubber pad at the end of the plunger rod.
4. Installing 60 durometer rubber bolt isolators between the rapper and counterweight arm, to reduce the vibration levels in the counterweight arm.
5. Enclosing the rapper and counterweight arm in a lead-vinyl cover.

Despite running these tests on a screen in a quieter part of the concentrator, background noise was still a problem below 500 Hz for some treatment combinations and below 1000 Hz for others. Figure 3.9 shows the results of the rank-ordering tests for the front position. These data show that a screen surface cover provided 8 dBA of noise reduction, indicating that noise radiation from the screen surface is the primary noise source. Additional benefit is derived from the remainder of the treatments. A total of 14 dBA reduction was achieved, bringing the equipment noise level down to the background level. These data showed that sufficient noise reduction could be obtained in front of the fine screen by the use of a screen surface cover.

Figure 3.9 shows that some of the rapper-counterweight arm noise at the rear of the fine screen influences the noise level at the front position; without the rapper enclosure, higher levels are measured in the 2 kHz to 16 kHz range, resulting in an increase of 2 dBA compared to the level of all five treatments. It was also found that 9 dBA of noise reduction could be achieved by using four of the five treatments, with the screen surface cover being the one left out. However, this approach required

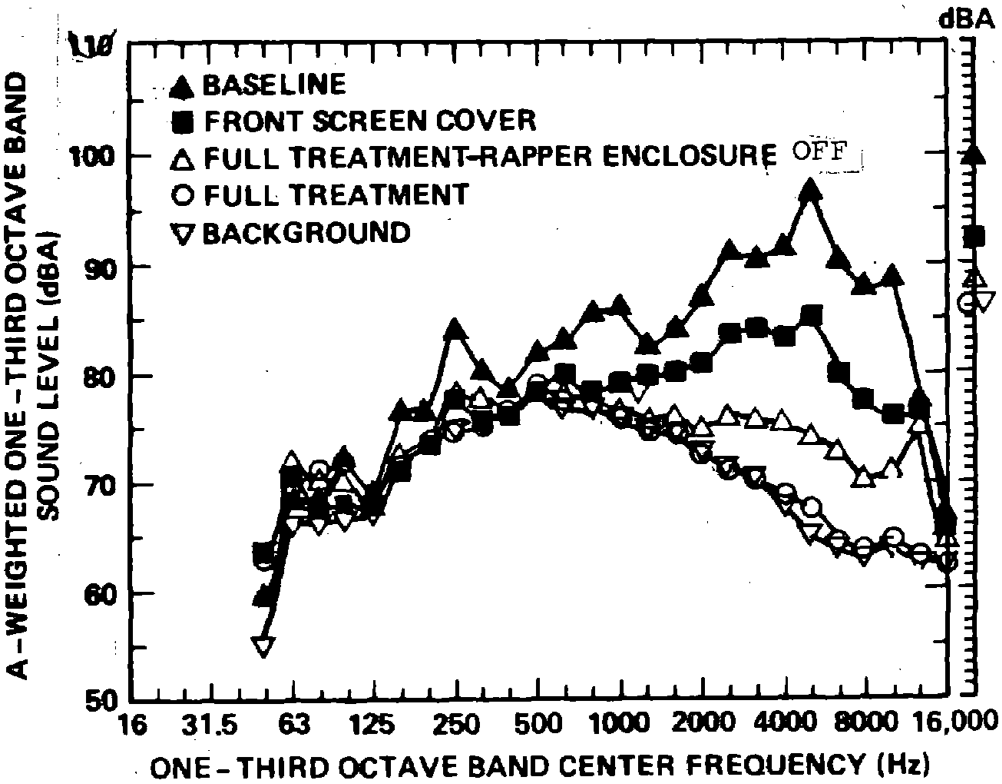


FIG. 3.9. RESULTS OF THE RANK ORDER TESTS FOR THE FRONT POSITION.

the use of a cushioned plunger rod to reduce the level of high frequency vibration and its effect on screen efficiency was unacceptable. Therefore, it wasn't a viable noise reduction technique and the authors concluded a screen surface cover would be required to achieve sufficient reduction at the front position.

Figure 3.10 presents the results of the tests at the high rear position (57 in. above the grating). The cover over the rear opening of the screenbox and the rapper enclosure are almost equally important treatments for this rear position.

Data are shown in Fig. 3.11 for the lower rear position (where the microphone is 2 ft above the grating). The vibration isolation treatment for the rapper provided a 6 to 7 dBA improvement by reducing the noise radiated by the counterweight arm. However, relatively high noise levels (on the order of 100 dBA)

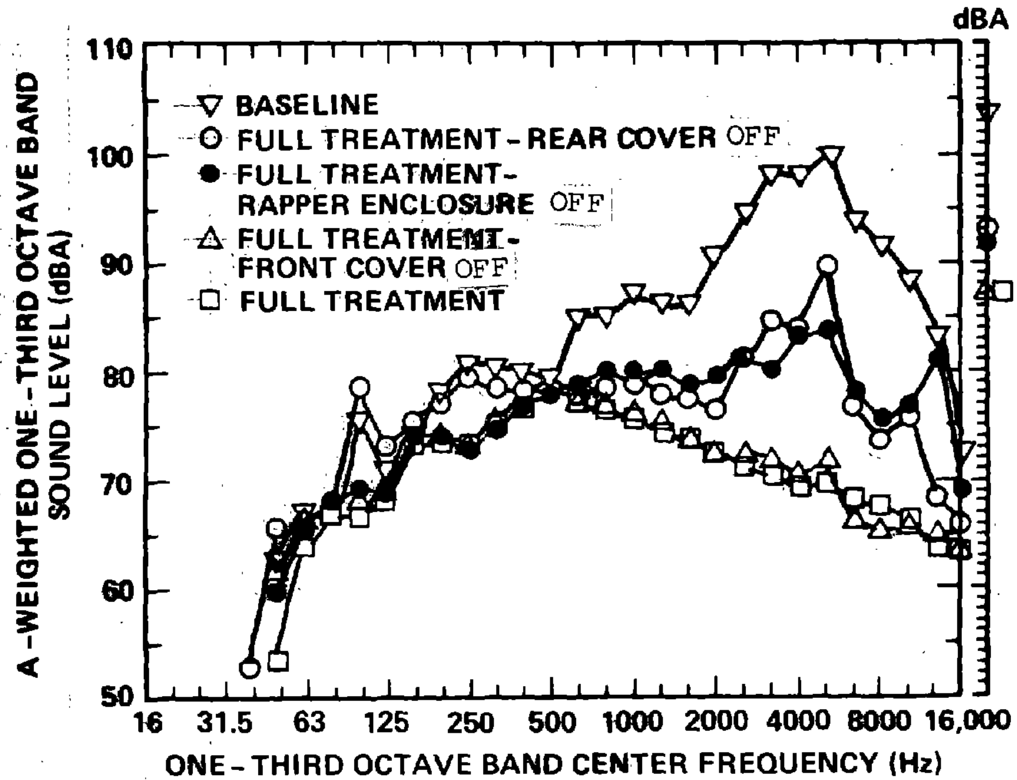


FIG. 3.10. RESULTS OF THE RANK ORDER TESTS FOR THE HIGH REAR POSITION.

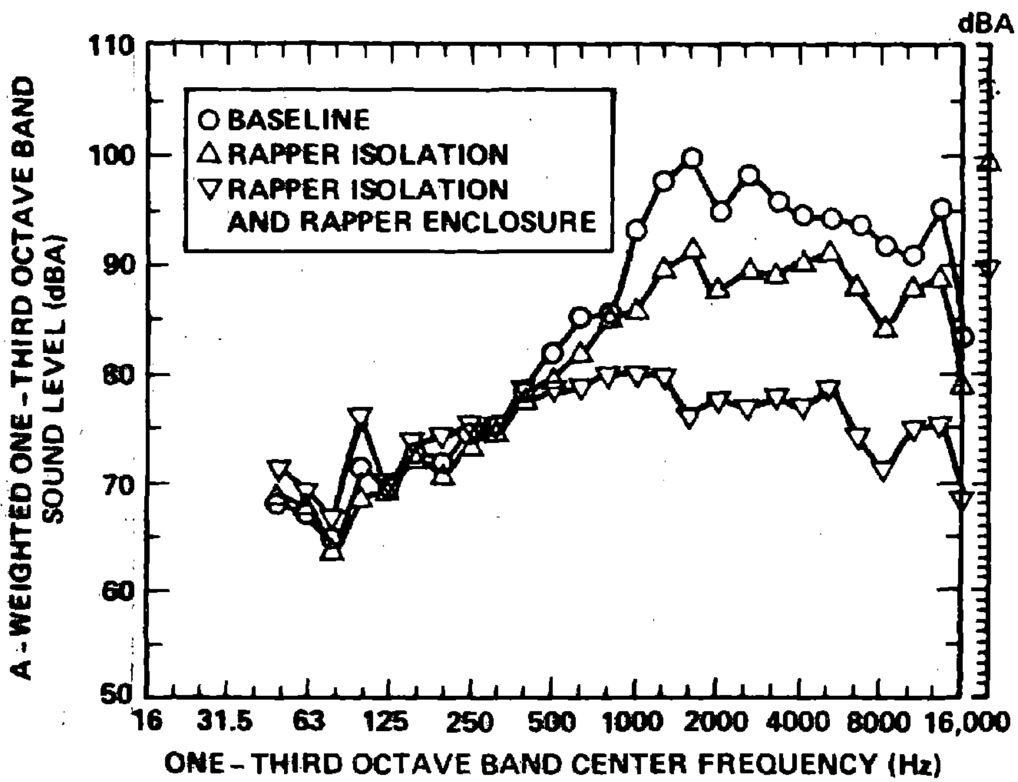


FIG. 3.11. RESULT OF THE RANK ORDER TESTS FOR THE LOW REAR POSITION.

remain from the impact of the rapper on the exposed end of the plunger rod, as evidenced by the additional reduction achieved when the rapper system is totally enclosed. These data indicate that, in addition to isolating the rapper from the counterweight arm, a cover over the rapper strikeplate and the exposed part of the plunger rod is necessary to achieve large reductions in sound level.

The overall results of the rank-order tests are summarized in Table 3.2 for each of the mechanic's positions. This table shows that two covers - one over the front of the screenbox and the other over the top opening at the rear of the screenbox - and modifications to the rapper/counterweight arm/plunger rod system are required.

TABLE 3.2. MAJOR NOISE SOURCES ON RAPIFINE SCREENS.

Position	Primary Noise Source(s)	Required Treatment
Front	Screen surface	Screen surface cover
High Rear	1. Screen surface radiation through rear opening at top of screenbox.	1. Cover over rear opening at top of screenbox.
	2. Rapper/counterweight arm/exposed plunger rod.	2a. Isolation of rapper from counterweight arm. 2b. Cover over rapper strike-plate and exposed plunger rod.
Low Rear	Rapper/counterweight arm/exposed plunger rod.	3a. Isolation of rapper from counterweight arm. 3b. Cover over rapper strike-plate and exposed plunger rod.

3.2.3 Other parameters affecting noise exposure

Four other factors can affect the noise exposure produced by the fine screens. Rapper air pressure and maintenance condition of the fine screens affect the magnitude of the sound level, and rapper duty ratio and rapper sequencing influence the duration of the mechanic's exposure to high noise levels.

Air Pressure

As shown in Table 3.1, the noise level produced by the rapper increases as the supply air pressure is increased. The data in Table 3.1 show the sound level at 6 ft from the rapper spanning the range from 95 to 105 dBA for the normal operating pressure range of 20 to 80 psi. These data are for the rapper only and were furnished by the manufacturer. To see how these rapper data transferred to actual operation on a fine screen, the SPL's between two rows of primary screens were measured at 30 psi, 50 psi, and 75 psi (see Fig. 3.12). The 50 and 75 psi

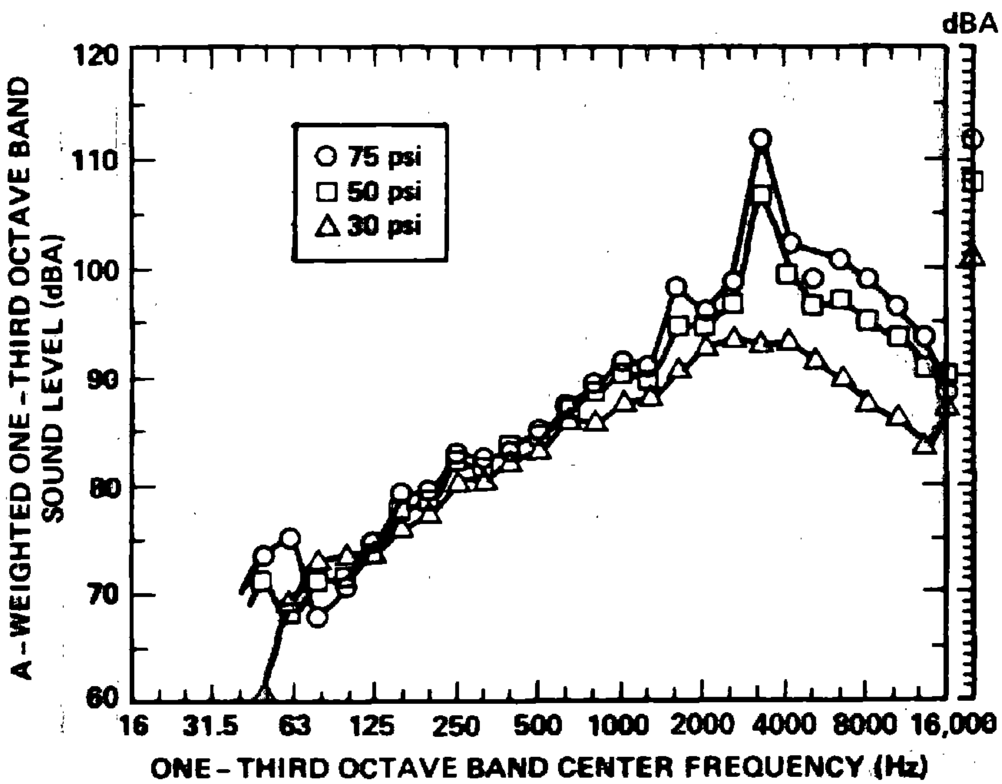


FIG. 3.12. EFFECT OF SUPPLY AIR PRESSURE ON SOUND LEVEL. SPL'S MEASURED BETWEEN TWO ROWS OF PRIMARY SCREENS.

spectra exhibit a peak at 3.15 kHz, while the 30 psi spectrum lacks the peak and consequently has a lower A-weighted sound level. These fine screens data follow the same tendency as the rapper data. Obviously, the mechanic's noise exposure could be reduced if the rappers were operated at the lowest possible air pressure. The minimum allowable pressure must be found by testing at each installation. If the pressure is too low, the rapper impact will not dislodge particles from the slots of the screen surface and blinding will occur. (If a higher-than-needed pressure is used, excessive noise results and screen surface life is reported to be shortened considerably.) The minimum allowable pressure is believed to be governed by the percent solids in the slurry, the volume flow over the screen surface, and the type of taconite ore. At the concentrator where the measurements were taken, plant personnel reported that 30 psi was insufficient to prevent blinding and that 50 psi was chosen for the operating pressure after considerable trial and error. They were, obviously, reluctant to change their operating practice. However, plant operators should be aware that the fine screens produce more noise at higher air pressures and the best operating pressure from a acoustic point of view is the lowest possible that effectively prevents blinding.

Duty Ratio

The duration of the "on" time for the rapper compared to the "off" time can be referred to as the rapper duty ratio. It is defined as

$$\text{rapper duty ratio} = \frac{\text{on time}}{\text{on time + off time}}$$

If the rapper is on for two seconds and off for two seconds, the duty cycle is 50%. Concentrator personnel can adjust the timing controls on the air supply to vary the duty cycle. Dorr-Oliver recommends an on time of 1/2 to 1 sec and using the off interval to adjust the duty ratio. The rapper will impact the screen surface between 30 and 60 times in the recommended interval and Dorr-Oliver believes this is a good starting point for finding the optimum duty ratio.

In the taconite industry, a wide variety of on times and duty ratios are used. In one typical concentrator, single impact rapping is used - the screen surface is hit once every 2.4 sec, as shown in Fig. 3.13. This is adequate to prevent blinding with the particular combination of percent solids, size distribution, and type of ore. In the concentrator where the noise control treatments were developed for this project, the duty ratio shown in Fig. 3.14 was used. The on time is 7.5 sec and the off time is 6.3 sec, which yields a duty ratio of 54%. This is significantly different from the plant which uses single impact rapping.

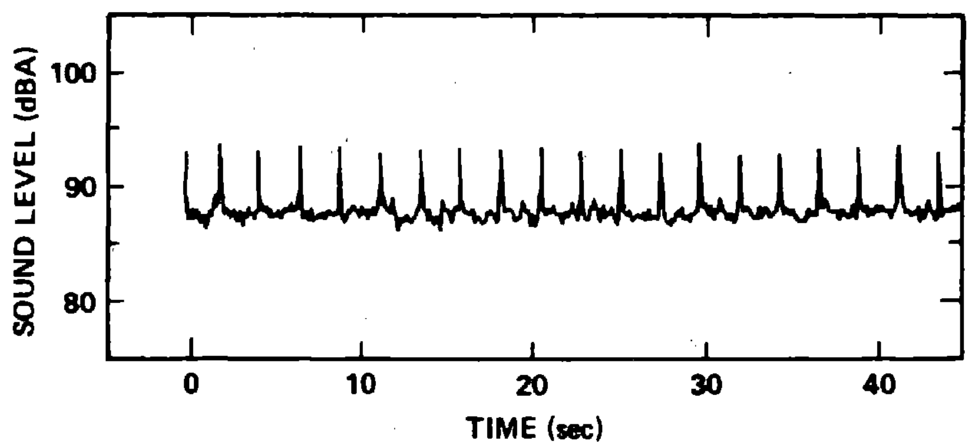


FIG. 3.13. TIME HISTORY OF SINGLE IMPACT RAPPING USED ON CONCENTRATOR NO. 1.

Industry-wide generalizations concerning the best rapper duty ratio cannot be made, since local conditions involving the type of ore and screen loading will be different at each mine, and what works at one mine will not necessarily work at another. Guidance can be offered, though, in recommending that concentrator personnel run tests to find the shortest on time and smallest

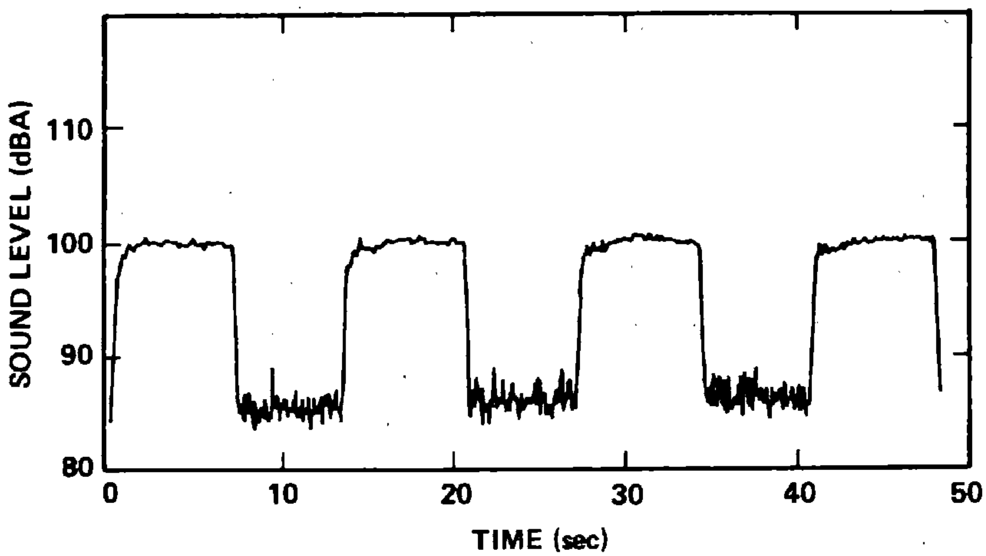


FIG. 3.14. TIME HISTORY OF RAPPER DUTY RATIO USED IN CONCENTRATOR NO. 2.

duty ratio that is acceptable from a blinding point of view. Rapping the screen surface longer than is absolutely necessary results in higher noise exposure for the mechanic, high air consumption, and shortens the life of the screen. (The screen surface is made from 304 stainless and work-hardens when it is struck. After a few months of operation, it becomes brittle.) Therefore, excessive rapping has two drawbacks and is to be avoided for optimum operation.

Rapper Sequencing

Once the optimum time and duty ratio have been determined, further reduction of the mechanic's noise exposure can be achieved by proper sequencing of the rappers. This applies only to areas where there is a high concentration of fine screens and the background sound level is below 90 dBA. In an area with two lines side by side, each with a front and back row and a primary and secondary stage, there could be 64 fine screens set up in 8 rows of 8 screens each. Each row is usually sequenced according to a timing pattern that places a reasonably steady load on the compressed air system. One row will come on for a few seconds and then shut off, followed by the second row, and then the third, etc. Although this results in a more manageable load for the air compressors, it has a negative effect on the fine screens mechanic's noise exposure, because it keeps the sound level above 90 dBA (thus contributing to his daily noise dose) almost continuously. Acoustically, the preferred timing pattern is to operate all the fine screens in one area at the same time and have them all off at the same time. This is beneficial because the MSHA regulation has each 5 dB trading ratio (i.e., the allowable exposure duration is halved for a 5 dB increase in sound level) and the noise of multiple rows of fine screens increases on a 3 dB basis (i.e., two rows of screens are 3 dB noisier than a single row). In a location where the mechanic is exposed to eight rows of screens, the noise level of all eight rows operating simultaneously will be at most 9 dB higher than a single row. The actual increase will be less than 9 dB because some rows will be more distant from the mechanic and, therefore, not all rows will contribute equally to the sound level. By phasing the rappers so that they are all on together, the mechanic will be exposed to higher noise levels for a short period of time, followed by a longer interval at the background sound level of less than 90 dBA. This concept is illustrated in Fig. 3.15.

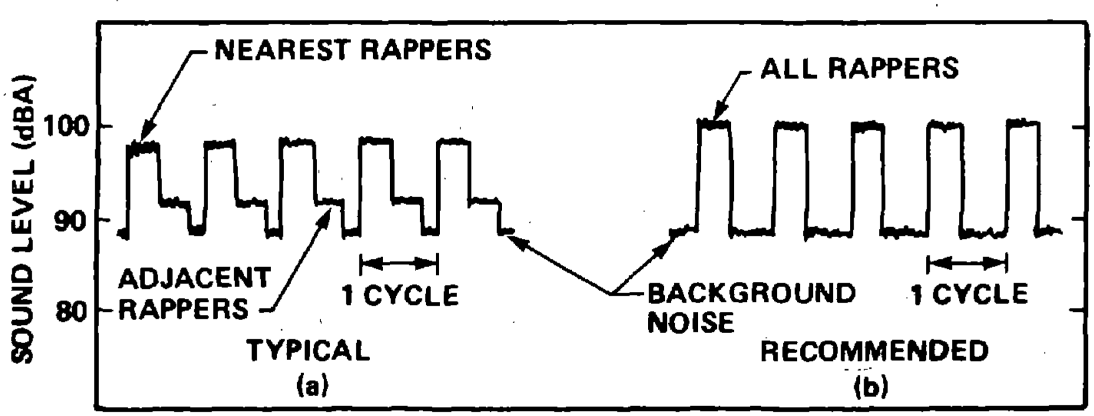


FIG. 3.15. TYPICAL AND RECOMMENDED RAPPER SEQUENCING FOR MULTIPLE ROWS OF FINE SCREENS.

The normal row sequencing found in most concentrators is shown in Fig. 3.15a. The nearest rappers produce the highest sound level and when they switch off, the noise level drops down to the noise produced by an adjacent row of rappers. When these adjacent rappers switch off, there may be a short period of background noise followed by the nearest rappers switching on again. The preferred timing scheme is shown in Fig. 3.15b. Here, all rappers are switched on and off together. This results in a few seconds of higher noise level, but is followed by a longer interval of background noise. If the background level is below 90 dBA, this interval does not contribute to the daily noise dose of the mechanic at all. The only contribution comes from the period when all the rappers are on. Since the sound level has been increased on a 3 dB basis, and the time spent above 90 dBA has been decreased on a 5 dB basis, the net change is an overall benefit. Modifications to the plant air system would probably be necessary in order to supply the large volume required when all rappers switch on simultaneously. The use of a large air reservoir would help to keep the air pressure from dropping from the initial surge demand.

Changes to the timing pattern, as the sole means of treatment, will not provide a very large reduction in noise exposure and are unlikely to bring the mechanic into compliance; however, a 20 to 30% reduction in noise exposure is realizable. For typical values of daily noise dose (on the order of 2.5 to 3.0 for the fine screens mechanic), this is equivalent to a 2 dB reduction in the time-weighted average sound level. Therefore, this treatment will not work by itself, but is best used in combination with other treatments.

Maintenance

As mentioned previously, maintenance is performed on a fine screen every three weeks. During the initial sound level measurements for this project, large variability was detected from screen to screen. It was found that the SPL's in the 2-kHz to 16-kHz range could vary by as much as 5 to 7 dB and the A-weighted sound level (which is controlled by the high-frequency SPL's) had a 3 dBA variability. Since small changes in sound level due to noise control treatments would be hard to detect with a 3 dB variability from screen to screen, some effort was made to determine the source(s) of the variability. Measurements of the acceleration level in the structure of the fine screen showed an even larger variation from screen to screen than the SPL measurements. This indicated that the impact force varied from rapper to rapper over quite a wide range. It was observed that the counterweight arm and rapper were "bouncing" on the end of the plunger rod on some of the fine screens. This "bouncing" excited more high frequency noise and resulted in higher A-weighted sound levels. An example of this "bouncing" condition is presented in Fig. 3.16. The sound level is increased by 2 dBA due to the bouncing counterweight arm. The fine screens mechanic had added large mill liner nuts to the balance end of the counterweight arm to try and eliminate the bouncing, but this has been only partially successful.

In order to keep the rapper-generated noise to a minimum, the counterweight arm should be properly balanced so that the rapper does not bounce on the plunger rod. Also, rapper maintenance should be performed on a regular basis to keep the impact force uniform in each fine screen.

3.3 Design Features of Treatment for Fine Screens

The results of the rank-order tests (described in Sec. 3.2.2) showed that two covers and treatment of the rapper/ counterweight arm system would be required. This section describes the design considerations involved in these three noise control items.

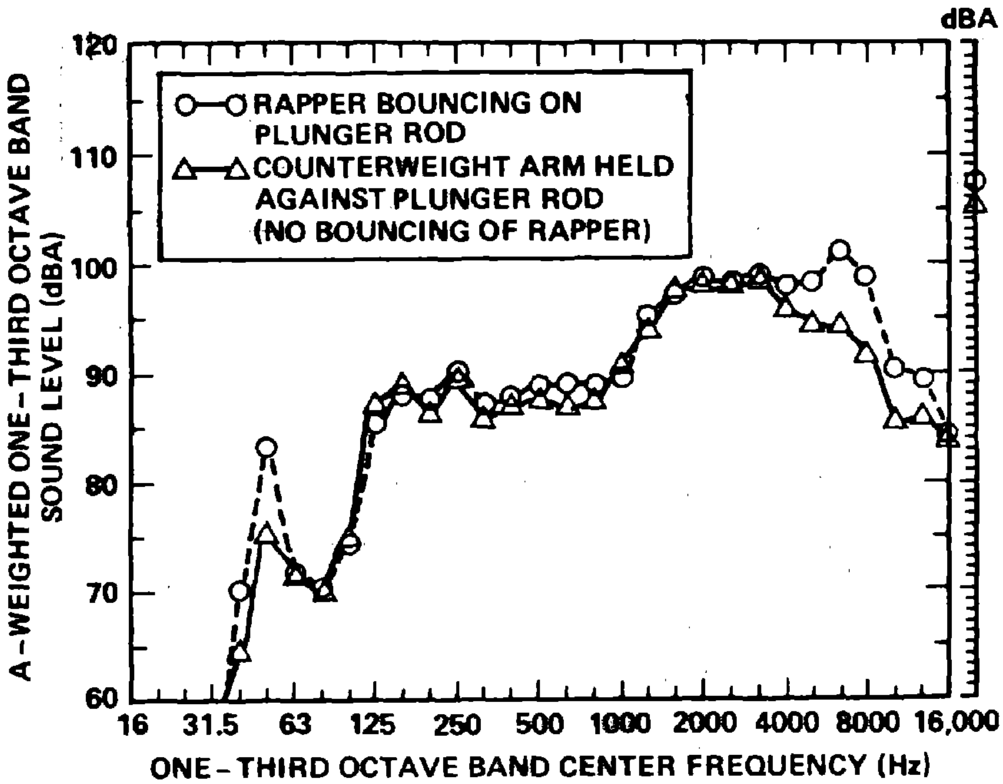


FIG. 3.16. EFFECT OF MAINTENANCE ON SOUND LEVEL. SPL'S MEASURED AT TOP OPENING AT REAR OF SCREENBOX.

3.3.1 Design of screen surface cover

A splash cover (Dorr-Oliver Part No. 01B12343-02) is provided for the bottom portion of the screen surface. The mechanical design of the screen surface cover was patterned after the Dorr-Oliver splash cover, which is 1/16-in. steel plate with angle stiffeners along the bottom and two sides. Location tabs along the bottom of the splash cover sit against the screenbox and keep the splash cover in proper position. The splash cover is 25-1/2-in.-wide and 24-in.-high and has a handle on the front to facilitate removal for screen surface maintenance. A new splash cover was constructed for the demonstration project.

The screen surface cover was also constructed of 16 ga. steel plate and was 25-1/2-in.-wide and 26-1/4-in.-high. The top 8-3/4-in. was bent at a 35° angle to cover the top part of the screen surface. The screen surface cover (installed with the splash cover) is shown in Fig. 3.17. It was also provided with a handle for easy removal and had two location tabs near the bottom corners. A rubber flap along the top edge of the cover contacts the feedbox to provide a good seal. This seal was reinforced with a steel back-up bar. The bottom edge was designed with an underlapping bar so that the existing splash cover rests against this edge when in place. The cover was stiffened by angle iron along the sides and across the diagonal. A 2-in.-thick layer of

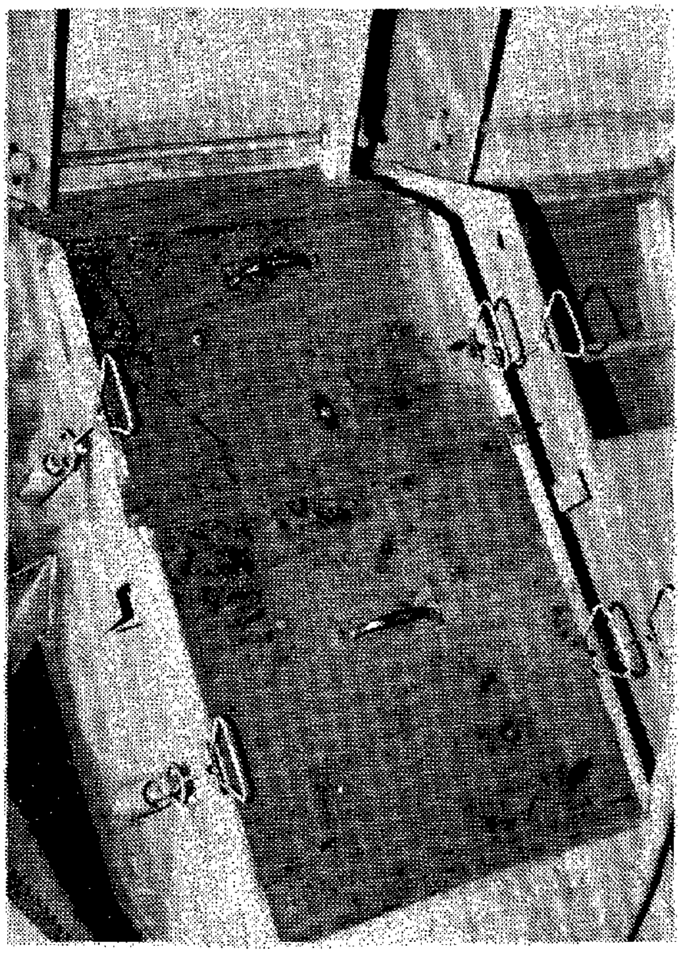


FIG. 3.17. SCREEN SURFACE COVER SHOWN INSTALLED WITH SPLASH COVER ON FINE SCREEN.

glass fiber sound absorption was fastened to the underside of the cover. Claremont Company Type 1613 'Tuffskin' was used for this application, as it has a vapor barrier of textured Mylar bonded to a 2-in. layer of glass fiber cloth. The Tuffskin kept the splashing water from soaking the glass fiber sound absorption material.

The Tuffskin sound absorption material was protected from mechanical damage by a layer of perforated metal. Perforated aluminum sheet, 20 ga. thickness and 40% open, was used. To attach the perforated metal/Tuffskin treatment, holes were drilled in the 16 ga. cover plate and punched in the Tuffskin and perforated aluminum. Bolts and stud buttons were used to secure the Tuffskin and perforated aluminum to the cover plate. (Two-in.-thick Tuffskin 1613 material and perforated aluminum were also added to the Dorr-Oliver splash cover.) A sketch of the screen surface cover is shown in Fig. 3.18.

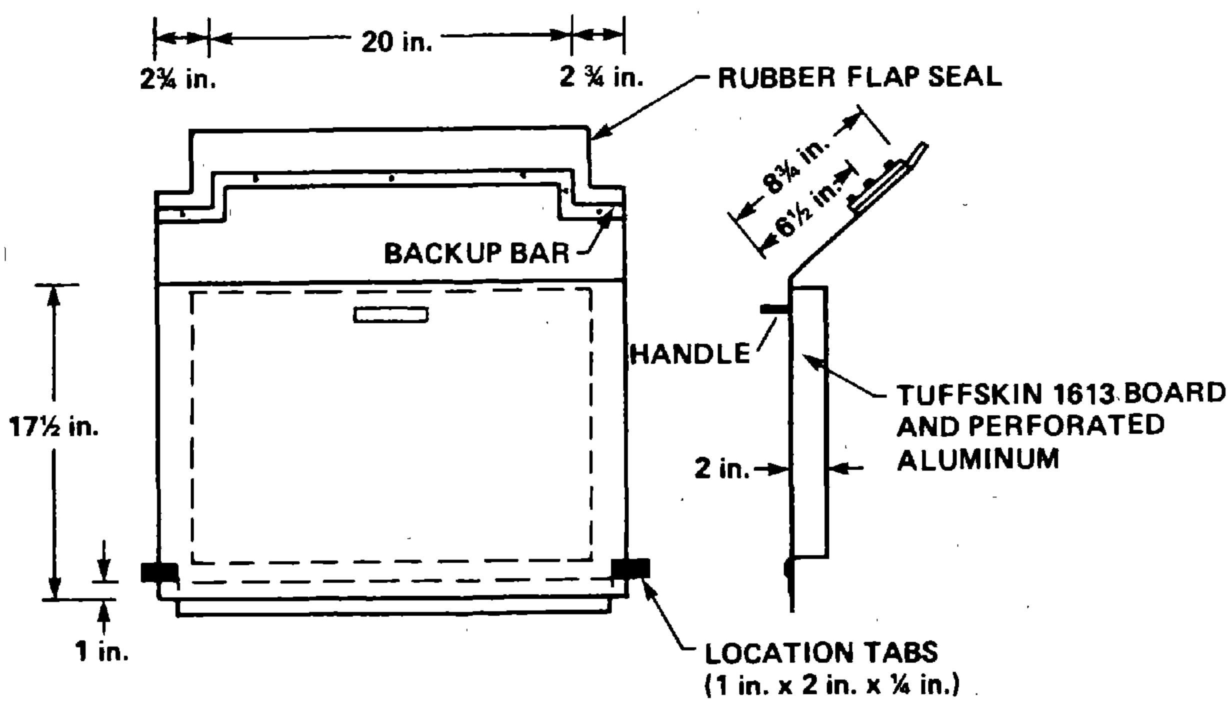


FIG. 3.18. DESIGN OF SCREEN SURFACE COVER.

3.3.2 Design of rear cover

The rear cover was designed and constructed along similar lines. It was 16 ga. steel plate, 21-1/4-in.-wide and 12-1/2-in.-deep. A rubber flap seal, reinforced with a steel back-up bar, was used along the front edge. A sketch of the rear cover is shown in Fig. 3.19 and the actual installation is shown in Fig. 3.20. A photograph of the same location, but without the cover, is shown in Fig. 3.21. The cover was supported along three sides - the left and right edges by two new support angles (2-in. × 2-in. angles, 11-in.-long, bolted in place), and the backedge by the existing feedbox flange. To accommodate the three large bolts and washers that hold the screenbox and feedbox together, three cutouts were made along the back edge of the cover. Two-in.-thick Tuffskin 1613 material was attached to the underside of the cover in the same manner as was used on the front screen surface cover. A handle was provided to facilitate the mechanic's inspection of the screen underflow.

A number of small openings had to be sealed after the two covers were installed. The rear cover sealed up the large opening at the top of the screenbox, but two long, thin gaps remained along the sides of the feedbox. The feedbox is narrower

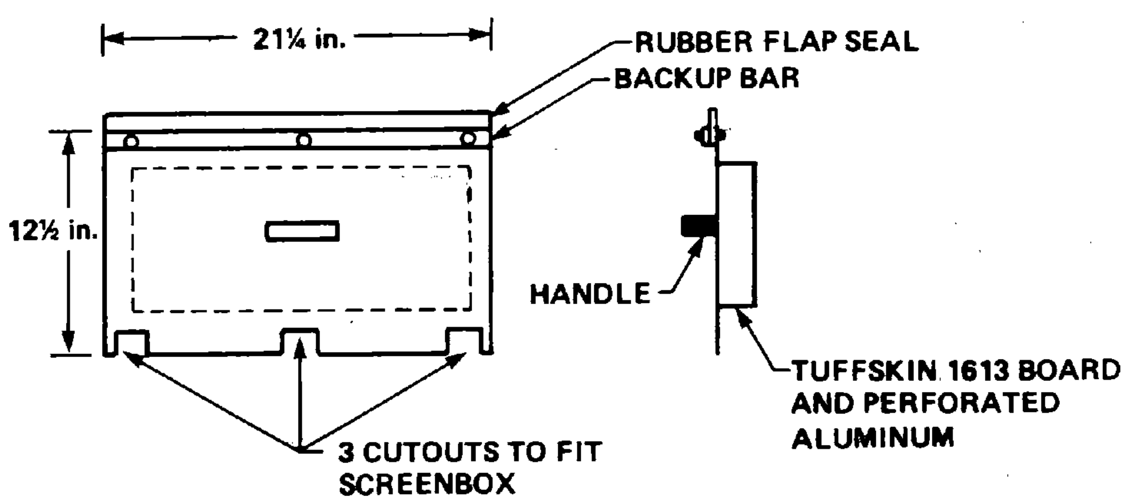


FIG. 3.19. DESIGN OF REAR COVER.

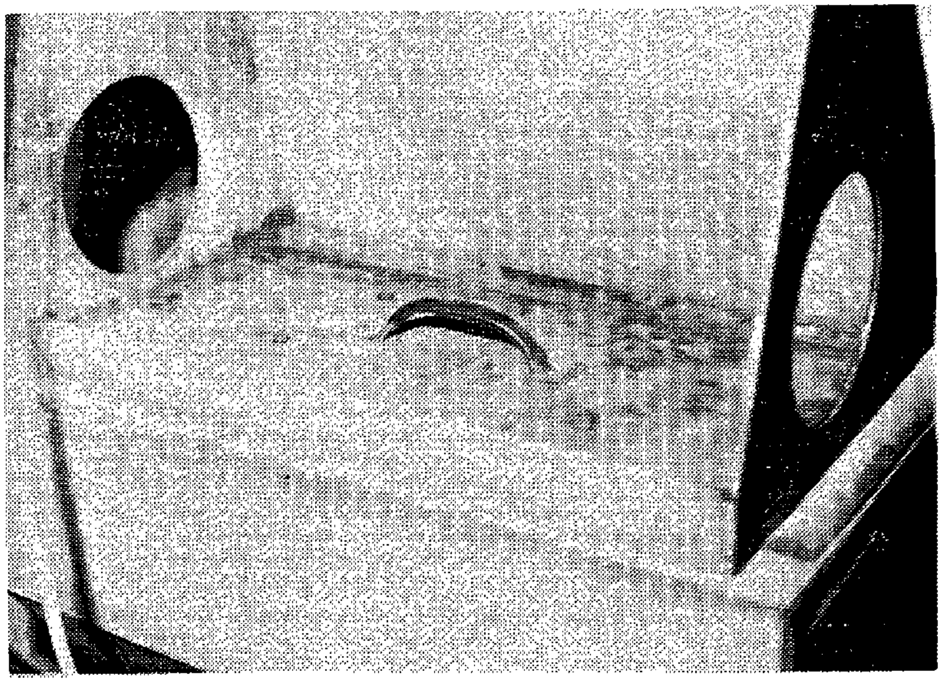


FIG. 3.20. REAR COVER INSTALLED ON FINE SCREEN.

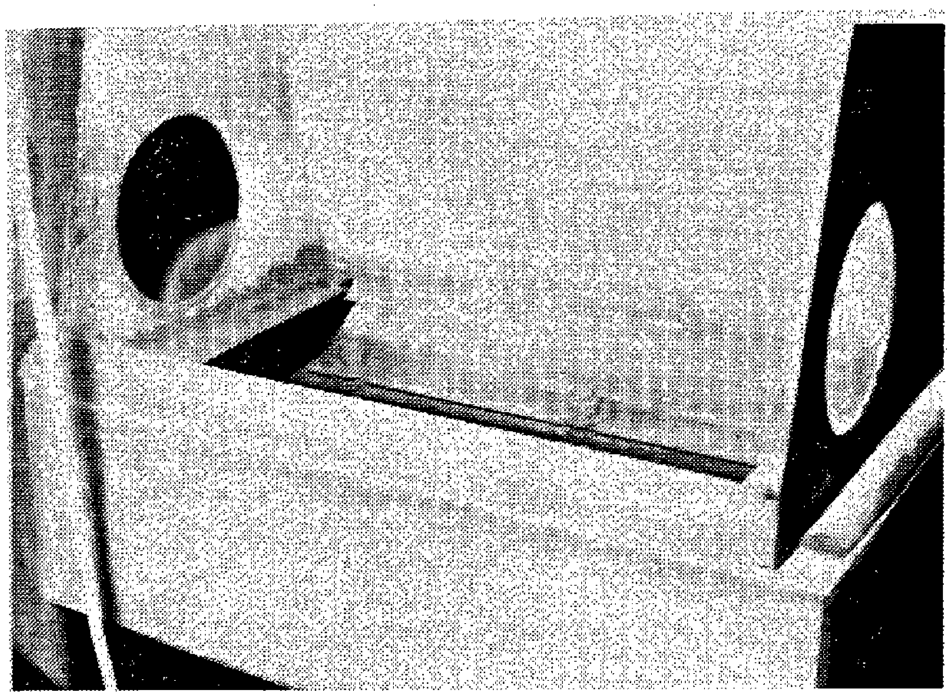


FIG. 3.21. REAR COVER REMOVED.

than the screenbox and this resulted in the two gaps. Closed cell foam pipe insulation, 5/8-in.-ID by 1/2-in. wall thickness, was cut to length and inserted in the gaps (see Fig. 3.22). The screenbox had four lift holes near the top and these were sealed with lead-vinyl and adhesive tape. If a welder had been available, 3-in. × 3-in. steel plates would have welded over the lift holes.

3.3.3 Design of rapper/counterweight arm treatment

The treatment for the rapper/counterweight arm system consisted of four items:

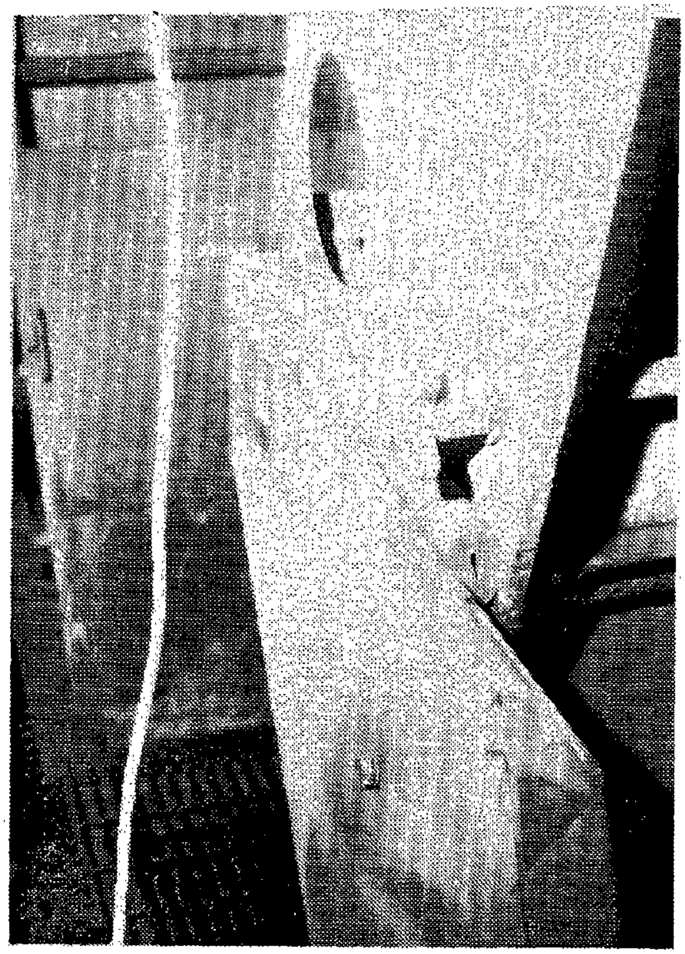


FIG. 3.22. CLOSED CELL FOAM PIPE INSULATION USED TO SEAL GAP BETWEEN FEEDBOX AND SCREENBOX (TREATMENT ALSO PARTIALLY VISIBLE IN FIGS. 3.20 AND 3.21).

1. Vibration isolation of the rapper from the counterweight arm.
2. Cover over the rapper strikeplate area.
3. Sheath over the exposed part of the plunger rod.
4. Muffler on the rapper exhaust.

Figure 3.23 shows three of the four treatments installed on the fine screen.

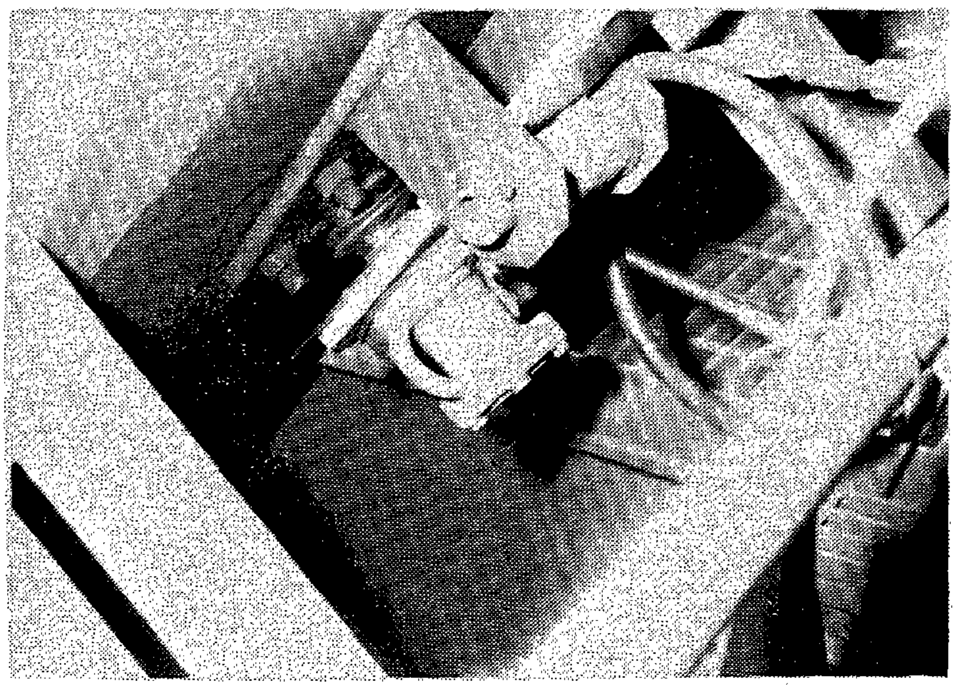


FIG. 3.23. PHOTOGRAPH OF RAPPER/COUNTERWEIGHT ARM TREATMENT (EXHAUST MUFFLER HIDDEN FROM VIEW.).

The vibration isolation consisted of substituting a "soft" rubber mounting system for the "hard" bolting arrangement. Rubber step bushings were made from Atlantic India Rubber Co. No. 2406 grommets; they were cut in half lengthwise to make two step bushings. The 1/2-in. rapper mounting holes were enlarged to 7/8-in.-diameter and the rapper was bolted to the counterweight arm as shown in Fig. 3.24. A layer of 1/4-in.-thick, 40 durom-

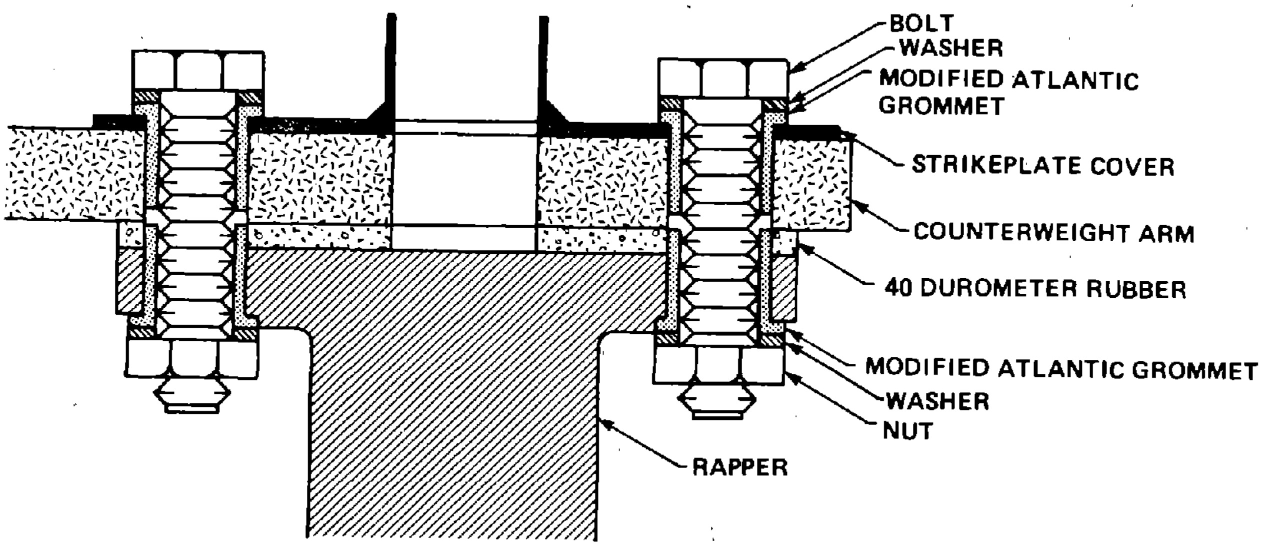


FIG. 3.24. DESIGN OF RAPPER VIBRATION ISOLATION SYSTEM.

eter rubber was used between the rapper and counterweight arm. No metal-to-metal contact was allowed so that the acceleration levels in the counterweight arm would be significantly reduced.

The rapper strikeplate cover was an 1/4-in. plate, 5-in.-wide and 7-1/2-in.-long, with a 1-1/2-in. hole in the center and a 5/8-in.-long collar welded over the hole. Two 7/8-in. mounting holes were drilled using the rapper bolt pattern. The strikeplate cover was attached to the counterweight arm using the rapper mounting bolts.

Various pieces of hardware were tested for the sheath to cover the exposed portion of the steel plunger rod. These included PVC hose, a rubber gearshift boot from an automobile, and a rubber bellows. The bellows worked best, providing a good seal at the strikeplate cover and the screenbox. A Minor Rubber Co. Type ZFD-1326 bellows was selected. It was fastened at both ends using hose clamps.

Exhaust mufflers were purchased from the rapper manufacturer. These turned out to be 1/4-in. pipe tap size, sintered-bronze mufflers with an adjustable flow control, such that the backpressure was adjustable. In this application, the flow control was adjusted to the least backpressure-maximum flow position. The mufflers supplied by Martin were identical to C.W. Morris Co. Quietaire Speed Control Silencers, Model MM-2. The muffler was installed on an extension pipe to remove the sintered bronze element from the grease and dirt in the vicinity of the rapper. (The air supply line has a filter to eliminate dirt from the compressed air.)

3.4 Effectiveness of Selected Treatments

3.4.1 Acoustical performance

Sound level measurements were taken at four positions around the quieted screen in order to test the effectiveness of the selected treatments. These measurements were obtained during a plant shutdown so that background noise from other equipment was not a problem. However, slurry wasn't available during the plant shutdown and in order to run the fine screen in a wet condition, a fire hose was placed in the top of the feedbox and clear water run over the screen. Because of the lack of slurry, the baseline sound spectra were not in perfect agreement with the operating baseline spectra; however, they remained dominated by the high frequencies.

Results for the front position where the mechanic performs screen reversal maintenance are shown in Fig. 3.25. The screen surface cover and the rear hole cover provided 2.5 dBA improvement over the baseline condition. When the rapper treatment was added, the sound level dropped to 84 dBA, a 10 dBA improvement over the baseline level. The sound absorption under the covers provided an additional 1 dBA reduction.

Similar results were obtained at the rear, ear-height position where the mechanic inspects for proper screen underflow; the spectra are shown in Fig. 3.26. The covers provided 3 dBA reduc-

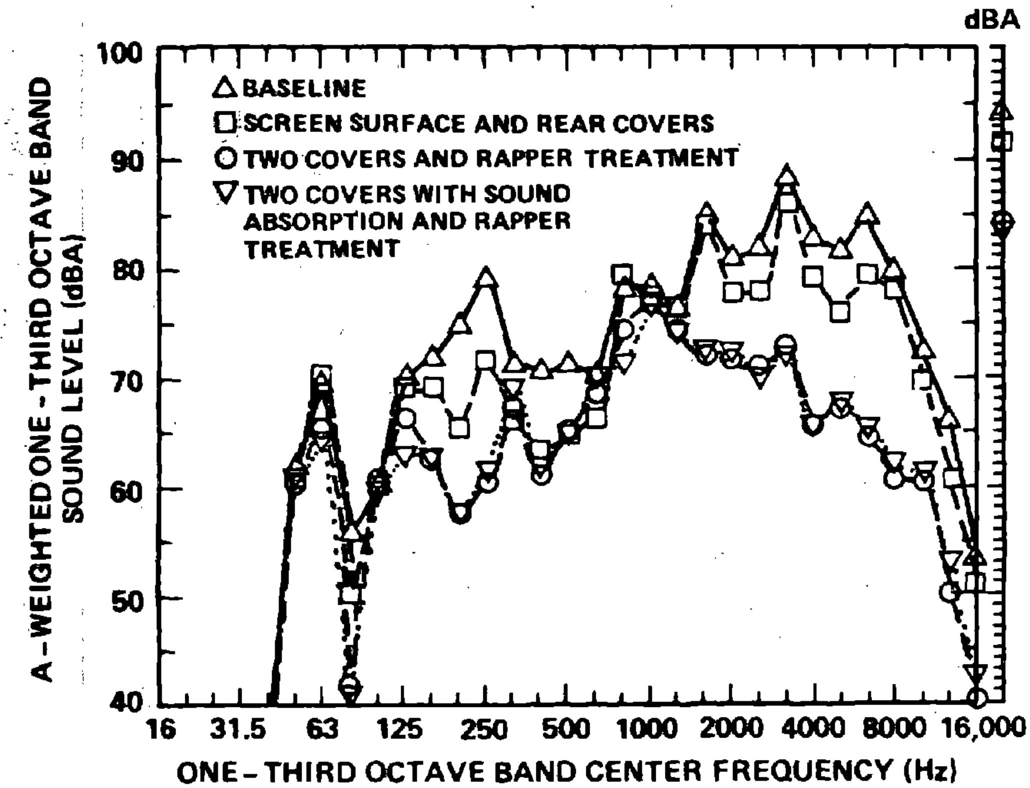


FIG. 3.25. SPL'S AT FRONT POSITION WITH NOISE CONTROL TREATMENTS INSTALLED.

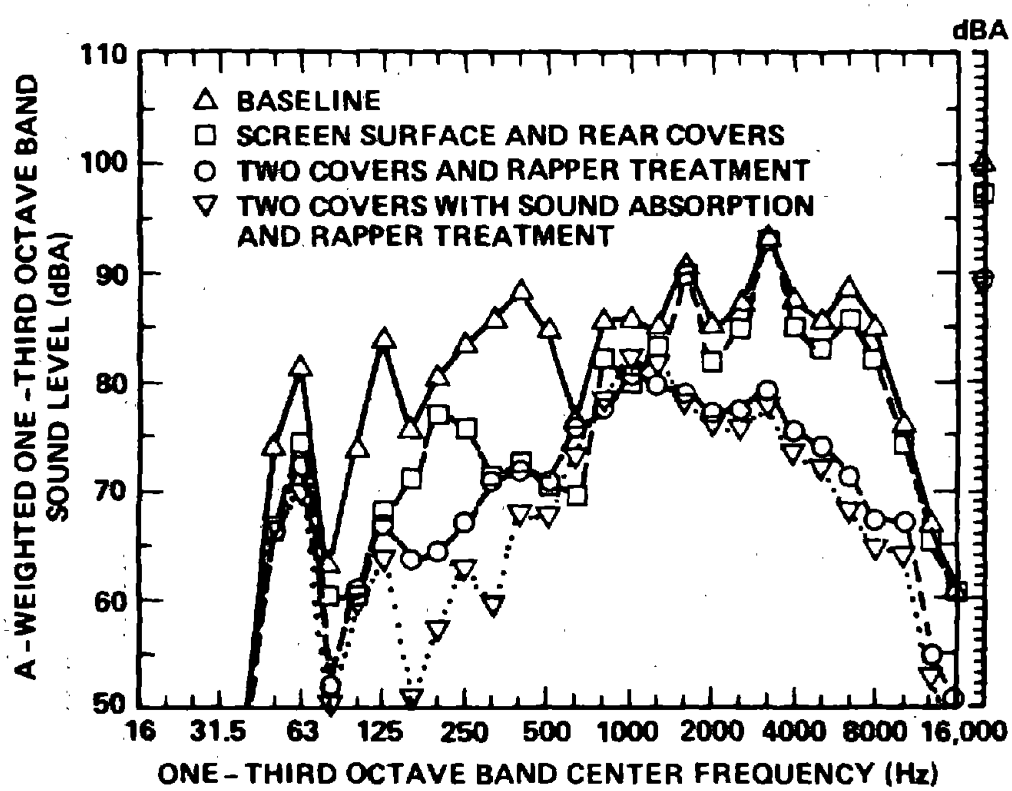


FIG. 3.26. SPL'S AT REAR, EAR-HEIGHT POSITION WITH NOISE CONTROL TREATMENT INSTALLED.

tion and the rapper treatment in combination with the two noise control covers yielded 11 dBA of improvement. The addition of the sound absorption treatment to the covers yielded another 0.5 dBA of reduction.

At the rear, low position near the rapper, the covers, as expected, provided only minor benefit (1.5 dBA). The most significant treatment was to the rapper and counterweight arm, which provided an additional 12 dBA reduction (see Fig. 3.27). Again, the sound absorption treatment on the two covers produced only small improvement (1 dBA) over the combination of the unlined covers and the rapper treatment.

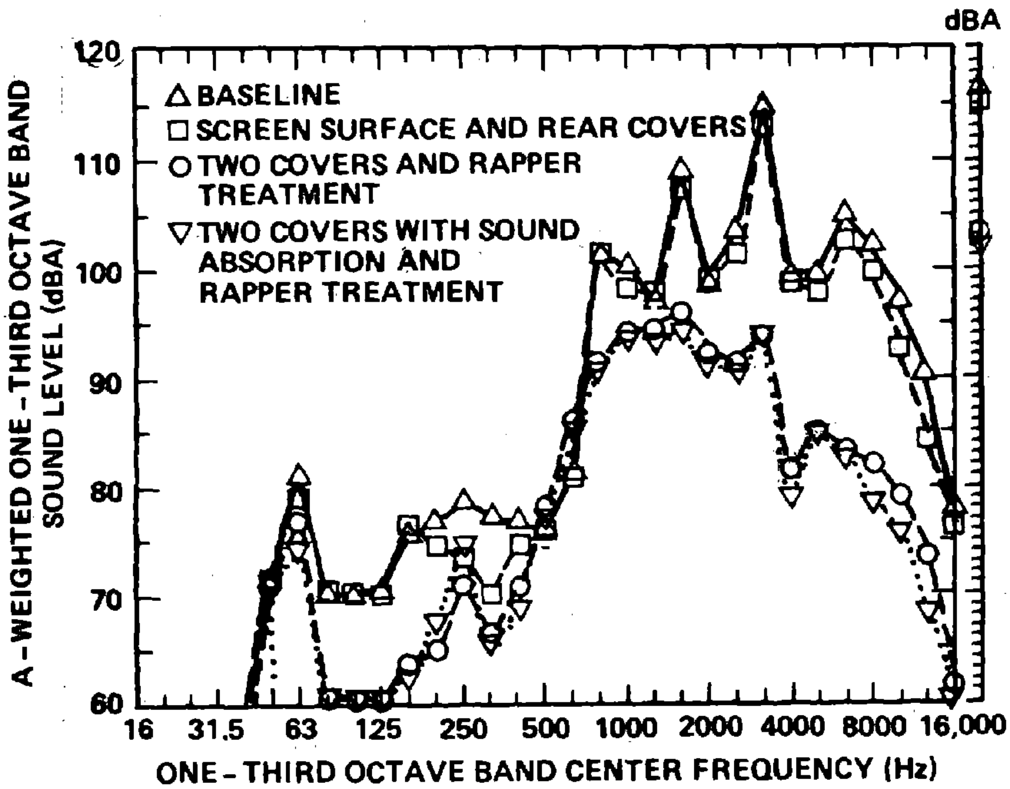


FIG. 3.27. SPL'S AT REAR LOW POSITION WITH NOISE CONTROL TREATMENT INSTALLED.

When the mechanic is on the secondary (bottom) level of a two-stage fine screens installation, he is exposed to the noise of the primary screens above, as well as to the noise of the secondary screens. The principal noise source on the primary screens is the rapper/counterweight arm system, which is located approximately 5 ft above the mechanic's head when he is standing between two rows of screens or behind the back row. The rapper treatment performs similarly when measured on the secondary level at the rear, low position. The reduction in sound level at the secondary level is 13.5 dBA, as shown in Fig. 3.28. The high-frequency components of the baseline spectrum were reduced considerably.

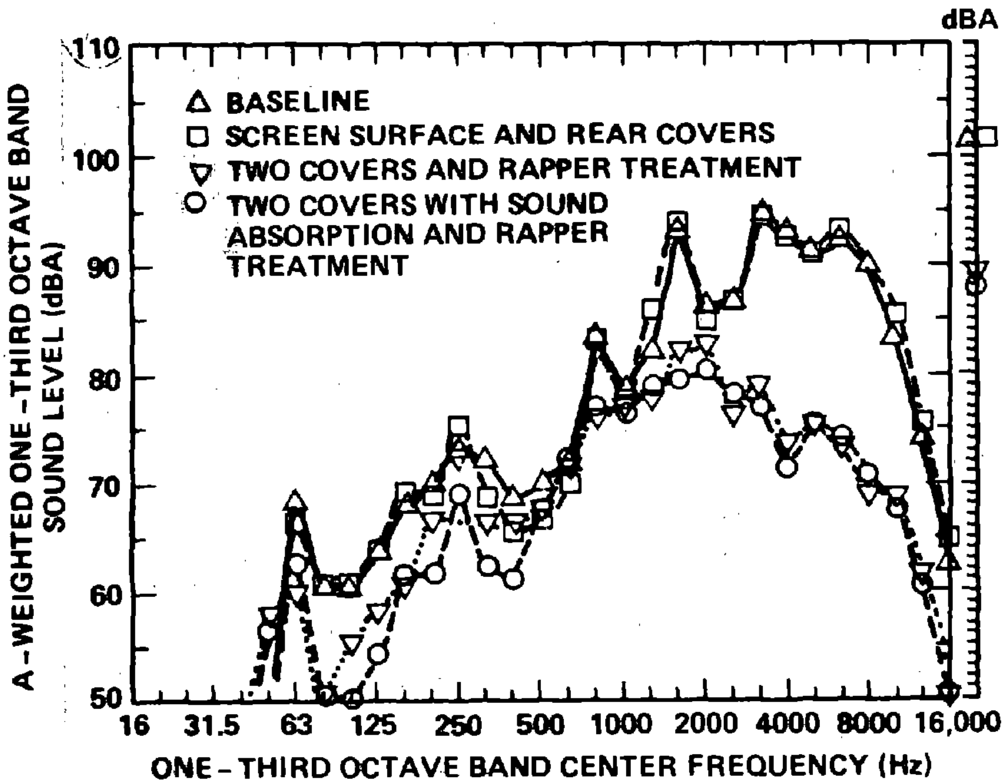


FIG. 3.28. SPL'S ON SECONDARY LEVEL WITH NOISE CONTROL TREATMENTS INSTALLED.

In summary, the selected treatments demonstrated a noise reduction of 11 to 13.5 dBA depending upon the measurement position. The fine screens mechanic had a 8-hr time-weighted average level of 97 dBA, which meant that a reduction of 7 dBA was required to bring him into compliance. Clearly, the required reduction was obtained. In fact, the sound absorption on the underside of the covers is not a necessary part of the treatment, since the 7 dBA noise reduction requirement is easily fulfilled by the remainder of the treatments.

3.4.2 Durability, maintenance and cost

Because of the depressed condition of the steel industry throughout the latter half of this project, the concentrator was not operated after all the demonstration treatments were installed. In terms of durability, some partial results were available. The screen surface cover and the rear hole cover were installed in March 1981, and by the end of 1982, had seen 11 months of operating service. In that time, no maintenance or repair of the covers was necessary, and when the full treatment was installed in September 1982, the two original covers were used in the evaluation of the complete package. However, due to the extended shutdown of the concentrator from April 1982 through the early part of 1983, no operating experience has been obtained on the rapper treatment.

Potential maintenance requirements are periodic cleaning of the sound-absorption-treatment's perforated metal cover and the sintered-bronze element in the rapper exhaust muffler. Should the holes in the perforated metal become clogged with taconite "mud," they would need to be washed out with the hose that the fine screens mechanic uses for screen maintenance. The sintered-bronze element should be cleaned and replaced or, if damaged, the whole muffler should be replaced. The rubber treatments are expected to last several years, but obviously will not have the service life of the steel components. Nitrile butadiene (Buna-N) rubber has a slightly superior oil/grease resistance than neoprene so that, for extended life, a Buna-N bellows should be used. However, the neoprene bellows service life is considered acceptable for this application.

The quantities and costs of the various materials used in the noise control treatment are detailed in Table 3.3. The total cost was $112.96; this includes the cost of making a new splash guard and the addition of the Tuffskin sound absorption treatment to it. In some cases, the minimum order requirements for certain items resulted in pieces of greater size than required for treating one demonstration fine screen. Therefore, the out-of-pocket expenses for quieting the one screen were greater than the $112.96 figure; however, each concentrator has many fine screens and the individual cost will be on the order of $113 per unit when many screens are treated at the same time. Estimates of the time required to fabricate the covers and rapper strikeplate cover, and then to install the full treatment on a screen range from 2 to 4 days. The higher figure represents the fabrication and installation time for the first screen treated by a mine and the smaller figure represents a more realistic estimate after mine personnel have gained some experience with the treatment.

TABLE 3.3. COSTS OF MATERIALS USED IN TREATMENT OF RAPIFINE DSM SCREEN.

Material Description	Size	Quantity Size	Unit Cost	Total Cost
• Steel sheet	16 ga sheet steel	10-3/4 sq ft	(E) 1.00/sqft	$ 10.75
• Angle iron	1 × 1 × 1/8 in.	23 ft	1.10/ft	25.30
• Handle	–	3	(E) 1.00	3.00
• Neoprene sheet for flap seals	1/4 in.; 40 durometer	1 sq ft	6.07/sqft	6.07
• Steel backup bar for neoprene seals	1 in. × 1/8 in.	5-1/2 ft	0.20/ft	1.10
• Misc nuts/bolts/washers for handles and backup bars	–	11 sets	(E) 0.20 ea.	2.20
• Sound absorption for covers	Claremont Co. Tuffskin Type 1613 2 in. thick	10 sq ft	2.34/sq ft	23.40
• Perforated aluminum	20 ga; 40% open	11 sq ft	1.50/sq ft	16.50
• Bolts	1/4 – 20 × 3 in.	13	0.20 ea.	2.60
• Stud buttons	1/4 – 20	13	0.20 ea.	2.60
• Paint	16 oz. can	1	2.75 ea.	2.75
• Angle iron	2 × 2 × 1/ in.	22 in.	1.20/ft	2.20
• Pipe insulation	Closed cell foam 5/8 in. ID × 1/2 in. wall thickness	5 ft	0.25/ft	1.25
• Step bushings	Modified Atlantic No. 2406 grommets 1-1/4 in. × 1-1/8 in.	2 pcs	0.41 ea.	.82
• Neoprene rubber sheet	1/4 in.; 40 durometer	0.25 sq ft	6.07/sq ft	1.52
• Steel sheet	1/4 in. sheet steel	0.25 sq ft	3.00/sq ft	.75
• Neoprene bellows	Miner Rubber Type 2FD-1326, 50 durometer double-dip	1	5.15 ea.	5.15
• Hose clamps	2 in.	2	.50 ea.	1.00
• Exhaust muffler	C.W. Morris Quietaire MM-2	1	3.75 ea.	3.75
• Extension pipe	1/4 in. NPT; 12 in. long	1	(E) .25	.25
Total				$112.96

*1981 costs.

(E) = estimated.

4. CRUSHER SCREENS

4.1 General Description

Vibrating screens are used in most ore cleaning and concentrating operations to separate ore according to particle size. Such separation is necessary because processing equipment is generally designed to operate most efficiently with a limited size range of ore.

As shown in Fig. 4.1, vibrating screens consist of one or more screen decks (either perforated sheet steel or woven wire) mounted on resilient supports (either steel springs or elastomer bushings). The oscillating forces are usually generated by an

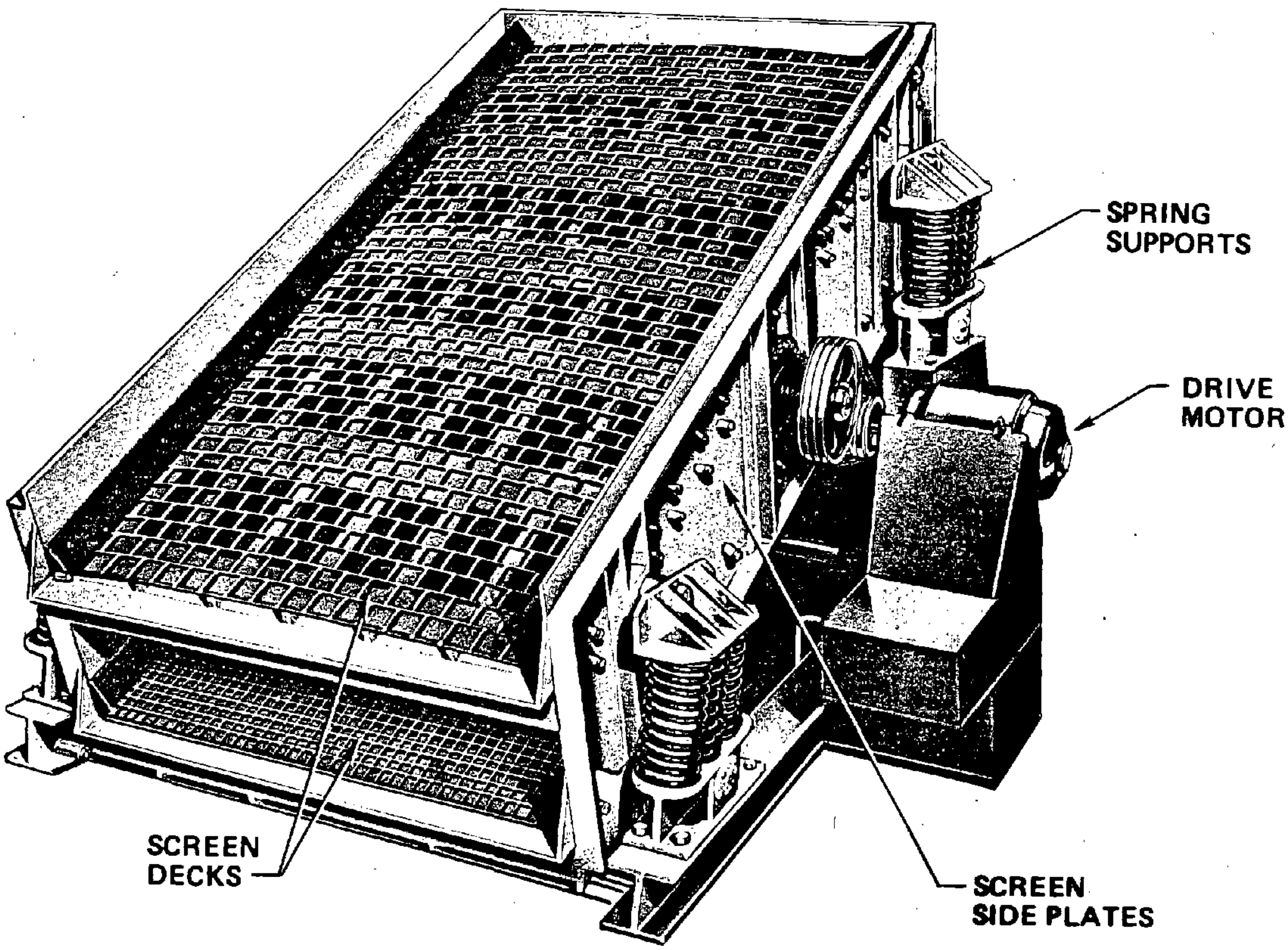


FIG. 4.1. BASIC COMPONENTS OF A VIBRATING SCREEN.

eccentric-weight shaft, driven by an electric motor through v-belts. As ore is fed to the top screen deck, the oscillation stratifies the material, allowing the small material to fall through the perforations. The oscillations also assist in transporting the oversized material along the top deck to a chute located at the discharge end of the top deck. The smallest material that passes through all of the screen decks is collected on a "bottom pan," which feeds a separate discharge chute.

One common application for vibrating screens in taconite processing plants is to screen the output of the secondary or tertiary crushers, as shown in Fig. 4.2. Here the screen following the secondary crusher removes the ore that has been sufficiently crushed from the feed to the tertiary crusher. This prevents unncessary crushing and enables the tertiary crusher to operate more efficiently. The screen following the tertiary crusher separates any oversized material and feeds it back to the tertiary crusher. The result is relatively efficient crushing without an excessive amount of "overs" or fines in the product.

Figure 4.3 shows a typical crusher screen used in taconite processing plants. The 6-ft × 14-ft screen is a Tyler Ty-Rock®, which uses a v-belt-driven, double-eccentric shaft to generate a circular vibratory motion. Both the screen box and eccentric shaft are mounted on "rubber-in-shear" mounts that are fastened to the stationary base frame.

This screen can be fitted with up to three screen decks. The upper deck is normally either a cast steel grizzly (such as that shown in Fig. 4.4) or a steel punch plate. The lower deck is usually woven wire (see Fig. 4.5) or a steel rod deck. While such screens can be mounted either horizontally or on an incline, inclined mounting is more common in crusher screens, with the angle of inclination typically between 20° and 25°.

Crusher screens are also often supplied with dust covers, of which there are three basic types. The first, shown in Fig. 4.3,

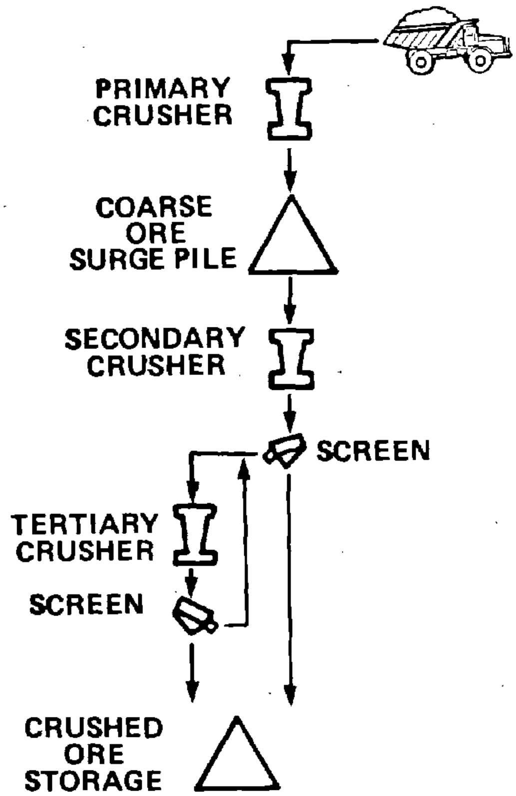


FIG. 4.2. TYPICAL CRUSHER CIRCUIT USING VIBRATING SCREENS.

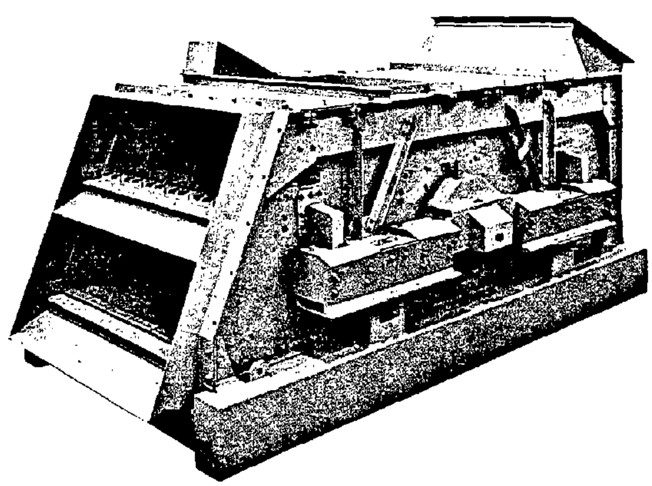


FIG. 4.3. TYPICAL CRUSHER SIZING SCREEN.

FIG. 4.4. CRUSHER SIZING SCREEN, GRIZZLY DECKS.

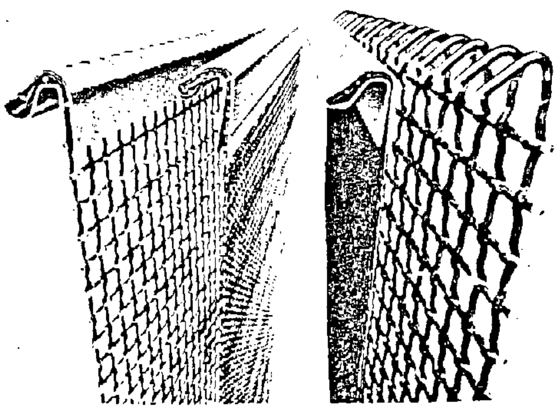


FIG. 4.5. WOVEN WIRE TYPE SCREEN DECKS.

uses a stationary steel cover for the top of the screen box and the screen side plates for the enclosure sides. The cover is supported by a steel frame mounted on the stationary base frame, and is fitted with flexible seals around the vibrating screen. The infeed and discharge chutes are also fitted with covers. Some screen manufacturers also offer a second type of enclosure, which fits over the entire screen mechanism. It is normally stationary, and fabricated from sheet steel. The third type, shown in Fig. 4.6, consists of rubber flaps either mounted

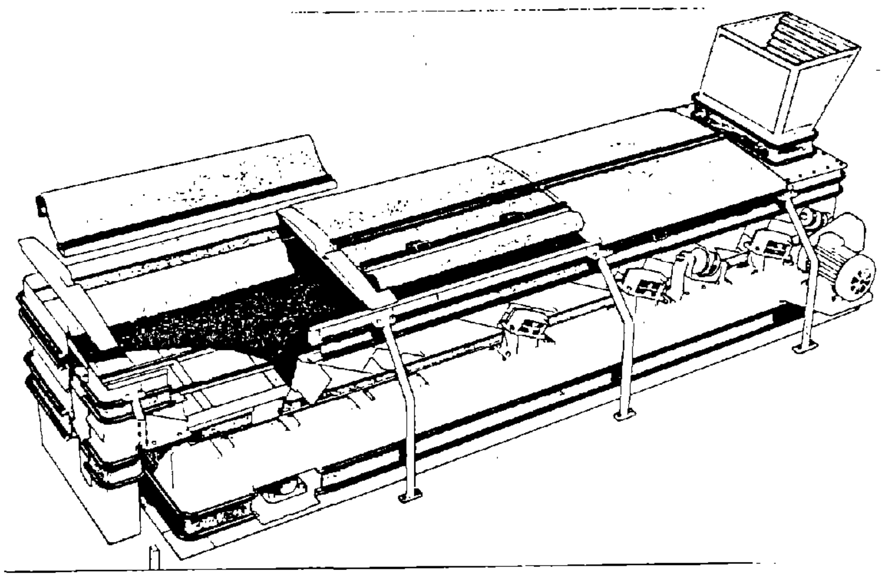


FIG. 4.6. FLEXIBLE DUST SEALS FOR A VIBRATING SCREEN (SOURCE: TRELLEBORG).

directly to the vibrating screen or to a stationary frame (which also requires flexible seals on the screen box). In either case the screen sideplates form the sides of the enclosure.

4.2 Noise Levels

Crusher sizing screens can easily generate sound levels in excess of 90 dBA, and often in excess of 100 dBA, at distances of 3 ft in typical plant environments. As a result of these high levels and the typical exposure times of plant personnel, crusher sizing screens were identified in Phase I of this research effort (U.S. Bureau of Mines Contract No. JO366055) as one of the major noise sources in the crushing areas of taconite processing plants. Although the crushers themselves were also identified as a major noise source, that issue is addressed in a separate report [13].

Figures 4.7, 4.8, and 4.9 show sound level data measured alongside crusher screens in three different plants visited during this project. The spectra are basically broadband, with the

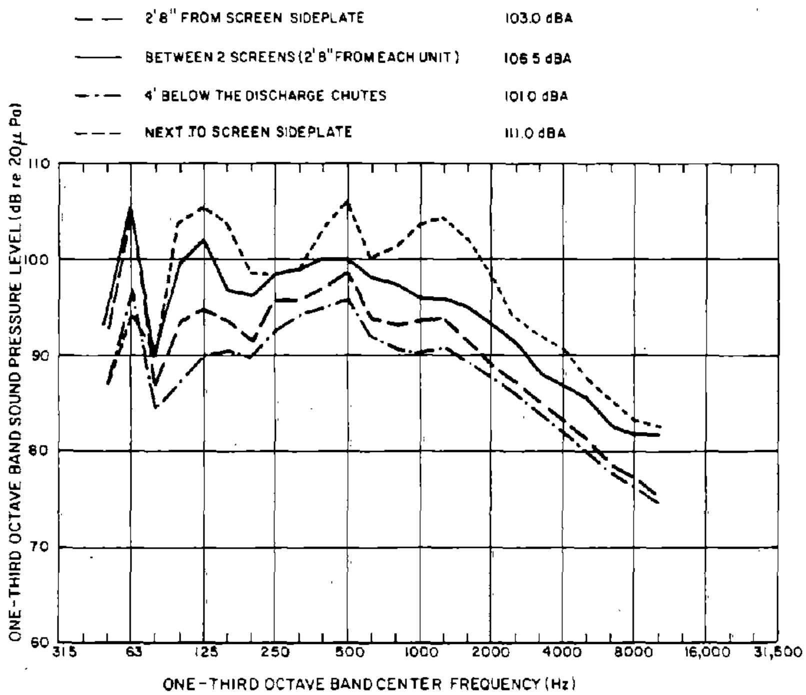


FIG. 4.7. ONE-THIRD OCTAVE BAND SPECTRA OF CRUSHER SCREEN NOISE - PLANT B.

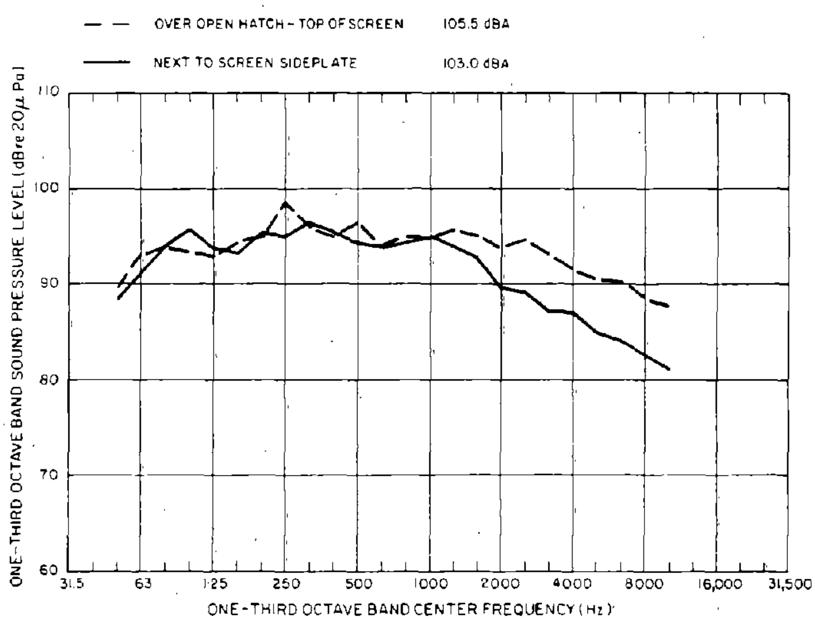


FIG. 4.8. ONE-THIRD OCTAVE BAND SPECTRA OF CRUSHER SCREEN NOISE - PLANT C.

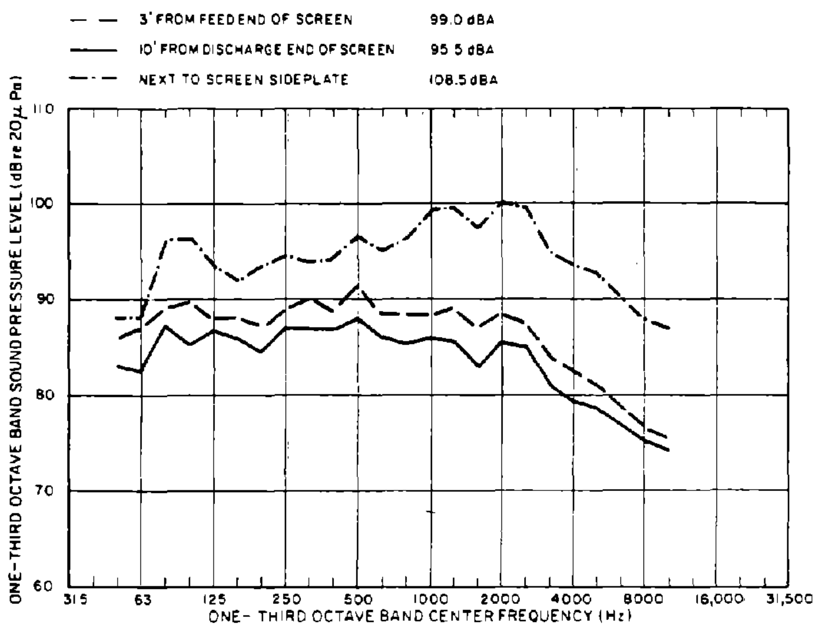


FIG. 4.9. ONE-THIRD OCTAVE BAND SPECTRA OF CRUSHER SCREEN NOISE - PLANT D.

exception of a peak in the 63-Hz band measured near the screens at Plant B. This peak resulted from some nearby electromagnetic feeders operating at 60 Hz.

While all of these screens tend to be broadband sources, the overall sound level near the screens does vary, as can be seen by comparing the sound levels measured next to the sideplates in Figs. 4.7, 4.8 and 4.9. The difference between these measurements, 8 dBA, occurs because of differences in screen loading and equipment layout, as well as in the basic screen noise.

4.3 Noise Source Diagnosis

There are two primary noise-generating mechanisms at work on vibrating screens such as crusher sizing screens: drive noise and flow noise. Drive noise is the inherent noise of the screen and is a function of the specific design of the screen structure, drive mechanism, and vibration isolation mounts. It can be either airborne, i.e., radiated directly from the motors, bearings, and gears (where applicable), or structureborne, i.e., converting the drive-induced vibrations in the screen decks and sideplates into sound. The sound of a screen operating alone, with no ore flowing over it, will sometimes provide a good approximation of the drive noise. However, it should be noted that the drive noise may change somewhat when the screen is loaded with ore.

Flow noise is the result of ore impacting itself as well as the screen deck and sideplates. It can be airborne i.e., radiated directly from the point of impact, or structureborne i.e., impacts result in vibration of the screen decks or sideplates. which are then radiated as noise.

Significant reduction of the airborne noise can be achieved by equipping the screen(s) with a dust cover.

Noise from the screen sideplates was evaluated by using data obtained through simultaneous measurements of screenplate vibration and "close-in" sound levels. The vibration data were used to calculate the sound pressure radiated by the screen sideplates using the following formula:

$$SPL_S = VL + 10 \log \sigma \tag{4.1}$$

where SPL_S is the sound pressure level at the surface (in dB re 20 μPa), VL is the velocity level (in dB re 3.3×10^{-8} ft/sec), and σ is the radiation efficiency of the sideplate. The velocity level was obtained from acceleration measurements made on the screen sideplates, which were integrated using the following formula:

$$VL = AL - 20 \log f + 150 \tag{4.2}$$

where AL is the acceleration level [in dB re 32.2 ft/sec^2 (1g)] and f is the frequency (in Hz).

The radiation efficiency of the screen sideplates was obtained using Ref. 12. Figure 4.10 shows the radiation efficiency calculated for one of the screen sideplates studied.

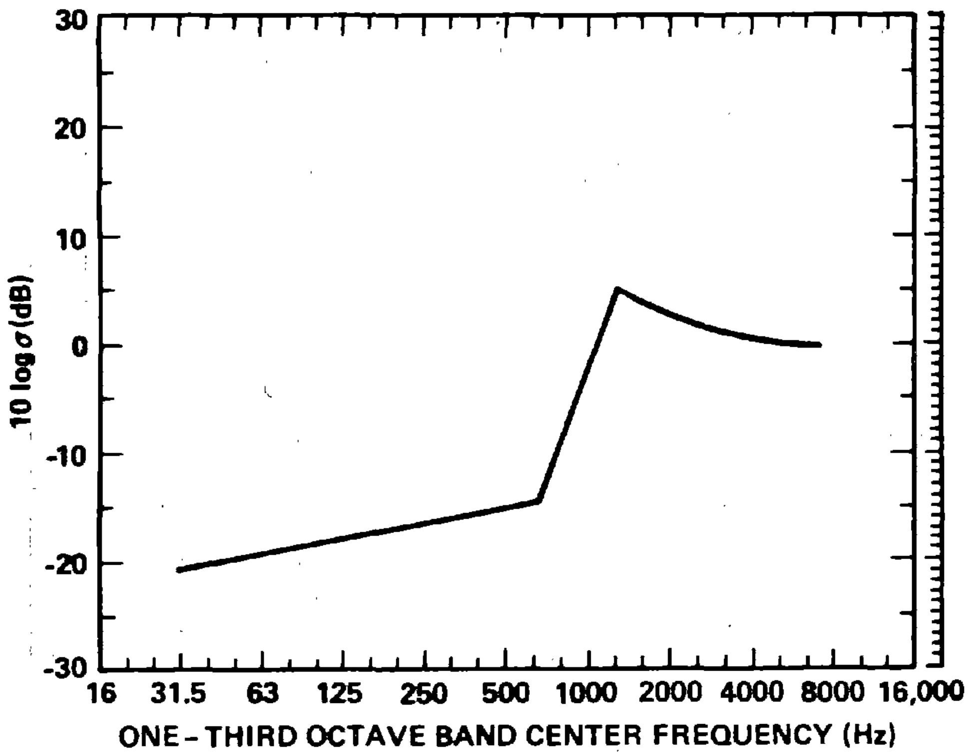


FIG. 4.10. RADIATION EFFICIENCY OF SCREEN SIDEPLATES.

Figures 4.11 through 4.15 are comparisons of the calculated sound pressure levels of the screen sideplates and the sound pressure levels measured close-in to the screen sideplates. These figures show that the agreement between the calculated and measured sound pressure levels is quite good. Furthermore, the magnitude of the levels indicates that radiation of the screen sideplates (due to the drive vibrations and ore impacts) is a major noise source on screens with dust enclosures (see Figs. 4.7, 4.8 and 4.9).

For screens without dust covers, the direct airborne sound from the ore impacts (i.e., airborne flow noise) can also be a noise problem. Figure 4.14 illustrates the effectiveness of the

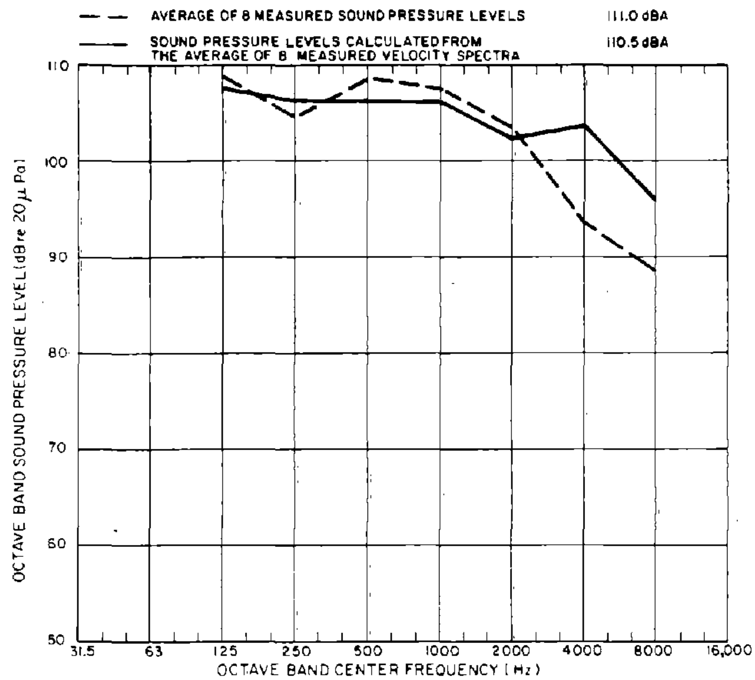


FIG. 4.11. OCTAVE BAND SOUND PRESSURE LEVEL AT THE SURFACE OF A TERTIARY CRUSHER SCREEN SIDEPLATE - PLANT B.

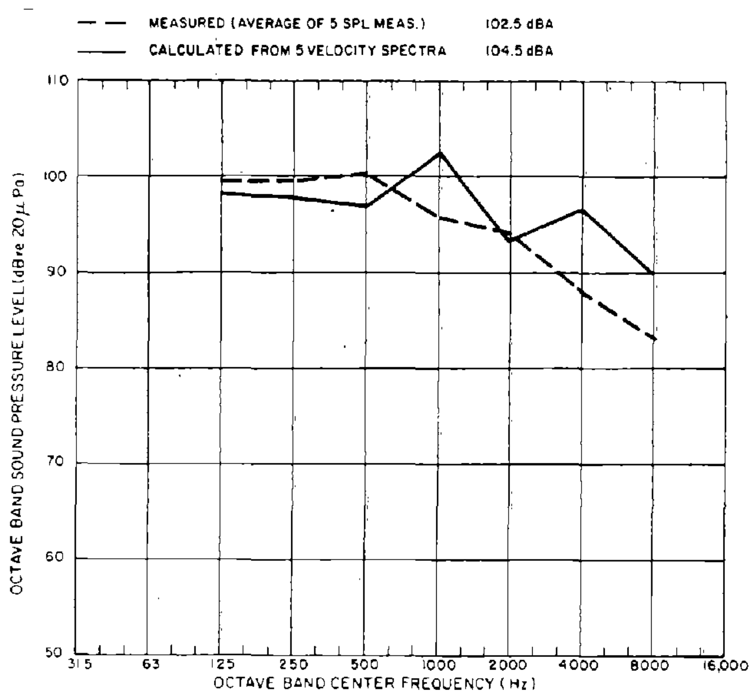


FIG. 4.12 OCTAVE BAND SOUND PRESSURE LEVEL AT THE SURFACE OF A TERTIARY CRUSHER SCREEN SIDEPLATE - PLANT C.

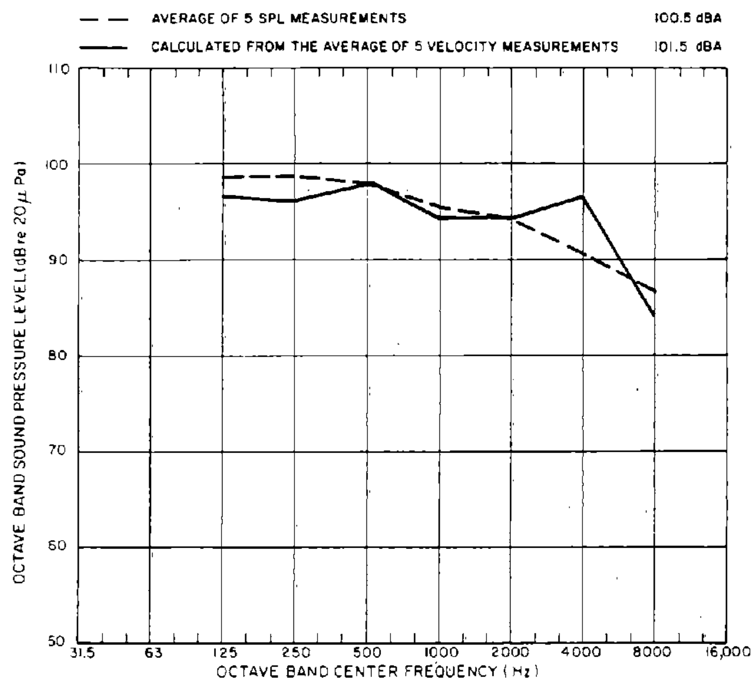


FIG. 4.13. OCTAVE BAND SOUND PRESSURE LEVEL AT THE SURFACE OF A TERTIARY CRUSHER SCREEN SIDEPLATE - PLANT D.

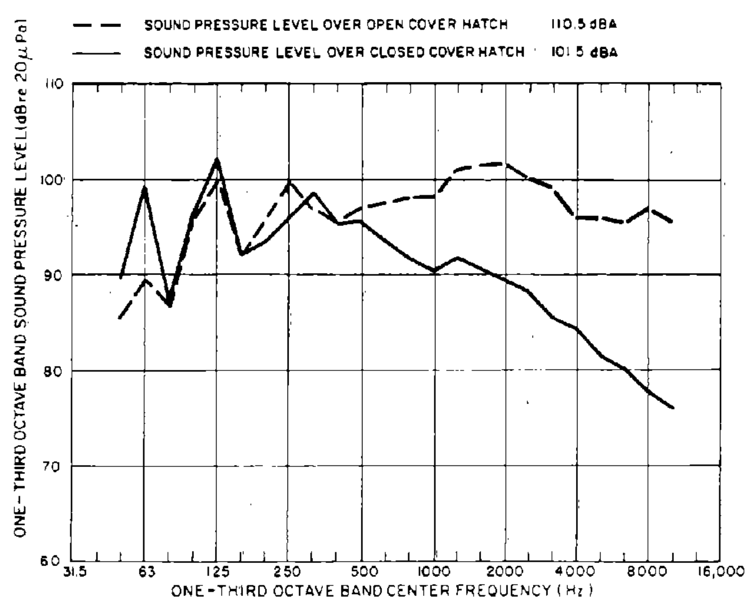


FIG. 4.14. SOUND PRESSURE LEVEL ON TOP OF CRUSHER SCREEN WITH DUST COVERS OPEN AND CLOSED.

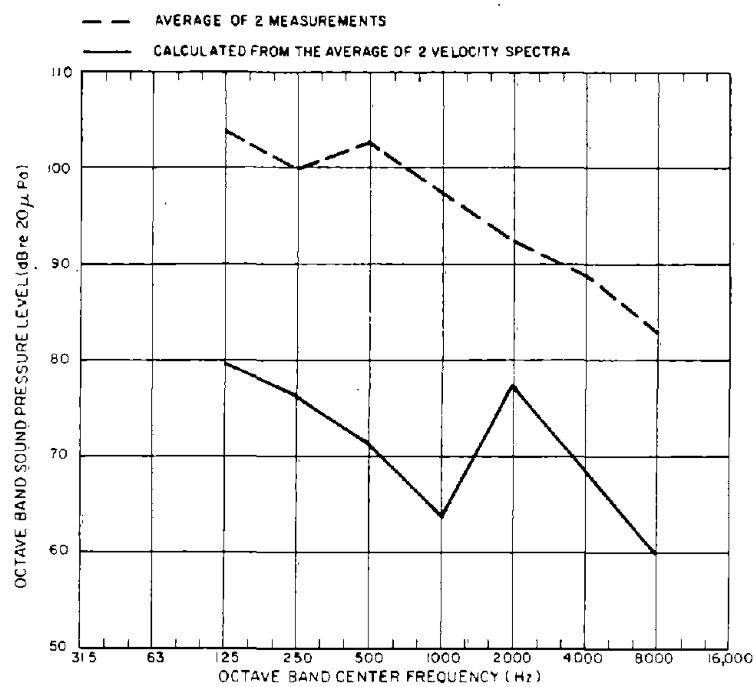


FIG. 4.15. COMPARISON BETWEEN CALCULATED AND MEASURED VALUES OF SOUND PRESSURE LEVEL AT THE SURFACE OF DUST COVER HATCH.

covers by comparing the results of a sound measurement made over a crusher screen with the covers open and closed. Clearly, the covers reduced the radiated sound energy (particularly at frequencies above 1 kHz).

Since the dust covers on these screens were stationary, they do not vibrate as strongly as the screen sideplates. To verify that the structureborne noise from the covers is not a major noise mechanism, simultaneous measurements were made of the acceleration levels and close-in sound levels for a screen cover in Plant B. Using the procedure described previously the structureborne radiation from the cover was calculated and compared to the measured levels. The results, shown in Fig. 4.15, indicate that the calculated levels are at least 15 dB below the measured airborne levels, verifying that the structureborne radiation from these stationary covers is not a major problem.

4.4 Noise Control Methods

Noise Reduction Requirements

Noise reduction goals were established for various plant areas in Phase I of this study. For areas around crusher screens a noise reduction of 8 to 9 decibels was considered necessary to meet current federal noise regulations.

Noise Reduction Alternatives

Because structureborne radiation from the screen sideplates is a major noise source, particularly for screens with dust covers, noise reduction effort should initially be directed to reducing the vibration of the sideplates. There are several methods of accomplishing this, including using a resilient screen deck, damping the sideplates, and installing vibration isolation between the screen deck and supporting cross-members.

While treatment of the sideplates is of primary importance, it will have only limited effectiveness if there are either no dust covers or if the seals around the dust covers are faulty. In the latter case, careful repair and maintenance can usually rectify the situation.

Screen decks with resilient top surfaces have been on the market for a number of years and have been used in a variety of mineral processing industries both in the United States and abroad. The resilient (elastomeric) top surface, either an abrasion-resistant grade of natural or synthetic rubber or urethane, reduces the impact of ore on the screen deck, thereby reducing the amount of vibration transfered into the screen's structure. Resilient screen decks are manufactured in a variety of configurations ranging from all-elastomer (rubber or urethane) to "clad" decks, where the elastomer is bonded to the top surface of a perforated steel or plastic support plate. For these clad decks, the elastomer can be vulcanized, cold-bonded, or, in the case of one rubber-over-plastic deck, fused to the support plate. Some of the so-called all-elastomer decks are also available with steel cables imbedded in them for strength and to allow tension adjustment.

A recent study performed for the U.S. Bureau of Mines by Allis-Chalmers [13] has documented the acoustical and operational performance of several types of resilient screen decks. The measurements were made on an Allis-Chalmers inclined screen at its test facilities in Wisconsin. While the measurements were made on open screens with dolomite, granite, and coal, the test results should also be representative of those obtainable with taconite. Furthermore, the noise reductions measured during these tests, typically 4 to 7 dBA, are consistent with data obtained from previous USBM research in operating coal preparation plants [14,15]. These in-plant tests have also shown that the full noise reduction potential of resilient screens can only be achieved if all of the locations of impact noise, such as the infeed and discharge chutes, are also treated.

Although resilient screen decks reduce the impact forces of ore thereby reducing vibration and noise, there are some operational tradeoffs to be considered. First is screening efficiency. While several of the screen decks tested by Allis-Chalmers actually resulted in an increase in screening efficiency while reducing the noise, most of the resilient decks tended to reduce efficiency along with the noise (typically by about 3% to 4%). There is also the question of cost. Initial material cost for resilient decking can run from 2 to 4 times (or more in some cases) than that of steel. The actual cost, however, must also take into account total service life, maintenance costs, etc. While the initial cost of resilient decks is higher than that of steel, deck manufacturers state that since the service life of the resilient decks is longer, the higher capital costs are off-set. In addition, screen replacement is often easier with the resilient decks, because of their reduced weight.

While early USBM studies of screen decks in coal preparation plants [14] uncovered durability problems (e.g., delamination of the elastomer layer) with some of the early elastomer clad-to-steel decks, more recent in-plant tests [15] have indicated that the service life of these decks has been improved. Now, some of the new urethane cast-to-steel decks have lifespans several times longer than those of the original steel punched plate, which typically lasted 6 months. In fact, in one test, the urethane cast-to-steel decks were removed after over 1.5 million tons of coal and 15 months of service because of cracks in the supporting steel punched plate rather than excessive wear of the urethane.

An alternative to using resilient screen decks is to vibration isolate the standard steel deck from the supporting cross-members. Although such isolators are not commercially available at the present time, they should be relatively simple to fabricate and have the advantage of not necessarily being directly exposed to the ore.

As part of the previously mentioned test program, Allis-Chalmers also investigated the effect of sideplate damping on the noise generation. In these tests various types of constrained layer damping were applied to the inside of an A-C low-head screen, covering approximately 66% of the sideplate surface. The tests indicated that the damping treatments increased the inherent loss factor of the sideplates from a baseline of 0.06 to as much as 0.12, resulting in a reduction of approximately 5 dB in the A-weighted noise level. The most effective of the treatment configurations was life-tested for 5000 hours, and individual treatments were tested for 25,000 hours in a salt-spray with no apparent degradation.

Finally, it should be mentioned that the noise in screening areas can often be attributed as much to ore impacting the infeed and discharge chutes, bottom pan, etc. as from the screening operation. Therefore, the treatments described above will only be effective to the extent that these other sources are also reduced.

5. ADMINISTRATIVE CONTROLS

Since the engineering noise controls considered in this program may be costly and will require field testing prior to general acceptance by the mining industry, some consideration was given to administrative controls. A review of previous employee noise exposure data was initially conducted, and the components of the total noise dose determined. Then, in conjunction with mine management, an assessment of the various administrative control methods was made.

5.1 Noise Exposures

Employee noise exposure data were obtained from four taconite mines in which the individual work scenarios and partial noise doses were measured. The mines are considered representative of the industry because they include one large semi-autogenous plant, two mid-sized plants (one with rod mills and one with autogenous mills), and one small autogenous plant. The total number of personnel employed in these four mines represented approximately 14% of the total industry population in the United States.

Since this program focused on noise control for processing equipment, only those employees who worked in the processing plant were considered. Mine personnel, office staff, and management were excluded from the present analysis. In the data base, approximately 55% of the employees were classified as plant personnel. Table 5.1 shows that the percentage of plant workers with a daily noise dose (DND) greater than unity was 32%. The different areas of the plant were divided up into sections (i.e., concentrator, pelletizer, shop areas, etc.); approximately 45% of plant personnel work in the concentrator section. The last column in Table 5.1 shows that approximately 27% of all plant personnel have a DND > 1.0, and spend part or all of their time in the concentrator section.

TABLE 5.1. EMPLOYEE NOISE EXPOSURES.

	% of Plant Employees With DND > 1.0	% Of Plant Employees Who Spend Part of Work Day in Concentrator Section	% of Plant Employees With DND > 1.0 and Who Work in Concentrator Section
Plant 1	39	43	32
Plant 2	33	40	28
Plant 3	35	47	26
Plant 4	20	51	20
Average	32	45	27

These data further illustrate the fact, identified in Phase I of this program, that the concentrator section provides the majority of the noise exposures.

A breakdown of the various job descriptions in the concentrator section is presented in Table 5.2. The workers were divided into two groups, operations and maintenance. Obviously, individual taconite plants may have slightly different job categories or numbers of employees in each group, but Table 5.2 is intended as an illustration of a typical plant. Also, the noise doses will vary from plant to plant depending on the equipment, etc. The jobs with the highest noise doses are typically those near the grinding mills, i.e., primary attendant, maintenance mechanics, and sweepers. Sweepers are cleanup personnel, who hose down areas around the mill and in the basement. Maintenance mechanics not only have high noise exposures, but also represent a large percentage of the workforce. Typical work scenarios and partial noise doses from the data base are shown in Table 5.3 for three job descriptions.

The maintenance mechanics get their largest noise dose from mill relining work. Mill relining is a noisy operation and it involves a large portion of the maintenance mechanic's work day.

TABLE 5.2. TYPICAL CONCENTRATOR SECTION NOISE EXPOSURES.

Job Description	% of Total Plant Employees	Typical Daily Noise Dose (DND)
Operations		
Control Room Operator	1	0
Primary Attendant	1	1.5 - 2.4
Secondary Attendant	1	1.3
Mill Helper	1.5	1.1
Shift Maintenance Mechanic	2.5	0.8 - 1.9
Sampler/Tester	2	1.6
Shift Laborer	3	0.6 - 2.0
Crane Operator	1	2.0 - 3.0
Sweeper	0.5	1.5 - 4.7
Foreman	1	0.9
Instrumentation Electricians	1.5	0.6
Others	2	0.5 - 1.3
Maintenance		
Maintenance Mechanic	10	1.3 - 2.0
Welder*	1	1.0 - 2.2
General Laborer	3	1.0
Carpenter and Helpers	0.5	0.5
Painters	0.5	0.6
Oilers	1	1.5 - 2.2
Plant Electrician	2	0.5 -1.6
Heating & Vent. Electrician	1	1.5

*Only includes welders assigned to concentrator.

TABLE 5.3. TYPICAL CONCENTRATOR EMPLOYEE NOISE EXPOSURES.

I. Maintenance Mechanic

	Min/Day	Partial Noise Dose
Concentrator, general	50	0.3
Concentrator pump bay	40	0.15
Primary, lining	90	0.8
Concentrator upper levels	70	0.25
Secondary mills	50	0.15
Booster station	20	0.1
Shops	60	0.05
Other	100	0.05
Total	480	1.85

II. Primary Attendant

	Min/Day	Partial Noise Dose
Mill floor	70	0.6
Upper levels	120	0.5
Cobber and conveyors	100	0.55
Basement	30	0.15
General	110	0.5
Other	50	0
Total	480	2.3

III. Welder

	Min/Day	Partial Noise Dose
Primary, lining	40	0.4
Upper mill levels	60	0.25
General concentrator	70	0.4
Shops	80	0.1
General, non-concentrator	40	0.05
Pellet plant	130	0.3
Other	60	0
Total	480	1.5

In contrast, the primary attendant receives the majority of his dose from four areas around the mills, and is subjected to more general plant noise.

5.2 Potential Administrative Noise Control

Noise exposure data presented earlier illustrated that in the concentrator section, most of the personnel are not confined to just one area or operator position. Personnel work and move around to all areas of the plant. Consequently, the concept of noise shelters or sound booths in the plant is not likely to be very beneficial. Some plants already have an acoustically quiet shift foreman's office located on the mill floor. This area is used by personnel for answering the paging system, obtaining work instructions, etc. There are not many instances when workers could perform their duties while protected by a sound booth.

Changing the work scenario so that the exposure time to high noise levels is minimized would, in most instances, be impractical. For example, it has already been shown that the primary attendant is exposed equally from four areas. Minimizing time in any one of these areas, if it were possible, would have only a small effect on the total noise exposure. Mill relining is one area that was identified as contributing a large portion of the daily noise dose for several categories of workers. However, changing liners has to be done at specific time intervals to ensure smooth production, and little change in the amount of time spent in relining can be foreseen. During the relining process the pneumatic bolter, used to tighten liner bolts, generates high impact noise levels. Maintenance mechanics not directly involved in rebolting should try to keep their distance from the bolter and minimize their noise exposure.

The concept of job swapping or using additional personnel is another method used in administrative controls. In most plants, other maintenance mechanics could theoretically assist with the relining process, thus helping to reduce the high noise exposure. However, in practice, this does not appear to be a feasible solution. First, within the job description of maintenance mechanic there are several categories or levels of employees. These categories are based on seniority and job performance. Consequently, swapping job functions among these different categories would not be possible under the current union contracts. In addition, heavy work is involved in the relining; this requires handling pieces of material that weigh up to 250 lb. Obviously, this type of work would not be suitable for all employees. The hiring of additional personnel for the high noise exposure areas would help minimize noise dosage, but in general this is not a feasible solution, even in good economic times.

Hearing conservation programs and hearing protectors have been used in taconite plants for many years. Most plants have signs posted, warning of the noise hazard and describing the required protective action that must be taken. Hearing protection devices (HPD) are already worn by employees, especially those working on mill relining. A large variety of industrial hearing protectors, if used correctly, have Noise Reduction Ratings (NRR) that range from 15 to 25 dB. However, recent studies [16], have shown that in real-world situations, the HPD's are not being used correctly by the majority of wearers. The typical noise reduction achieved in the real world is suggested to be 10 dB less than the stated NRR. In addition, there are also the problems of enforcing the use of HPD's, and of ensuring cleanliness of the devices in a plant environment that is very dusty and dirty.

6. CONCLUSIONS

This section summarizes the work performed on the three pieces of equipment studied during this research project. Conclusions and recommendations for future research are also provided.

6.1 Grinding Mills

Extensive measurements were obtained on semi-autogenous, autogenous, and rod mills. These data were used to determine how the process parameters affect the noise level produced by grinding mills. These findings are presented in Sec. 2.2.3 of this report.

Novel diagnostic measurements were made to measure the noise and vibration produced by grinding mills. Special diagnostic techniques had to be used, because grinding mills are always arranged in multiple units, and taconite processing plants present a very reverberant sound field in which to make measurements. In addition, production requirements preclude measuring one grinding mill by itself, thus noise measurements usually include the contribution from other mills. A multi-channel, time-multiplexing, radio-linked data acquisition system was designed to obtain acceleration measurements on a rotating mill during actual production conditions. These acceleration data were used to conclude that, in the case of semi-autogenous and rod mills, high levels of impact noise are produced on only a small area of the falling side. This area is where the grinding media and the charge drop onto the mill shell. These data were further used to determine the noise reduction potential of installing a partial barrier around a mill.

Several noise control options were discussed with mill manufacturers and it was concluded that the most promising techniques were partial barriers for semi-autogenous and rod mills, and rubber liners for autogenous mills.

A partial barrier was built around the falling side and both end caps of a semi-autogenous mill. A noise reduction of 6 dBA was achieved on the falling side of the mill (where the levels were highest). It was concluded that partial barriers are a useful technique to reduce the noise levels from semi-autogenous and rod mills, where high levels of impact noise are involved. In the case of autogenous mills, the levels of impact noise are lower and the concept of rubber liners was investigated. Two plants were visited where at least one mill was fitted with rubber liners and noise level measurements indicated a noise reduction of at least 4 dBA due to the rubber liners (compared to the standard steel liners). Because of the high background noise levels, it was not possible to measure the full performance of the rubber liners. In terms of wear and durability, the results were mixed. At one mine, the rubber liners looked very promising in terms of durability, but at the other mine the results were disappointing. We recommend that additional research should be conducted into the wear life of rubber mill liners with different types of taconite ore.

This research project also determined that at some plants, noise exposures of mill personnel can be influenced by mill drive equipment and grinding circuit screening equipment. Therefore, the noise of the drive motor and the gearbox should be considered along with the actual mill noise in an overall noise reduction program. In addition, some plants have vibrating screens located next to the mill discharge and these can also produce high noise levels. Treating the mill noise, but overlooking the ancilliary equipment, may not result in a significant noise level reduction in the grinding mill areas.

In summary, the trend for grinding mill noise control is to pursue the use of rubber liners for autogenous mills and partial barriers for semi-autogenous and rod mills.

6.2 Fine Screens

Noise and vibration measurements were acquired on Dorr-Oliver Rapifine DSM screens and it was concluded that the major source of noise for the fine screens mechanic was the high level of vibration imparted to the screen surface by the pneumatic impact rapper. Although the rapper impact frequency was on the order of 50 Hz, the analysis of the equipment noise showed that it was high-frequency dominated. A low-pass mechanical filter, consisting of a rubber block sandwiched between steel surfaces, was designed and tested. The filter allowed the low frequency impact vibration to pass with virtually no attenuation, but reduced the high frequency vibration that is responsible for the occupational noise problem by 10 to 20 dB. However, the mechanical filter increased screen blinding during actual production tests and the technique was discarded. These tests determined that the high frequency vibrations in the screen surface are important to prevent blinding.

Four locations were chosen to represent the positions where the fine screen mechanic performs maintenance, and tests were conducted to rank order the various sources at each location. These measurements found that different sources control the noise level in front of a Rapifine screen, than do behind or below the screen. Noise control treatments were designed, fabricated and tested for these sources. These treatments consisted of a front screen surface cover, a rear cover, and rapper system modifications. The modifications included vibration isolation between the rapper and the counterweight arm, an exhaust muffler, a cover over the rapper strikeplate area, and a rubber bellows over the exposed part of the plunger. These noise control treatments were installed on a production fine screen and a 11 to 13.5 dBA noise reduction was measured. With regard to durability and maintenance, little operating experience was obtained on the installed treatments due to extended plant shutdowns at the cooperating mines. The cost of materials for the treatment was $113 and an estimated 2 to 4 days of mine labor were needed to fabricate and install the treatments.

These tests showed that Rapifine screens should be operated at the lowest air supply pressure that gives adequate screening performance, because the screen noise level increases as the pressure is raised. Additionally, screen surface life is reduced if operated at a high-than-needed pressure. It was also concluded that the duty ratio should be kept to a minimum, since the period of time the rapper is vibrating influences the mechanic's noise dose. Sequencing of multiple banks of fine screens should be timed so that all banks vibrate simultaneously and then this is followed by an interval of quiet when all the rappers are off. This sequencing operation is preferable to one bank vibrating, followed by another, followed by another, etc. so that there is no period of quiet. The rationale for the preferred sequencing scheme is described in Sec. 3.2.3. It was also found that the maintenance condition of fine screens affects the noise level they produce; poorly maintained screens, as expected, produced higher noise levels. The researchers concluded that efficient operation of properly maintained fine screens at the optimum pressure and duty ratio and with proper sequencing can minimize the full-shift noise exposure of plant personnel.

6.3 Crusher Screens

Sound level data measured near crusher screens in three plants are well above 90 dBA and can be greater than 100 dBA. The spectra are essentially broadband and result from a combination of drive noise and flow noise. Drive and flow noise can be either airborne or structureborne; dust covers can reduce the airborne component of the flow noise significantly. Structureborne noise radiation by the screen sideplates was found to be the major noise source on screens that have dust covers. For screens without dust covers, airborne sound from ore impacts can also be a noise problem.

Reducing the vibration level in the crusher screen side-plates was chosen as the primary approach to noise control. The use of resilient screen decks would accomplish this; other studies have shown noise reductions can range from 4 to 7 dBA. Due to the permanent shutdown of the plant at which a cooperative agreement had been signed, it was not possible to install a resilient screen deck on a taconite crusher screen during this program. Other possible noise control approaches include installing vibration isolators between the standard steel deck and the supporting cross-members, and increasing the damping of the screen sideplates so that they are less efficient noise radiators. Unfortunately, vibration isolators of the type required are not commercially available at the present time. Sideplate damping treatment has been tried by one screen manufacturer and achieved a 5 dBA noise reduction. However, this is not a commercial product either at the present time.

Thus, using resilient screen decks holds the most promise for crusher screen noise control. Additional benefits might be lower life-cycle costs, since the service life of resilient decks is claimed to be longer than steel, and easier screen replacement due to their lower weight. A potential disadvantage is a negative effect on screening efficiency.

6.4 Administrative Control of Noise Exposure

Personnel noise exposure data and job responsibilities were obtained at four plants. The components of the individual noise exposures were determined and assessments were made with mine management personnel of the various administrative control methods available.

The noise exposure data for plant personnel (pit and management personnel were excluded from study) indicated that the concentrator was the plant area that provided the majority of the noise exposure. Fines crushing and pellet plants were less important than the concentrator. The discussions with plant management revealed that concentrator personnel were divided into operation and maintenance groups and that most personnel were not confined to one area.

The administrative control techniques considered were changing the work scenarios and job swapping. Changing work scenarios is only possible in a limited way before conflict arises with production requirements. Equipment has to be checked by operations personnel on a regular basis and altering this monitoring function in a significant way would impact on smooth plant operation. Slurry density must be sampled, mills must be charged, gages must be monitored - all on a regular schedule if the plant is to function efficiently on a full-time basis. Without increased automatic process control, work scenarios are very difficult to change. Job swapping also has limitations in the unionized taconite plants. Job assignments are based on a seniority system and work rules are controlled to some extent by the union contract. To be effective, job swapping requires having multiple personnel with the same job classification so that Worker B can substitute for Worker A after he has reached the limit of his noise dose for that day. Even in the large taconite plants, this duplication of personnel is not present. In some maintenance positions (particularly mill relining which involves handling liner sections weighing 250 lb), not all personnel are able to perform those duties.

The use of hearing protection devices (HPDs) is widespread in the industry. Warning signs are commonly posted in high noise areas and HPDs are available at plant stockrooms. Use of HPDs is obviously beneficial to some extent, but their total effectiveness is unknown. Many studies have shown that due to poor fit in everyday use, the labeled performance of HPDs is not being achieved and the actual protection can be as much as 10 dB lower than the HPD rating. Compounding the problem of poor fit, using plug-type HPDs in the dirty, dusty environment of a concentrator can also lead to possible ear infections.

To summarize, using administrative controls to limit employee noise exposure in taconite plants, while theoretically possible, is limited by the requirements for efficient production and the lack of multiple personnel with the same qualifications and job classifications. Worker use of hearing protection devices, although not a permanent solution to the overexposure problem, should continue to be encouraged; however, the problems of poor fit and cleanliness should not be overlooked.

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APPENDIX A: PERMISSIBLE NOISE EXPOSURES IN TACONITE PLANTS.

The Federal Mine Safety and Health Act of 1977 (P.L. 95-164) specifies the permissible noise exposure for workers in the metal and non-metal mining industries. More specifically, the surface taconite processing plants are covered under the Code of Federal Regulations, 30 CFR 1.1, Subchapter N, Part 55, Section 55.5, which details the following noise exposures that are not to be exceeded:

Permissible Noise Exposures

Duration Per Day, Hours of Exposure	Sound Level, dBA, Slow Response
8	90
6	92
4	95
3	97
2	100
1-1/2	102
1	105
1/2	110
1/4 or less	115

No exposure shall exceed 115 dBA. Impact or impulsive noises shall not exceed 140 dB, peak sound pressure level.

Note that the regulation allows an eight-hour time-weighted average (TWA) level of 90 dBA and has a 5 dBA trade-off ratio (i.e., for every 5 dBA increase in sound level, the allowable duration is reduced by a factor of 2).

APPENDIX B: MILL VIBRATION DATA ACQUISITION SYSTEM

In order to acquire broadband vibration data on large rotating grinding mills, a design was made for a time-multiplexed, radio-linked data acquisition system. The design goals were to acquire multiple channels of vibration data, using either integral-circuit BBN accelerometers or charge-coupled devices marketed by Bruel and Kjaer, Endevco, and others.

Mill rotation periods are approximately 5 to 6 seconds. Data on each channel were to be acquired for up to one minute, and then the system incremented to the next channel with an identifying code. Rather than command the channel address, which would have required a two-way radio communication, an internal clock was incorporated that incremented channels at a pre-set interval, and simultaneously mixed in an 8-kHz tone burst with the data at the onset of a new channel. The onset of data on channel 1 was identified with a prolonged tone burst. The system components and an example of the data multiplexing are illustrated in Fig. B.1.

The multiplexing electronics were all low-power CMOS, self-contained, and battery-powered. The complete system, including multiplexor and telemetry link, was contained in a NEMA-12 enclosure, with BNC connectors for channel access. The telemetry system was a Ceta-Vega Model 77B/66 transmitter and receiver. The system was specially modified by the factory to improve vibration sensitivity and increase dynamic range. The unit is a crystal controlled FM UHF radio link operating at 169 MHz, with a line-of-sight range of 1000 ft. Radio system performance was measured at BBN and had a 65 dB dynamic range, with a flat frequency response to ±0.1 dB from 100 Hz to 10 kHz. The nonlinearity was ±0.1 dB over the measurement range and transient response was excellent. The radio transmitter was only 1 in. × 3 in. × 4 in. and was packaged inside the multiplexor box.

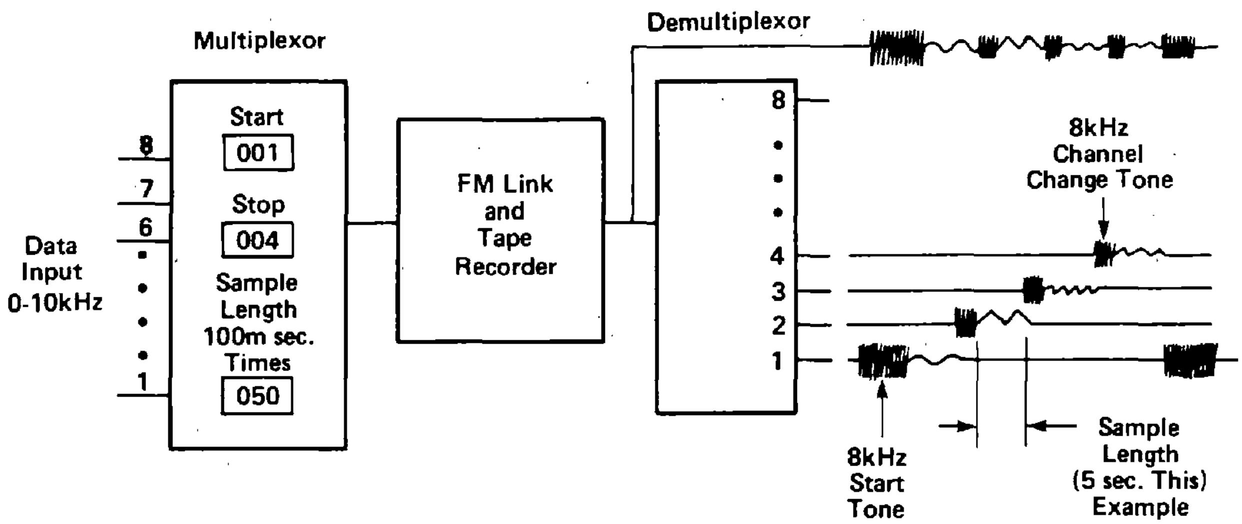


FIG. B.1. MULTIPLEXOR SYSTEM COMPONENTS.

The multiplexor schematic is illustrated in Fig. B.2. The hex inverter CD4049 chip is used to generate two clock pulses, an 8 kHz channel identification, and a long-period clock pulse used to increment channels. The eight-channel multiplexor is a CD4051 chip, in which the system inputs on the right are switched to the output according to the address on the input pins A, B, C. These addresses are provided by the binary counter CD4029 chip, which is clocked by the slow-pulse train from the CD4049.

Each time a channel increments (i.e., Q1 of CD4029 transitions low), the monostable M1, which is one half of an MM74C221, emits a low-state signal for approximately 1 second. Switch CD4066 then disconnects data, and connects the identification tone code into the output through the divider network R5 and R6. Similarly, whenever Q3 of CD4029 transitions low, indicating the onset of the channel 1, the monostable M2 is fired, which again switches the 8 kHz tone into the output, for approximately 5 seconds. The system can be configured to multiplex up to eight channels of data. The configuration shown in Fig. B.2 is for four channels.

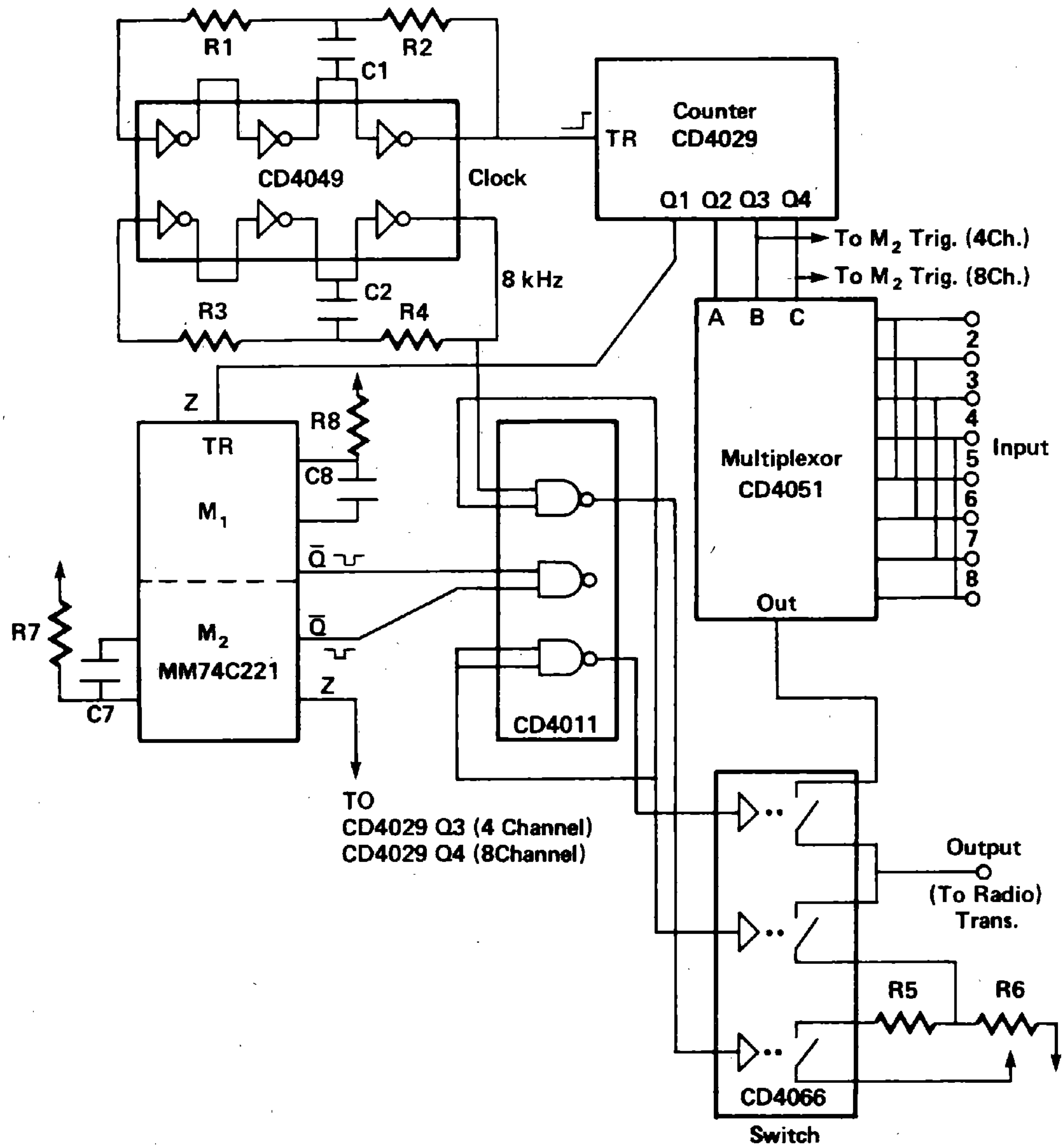


FIG. B.2. SCHEMATIC OF MULTIPLEXOR UNIT.

APPENDIX C: CALCULATION OF RELATIVE ACOUSTIC SOURCE STRENGTHS FOR MILLS

This appendix presents the details of the relative sound power level calculations, which were used in Sec. 2.2 to predict the potential mill noise reductions with a partial barrier. The concept of an equivalent radiating area was developed to utilize the measurements obtained from the directional microphone and the mill vibration data. These data indicated that relatively large and distinguishable areas of the mill had uniform nearfield sound pressure levels. Hence, the equivalent radiating area (E) is calculated as follows

$$E = 10^{RPWL/10}$$

$$RPWL = 10 \log S + RSPL$$

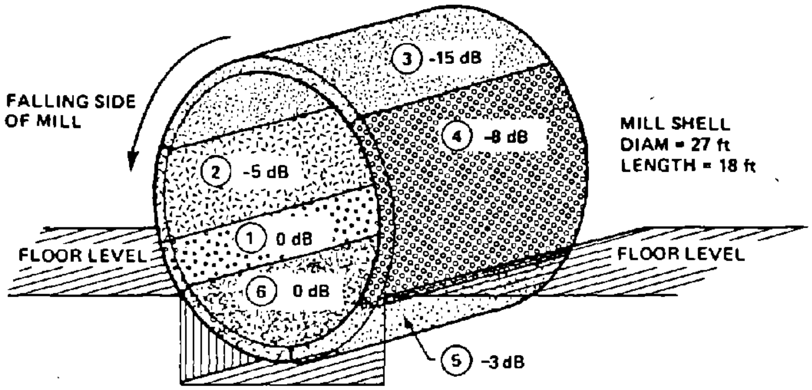
where RPWL is the relative sound power level of the area S. S is the area of the mill shell under consideration and RSPL is the nearfield sound pressure level relative to 0 dB at the falling side of the mill.

The table in Fig. C.1 shows the relative power level (RPWL) for each section of the mill calculated using each shell area and RSPL. The equivalent radiating area of the mill shell $E_{i(shell)}$ is 540.8. The same calculation is performed for the discharge and feed ends of the mill, see Fig. C.2. Therefore, the total equivalent power level of the mill is calculated

$$RPWL_{total} = 10 \log \Sigma E_i = 10 \log (540.8 + 188 + 211)$$

$$RPWL_{total} = 29.7 \text{ dB} \quad .$$

Using this relative power level one can then calculate the noise reduction that could be achieved if the noise from a



AREA		1	2	3	4	5	6	TOTAL
SHELL AREA	Sq ft	76	169.5	551.5	264	233	233	1527
RSPL	dB	0	-5	-15	-8	-3	0	
RPWL	dB	18.8	17.3	12.42	16.22	20.7	23.7	
$E_{i(SHELL)}$		75.86	53.7	17.46	41.88	117.5	234.4	540.8

FIG. C.1. MILL SHELL DIVIDED UP INTO RELATIVE POWER LEVELS FOR SPECIFIC AREAS OF THE MILL.

particular section of the mill is eliminated, i.e., through the use of a partial barrier.

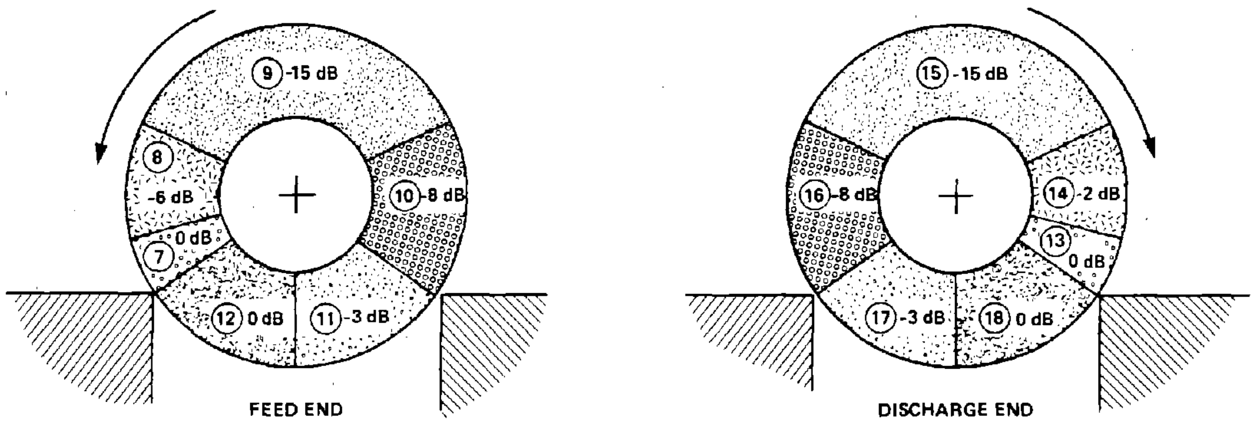
It is assumed that a barrier would provide a relative sound pressure level of -10 dB at the area covered by the barrier. The three test cases were calculated as follows.

Case 1: Barrier Over Mill Shell Sections 1 and 6 (falling side)

Set -10 dB at sections 1 and 6

$$\Sigma E = 281.6 + 188 + 211$$

therefore RPWL = 28.3 dB



AREA		7	8	9	10	11	12	TOTAL
SHELL AREA	Sq ft	27.2	60.4	196.2	94	83.1	83.1	544
RSPL	dB	0	-6	-15	-8	-3	0	
RPWL	dB	14.4	11.8	7.9	11.7	16.2	19.2	
$E_{i\,(FEED)}$		27.2	15.14	6.2	14.8	41.7	83	188

AREA		13	14	15	16	17	18	TOTAL
SHELL AREA	Sq ft	27.2	60.4	196.2	94	83.1	83.1	544
RSPL	dB	0	-2	-15	-8	-3	0	
RPWL	dB	14.4	15.8	7.9	11.7	16.2	19.2	
$E_{i\,(DISC)}$		27.2	38.02	6.2	14.8	41.7	83.1	211

FIG. C.2. FEED AND DISCHARGE ENDS DIVIDED UP INTO RELATIVE POWER LEVELS FOR SPECIFIC AREAS OF THE MILL.

i.e., approximately 1.4 dB noise reduction.

Case 2: Barrier Over Mill Shell 1 and 6, and End Caps 7, 12, 13, and 18

Set −10 dB at sections 1, 6, 7, 12, 13, and 18

$$\Sigma E = 281.8 + 88.9 + 111.7$$

therefore RPWL = 26.8 dB

i.e., approximately 2.9 dB noise reduction.

Case 3: Barrier Over Lower Third of Mill and Both End Caps

Set −10 dB at sections 1, 5, 6, 7, 11, 12, 13, 17, and 18