COAL CUTTING NOISE CONTROL

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ABSTRACT

The results of a Bureau of Mines laboratory investigation of coal cutting mechanics and noise is presented. Some basic theoretical aspects of coal cutting mechanics and noise generation are discussed, and the results of the laboratory experiments are used to formulate analytical models of the coal cutting forces and noise. It is concluded that by using deeper depths of cut, slower cutting speeds, and more efficient cutting tools, it is possible to reduce the level of coal cutting noise as well as provide benefits to other important areas of underground health and safety. In addition to these basic operational parameters, fundamental structural design criteria for reduced-noise coal mining machine cutting heads are also presented.

INTRODUCTION

In response to the Federal Coal Mine Health and Safety Act of 1969, which established maximum noise exposure levels permissible for mining personnel, the Bureau has undertaken a number of research programs aimed at reducing the noise associated with mining operations. One of the more serious noise problems in the coal mining industry is associated with the operation of continuous miners in underground coal production.

A noise survey was conducted on a representative sample of continuous miners (1),[3] and a summary of these results is presented in figure 1. These data show that the vibration of the cutterhead and conveyor are the major continuous miner noise sources. The drive train and hydraulic system, on the other hand, are secondary noise sources, because their contribution to the overall noise level is smaller.

The problem addressed in this paper is the noise generated by underground coal cutting operations. The noise sources directly associated with the cutting of coal are diagramed in figure 2. The available experimental evidence indicates that coal fracture noise and face radiated noise generated during the operation of the typical continuous miner are well within present MSHA noise regulations (2). The order of importance of the remaining coal cutting noise sources is primarily determined by the design and operation of the cutterhead.

Several fundamental design and operational concepts for low-noise coal cutting are presented in this paper. These concepts apply to any type of continuous miner that uses bits to break coal from the face. Understanding how the design concepts result in low-noise coal cutting requires a limited background in structural dynamics. The "Background" section should provide the needed basics for the uninitiated reader.

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[3]Underlined numbers in parentheses refer to items in the list of references at the end of this paper.

BACKGROUND

The fact that a structure will vibrate when struck and that the vibration eventually stops is a matter of common experience. Structural dynamics is the term used to describe the area of scientific study that seeks to mathematically describe this common experience. The way in which a structure vibrates in response to an applied force is an inherent attribute of the structure.

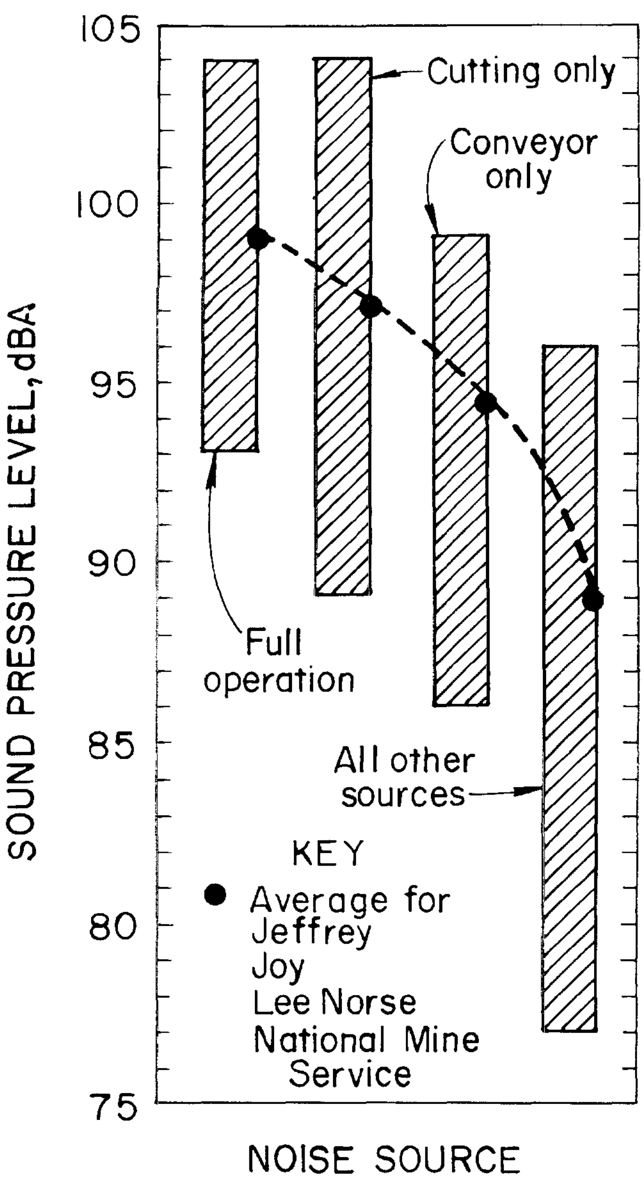


FIGURE 1. - Contributions of major sources of noise for continuous miners.

The response of a structure can be more easily examined in the frequency domain. Frequency is a period of oscillation or the rate at which specific action or event repeats itself. If an event or motion is repeated 20 times a second, the frequency of oscillation is said to be 20 Hz. A pure tone of music is a pressure variation in the air that strikes the ear at regular intervals. Any signal time history, such as the force time history in figure 3, can be thought of as the sum of an infinite number of single frequency signals. Each frequency has a specific amplitude and starting time relative to the other frequencies. To continue with the music or sound example: If the force signal of figure 3 were a noise signal, it could be exactly re-created by a very large number of whistlers. Each whistler would be assigned a specific tone or frequency to whistle, a specific loudness to whistle, and a specific time to start whistling. If done just right, the sum of all the whistlers' single frequency inputs would result in the specific noise time history desired.

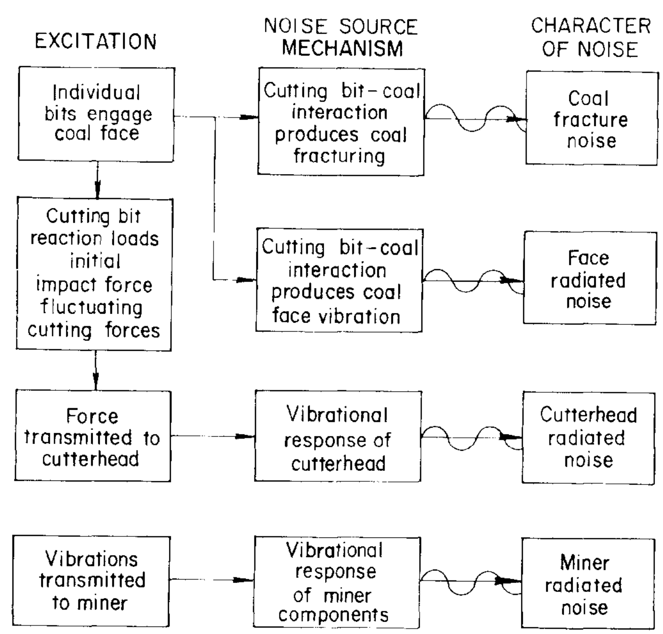


FIGURE 2. - Coal cutting noise sources.

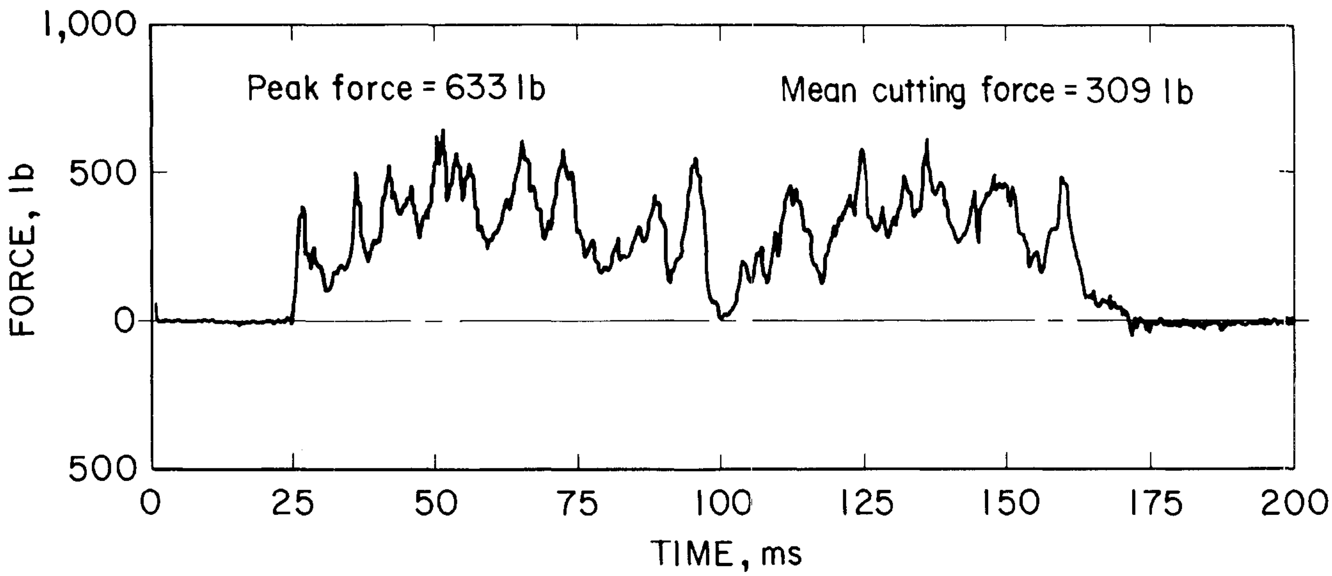


FIGURE 3. - Force time history of single auger bit cutting coal at 60 in/s.

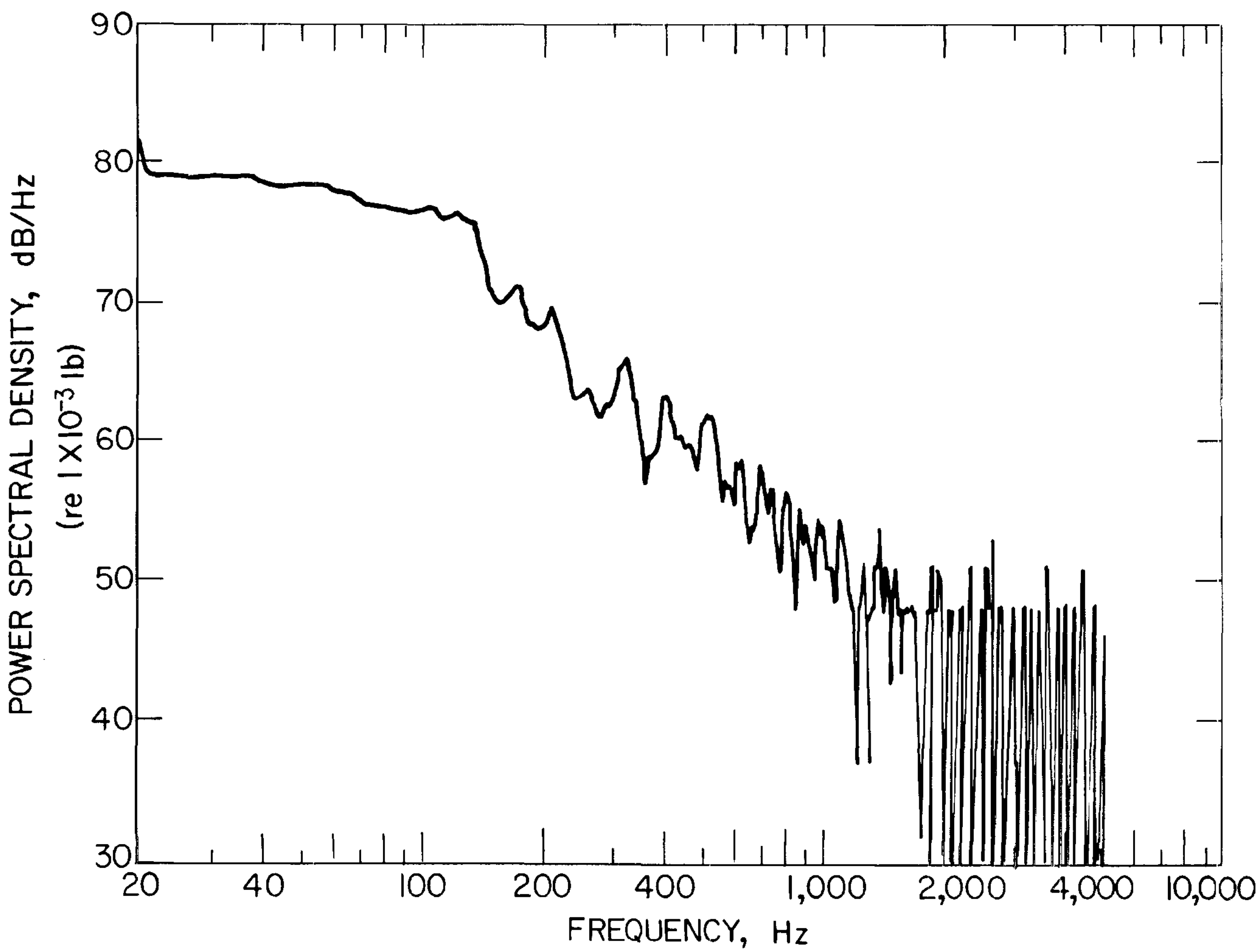


FIGURE 4. - Force power spectral density of single auger bit cutting coal at 60 in/s.

Of course, the signal does not have to be noise. The signal can be force, velocity, displacement, or any measurable quantity. The frequency domain representation of a time history simply describes the magnitude that each frequency component must have and the time at which the frequency must begin relative to the other frequencies. The force time history of figure 3, therefore, is made up of the frequency domain force components of the magnitudes shown in figure 4.

The vibratory response of a structure to an applied force is strongly dependent on the frequency at which the force is applied. The response characteristic of a structure at a specific frequency is found by measuring the motion caused by a unit force applied to the structure at that frequency. If measurements are made over a range of frequencies, the frequency response characteristics of the structure are, of course, more fully defined. Figure 5 shows the response characteristics at the free end of a beam fixed at one end (like a flagpole). The characteristics are determined by the specific configuration of the structure. Perhaps the greatest emphasis in structural dynamics is the development of methods to predict the response characteristics of a structure.

It is clear in figure 5 that the structural response to a unit force is much greater at some frequencies than at others. This phenomenon is an important aspect of structural dynamics. It results from the combined effects of the distributed mass and stiffness of a structure. At these high-response

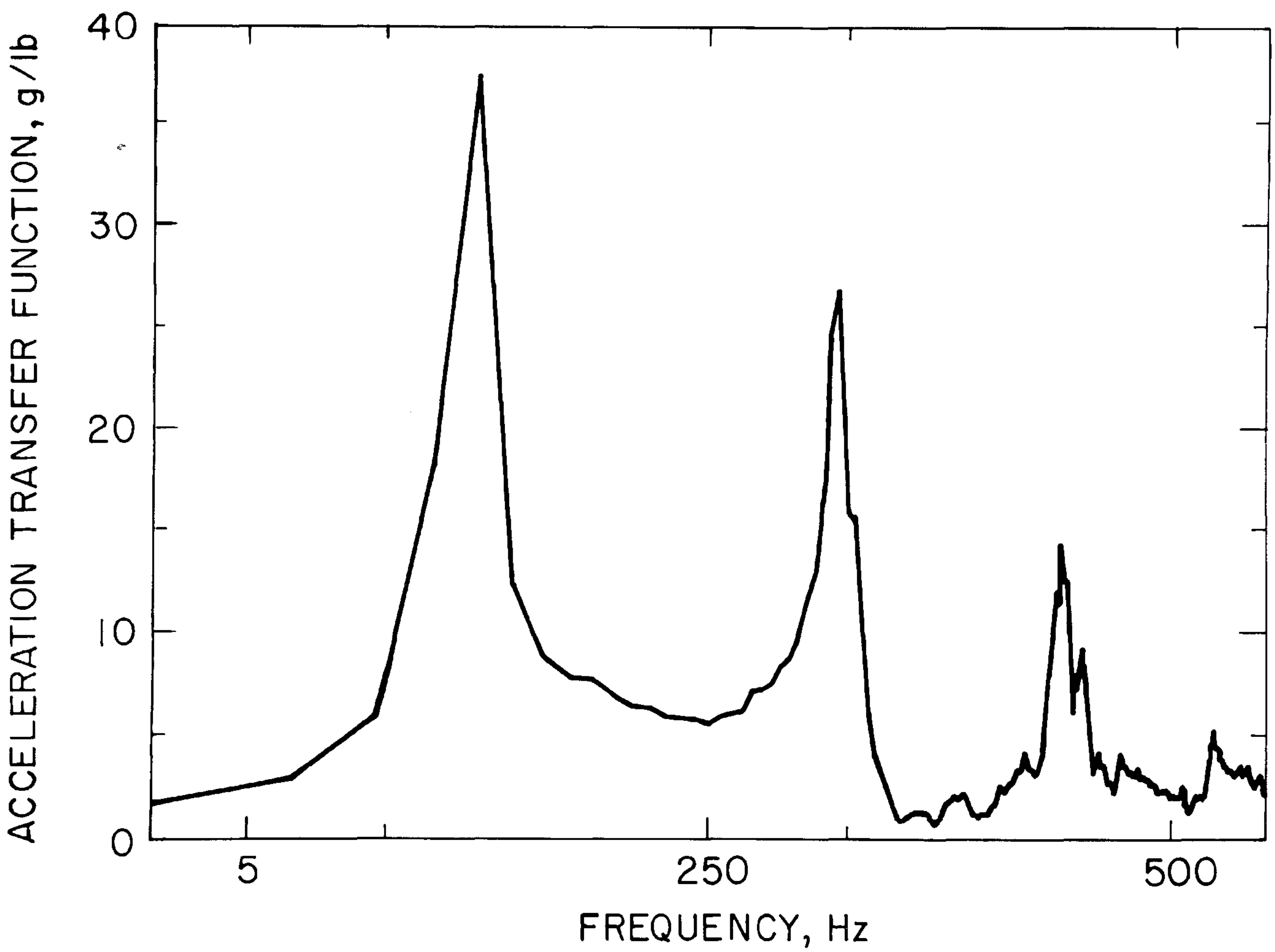


FIGURE 5. - Structural response characteristics.

frequencies, the motion of the structure is limited only by its energy dissipation (damping) characteristics. The frequencies at which this occurs are called the "natural" frequencies, or resonances, of a structure. The response at the natural frequencies of a structure can easily be 100 times the response of the structure to the same force magnitude applied at a nonresonant frequency. When one pushes a child in a swing, one applies a periodic force at the first natural frequency of the swing. The energy dissipation mechanism of the swing is primarily the wind drag against the body as it moves through the air. One final example is a flagpole. One can cause the tip of the flagpole to swing widely from side to side by pushing on the pole at a specific rate--the rate at which the pole naturally sways back and forth. The energy dissipation (damping) of the pole limits the peak deflection of the pole and causes the pole to come to rest after the periodic force is removed. All structures have an infinite number of natural frequencies. The frequencies at which these resonances occur is solely a function of the structural configuration (mass, stiffness, and damping distribution).

That a vibrating structure cruses noise is another fact of common experience. Some of the energy of vibration is expended moving air. The amount of energy transferred to the air from a vibrating structure is proportional to the average surface velocity of the structure. The proportionality is the efficiency at which the vibrating structure can pass energy to the surrounding air. It is often called the radiation efficiency. The radiation efficiency is a strong function of both frequency and structural configuration. It tends to be quite small at lower frequencies and rises to a constant maximum value as the frequency increases.

NOISE CONTROL CONCEPTS

Each noise control concept presented in this paper takes full advantage of the basic nature of the dynamic forces generated during coal cutting to reduce coal cutting noise. Continuous mining machines exploit the brittle fracture characteristics of the coal. When a cutting bit engages the coal face, the coal resists its advance in a manner analogous to a spring. The applied force increases until the stress in the coal initiates localized brittle fracture. The coal fracture propagates for some characteristic distance from the point of force application. The cutting bit continues to advance, meeting steadily decreasing resistance as it clears out the fractured coal. When the bit again contacts unfractured coal, the process is repeated.

A representative force time history (fig. 3) and force power spectral density (PSD) (fig. 4) of a rigidly mounted bit cutting coal clearly show the highly impulsive nature of the process. It is interesting to note that the impact force generated as the bit initially enters the coal is no more impulsive than the subsequent individual fracture events. The extensive experimental program from which figures 3 and 4 were taken has shown that both the stress required to initiate fracture and the characteristic distance of fracture propagation are relatively independent of cutting bit velocity. Hence, for a given type of coal, depth of cut, and bit configuration, the peak force that initiates coal fracture and the total number of fracture events that occur in a prescribed length of cut are velocity independent.

All continuous mining machines experience this type of excitation. In the frequency domain, the cutting force PSD is flat until a cutoff frequency, after which it declines with frequency at a rate that is inversely proportional to the square of the frequency (see figure 4). The cutoff frequency is primarily a function of the coal type and the cutting bit velocity.

The peak and mean cutting forces are a function of the material being cut, bit type, and depth of cut. While not extensive, cutting force data have been obtained for a variety of coal, shale, and rock types at several different depths of cut (3-6). Table 1 summarizes the generally available data from both laboratory and in situ cutting tests.

Recall the essential characteristics of coal cutting: First, coal resists advance of the bit in a manner analogous to a spring. Second, localized brittle fracture occurs when the stress in the coal reaches a critical value. Third, the brittle fracture propagates for some characteristic distance from the point of force application. Fourth, bit velocity does not affect the first three characteristics. Of course, since real coal is not homogeneous, the above is only true in a very rough statistical sense. These characteristics are represented in the very simple but quite useful single-degree-of-freedom coal model illustrated in figure 6.

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

TABLE 1. - Laboratory and in situ cutting force data

Material	Bit type	Depth of cut, in	Mean force, lbf	Standard deviation, lbf	Peak force, lbf
Blue Creek coal............	Plumb bob.........	0.5	607	325	1,608
	...do.............	1	925	541	1,994
	...do.............	2	1,500	950	3,309
Pittsburgh No. 1 seam coal..	...do.............	.5	515	245	1,082
	...do.............	1	958	590	2,440
Pittsburgh No. 2 seam coal..	...do.............	.5	773	406	2,088
	...do.............	1	1,037	646	2,727
	...do.............	2	1,489	929	3,679
Illinois No. 6 seam coal....	...do.............	.5	694	329	1,704
	...do.............	1	1,112	525	2,495
	...do.............	2	1,310	642	2,555
Illinois shale.............	Wedge.............	1	866	673	2,918
Bruceton synthetic shale....	...do.............	.5	616	472	3,163
	...do.............	1	1,739	1,127	6,780
Blue Creek shale...........	Plumb bob.........	1	925	541	1,994
	Auger point attack	1	650	337	1,494
	Wedge type 1......	1	689	533	3,330
	Wedge type 2......	1	667	511	2,303
Sandstone..................	Point attack......	.5	1,385	NA	NA

NA Not available.

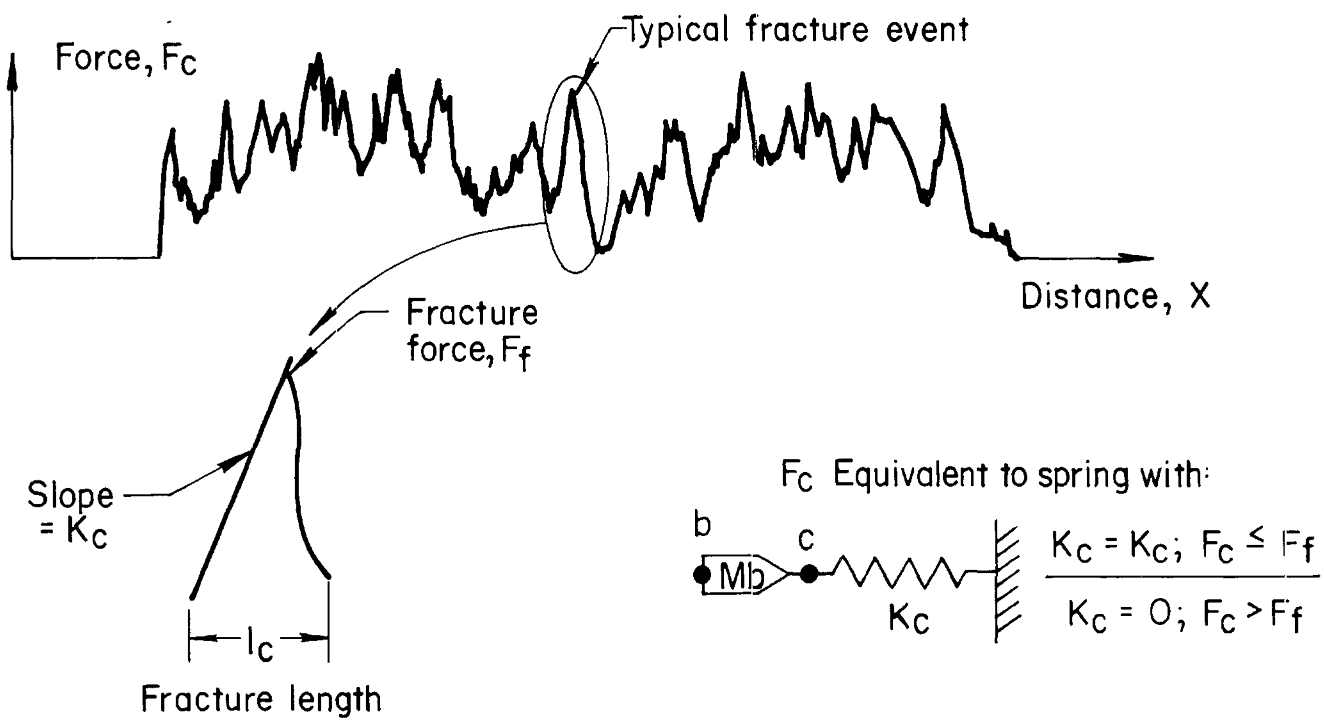


FIGURE 6. - Coal cutting force model.

NOISE CONTROL BY REDUCING BIT VELOCITY

The first principle of cutterhead design and operation is: *Keep the bit velocity as low as possible.* The characteristic fracture length as described in the coal cutting model is independent of velocity. Therefore, the impulse time (or duration) of an individual fracture event is inversely proportional to the bit velocity. The typical impulse time determines the cutoff frequency of the force spectrum. The shorter the impulse time, the higher the cutoff frequency. Experiments with a wide range of coal, shale, and synthetic coals have shown that the one-third octave-band force level decreases about 1.5 to 2 dB per band after the cutoff frequency (fig. 7). This is typical of the spectrum associated with a triangular pulse shape. Slower cutting, therefore, lowers the high-frequency force levels by reducing the cutoff frequency.

Several factors combine to make the low frequency (below 200 Hz) dynamic cutting forces of little importance as a source of harmful radiated noise. Below 200 Hz, most cutterhead and miner structures are generally in the stiffness-controlled region of their structural response. Hence, the magnitude of the structural vibration due to a given unit of force is not significantly amplified. The transfer of energy from the vibrating structure to the air is also very inefficient in the frequency range below 200 Hz for the typical cutterhead, miner size, and configuration. Finally, the noise that is radiated below 200 Hz is not as damaging to the ear as the higher frequency noise. The A-weighting of noise spectra (fig. 8) is typically used to represent this fact. MSHA workplace noise regulations are written in terms of A-weighted noise.

The bit velocity can be reduced while maintaining coal production by increasing the depth of cut and/or the number of bits per cutting line. The hypothetical characteristics given in table 2 and the following paragraph illustrate and explain the noise control advantage of reduced bit velocity.

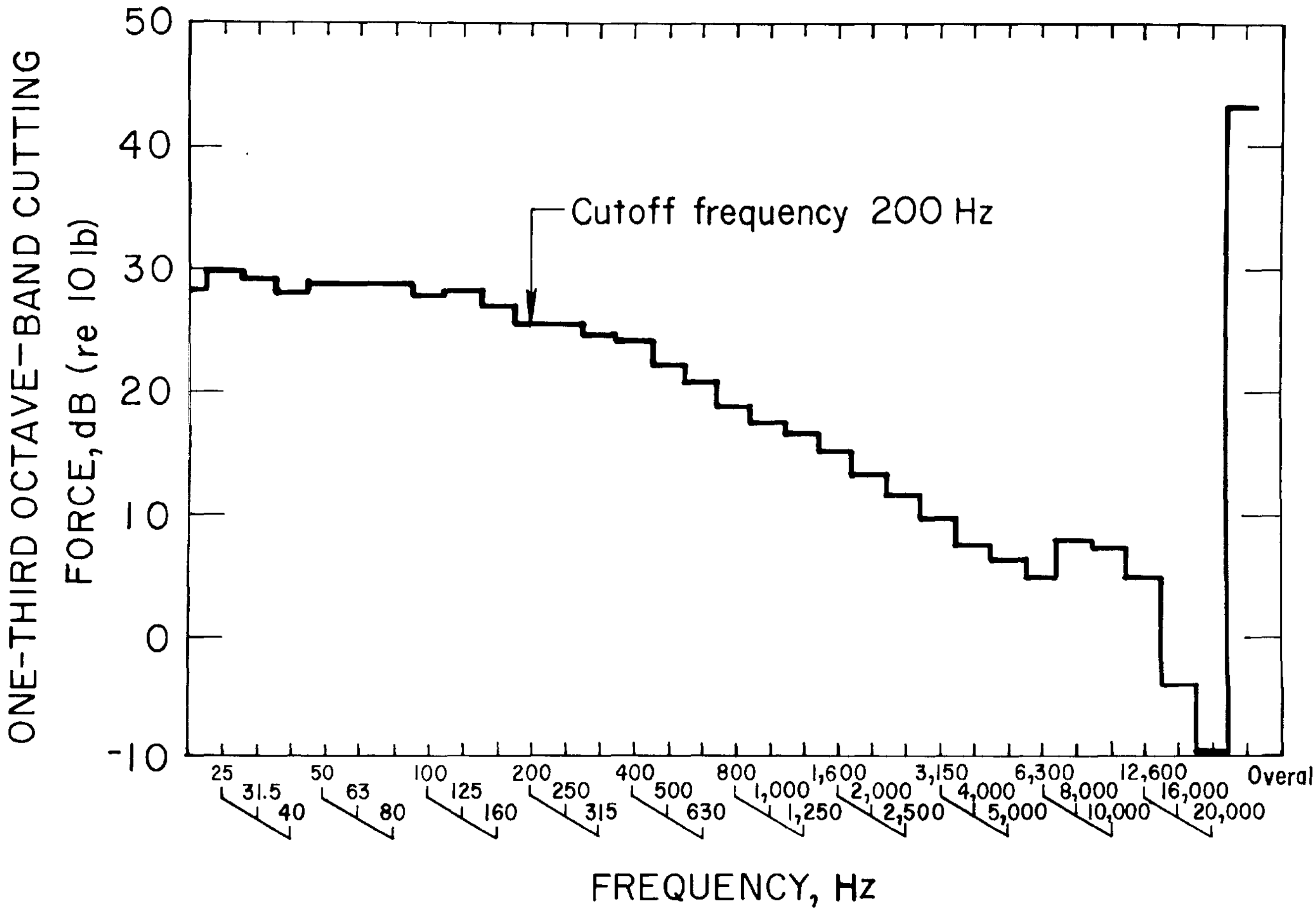


FIGURE 7. - Typical one-third octave band coal cutting force spectrum.

TABLE 2. - Hypothetical drum characteristics

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

[1]2-start drum.

[2]Doubles force level on the bit over the force level of a 1-in-deep cut.

Assume a baseline or control cutterhead, A, with the characteristics listed in table 2. A comparable cutterhead, B, would have the same depth of cut as drum A but operate at one-half the bit velocity. A cutterhead that must operate at twice the depth of cut of drum A to maintain the same production would have to have the characteristics listed for drum C in table 2. The force spectrum acting on the bits of each drum type is drum type is postulated in figure 9. The force scale of the figure is arbitrary since it is the relative force magnitude of the three drums that is of interest. The dynamic force on each bit of drum B is 6 dB less than on each bit of drum A in the frequency region above the cutoff frequency of drum A. Reduced force on a cutting bit translates directly to reduced miner structural vibration due to that bit. The total vibration of a continuous mining machine is the sum of the vibration caused by each bit that is in contact with the coal. Since drum B has twice the number of bits as drum A, a 6-dBA reduction in the force on each bit would result in a 3-dBA overall structural vibration and noise reduction.

Drum C has the same number of bits as drum A. Drum C operates at one-half the cutting speed but twice the cutting depth to maintain the same production rate as

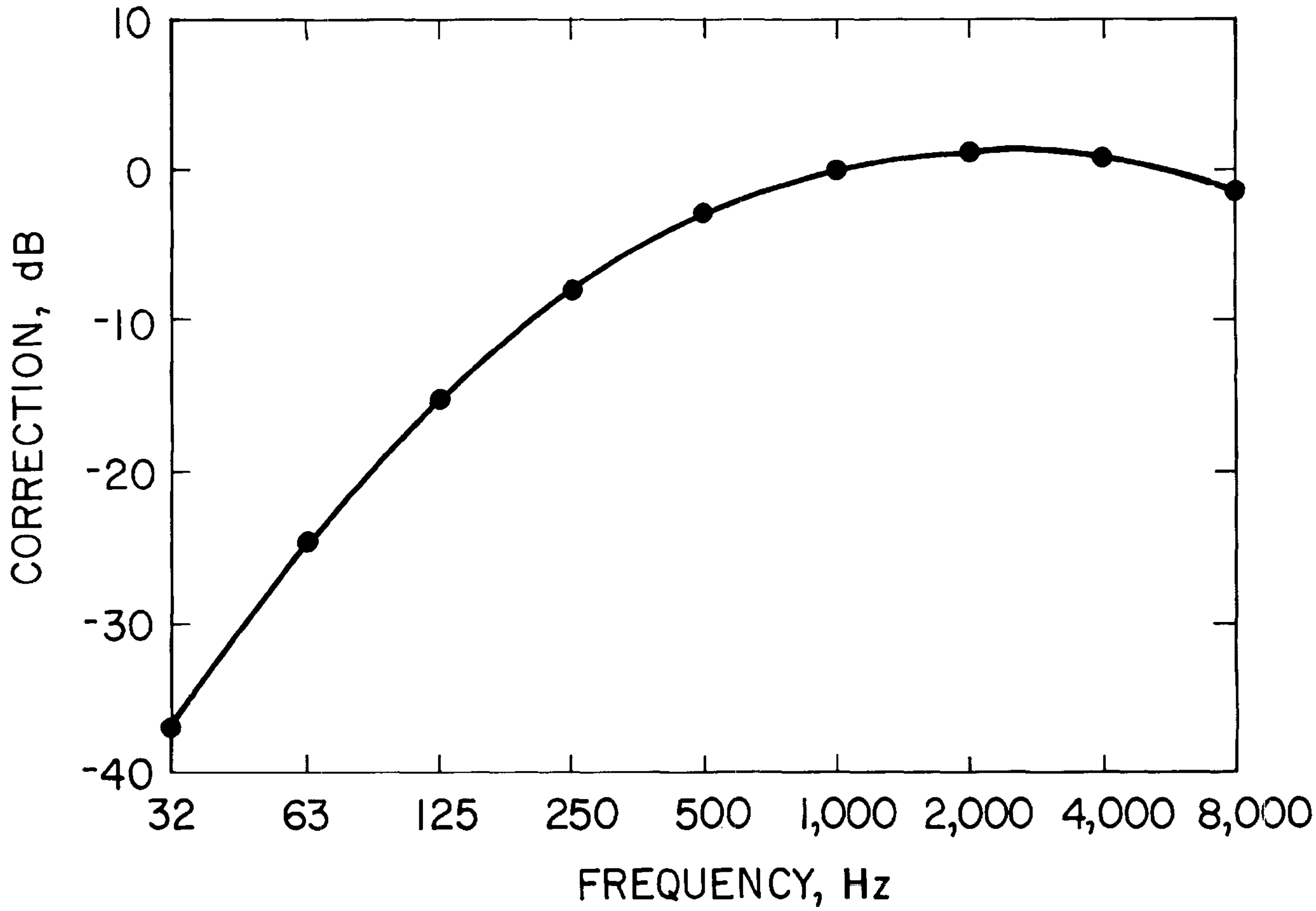


FIGURE 8. - A-weighting of noise spectra.

drum A. As noted in table 2, the cutting force magnitude of drum C is assumed to double with a doubling of cutting depth. This is actually somewhat conservative because data (table 1) often indicates only a 1.5-to 1.7-fold increase in force per doubling of cutting depth. The 3-dB force reduction (fig. 9) achieved above the cutoff frequency of drum A should result in a 3-dBA reduction in coal cutting related noise.

Reference 3 contains quantitative experimental support for this analysis method. Coal cutting noise, as determined in the laboratory, for a 1-in depth of cut and 96-in/s cutting speed averaged 101.5 dB as measured on a flat (non-A-weighted) scale. Coal cutting noise for a 1-in depth of cut and 16-in/s cutting speed averaged 88.2 dB, a 13.3-dB reduction. Based on the cutting speed reduction, the model presented here predicts a force reduction of 12 to 16 dB above the cutoff frequency of the 96-in/s cut. Again, because the A-weighting deemphasizes the lower frequency force components, the force reduction should result in a 12-to 16-dBA noise reduction.

Although an overall noise reduction of around 3 dBA is not sufficient to solve all cutterhead noise problems, the reduction can be achieved for very little cost. In addition, there is strong evidence that slower and/or deeper cutting produces less dust and also reduces the likelihood of spark ignitions of methane (7-9).

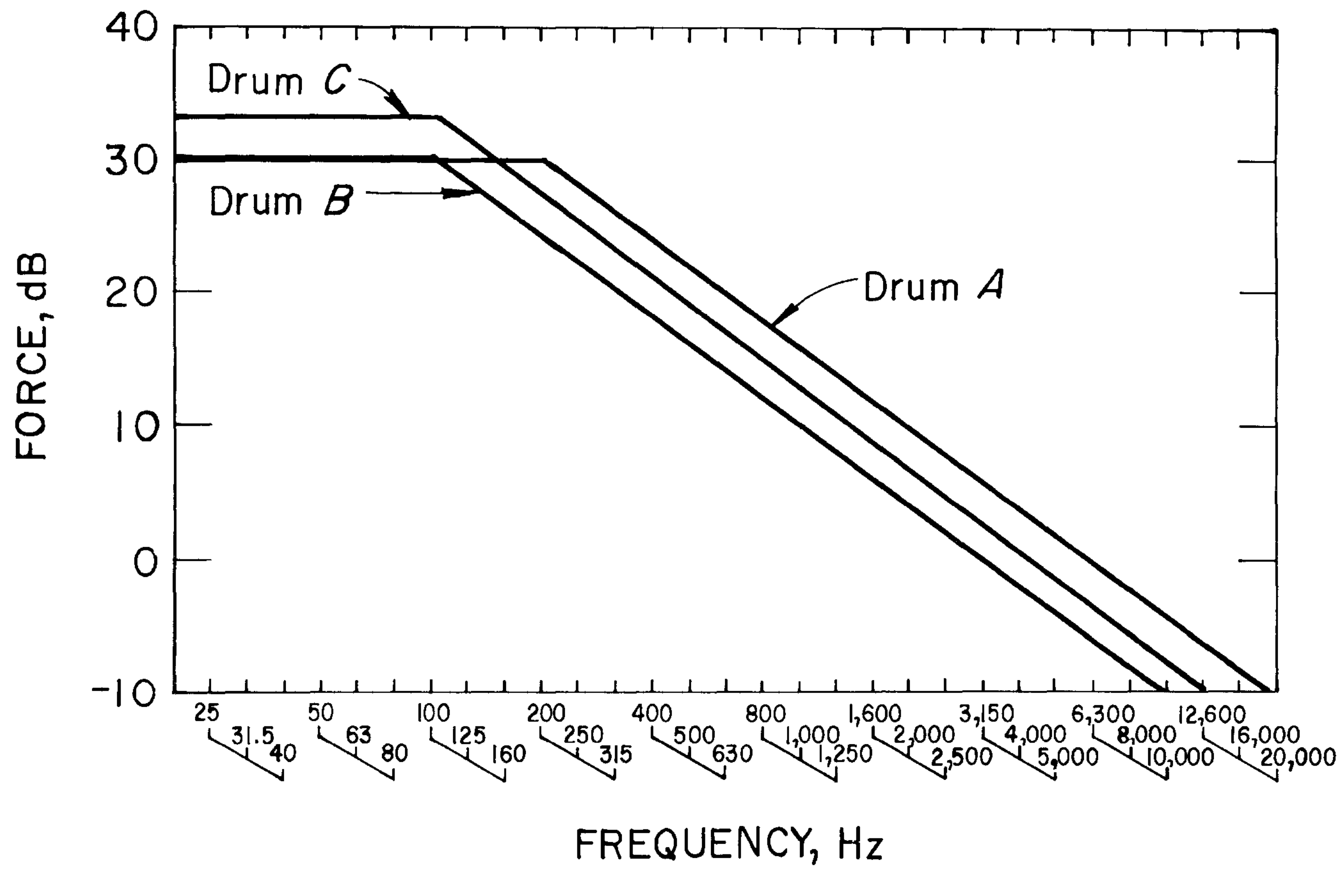


FIGURE 9. - Estimated force spectra.

NOISE CONTROL THROUGH CUTTERHEAD STRUCTURAL RESPONSE ALTERATION

The dynamic forces at each bit-coal interface are reacted by the cutterhead structure. The resulting vibration of the cutterhead structure is generally the major coal cutting noise source. The second principle of cutterhead design is: *Keep the frequency of the first structural mode as high as possible.* The third principle of cutterhead design is also related to the response characteristics of the cutterhead structure: *Keep the structural damping as high as possible.* Because the excitation also passes through the cutterhead to the other miner components, these rules apply to all miner structures. The purpose of the rules is to reduce the resonant response of the structures to the dynamic excitation of the coal cutting forces.

Because the frequency spectrum of the cutting force decreases rapidly as the frequency increases, the coal cutting noise above about 2,000 Hz is generally below the level of concern. Indeed, noise below the 1,000-Hz band generally controls the overall level. The second design principle limits the structural response to the cutting forces primarily by eliminating structural resonances in the "problem" frequency range. Those structural resonances that cannot be eliminated from the band of concern should be well damped (design principle 3). Damping, of course, limits the magnitude of the structural response at the resonances.

NOISE CONTROL THROUGH ISOLATION OF RADIATING STRUCTURES FROM THE DYNAMIC CUTTING FORCES

The dynamic cutting forces cause vibration not only of the cutterhead but also of the other miner components. The vibrations propagate throughout the mining machine. The fourth principle of cutterhead design is: *Keep the higher frequency dynamic cutting forces isolated from as much of the miner structure as possible.* As previously discussed, the higher frequency coal cutting forces cause the unacceptably high noise levels. The higher frequency coal cutting forces can, in principle, be isolated from the major radiating surfaces of the cutterhead and the miner. This has been demonstrated by both analysis and experiment. Two analytical models have been developed under an ongoing Bureau contract (10). The first model was capable only of evaluating the feasibility of the concept. The second model allowed the detailed evaluation of candidate isolated cutting tool designs. The analytical feasibility evaluation involves a rough approximation of the cutting force characteristics in combination with simple frequency domain force isolation mathematics.

The peak forces seen in the force-time history of figure 3 produce the stress required to initiate brittle fracture of coal. The peak force, not the mean force, cuts the coal. Therefore, if coal cutting is to occur, the force-time history at the tip of an isolated cutting tool should be very similar to that of a rigidly mounted tool. The force peaks must be reached. The initial loading rate of the bit tip at each fracture event may be somewhat smoothed by the softer response of the isolator. This effect is, however, limited by the number of fracture events that must occur in a given length of cut. Therefore, the force on the bit tip can be considered, in this rough analysis, to be independent of the response characteristics of the cutting tool.

A simple single-degree-of-freedom isolated cutting tool is shown in figure 10. Assume that--

The mining machine has sufficient power to maintain a constant cutting head velocity, V_h, regardless of the force generated at the bit-coal interface, F_c.

The cutting head structure is rigid compared to the isolator.

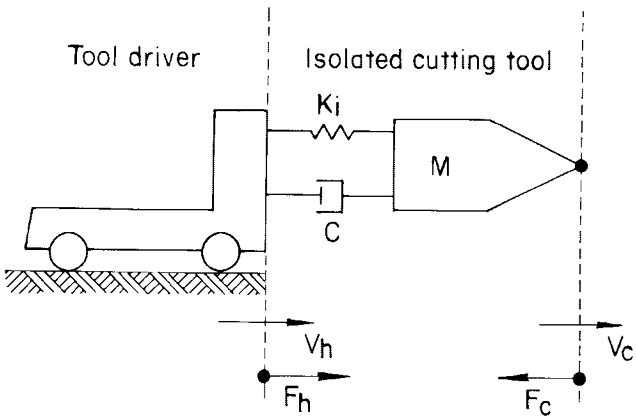
The cutting bit and tool holder are rigid compared to the isolator.

The isolator is massless.

A frequency domain dynamic force balance can be written to calculate the force transmissibility, T, of the isolator. Force transmissibility at any given frequency, $\omega(\omega=\omega^2\pi^f)$, is defined as the ratio of force at the base of the isolated cutting tool to the force at the tip. With the definition of symbols as in figure 10, the result is

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

This equation can be used to evaluate the gross feasibility of the isolated cutting tool. It also clearly demonstrates the concept and advantages of force isolation. Figure 11 gives the transmissibility at various levels of damping over a range of frequency ratios, f/f_n, where f is the exitation frequency and f_n is the natural frequency of the isolated cutting tool. The isolated cutting tool simply allows the mass and acceleration of the tool to react a portion of the coal cutting force.



KEY

K_i Isolator spring stiffner
C Isolator damping coefficient
V_h Velocity of cutting head
F_h Force at cutting head
M Mass of isolated cutting tool
V_c Velocity of cutting tool tip
F_c Force at the cutting tool – coal interface

FIGURE 10. - Single-degree-of-freedom isolated cutting tool.

A mineworthy isolated cutting tool must have the following characteristics:

1. The highest natural frequency of the tool isolation system must be less than 170 Hz to assure that isolation occurs above 250 Hz.

2. Elastomeric isolator elements must not be allowed to deflect more than 10 pct of the isolator thickness under shale cutting force levels.

3. Cutting tool space constraints must not be violated.

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

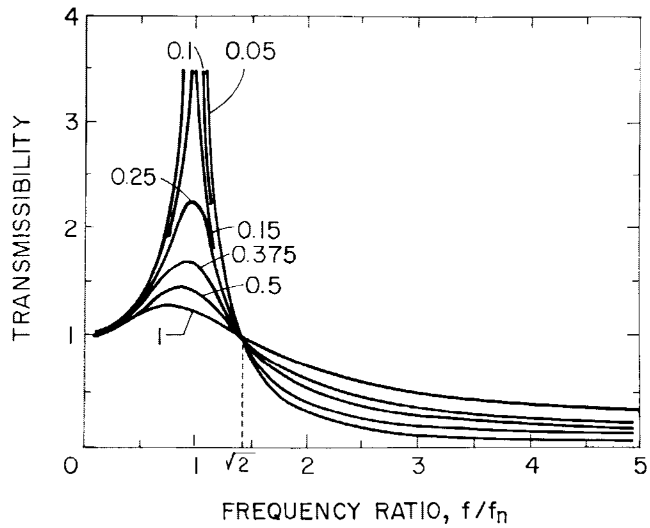


FIGURE 11. - Typical transmissibility curves.

For the purpose of rough feasibility calculations, assume a maximum allowed isolated cutting tool deflection, d_m of 0.15 in, damping, ζ, of 8 pct and natural frequency, f_n, of 170 Hz. Table 3 gives details on shale cutting force (5). The design values are the maximum force levels measured during the experimentation. Because the excitation of the isolated cutting tool is a continuous dynamic process, the peak deflection of the isolator will occur at the resonance frequency, ω_n. The transmissibility equation and the equations defining $\omega_n (\omega_n = \sqrt{K_i/m})$ and $\zeta (\zeta = C/2\sqrt{K_i m})$ can be used to help calculate the peak deflection of the isolator at resonance. At resonance, the transmissibility equation reduces to

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

Using the design parameters given in table 3, the total equivalent peak static force, F_E, on the isolator at resonance is

$$F_E = (FB * CF * 1/2) + F_m$$

$$= 15{,}000 \text{ lbf.}$$

TABLE 3. - Shale cutting parameters for feasibility calculations

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

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

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

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

$$m = \frac{(15{,}000/0.15)}{(2\pi 170)^2},$$

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

These simple calculations indicate that the isolated cutting tool concept is quite feasible and that the concept deserves much more rigorous scrutiny.

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

In addition to the frequency domain analysis, the program has time history analysis capability. A coal seam model that may include layers of rock can be defined. The basic coal and rock model are as previously described. The time history model maps the deflection of all points of interest through a full rotation of a cutterhead of specified cutting diameter, advance rate, and rotational speed. The program is a fast and effective way of evaluating and optimizing the response characteristics of an isolated cutting tool design.

The first experimental evidence that isolated cutting tools could cut coal and achieve significant isolation was obtained with a linear cutting apparatus (LCA). The LCA (fig. 12) was designed specifically for controlled coal cutting force measurements. An isolated cutting tool was built for mounting on the LCA (fig. 13). A rigid cutting tool was also built (fig. 14). Coal was cut at 60 in/s from opposite sides of the same coal sample (fig. 15). One side was cut with the rigid tool and the other side with the isolated tool. Cuts were made on several blocks of coal in this manner.

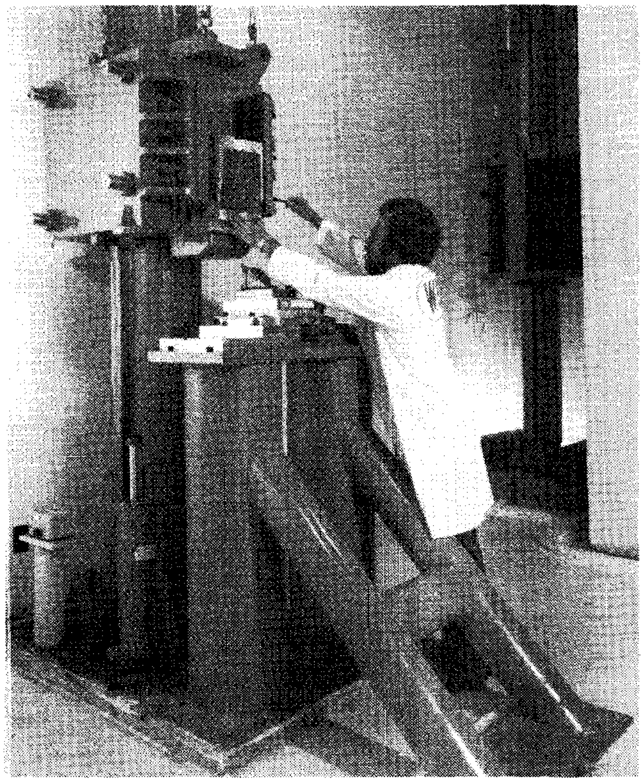


FIGURE 12. - Linear cutting apparatus (LCA).

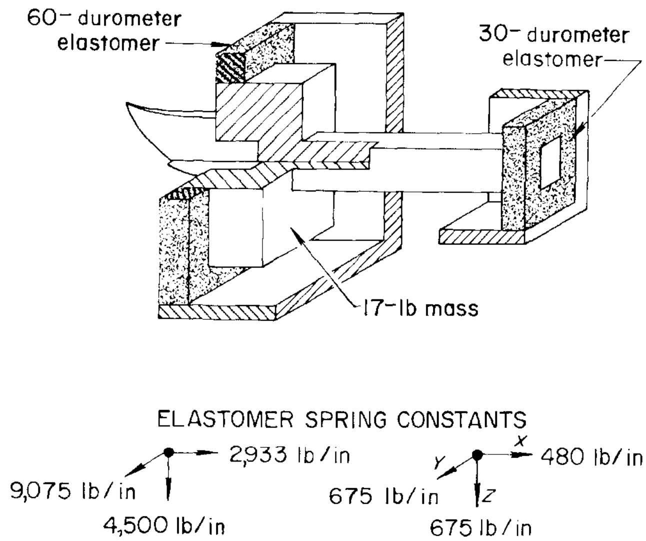


FIGURE 13. - Isolated cutting tool for LCA tests.

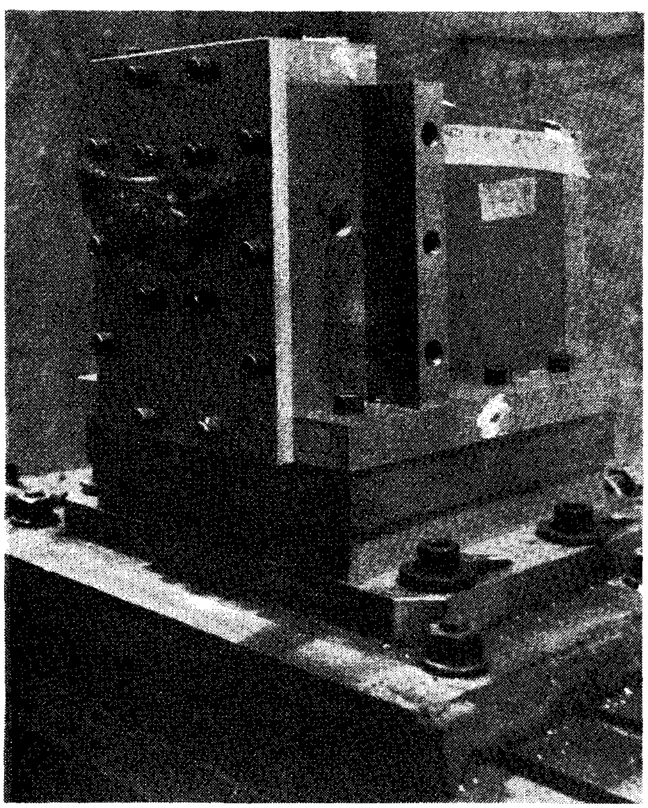


FIGURE 14. - Rigid cutting tool for LCA tests.

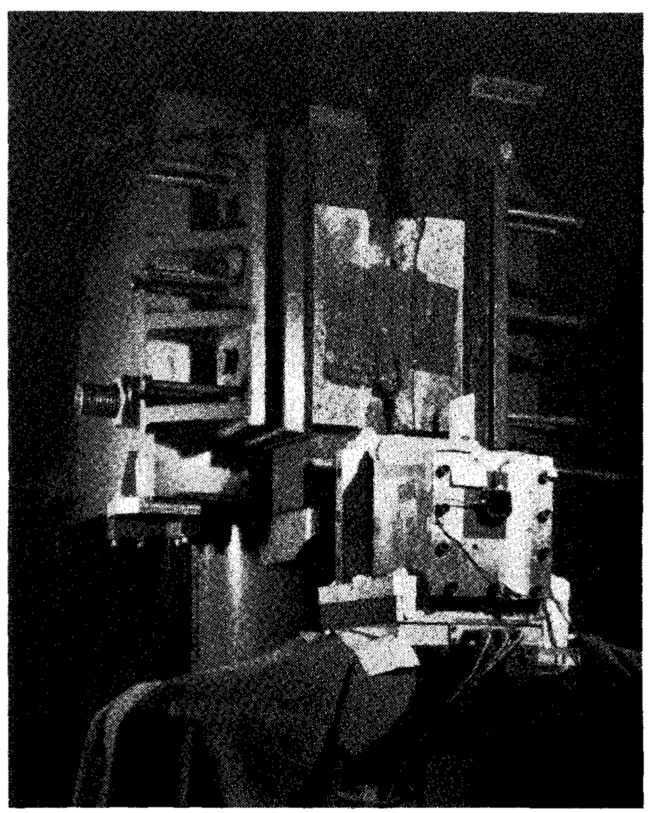


FIGURE 15. - Typical coal sample for LCA tests.

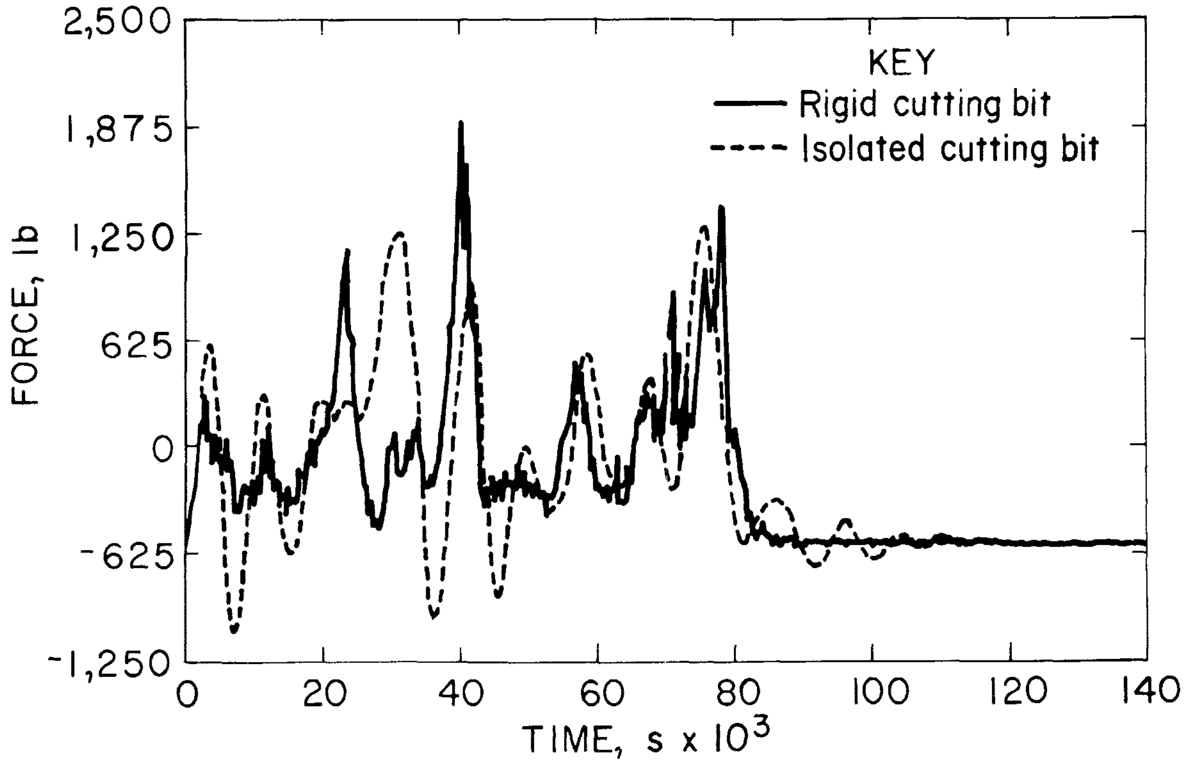


FIGURE 16. - Force time histories for LCA tests.

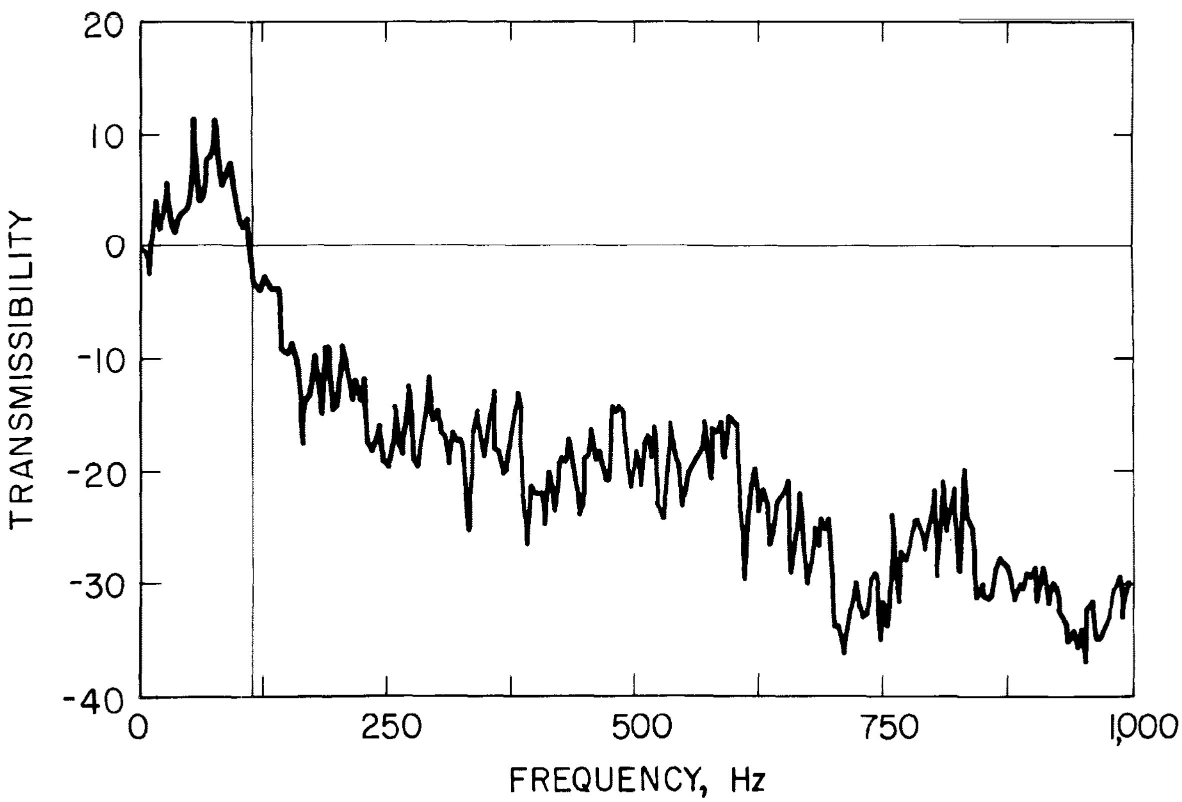


FIGURE 17. - Isolator transmissibility for LCA tests.

Figure 16 gives the force time histories experienced at the cutting tool mount for both tools. The force transmitted to the mount by the isolated cutting tool is clearly smoother. Figure 17 gives the frequency domain results in the form of isolator effectiveness (force at base for isolated tool divided by the force at the base for the rigid tool). The curve is rough because of the statistical differences between coal along each separate cutting line. The weight and particle size distribution of the coal cut by the isolated bit were not statistically different from those of the rigidly mounted bit. These analytical and experimental results clearly demonstrate that coal cutting forces can be isolated.

APPLICATION

Recall the three basic coal cutting noise control concepts discussed in this paper: (1) reducing bit velocity, (2) altering the structural response of the cutting head, and (3) isolating the cutting head from dynamic cutting forces. The first concept, reducing bit velocity, can be applied to almost any continuous mining machine provided that other design changes (e.g., deeper cutting, bit mounting design, etc.) are made to maintain production levels. The second and third concepts have been applied in several recent and ongoing Bureau research programs, as described in the following section.

ALTERING THE STRUCTURAL RESPONSE OF THE CUTTING HEAD

Auger Miner Cutting Head (11)

The helix of the standard auger miner cutting head (fig. 18) is the primary coal cutting noise source of the auger mining system. Following the structural

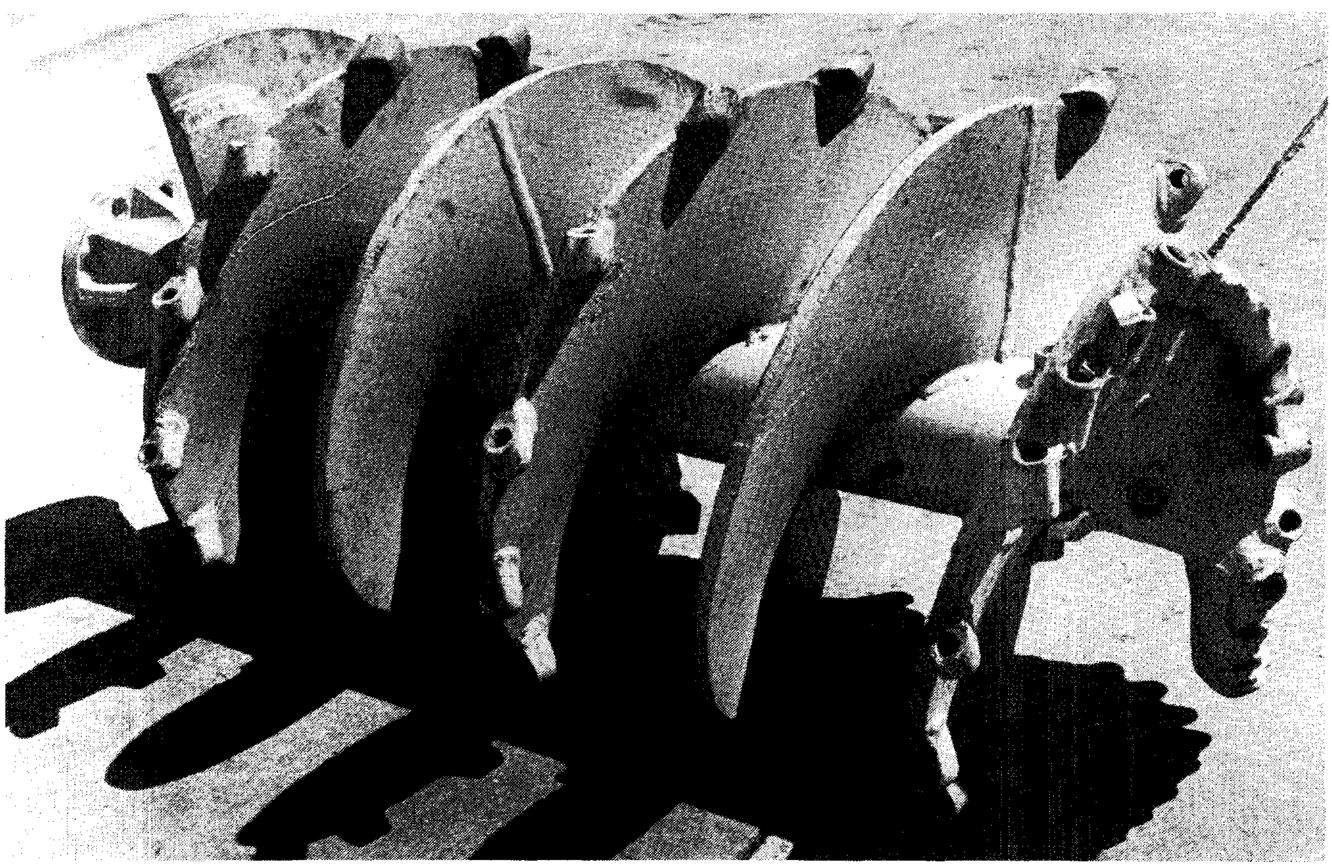


FIGURE 18. - Standard auger miner cutting head.

design precepts, the helix was stiffened and damped. The dual goals of achieving a high first natural frequency and high damping of the auger helix were obtained with a sand-filled, conical-helix stiffener. The stiffener can be seen on the prototype reduced-noise cutting head in figure 19. The cavity created by the stiffener and the helix was filled with sand and sealed. To further stiffen the helix, the core size was increased. This configuration achieved a fourfold increase in helix first natural frequency and a tenfold increase in helix damping at the resonances below 2,000 Hz. The full operational capability of the design was verified by a 6-month-long in-mine test at Mears Coal Co., Dixonville, PA. In-mine noise measurements, taken during the operational testing, demonstrated a 10-dBA noise reduction at the worker position nearest the cutterhead. Controlled laboratory cutting tests on synthetic coal verified the in-mine noise reduction results (fig. 20).

Longwall Shearer Cutting Head (12)

The longwall shearer cutting head is very similar to that of the auger miner in that the spiral, platelike helix is the primary coal cutting noise source of the mining system. Therefore, the longwall shearer helix was stiffened and damped in the same manner as the auger cutterhead helix. Although, the first resonance of the helix was significantly increased, the damping technique only slightly increased the damping of the very massive longwall helix. Laboratory tests predicted a noise reduction of 4 dBA for the stiffening and damping procedure. Although significant, this noise reduction is not sufficient to eliminate coal cutting noise as a potential longwall workplace hazard.

The damping of the longwall shearer helix could possibly be doubled by the use of a different damping material, thus resulting in an additional 3-dBA reduction in the noise radiated by the helix. Because of the helix configuration, the first resonance of the helix is probably near the practical maximum. If significantly larger noise reductions are to be achieved, therefore, it is evident that a radically different shearer head design is required.

The helix of the traditional longwall shearer head performs two functions: It supports the cutting bits in the standard helical pattern, and it moves the cut coal to the conveyor. The dual function forces a configuration that is inherently highly resonant and lowly damped in the "problem" frequency range (200 to 2,000 Hz). As previously discussed, attempts to stiffen and damp the helix results in only moderate noise reductions. The basic geometry limits the increase in stiffness and resonant frequency that can reasonably be achieved. Increases in damping are limited by both geometry and material properties.

A new concept in bit lacing has been developed to eliminate the necessity of a continuous, two-function steel helix. In this lacing concept, called staged bit lacing, the bits are arranged in sets of cutting bit arrays. Each cutting bit array consists of several bits mounted on a single stand. The bits of each cutting array are set adjacent to each other. The individual stands have no natural frequencies within the range of interest. Each stand is very bulky (that is, all dimensional ratios approach unity), which further reduces the efficiency at which the vibrational energy of the structure is passed to the air.

The conveying function of the steel helix is performed by a helix of wear-resistant, high-density polyurethane. The material selected is sufficiently stiff to bridge the distance between the bit array stands but has extremely high damping. It will not support significant resonant vibration. The material has twice the wear resistance of wear-resistant steel; in fact, it is used as conveyor lining and at chute transfer points in many industries, including coal preparation plants.

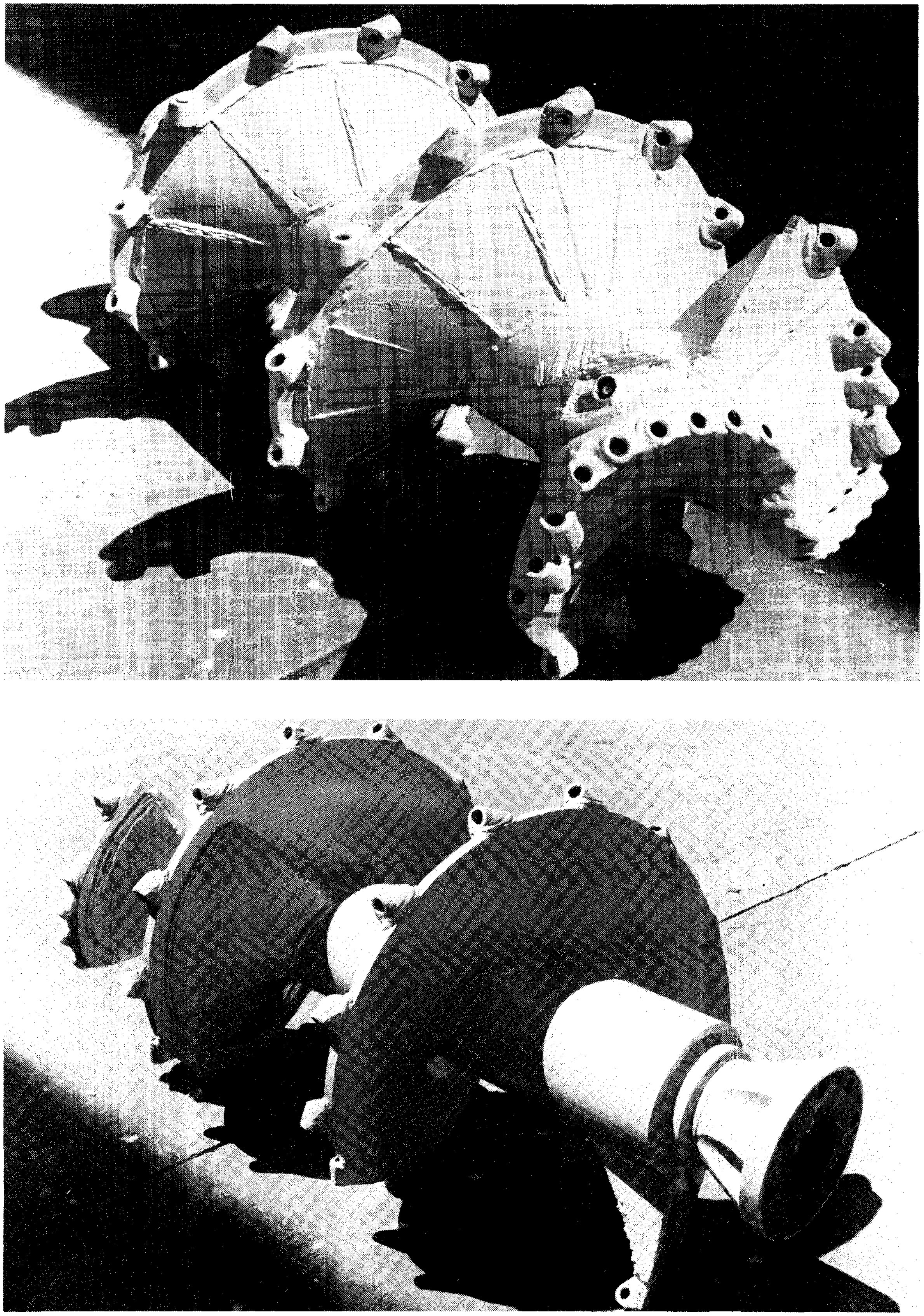


FIGURE 19. - Reduce-noise auger miner cutting head.

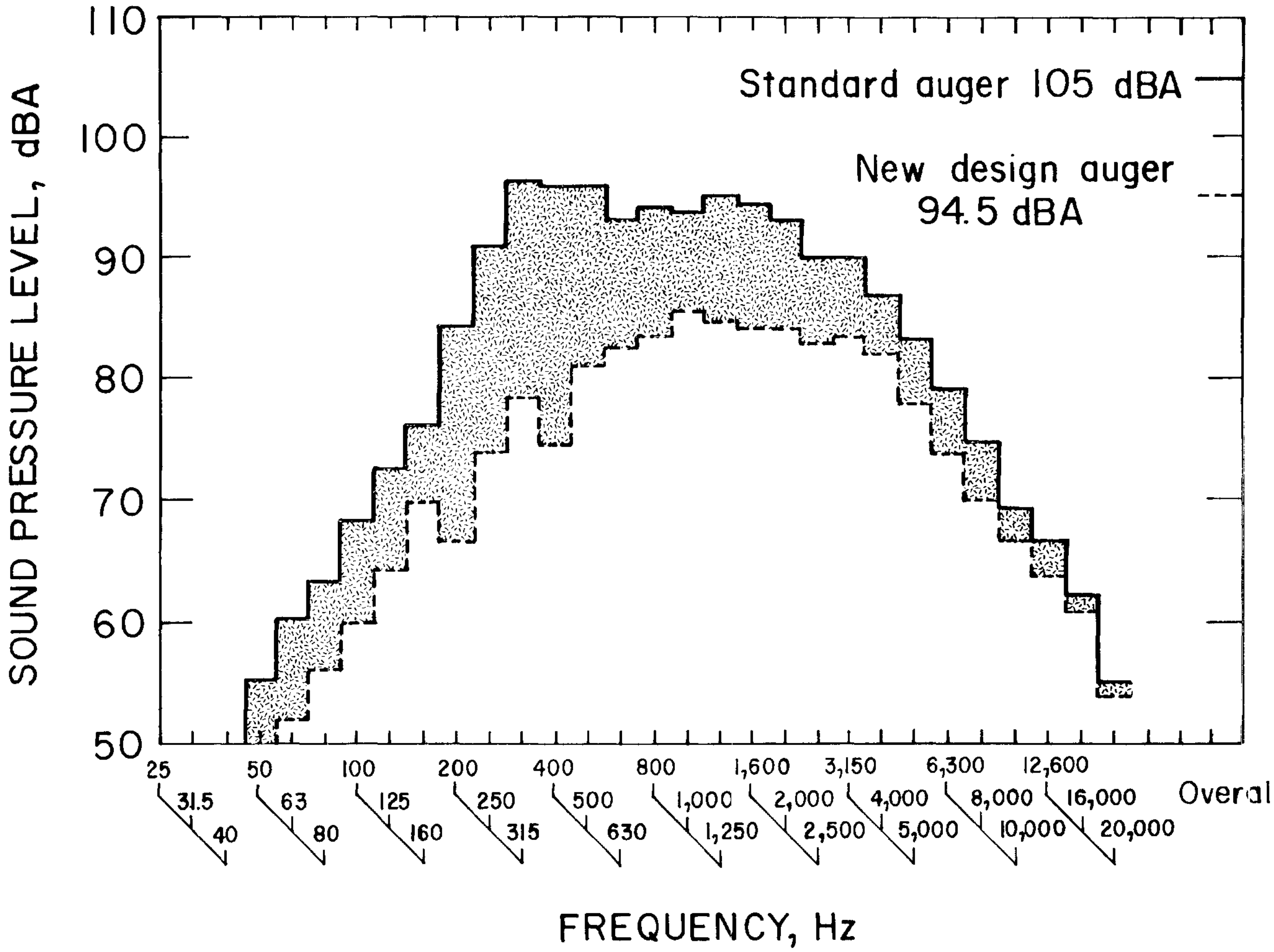


FIGURE 20. - Noise from laboratory test of auger miner cutting head.

This structurally altered shearer head (staged bit lacing and nonmetallic helix) has been fabricated and is now being tested at the Bureau's Pittsburgh (PA) Research Center. Extensive in-mine tests of the new shearer head design are also planned.

Isolation of Coal-Cutting Forces (10)

Figure 21 shows the basic design configuration of an isolated cutting head for a drum-type continuous miner. The "cutting-drum" actually consists of 12 separate 7-in-wide, 22-in-diameter rings. The rings are mounted side-by-side around a 7.5-in-diameter central drive shaft to form a cylindrical, drum-shaped cutting head. Three to five bit blocks can be mounted on the outer surface of each ring in either a standard scroll lacing pattern or, as shown in figure 21, in a side-by-side (staged) lacing pattern. The rings are mounted to the central drive shaft through elastomeric bushings and are separated from each other by 1/2-in air gaps. The bushings isolate the cutting head (and the remainder of the machine) from all coal cutting exitation forces above 140 Hz. Theoretically, this would render coal cutting noise insignificant when compared with noise from other sources on the miner (see figure 1). This cutting head is now being fabricated for testing in an operating underground mine.

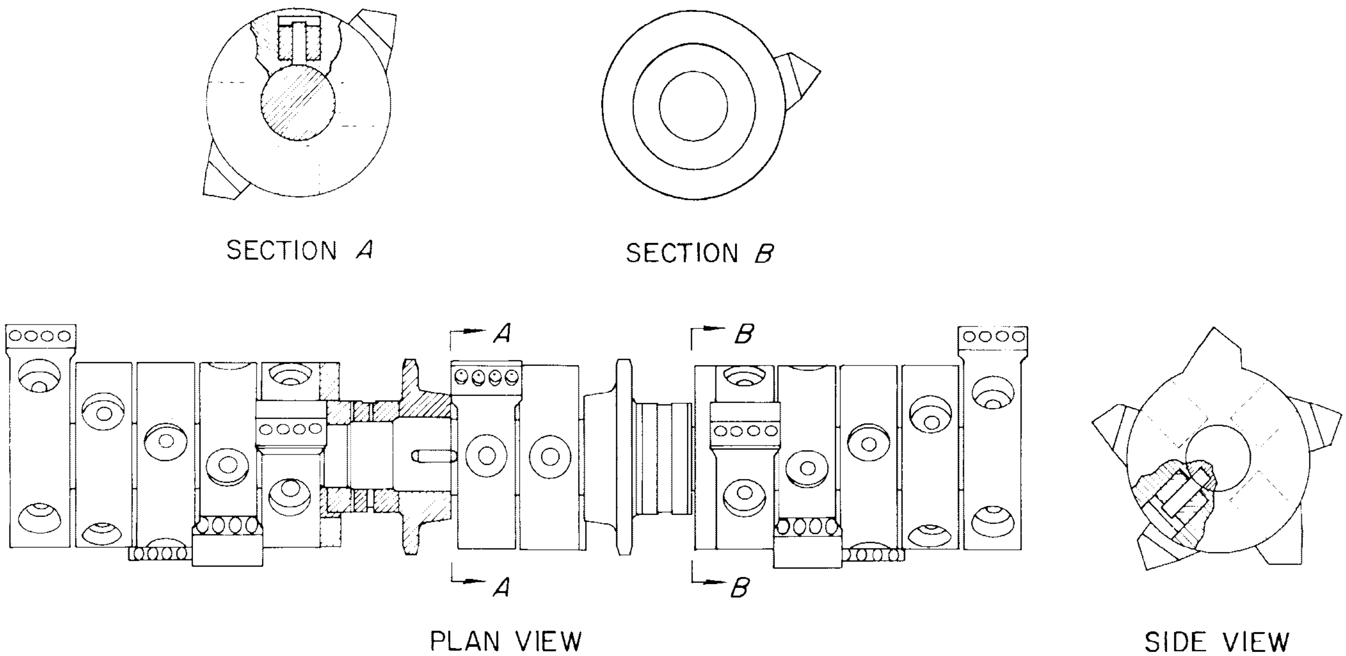


FIGURE 21. - Continuous miner cutting head with isolated cutting tools.

CONCLUSION

Investigations into the mechanics and noise associated with coal cutting have revealed three basic methods for reducing coal cutting noise. First, the cutting bit velocity should be kept as low as possible; second, the cutting head structure should have a very high natural frequency and be highly damped; third, the coal cutting forces should be isolated from the remainder of the cutting head structure. The first method can be used on any type of coal cutting machine. The second method has been used successfully by the Bureau on an auger-type continuous miner and is now being tried on a longwall shearer drum. The third method will soon be tested underground on a standard drum-type continuous miner.

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