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NOISE CONTROL TECHNOLOGY

FOR

SELECTED WOODWORKING MACHINERY

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Methods to minimize noise produced by vibrations and aerodynamic phenomena in the woodworking industry (SIC-2511) are examined. Sources of woodworking machinery noise and associated control procedures are described. The control of workpiece vibration noise, tool vibration noise, and machine vibration noise is considered. It is noted that while total acoustical enclosures are useful in many instances, they can be cumbersome and yield reduced efficiencies in certain cases. Two cases for which considerable difficulty arises in the use of total enclosures are the surfacing of long materials where it is not practical to totally enclose the entire source, and the circular sawing operation for which total enclosures are often impractical. These two cases are discussed in considerable detail. Scaling laws for long cutterheads are provided, and noise reduction due to parameter changes are considered. Some of the results of an ongoing parameter study concerning the aerodynamic noise produced by circular saw blades are reviewed. A noise control guide for woodworking machinery is included. This guide relates typical noise sources to the various abatement techniques presented in the text.

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PREFACE

This report on noise control technology for selected woodworking machinery has been prepared through a grant (OH00417-02) from the National Institute for Occupational Safety and Health. The purpose of the report is twofold: (a) to propose new control techniques for woodworking machinery noise produced by structural vibration and aerodynamic phenomena and (b) to establish a unified approach to the utilization of existing technology for woodworking machinery noise control. Primary emphasis is placed on techniques that supplant the dependence on total acoustical enclosures. Enclosures can be used for many applications, but such an approach can be cumbersome and result in reduced efficiency in some cases. Two cases for which considerable difficulty arises in the use of total enclosures are the surfacing of long materials where it is not practical to totally enclose the entire source, and the circular sawing operation for which total enclosures are often impractical. These two cases are treated in considerable detail in this report. The production of noise by aerodynamic sources is also considered, and empirical scaling laws for both cutterheads and saw blades are presented. Some of the results of an ongoing parameter study concerning the aerodynamic noise produced by circular saw blades are reviewed.

The methodology described in this report is extended to various other woodworking machinery applications through a noise control guide which relates typical noise sources to the various abatement techniques presented in the text.

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

LIST OF FIGURES

LIST OF FIGURES (continued)

Page

Page

I. INTRODUCTION TO WOODWORKING MACHINERY NOISE SOURCES AND CONTROL PROCEDURES

Noise Generation Mechanisms

Noise radiation by woodworking machinery can be classified as resulting from two main mechanism categories -- structural vibration and aerodynamic sources. There is considerable variation in different machines as to which component in the system usually dominates the noise energy produced. These sources are classified and discussed in succession.

Noise Due to Structural Vibration

There are three main system components that can generate noise: workpiece vibration which for our purposes is considered to be part of the machine/process system, cutting tool vibration, and vibration of machine frame members and cover panels. Vibrations of these structures produce sound by causing air disturbances propagated from the surface of the structure. The impulsive nature of the wood machining operation results in favorable conditions for structural vibration produced noise.

Workpiece Vibration. The usual woodworking process involves passing a wooden workpiece by a tool (or moving the tool by a stationary workpiece) for the purpose of cutting, milling, or shaping. These processes require the removal of material which involves an energy transfer between the cutting unit and the workpiece. The operational speed of many woodworking machines is such that excitation frequencies occur that can match resonance frequencies of the workpiece resulting in large amplitude vibrations along with the attendant noise generation.

Cutting Tool Vibration. There are relatively few cases in which vibration of the cutting tool itself is a major source of noise. For the most part, tool vibration cannot be permitted due to performance requirements. Further, the dimensions, natural resonances, and excitations combine to produce very low acoustic radiation efficiencies. There is, however, one notable exception — the circular saw blade which can produce significant noise due both to forced vibration and resonant response.

Machine Vibration. The vibration of structural components of machines can be a major source of noise. The sources of excitation for machine vibration include: 1) forces transmitted to the machine frame from the cutting tool, 2) forces generated due to rotating unbalances, and 3) forces due to worn or improper meshing of gears.

Noise Due to Aerodynamic Phenomena

Aerodynamic noise is generated when fluctuations about the normal atmospheric pressure are caused to occur. Aerodynamic noise includes noise generated as a result of instabilities in such airflow conditions as turbulence in wind, high velocity mixing as in turbulent jets, and flow over surfaces either moving or stationary[1].

There are several types of aerodynamic noise sources that can be identified with woodworking machinery. When the mass of a fluid (air) in a fixed region of space is caused to fluctuate, the noise producing source is called a simple source or a *monopole*. A second noise source classification is the *acoustic dipole* which conceptually can be thought of as two monopoles or simple sources operating close together 180° out of phase. Physically, this source results when a force varying in magnitude of direction causes the momentum (mass times velocity) in a fixed region to vary with time. The third source category is the *acoustic quadrupole* which can be thought of as the combination in close proximity of two dipoles. Each of the source mechanisms discussed can occur in woodworking machines and are usually associated with rotation of cutting tools.

Free Rotating Cutting Tools. Cutting tools usually incorporate irregular surface geometries, sharp edges, and cavities which move rapidly through the air. Based on the preceding discussion, it is not difficult to visualize the kind of fluctuations associated with the simple source, the dipole source, and the quadrupole source.

Rotating Cutting Tools and Nearby Stationary Surfaces. Stationary surfaces close to rotating cutting tools are a common situation in woodworking machinery. The clearance between cutting edges and fixed machine components can have a dramatic effect on noise generation.

Rotating Cutting Tools with Induced Air Flow. Most woodworking machinery installations have chip/dust removal systems. Air is drawn from around the cutting area by a vacuum system, and the chips/dust are entrained in the air and conveyed to central depositories. This air flow over cutting tools can have an effect on noise production, causing the noise to increase.

It is clear that the irregular and complex geometry found in cutting tools, along with the presence of stationary surfaces with imposed airflow, have the potential for producing noise through all known mechanisms. The understanding of which source mechanism dominates noise generation in a given situation is, of course, an aid in the noise control procedure.

Noise Control Methodology

A general discussion has been given of noise generation mechanisms. Before considering each noise generation mechanism and control method in detail, an overview summarizing the methodology set forth in this report is presented.

Structural Vibration Noise Reduction

Noise associated with the vibration of workpieces, cutting tools, and machine components can be controlled by reducing vibration amplitudes and reducing the effectiveness of the system converting vibratory energy into acoustic energy. The means employed to accomplish this include: a) changes in system stiffness and mass for controlling vibration amplitude. b) Application of a damping system to reduce the amplitude of vibration. A portion of the input energy which would otherwise increase the vibration amplitude is then dissipated due to damping. c) The use of mechanical vibration suppression to inhibit flexural wave propagation. d) Detuning the vibratory system by either modifying system resonances or changing the exciting frequencies. e) Isolation of the source of vibratory energy from radiating machine components.

Aerodynamic Noise Reduction

The three main categories of aerodynamic noise in the woodworking industry are: 1) long rotating cylinders with nearby stationary surfaces, 2) intermediate length cylinders with or without stationary surfaces, and 3) thin disks of relatively large diameter such as circular saw blades. The parameters that have the most dramatic effect on aerodynamic noise for long cylinders (cutterheads) are reduction in tip speed, an increase in clearance between knife tips and table lip or other stationary components, and the modification of the table lip geometry. The intensity of aerodynamic noise generated by long cylinders depends on tip speed and varies between a fourth power dependence and a sixth power dependence depending on operational conditions[2].

Intermediate length cylinders are defined as those with diameter to length ratios between 1 and 10. For these cases, the effective knife projection from the cylinder body has a significant effect on noise in addition to the clearance between the cutterhead and nearby stationary surfaces.

The limiting case of short cylinders is the thin disk which for this discussion is the circular saw blade. Aerodynamic noise from saw blades depends on speed of rotation, tooth and gullet geometry, and blade thickness. While the optimum saw blade from the standpoint of noise is not defined, it has been determined that a favorable relationship between gullet width and blade thickness is of major importance in the reduction of aerodynamic noise[3].

Noise Reduction through Tooling Changes

Significant developments in tooling design have provided dramatic reductions in noise due both to vibration and aerodynamic sources. These tooling changes are based on the recognition that the intensity of noise radiated depends on the time rate of change of air mass fluctuation, force fluctuation, and correlation of such fluctuations along extended surfaces.

New tooling concepts have been developed for noise control of woodworking machinery, and it can be expected that additional developments will become available as new technology results from continuing research efforts. In some cases, a tooling change may be the most satisfactory alternative; in other cases, a combination of tooling changes and acoustic enclosures may be most appropriate. There are situations where acoustic enclosures alone give satisfactory results. The specific decision to use any of these combinations depends on a number of factors and should be made only after all suitable alternatives have been explored.

The following sections of this report deal quantitatively with procedures that have been found to reduce woodworking machinery noise. In Section II, vibration generated noise is treated while aerodynamic noise is discussed in Section III. Section IV is a compendium of procedures for noise control of selected machines and represents the current state-of-technology for reduction of noise from woodworking machinery.

II. WOODWORKING MACHINERY NOISE PRODUCED BY STRUCTURAL VIBRATION

Many woodworking processes involve engaging a workpiece with a rotating cutting tool to be cut, surfaced, or shaped. The result is usually structural vibration of the workpiece, the tool, or both. Under certain conditions, this structural vibration couples with the surrounding air to produce sound. The process involved is illustrated in Figure 2.1 for sound produced by vibration of the workpiece.

Figure 2.1 depicts an excitation/response/radiation flow system. The excitation is provided by the rotating tool which usually incorporates a number of cutting blades or teeth. Excitation energy is available over a large frequency range since energy contributions occur at the rotational frequency, the blade passage frequency, and various harmonic frequencies depending on the particular force-time characteristics of the excitation. As the rotating cutting tool makes contact with the workpiece, an energy transfer occurs resulting in vibrational energy being delivered to the workpiece. The magnitude of this energy transfer depends on a number of factors involving the matching between excitation and structural response characteristics. The transfer is usually quite effective in systems where the excitation frequencies of the energy input overlap natural resonant frequencies of the workpiece. Structural vibration is easily excited by external force systems with energy contributions near natural frequencies of the workpiece. Vibratory energy is also transmitted to the workpiece due to the excitation force when the frequencies contained in the input energy do not match the resonance frequencies.

Energy stored in the workpiece in the form of structural vibration is dissipated by mechanical damping (internal or through physical contact with other structures) and through a transfer of energy to the surrounding air. The effectiveness of this energy transfer in the form of radiated sound depends on the degree to which vibrations of the workpiece produce compressions in the surrounding air that propagate as sound. This coupling between the vibrating workpiece and the air depends on the relationship between the structural vibrational wavelength and the acoustic wavelength at the same frequency. Structural wavelengths which are longer than the acoustic wavelength result in the radiation of sound. In cases where the acoustic wavelength is greater than the structural wavelength, a hydrodynamic short circuit is developed (air is not compressed) and no sound is radiated. The frequency at which the wavelength in the structure is equal to the wave length in air is known as the critical frequency.

The system described in Figure 2.1 consists of a cutterhead and a workpiece which is set into vibration by the impact of blades mounted in the rotating cutterhead. As noted previously, blade impact results in a large amount of energy available to the workpiece over a wide range of frequencies. The critical frequency of the workpiece is usually relatively low, making the conditions for sound radiation quite favorable.

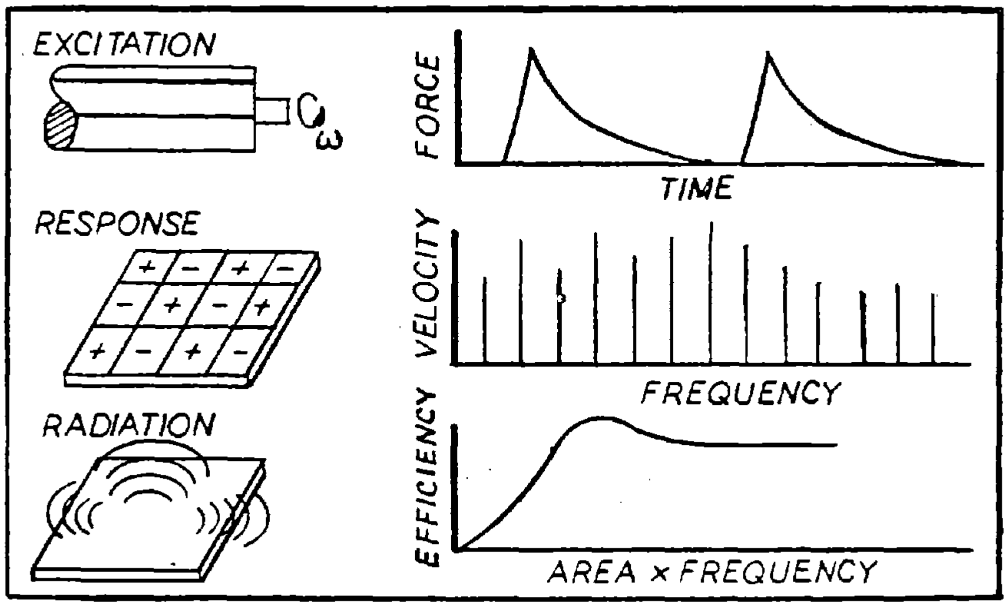

Figure 2.1 Noise produced by workpiece vibration

A second excitation/response/radiation system is illustrated in Figure 2.2 in which the tool itself experiences a vibration response. The tool illustrated in Figure 2.2 is the circular saw blade, commonly found in the woodworking industry. The excitation possibilities are numerous, some of the more important being aerodynamic, rotational dissymmetry, and tooth impact. Obviously, the excitation energy input encompasses a wide range of frequencies, while the response of the tool to these energy sources depends primarily on tool geometry. The circular saw blade has many natural resonant frequencies in the audible range which account for a strong vibrational response and accompanying radiation of sound.

In this discussion, cases in which either workpiece vibration or tool vibration clearly dominate the sound production have been considered. Implicit in the discussion is the assumption that vibration energy is stored throughout the tool or workpiece and dissipated through damping mechanisms or the radiation of sound. For this situation to exist, only a relatively small amount of damping can be present in the system. In effect, this assumption allows the additional radiation from the flexural nearfield in the immediate vicinity of the tool/workpiece system to be neglected in comparison to other sources. In cases where neither workpiece nor tool vibration sources dominate, sound radiation from the flexural near field of the tool and workpiece must be considered. This case is illustrated in Figure 2.3.

The sources illustrated in Figures 2.1, 2.2, and 2.3 occur to varying degrees in most wood machining processes. In arriving at effective techniques for controlling structural vibration and the resulting noise, it is essential that the various elements of the excitation/response/radiation system be properly identified and understood.

Control of Workpiece Vibration Noise

Workpiece vibration produced noise is characterized by Figure 2.1 and results from the acoustic coupling of vibrations of the workpiece with the surrounding air. Means by which this noise can be reduced include: 1) reduction of workpiece excitation, 2) reduction of workpiece response, and 3) reduction of workpiece radiation. Reduction of excitation involves changes in operational conditions or tooling while reduction of workpiece response involves the use of restraint systems or damping techniques since the workpiece itself usually cannot be altered. The reduction of workpiece radiation in this context refers to acoustic shielding techniques. It is recognized that radiation characteristics can also be altered by changes in the system mass and stiffness.

Reduction of Workpiece Excitation

The major source of workpiece excitation is the energy transfer that occurs when the rotating cutterhead impacts the workpiece. The workpiece is usually relatively weak structurally in one or more planes, resulting in easily excited vibration response. Since alterations of the workpiece are difficult at best, the reduction of excitation is of major importance in reducing radiated noise.

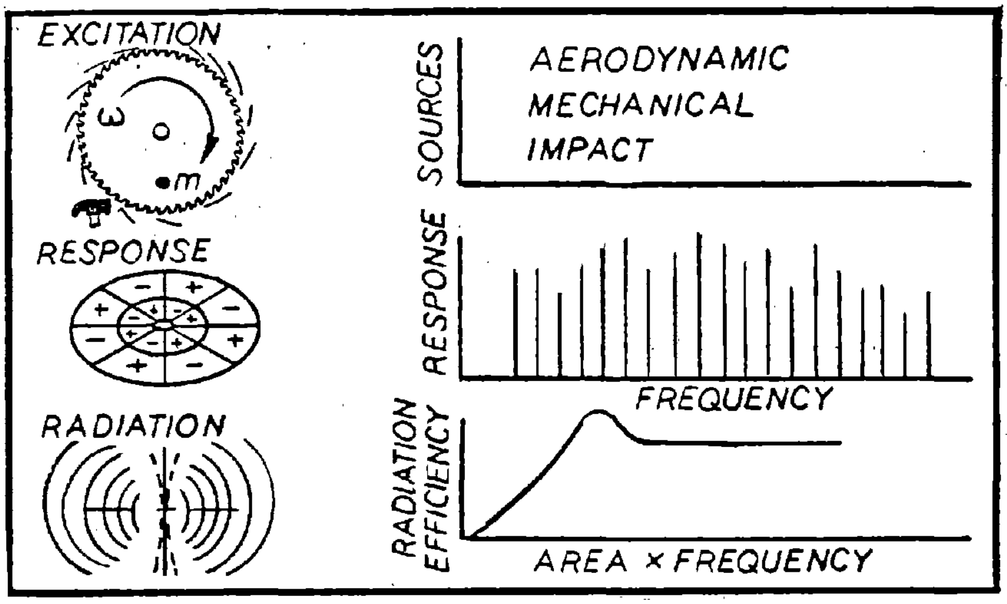

Figure 2.2 Noise produced by cutting tool vibration

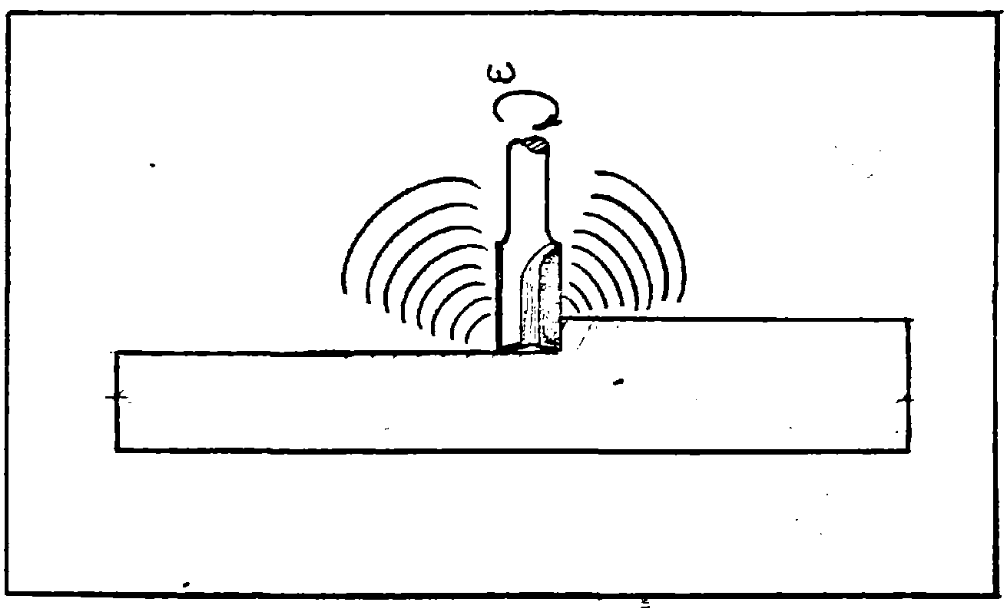

Figure 2.3 Near field noise radiation

The mechanism of energy transfer from the cutterhead to the workpiece which results in flexural vibrations and sound radiation is illustrated in Figure 2.4. For conventional woodworking cutters consisting of a circular cylinder with knives mounted in the periphery, impact occurs as each knife engages the workpiece (see Figure 2.5). The force experienced by the workpiece is periodic and occurs at the blade passage frequency (N RPM/60; N = number of knives and RPM = revolutions per minute). The shape of the force pulse is, in general, unknown but can be represented by a Fourier series for the force $F(\omega)$ having energy contributions at the blade passage frequency (ω_o) and harmonic frequencies ($n\omega_o$) as

$$F(\omega) = \sum_{n=0}^{\infty} A_o(n\omega_o)\delta(\omega - n\omega_o) \tag{2.1}$$

where A_o* governs the amplitude and δ is the dirac delta function. Thus, for any type of periodic impact, the resulting force pulse can be subdivided into a series of pure tone signals which are harmonically related, i.e., all frequencies are multiples of the fundamental frequency. In this case the contributing frequencies are given by

$$f_n = (N\ RPM/60)n$$

The amplitude at each frequency is the Fourier transform of the signal divided by the period ($F(\omega)/T$).

The reduction of $F(\omega)/T$ involves an assessment of the variation in $F(\omega)$ with T since the two are not independent. The quantity $F(\omega)$ involves momentum upon impact and is known to decrease with decreasing period (T). Several of the important factors affecting $F(\omega)/T$ are listed in Table 2.1.

Table 2.1 Factors Influencing Excitation

Force pulse amplitude
Force pulse duration
Force pulse period

The effect of each of the parameters listed in Table 2.1 can best be described by considering a hypothetical force-time history, shown in Figure 2.6*.

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*The case under consideration is that of the conventional straight knife cutterhead commonly found in the woodworking industry.[3] The general nature of the force-time history is known for this case, however, other type cutters, abrasive wheels, drums, sanders, etc., can be treated in a similar way provided the nature of the force-time history is known.

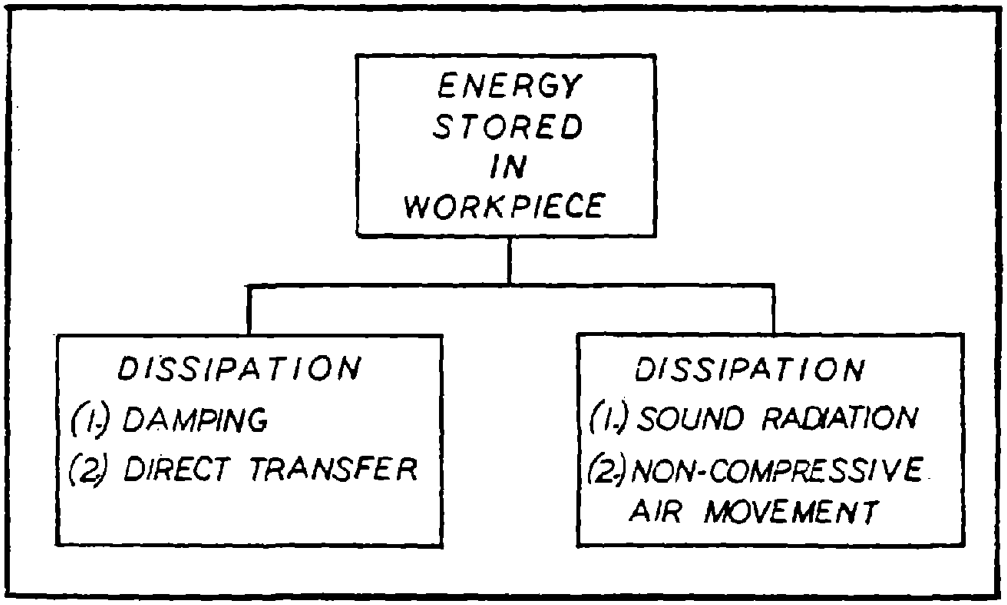

Figure 2.4 Energy dissipation mechanisms

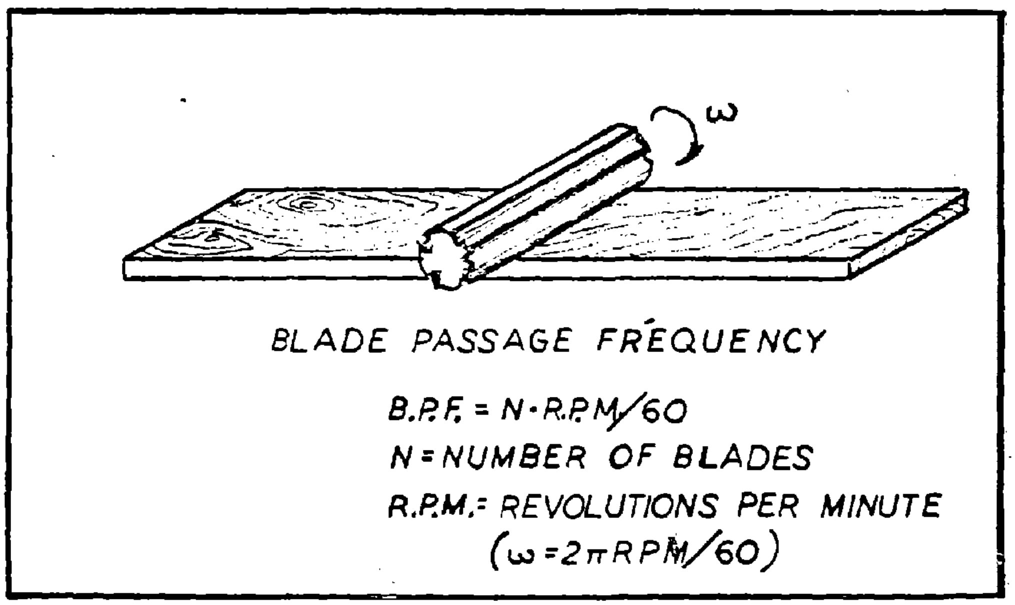

Figure 2.5 Cutter-workpiece interaction

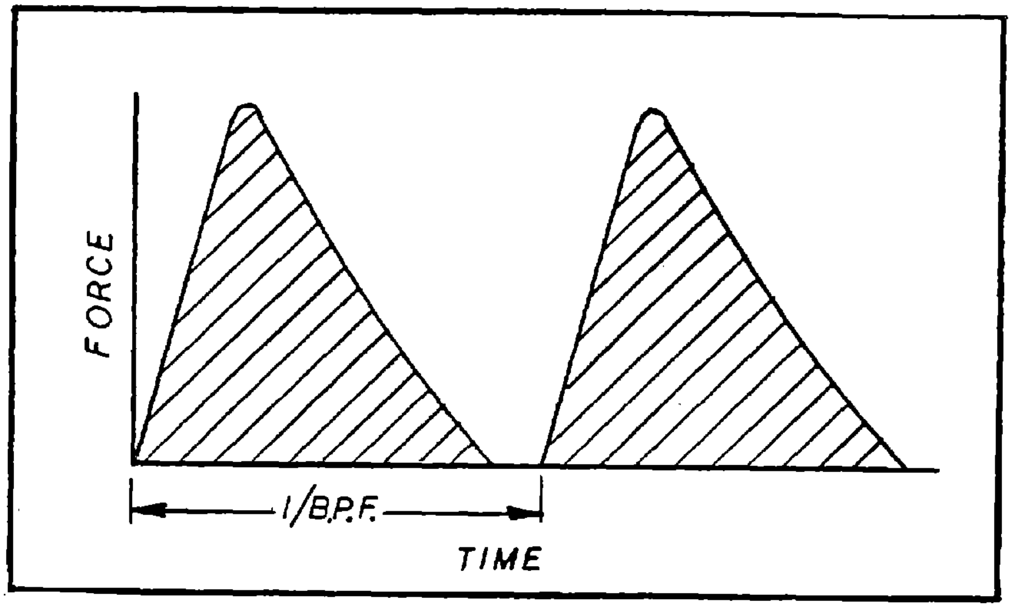

Figure 2.6 Excitation force-time history

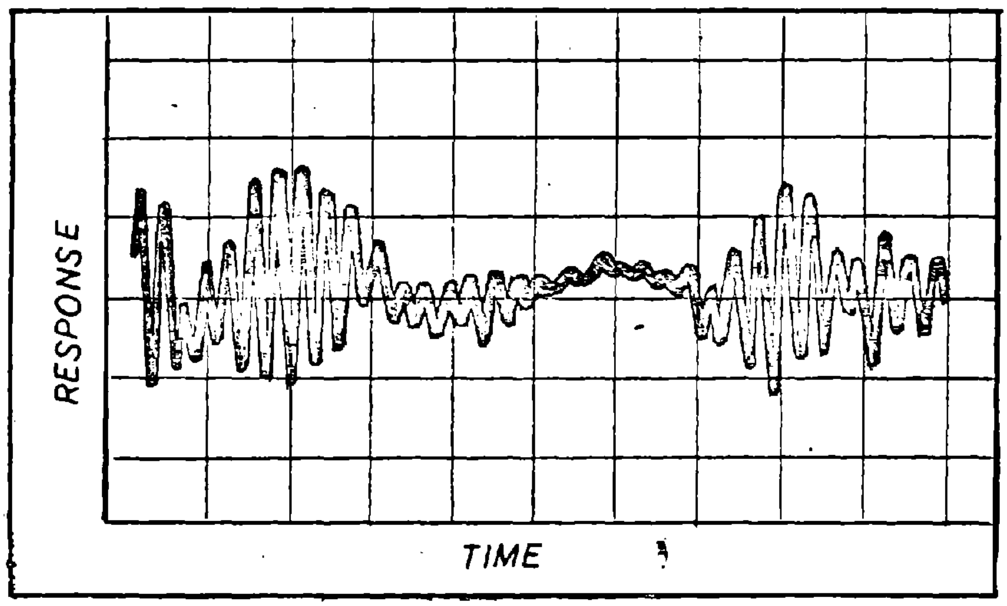

Figure 2.7 Response-time history of chip breaker

The variation in force with time (df/dt) is of considerable importance and in some cases is actually of greater importance than the force magnitude. As pointed out, the exact shape of the force pulse is not known; however, it has been determined experimentally that the pulse duration is related to the individual knife engagement time and the pulse spacing corresponds to the blade passage frequency. An oscilloscope trace[5] taken from the chipbreaker mechanism (which is in direct contact with the workpiece) is shown in Figure 2.7. The superimposed oscillation corresponds to a resonant frequency of the chipbreaker which was high enough to provide a reasonably clean trace.

It is apparent that reduced excitation of the workpiece requires a reduction in the time rate of change of force acting on the workpiece. This can be accomplished through a reduction in impact magnitude or a more continuous cutting action. An approach that effectively smooths out the force-time variations acting on the workpiece is abrasive processes, such as sanding, which essentially results in a zero df/dt due to stock removal. Another approach is the spiral (helical) cutterhead design[4] shown in Figure 2.8.

In reference 4 the vibrational velocity of the workpiece due to cutterhead impact was expressed in terms of the fluctuating force for straight knife and helical cutterhead designs. For the straight knife cutterhead, the fluctuating force was found to vary directly with workpiece width ($F_{rms} \sim w$) as contrasted to the helical case where the fluctuating force does not depend directly on workpiece width. The magnitude of the fluctuating force for each case depends on operational conditions as well as the factors listed in Table 2.1. Figure 2.9 illustrates the improvement in the force-time history for the helical design as opposed to a straight knife cutterhead. The importance of multiple point contact in the helical design is illustrated in Figure 2.10 which shows the fluctuating force acting on the workpiece as a function of a contact ratio (workpiece width (w) to the distance between adjacent knife tips (L)). For helical cutterheads it is necessary to distinguish between cases in which more than one knife is engaged in the cut and cases in which only one knife is in contact at any given instant. The condition that more than one knife be fully engaged is satisfied whenever w/L is greater than unity. Figure 2.10 indicates that for effective workpiece vibration control the ratio of w/L should be greater than unity and ideally as large as possible.

It is possible to set a lower limit on the distance L by examining the radiation efficiency of workpieces of various widths radiating near and above the critical frequency. Utilizing a simple piston radiation model[5], it can be shown that values of the ka parameter (the product of frequency and workpiece width) of about 2 or less result in a poor acoustical efficiency. This is illustrated in Figure 2.11 which compares the radiation characteristics of workpieces of various widths. Thus, cutterhead design should be such that $w/L > 1$ for all workpiece widths of efficient radiating area.

Experimental results reported in reference 4 indicate a substantial decrease in workpiece vibration and a corresponding reduction in sound generation. Frequency spectra are presented in Figures 2.12 and 2.13 comparing the helical cutterhead with a conventional straight knife cutterhead. Workpiece vibration levels produced by this helical cutterhead are substantially reduced,

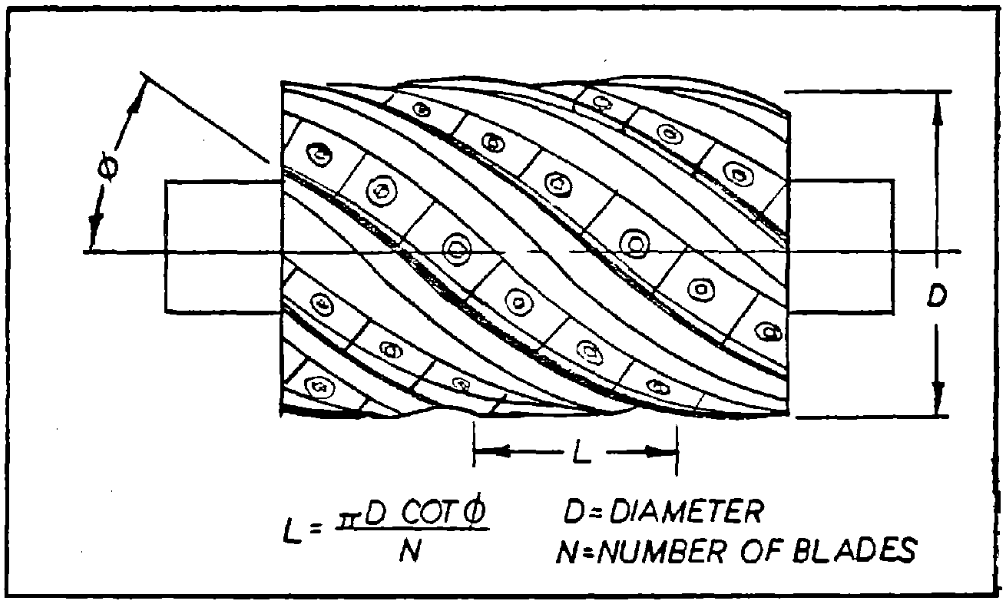

Figure 2.8 Helical cutterhead

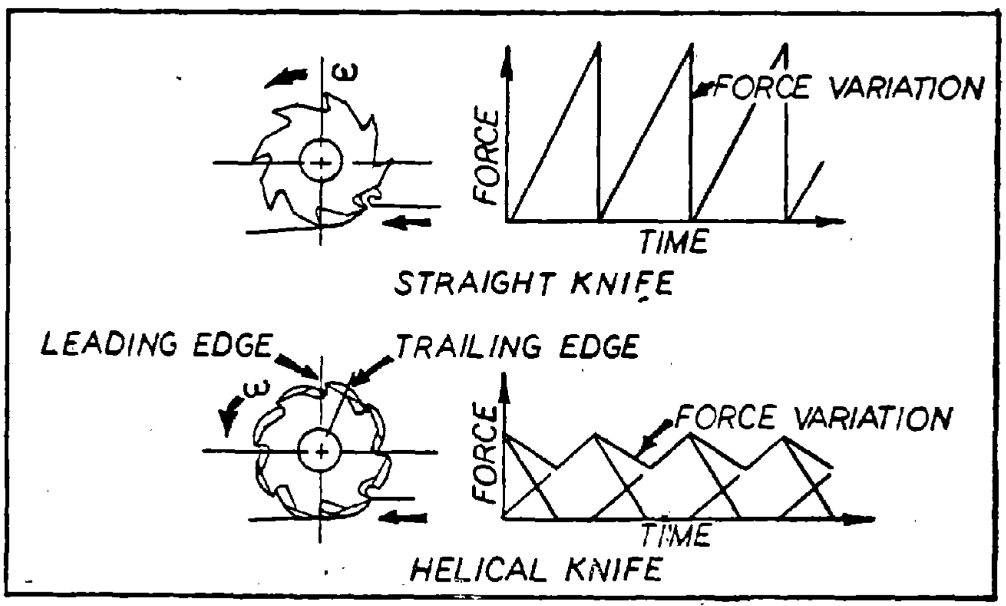

Figure 2.9 Comparison of force-time fluctuations

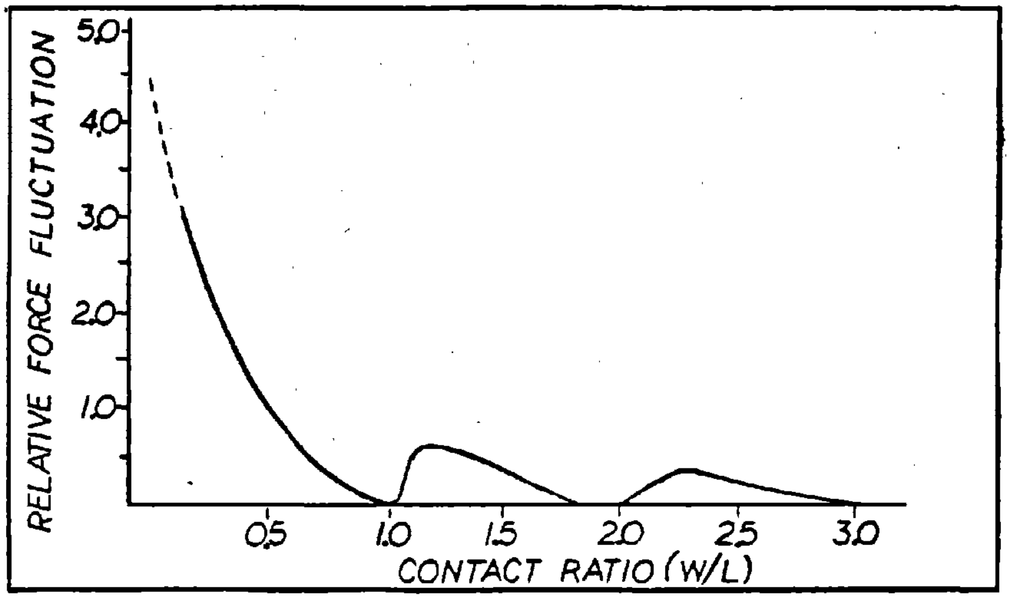

Figure 2.10 Cutting force fluctuation versus contact ratio

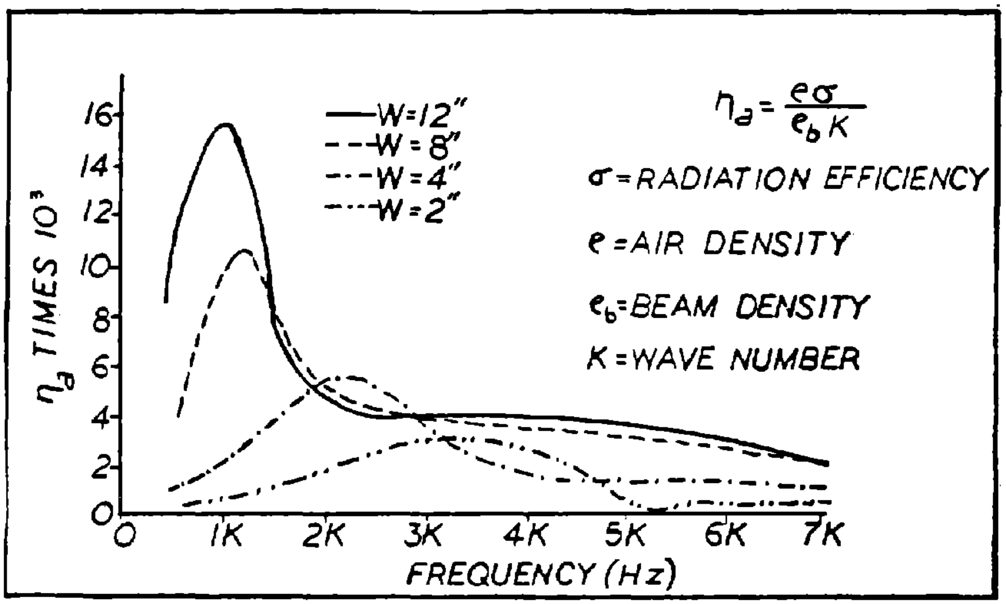

Figure 2.11 Workpiece radiation versus frequency

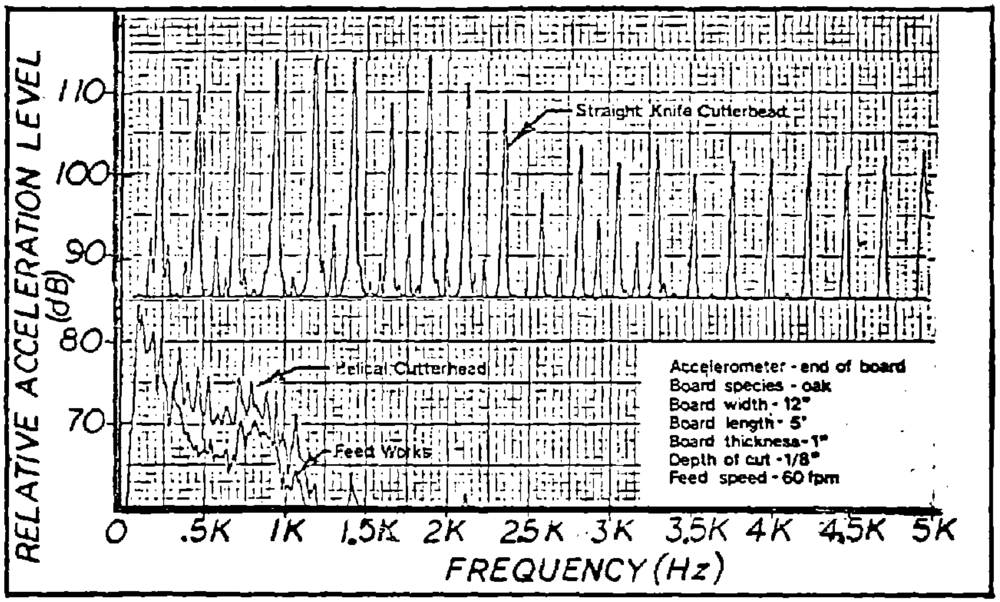

Figure 2.12 Comparison of workpiece vibration spectra

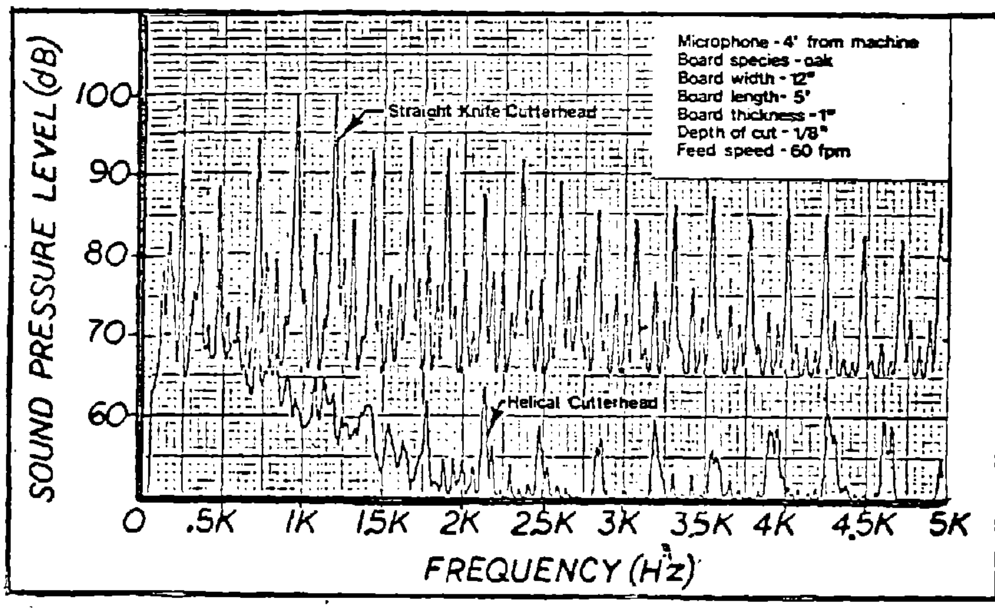

Figure 2.13 Comparison of sound spectra

and frequency spectra do not exhibit the resonant workpiece response near multiples of the knife passage frequency. It is noted that the vibration spectrum for the helical case is only slightly different from the vibration spectrum produced by the action of the feeding mechanism alone.

The radiated acoustic power for straight knife and helical cutterheads is given as[4]

$$W_{straight\ knife} = \rho F_{rms}^2/\eta\rho_m^2 t^2 C_B \qquad (2.2)$$

$$W_{helical} = \rho\omega w F_{rms}^2/4\rho_m^2 t^2 C_B^2 \eta \qquad (2.3)$$

where $C_B = \{\omega^2 E t^2/12\rho_m\}^{\frac{1}{4}}$; (the bending wave speed),
ρ = density of air,
F_{rms} = fluctuating force acting on workpiece,
η = damping factor,
ω = circular frequency,
E = Young's Modulus,
t = workpiece thickness,
ρ_m = density of workpiece,
w = workpiece width.

For the straight knife case, Frms is proportional to workpiece width so that

$$F_{rms}^2 \sim Kw^2, \text{ where } K = \text{constant}.$$

The power expression from equation 2.2 becomes

$$W_{straight\ knife} = \rho K w^2/\eta\rho_m^2 t^2 C_B^2 \sim Kw^2. \qquad (2.4)$$

For the helical case, the fluctuating force (Frms) does not depend directly on workpiece width but varies with the contact ratio w/L (Figure 2.10). For cases where w/L is greater than unity, the fluctuating force shown in Figure 2.10 can be approximated by a constant K_H so that equation (2.3) becomes

$$W_{helical(w/L>1)} = \rho\omega w K_H/4\eta\rho_m^2 t^2 C_B^2 \sim K_H w \qquad (2.5)$$

Sound pressure levels for these two cases differ from power levels by a constant depending only on room characteristics. Neglecting the operational constants K and K_H, the expressions for sound pressure level become

$$L_{p(\text{straight knife})} \sim 10\log_{10}(w^2/w_o^2) \sim 20\log_{10}(w/w_o) \qquad (2.6)$$

$$L_{p(\text{helical } w/L>1)} \sim 10\log_{10}(w/w_o) \qquad (2.7)$$

where w_o = reference width. The predicted 6 dB increase in sound pressure level for a doubling of workpiece width in the straight knife case, and the 3 dB per doubling for the helical case (w/L>1) are shown in Figure 2.14. The effect of aerodynamic and other secondary sources is apparent as well as differences in level magnitude governed by the constants K and K_H.

Field test data for a production version of the helical cutterhead are compared with noise levels produced by conventional cutterheads in Table 2.2.

Table 2.2 Comparison of Noise Levels Present at the Operator Position for Planers Equipped with Standard and Helical Cutterheads

Material Width	Standard Cutterhead (dBA)	Helical Cutterhead (dBA)
idle	91	78
2"	90	81
4"	95	81
6"	98	80
8"	101	81
10"	103	82
12"	104	85
24"	108	87

Some of the advantages of continuous contact between the cutterhead and the workpiece present in the helical design can be obtained by utilizing small sections arranged to form a breakup type unit. The reduction in excitation is usually less for these type heads than for the true continuous helical design. In instances where multiple point contact cannot be practically achieved with a helical design, a breakup unit consisting of a number of thin saw type cutters stacked together to form an apparent helix is quite effective. Such a cutterhead is shown in Figure 2.15. An important consideration in the design of segmented type cutters is the effective contact length of each segment and the number of segments acting on the workpiece at any given instant.

The means employed to reduce excitation of the workpiece reported in this section have concentrated on alteration of the force-time history by smoothing out the total fluctuating force acting on the workpiece. Other techniques include reductions in RPM and diameter to reduce impact velocity and increased

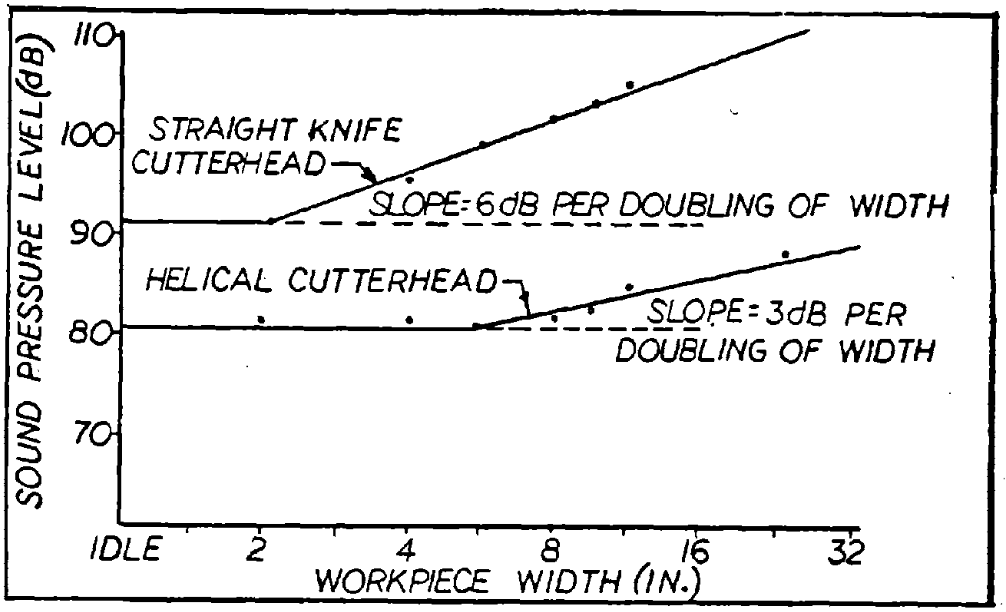

Figure 2.14 Sound pressure level versus workpiece width

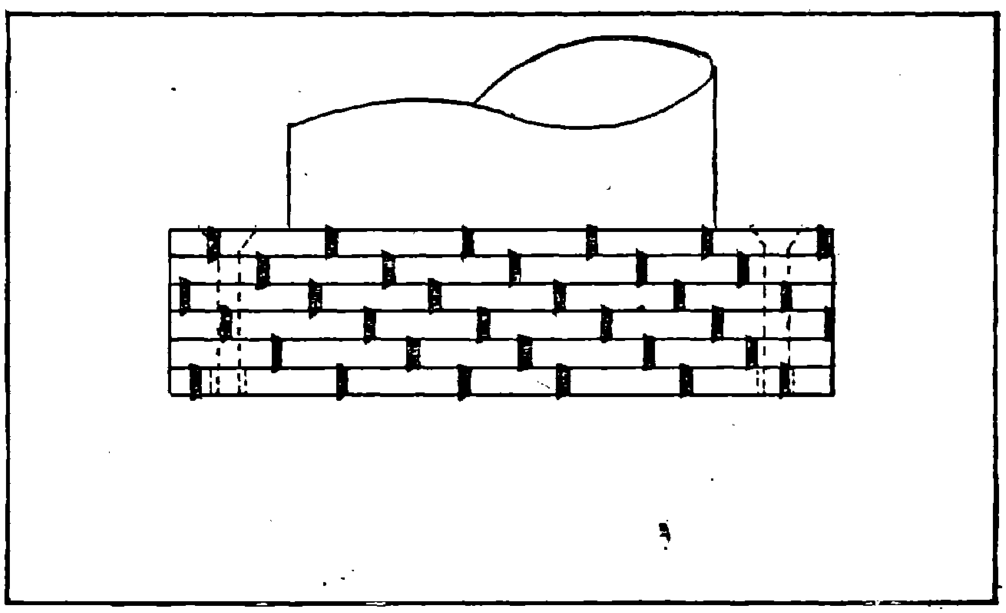

Figure 2.15 Segmented cutterhead

edge sharpness to minimize the resistance upon impact. Techniques for reducing input excitation through resilient cutterhead knife mountings, etc., are also possible; however, the effect of such alterations on cutterhead performance and maintenance characteristics must be carefully considered. From a practical standpoint, the best means of reducing workpiece excitation appears to be reduced force fluctuation achieved by continuous multipoint cutting action. Such cutting characteristics can be achieved through cutterhead design changes described in this section or by abrasive processes.

Reduction of Workpiece Response

Workpiece response to excitation occurs at vibrational frequencies governed by the physical characteristics of the workpiece, the nature of the support system, and the nature of the excitation. For the special case of a uniform beam subjected to a periodic force signal, the response can be expressed as[5]

$$Y(x,\omega) = 2/M \sum_{n=1}^{\infty} \frac{\sin(n\pi x/\ell)\sin(n\pi x_o/\ell)}{\omega_n^2} \left\{\frac{1}{(1-(\omega/\omega_n)^2)^2+\delta_i^2}\right\} F(\omega) \qquad (2.8)$$

where

$Y(x,\ )$ = displacement at position (x) and frequency (ω),
M = beam mass,
ℓ = beam length,
x = lengthwise coordinate,
x_o = location of force application,
ω_n = natural frequency of nth mode,
δ_i = internal damping,
$F(\omega)$ = Fourier representation of forcing function

This steady state solution contains harmonic components at each of the forced frequencies present in the Fourier series representation of the excitation force. The term $\sin(n\pi x/\ell)$ is the expected sinusoidal variation in the response, while the term $\sin(n\pi x_o/\ell)$ represents a suppression of frequencies in accord with the location of the force on the beam. The excitation term $F(\omega)$ dictates the frequencies at which the response takes place. For periodic excitation, such as that found in most cutting operations, the excitation pulse can be resolved into a series of pure tone components which are harmonically related, i.e., all frequencies are multiples of the fundamental frequency. In the case of a straight knife cutter, this corresponds to vibration excitation at the blade passage frequency and integer multiples of this frequency. From equation (2.8) the vibrational amplitude is seen to depend on the beam mass (M), the proximity of the forced frequency components to natural frequencies (ω/ω_n), and the damping present in the system (δ_i). Alteration of $F(\omega)$ or ω was considered in the previous section so that only other means of altering responses are considered herein. In practice, it is quite difficult to physically alter the workpiece itself since the end product is of primary importance. Thus, techniques for increasing stiffness, altering frequencies, and increasing energy absorption are limited to those which can be accomplished artificially through changes in excitation, changes in support systems, and contact systems having energy absorption characteristics.

Increased Stiffness

Methods for increasing workpiece stiffness usually involve area contact between rigid support structures and the workpiece. In cases where the workpiece is not "fed through the machine", solid clamping can be used effectively to increase stiffness. For operations in which the workpiece passes through the machine, contact shoes, chain belts, and other stiffening systems can be utilized (Figure 2.16). Unfortunately, nonresilient stiffening systems which make area contact with the workpiece often hinder the feeding operation and seldom achieve true area contact due to surface irregularities in both the workpiece and the stiffening mechanism. Conditions under which vibratory energy is transmitted and reflected are discussed later in this report.

Increased Damping

The response of the workpiece given in equation (2.8) is also dependent on the damping present in the system (δ_i). For practical purposes, this damping can be thought of as resulting from three mechanisms: (a) internal damping, (b) damping due to contact with other structures, and (c) radiation damping. The mechanism of damping is such that for a constant energy input to the system, the vibration response reaches steady state so that the additional energy supplied to the system with time evidently must be transformed into another form. The mechanism of internal damping is generally accepted as having to do with heat conduction between differently strained regions in a material and is defined in terms of a loss factor which depends on the energy dissipated per cycle of vibration. The damping loss factor of several common materials is given in Table 2.3.

Table 2.3 Representative Damping Loss Factors

Material	Loss Factor
Steel	.0004
Oak	.0100
Rubber	1.0000

In addition to limiting the amplitude of vibration response, especially near resonances, damping also directly affects the attenuation of bending waves with increasing distance from the source of excitation. This is an especially important aspect when dealing with extremely long beams having vibration fields

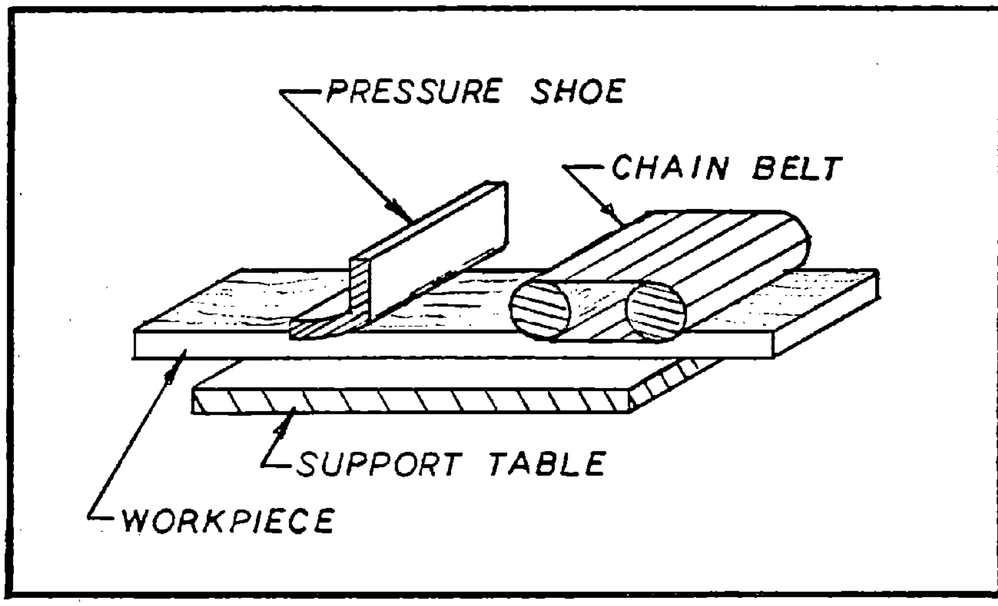

Figure 2.16 External stiffening systems

which cannot be properly categorized as reverberant. The requirements for appreciable attenuation in such cases are high damping, small wavelength (high frequency), and long beam lengths. In most instances, these parameters can only be altered through the use of externally applied systems. Several externally applied damping techniques are listed in Table 2.4.

Table 2.4 Externally Applied Damping Techniques

Attached Layers
Resonant Systems
Air Pumping Systems
Granular Systems
Rattling Systems

The most notable means of achieving large amounts of structural damping is the use of attached viscoelastic layers. These layers take advantage of the energy storage properties of viscoelastic materials when subjected to shear deformation. Such treatments may be single layers, multi-layers, or layers sandwiched between constraining plates. Energy absorption by free (uncovered) layers is primarily due to flexure and extension of the layers. The loss factor of such a system depends on the relative thicknesses and elastic modulus of the viscoelastic layer and is illustrated in Figure 2.17[7].

When a constraining cover plate is attached to the viscoelastic layer, the bending motion also produces shear in the viscoelastic layer. The effectiveness of such a system depends on how well the viscoelastic layer couples the flexural motions of the two plates. In practice, the ratio of plate stiffness to damping layer stiffness, the bending wavelength, and the loss factor of the constrained layer must be considered when using such an arrangement. The damping of such a system is frequency dependent and consequently systems must be designed for each application. The use of the damping techniques presented in this section is complicated by the requirement that the damping system be securely attached to the vibrating structure. The major difficulty with such systems in conjunction with a moving workpiece is attachment so as to insure proper energy transfer. A constrained layer system for possible use in conjunction with a vibrating workpiece is illustrated in Figure 2.18.

In cases where damping cannot easily be achieved through the use of composites, it is possible to transfer vibratory energy into tuned resonators. Such resonators may consist of spring mass systems attached to the surface of the vibrating structure and usually provide damping in a rather narrow frequency band as opposed to the broad band effects possible with composites. Unfortunately, the selection of optimum tuning and damping parameters requires a specific knowledge of attachment points, resonant beam frequencies, etc., which is not usually practical for random workpieces moving through a machine. A detailed treatment of the use of dynamic vibration absorbers for various beam structures is presented in reference 6. Two examples of such systems are properly tuned spring mass systems in contact with the vibrating structure and the attachment of thin plates normal to the vibrational axis of the structure, as shown in Figure 2.19[6]. The broad frequency response shown in Figure 2.19 is obtained by employing damped plates of varying length and indicates a possi-

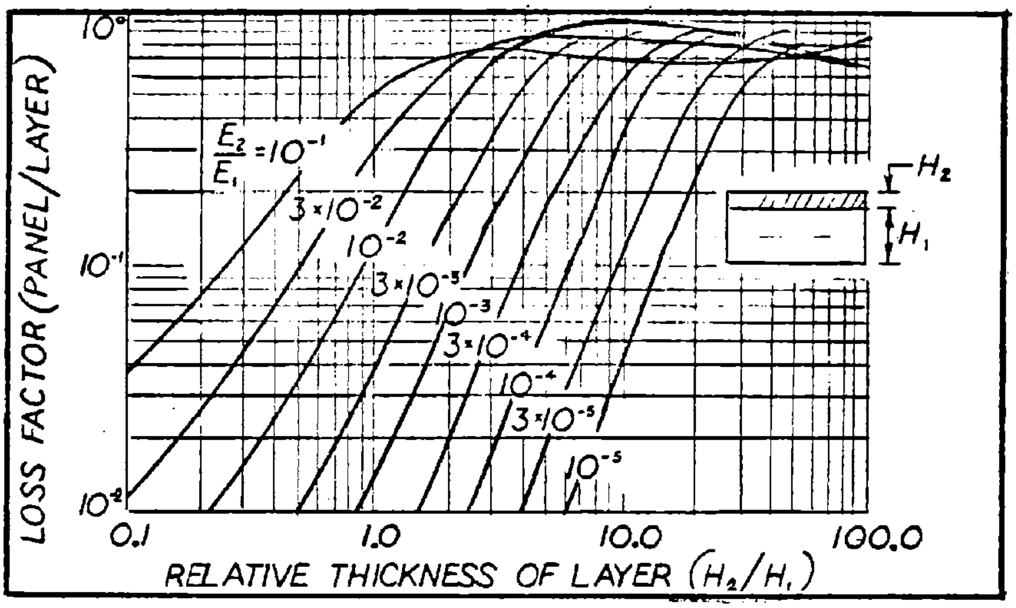

Figure 2.17 Loss factor versus layer thickness (after reference 7)

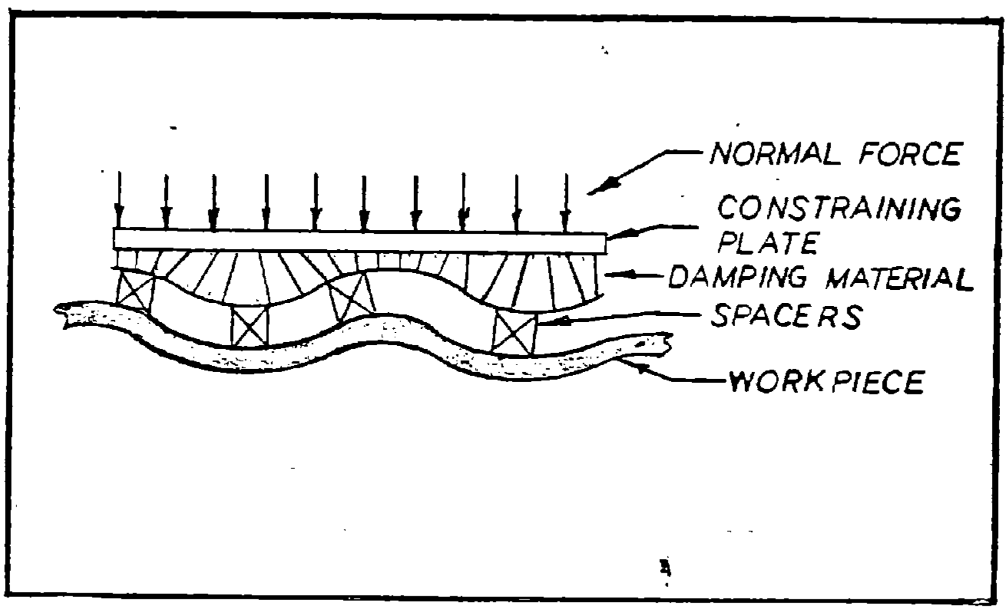

Figure 2.18 Constrained layer damping system

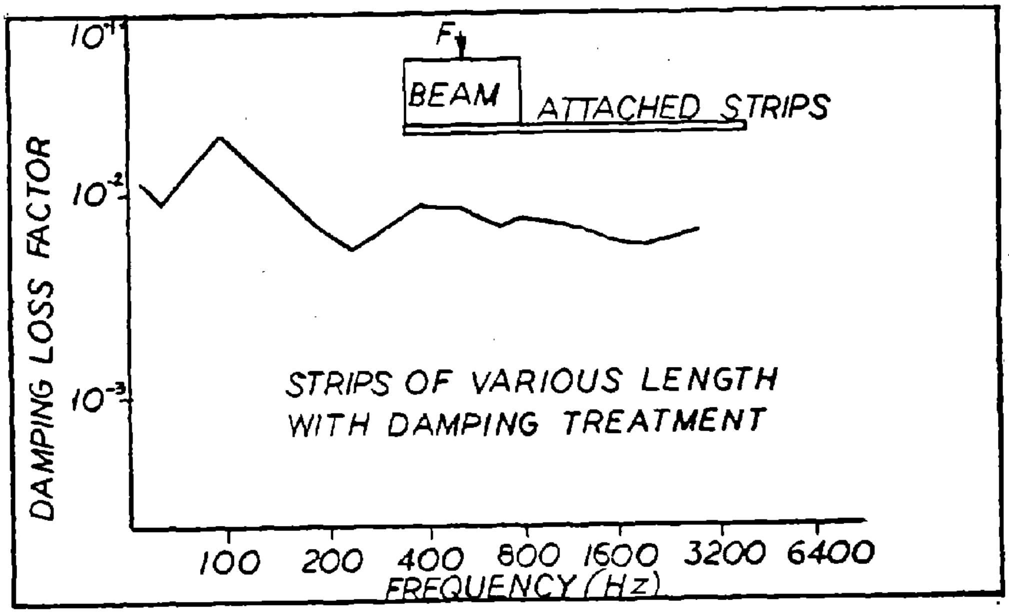

Figure 2.19 Resonant damping system (after reference 6)

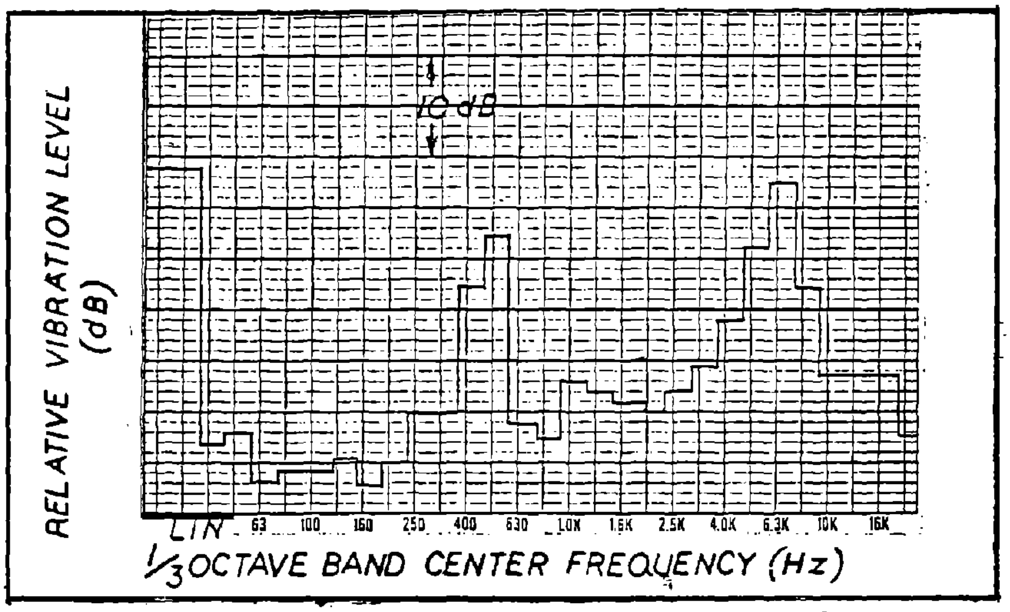

Figure 2.20 Chipbreaker frequency response

bility of interesting vibration response reduction in a practical sense.

The effect of an externally applied resonant system has been observed experimentally for the chipbreaker mechanism used on wood planers[5]. The chipbreaker consists of a number of spring loaded pads which contact the workpiece. The chipbreaker response (Figure 2.20) is seen to occur at frequencies of approximately 500 and 6000 Hz and is dependent on the particular construction details. The natural vibrational frequency of the individual spring loaded pads can be approximated by

$$f = \frac{1}{2\pi}\sqrt{k/m}$$

where k = individual spring stiffness,
m = mass of shoe.

The resonant response shown in Figure 2.20 at 500 Hz falls in the range of typical chipbreaker design. A comparison of planer noise levels for operation with and without the chipbreaker is shown in Figure 2.21. The large difference observed suggests that properly tuned resonant systems similar to the chipbreaker could be used effectively to reduce workpiece response and associated noise.

The effect of edges and other discontinuities on the system damping loss factor has been treated in reference 6 for the case of panels. In effect, a discontinuity "absorbs" a portion of the incident energy and is defined through an absorption coefficient involving the fraction of the bending wave energy which is not returned to the panel. The energy that can be dissipated by a structural member in contact with the vibrating system depends on the nature of contact which is governed by the fastening method employed. In general, multiply fastened or contacting systems add an additional damping mechanism due to air compression between adjacent surfaces. The effectiveness of the system is proportional to the width of the discontinuity and depends primarily on a ratio of the distance separating connection points to the flexural wavelength of the vibrating structure. For the case of a beam, the vibrational field is such that the wave motion is in phase across the beam width, thus the orientation of the damping structure must be in the direction of wave propagation.

The use of granular materials (such as sand) for damping is quite common, primarily for economic reasons. The material is usually placed in voids or specially designed cavities within a structure. The damping mechanism involved is believed to be the conduction of energy into the sand where the energy is in turn dissipated. It can be shown[6] that system resonances, and hence high losses, occur when the thickness of the granular layer is an odd multiple of a quarter of a structural wavelength. The structural wavelength for workpieces varying from 1/2" to 2" in thickness in the frequency range from 500 Hz to 5000 Hz ranges from about 30 inches to 8 inches. In practice the granular layers should be nonuniform in thickness so that damping can be achieved over a wide range of

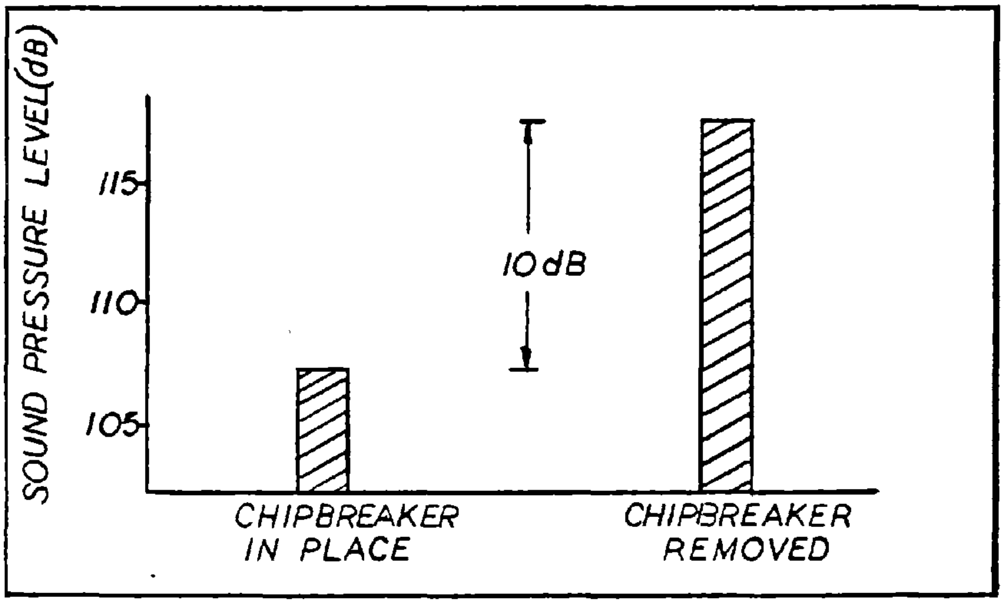

Figure 2.21. Effect of chipbreaker on planer noise

frequencies rather than just at or near resonances.

The final damping mechanism to be considered is an energy removal process involving intermittent impulsive contact between a vibrating structure and another body. This "rattling" mechanism involves energy transport due to energy losses at contact points, where large contact stresses may be present. An example of such a mechanism is the notable increase in plate damping when loose rivets or plugs are utilized as opposed to the case of tight plugs.

Alteration of Wave Propagation Characteristics

In the preceding sections the effects of stiffness and damping have been discussed along with means of altering these parameters. In addition, wave propagation along a vibrating structure can be drastically altered by discontinuities causing incident energy to be reflected to some degree, thus reducing the energy proceeding past the discontinuity. Several methods for reducing vibration transmitted beyond a certain point are listed in Table 2.5.

Table 2.5 Techniques for the Reduction of Transmitted Vibration

Changes in cross section
Corners or branches
Blocking masses
Effective termination

Although the attenuation characteristics of each method will be discussed individually, it is quite difficult to distinguish between mechanisms since the addition of an external system has an effect on stiffness, mass, resonant frequencies, and damping as well as the reflection of energy due to the discontinuity. In analyzing a system designed to prevent the transmission of vibratory energy beyond a discontinuity, it is helpful to refer to the diagram of Figure 2.22.

When waves propagating along a structure encounter changes in cross section or material, a portion of the incident energy is reflected. In practice, changes in cross sectional area on the order of 50 to 1 are required to achieve appreciable transmission loss. For workpiece vibration, this technique is of little practical interest.

The transmission loss associated with the junction of two structures in a manner conducive to the transmission of bending moments from one to the other is of some practical interest in the reduction of workpiece vibration transmission. The method of analysis of corners at right angles applies to the bending wave energy transmitted past a simple roll support which is of major interest in the present analysis. For the boundary conditions that apply to simple supports (equal angular velocity and bending moments on either side of the support and zero transverse velocity at the support) it can be shown that 50% of

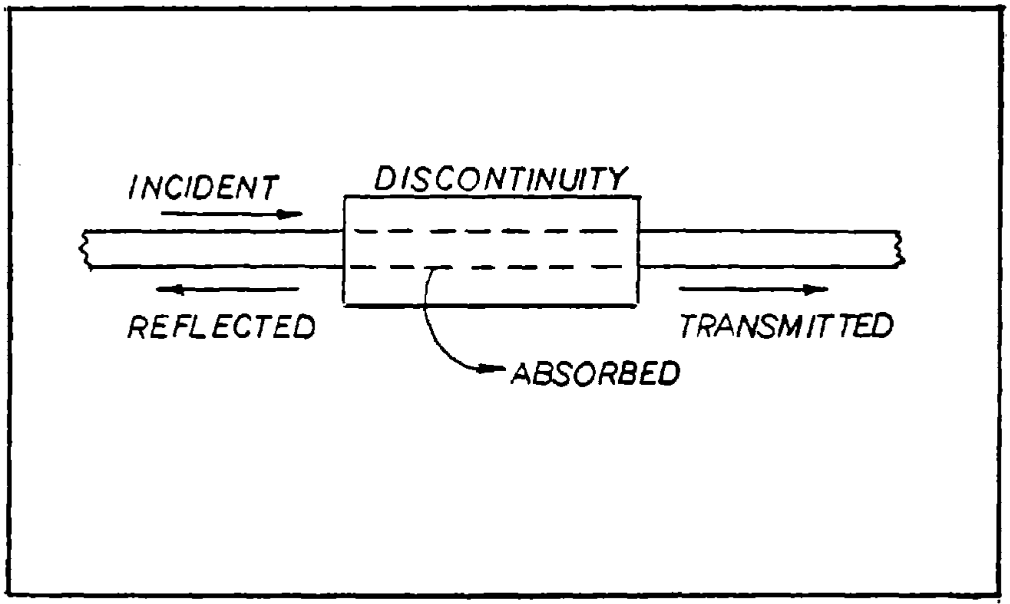

Figure 2.22 Effect of discontinuity on energy flow

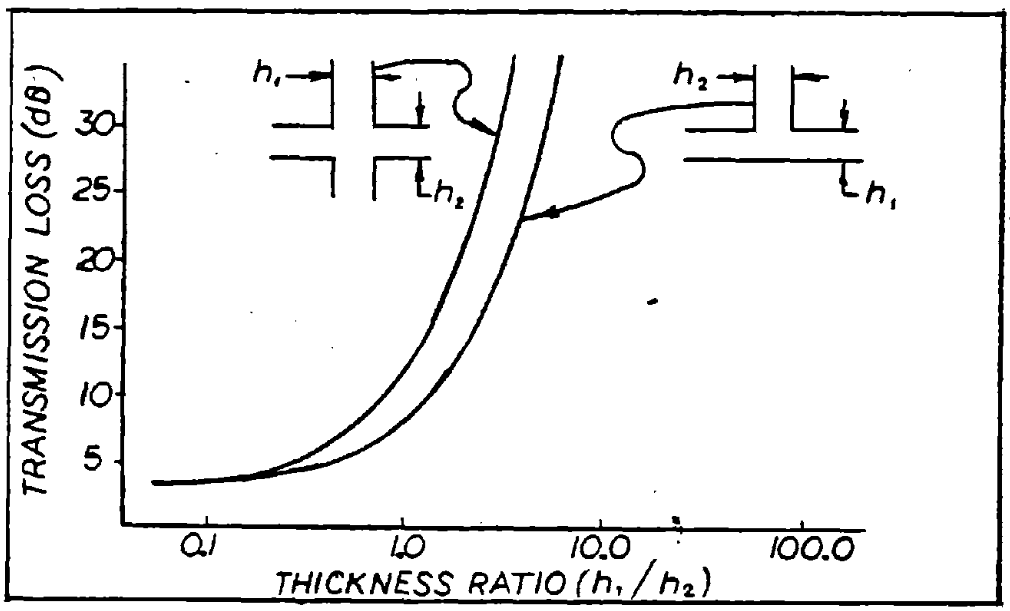

Figure 2.23 Transmission loss at structural intersections
(after reference 6)

the incoming energy is reflected. Unfortunately, the resulting transmission loss from such a system is quite sensitive to the nature of contact, and any deviation from a flexurally rigid junction can cause large discrepancies. It is interesting to note that the workpiece-pressure bar configuration commonly found in woodworking machinery resembles to some degree the "T" junction shown in Figure 2.23. As the thickness of the intersected plate is increased, the attenuation approaches the case of a rigidly clamped support (termination). As in the case with structural damping, a major difficulty exists regarding the proper contact mechanism.

When structure borne waves impinge on structural supports (ribs, etc.), the supports act as attached masses which can block the propagation of waves. This blocking effect occurs for bending waves even in the absence of zero forces and moments at the point of attachment. This results from the requirement that the forces and moments need only to interact so as to produce a near-field condition preventing the transmission of bending waves. The conditions for total transmission past a single blocking mass are met at a lower frequency than the frequency of total attenuation so that a blocking mass behaves basically as a low pass filter.

In analyzing the attenuation of bending waves by attached blocking masses, two ratios are found to be of primary importance. These involve a mass parameter and a structural parameter and are given by[6]

$$\mu = \frac{2\pi}{\lambda} m/m' \text{ (mass parameter)} \tag{2.9}$$

$$\nu = \{m'/m\}\sqrt{H/m} \text{ (structural parameter)} \tag{2.10}$$

where λ = bending wavelength,
m = blocking mass,
m' = mass per unit length of beam,
H = mass moment of inertia of attached mass.

The transmission loss provided by a blocking mass is somewhat complex and is given in reference 6; however, several important characteristics of blocking masses which depend on these ratios are listed.

(i) At low frequencies the transmission becomes complete.

(ii) At high frequencies the product of transmission coefficient (t) and μ approaches 2. This implies that at high frequency (short wavelength) t is small provided μ is large.

(iii) Between the frequencies given by (i) and (ii), total transmission occurs when

$$\mu^3 + 2\mu^2 - 2/\nu^2 = 0.$$

(iv) Above the frequency given by (iii), total attenuation occurs when

$$4 + \mu + \nu^2\mu^3 = 0.$$

(v) Above the total attenuation frequency, the transmission loss may be approximated by

$$R = 20\log_{10}(\mu/2)\ \text{dB}.$$

If 4 can be neglected in the expression for total attenuation, then $\mu \simeq 1/\nu$ for total attenuation and

$$\lambda \simeq 2\pi\sqrt{H/m}\ .$$

In this discussion it is assumed that the added mass is rigid so that all points participate in the rotational motion. The obvious deviation of light bodies with large radii of gyration from the ideal rigid case adversely affects the attenuation. It is quite interesting to note that the case of total transmission occurs when the rotational inertia of the blocking mass is disconnected as would be the case with a mass attached flexibly to the primary structure. At high frequency, the mass remains at rest, and the situation is identical to a beam supported at a single point where 50% of the incident energy is reflected. The cases for rotational participation and rotational disconnection are shown in Figure 2.24.

In practice, the conditions just discussed differ from the ideal case primarily because of the existence of reflected (returning) waves. The primary problem with reflected waves is the possibility of resonance of the portion of the structure beyond the blocking mass. This means that the effectiveness of a blocking mass is sensitive to whether or not the primary transmitted wave adds in phase with the reflected wave. In the resonant case the portion of the beam beyond the blocking mass may accept more energy from the source than the connecting elements, leading to a condition worse than the case without a blocking mass. Likewise, if antiresonance occurs, additional attenuation may be observed. An auxillary movable blocking mass can be employed to accomplish the antiresonant condition at a particular frequency. The problem of reflected waves is difficult to overcome when a large number of frequencies (and hence structural wavelengths) are involved. The results of attaching blocking masses to systems having reflective terminations are difficult to predict, and experimental data exhibit considerable scatter[6]. For long workpieces having fairly large damping, it is possible to achieve substantial transmission losses even though the end of the workpiece is reflective in nature.

The simple case of a blocking mass attached to a vibrating workpiece can be analyzed by using equations (2.9) and (2.10) provided the requirements of point attachment and participation of the mass in the angular motions of the

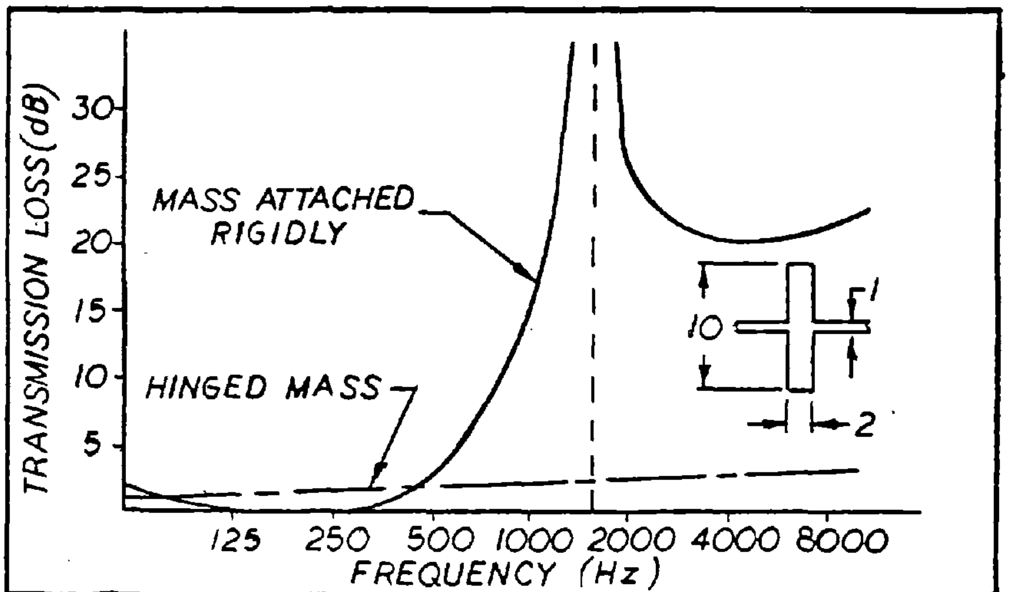

Figure 2.24 Effect of rotational inertia on transmission loss

workpiece are met. These formulations can also be used in cases where the width of contact is small compared to the bending wavelength. Cases where the width of the attached mass exceeds the point contact limitation must be analyzed subject to different boundary conditions as will be discussed in the next section. In practice, the existence of reflected waves from the end of the workpiece and the difficulties involved in achieving point contact without rotation make the utility of blocking masses as a means of significant attenuation of bending waves questionable.

As a result of difficulties with reduction of workpiece response through changes in stiffness, damping, and the previously discussed wave attenuation methods, a theoretical and experimental program was undertaken[8]. The program was aimed at modeling and experimentally investigating the effect of various external systems on the reduction of vibration transmitted beyond the point of application. This approach involved the artificial termination of the vibrating structure so far as bending wave propagation is concerned. In effect, the system is such that bending moments and angular velocities in addition to transverse motion are suppressed over an area sufficient to prevent bending wave propagation beyond the discontinuity.

It is recognized that for the reverberant type vibrational fields under discussion, complete reflection can result in higher vibration levels within these constraints. Thus, to accomplish noise control, the system must be used in conjunction with an acoustical enclosure. The advantage is that only that portion of the vibrating workpiece between the constraints need be enclosed as opposed to enclosing the entire vibrating workpiece. The utility of such a system becomes apparent in view of the common practice of sawing and planing boards of 10 to 20 feet in length prior to cutting, which would otherwise necessitate enclosures in excess of 40 feet in length. Referring to the flow diagram presented in Figure 2.22, the system absorbs, transmits, and reflects portions of the incident energy. The requirements for absorption are met when there is an impedance match and can be analyzed in terms of a dynamic absorption system as discussed in the previous section. The impedance characteristics of a particular discontinuity are known to depend strongly on the effective mass and stiffness of the discontinuity as well as the length and nature of contact.

In reference 8, vibration suppressors attached to a vibrating beam were studied for contact areas exceeding one half of a structural wavelength. The stiffness properties of the system were modeled as a series of linear springs, and lumped masses were used to approximate the effect of mass loading. The two models are illustrated in Figure 2.25. The coverage of at least one half of a structural bending wavelength results in complete termination of the propagating wave for the limiting case of infinite stiffness or infinite mass. In the stiffness model of Figure 2.25(a), the springs can only transmit compressive forces which are assumed to be proportional to the deflections occurring at the spring contact points. The mass loading effect is analyzed by means of lumped masses which are connected to the beam in a manner that permits rotational as well as transverse motion. Typical normalized results for the stiffness and mass models are shown in Figure 2.26 and 2.27 for a stiffness ratio (the ratio of the distributed spring stiffness to the beam stiffness) and a mass ratio (the ratio of the lumped mass to the mass of the beam

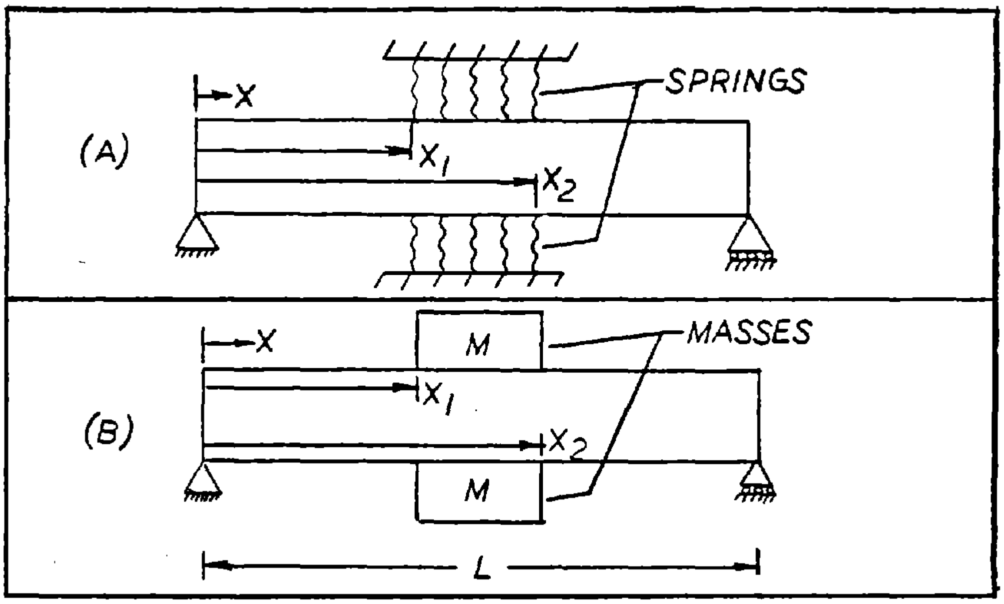

Figure 2.25 Vibration suppression models

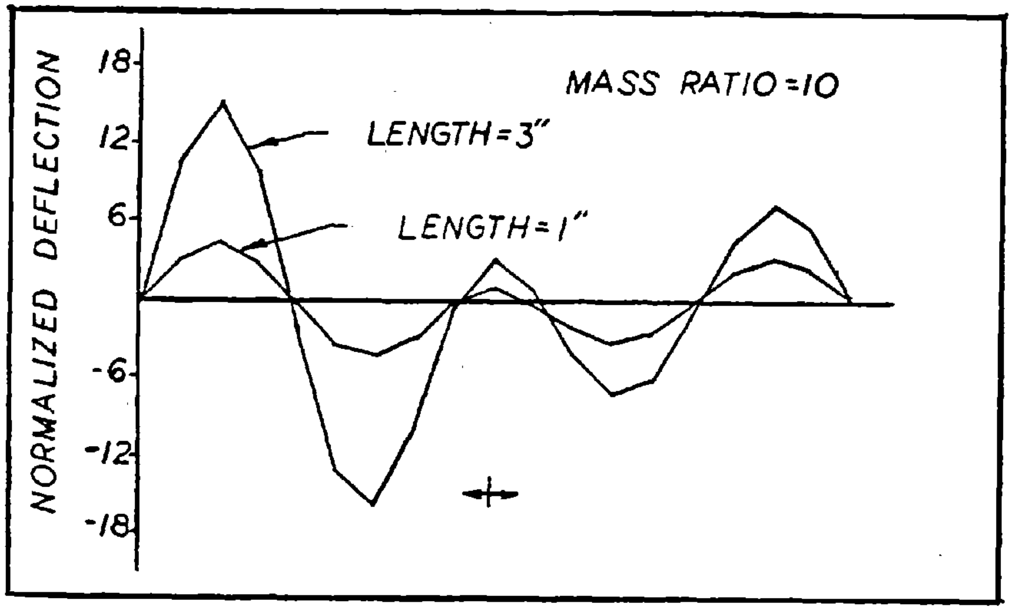

Figure 2.26 Effect of mass on suppression efficiency

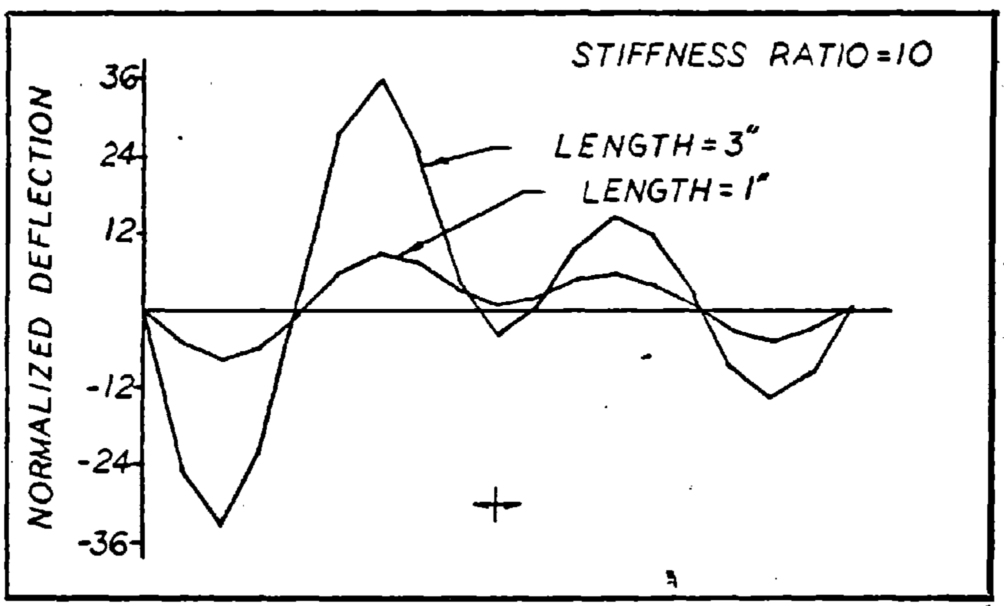

Figure 2.27 Effect of stiffness on suppression efficiency

per unit length) of ten. As expected, a general tendency of increasing attenuation as the system approaches the case of zero displacement along a distance of one half of a structural wavelength was observed.

An experimental investigation was performed in reference 8 in which the effect of stiffness and mass loading systems on vibration attenuation for beams was examined. Although it was quite difficult to separate and identify the effects of changes in stiffness, mass, damping, etc., in addition to the problems associated with a reflective termination, the investigation did identify important parameters which are helpful in the design process. The experimental arrangement utilized in this study is shown in Figure 2.28. The test facility was equipped with a hydraulic jack system which provided the required range of contact pressures. Beams of two different materials were utilized for testing purposes, mild steel and red oak. The physical properties and dimensions of the test beams are given in Table 2.6.

Table 2.6 Workpiece Models and Related Parameters

Material	Dimensions (in)	Young's Modulus (lb/in^2)	Density (lb/in^3)	Critical Frequency (Hz)
Steel	1 x 1/8 x 36	29.00×10^6	0.282	3963
Oak	1 x ½ x 36	1.78×10^6	0.028	1277
Oak	1 x 1 x 36	1.78×10^6	0.028	638

These particular beams were chosen to provide the range of critical frequencies and beam thicknesses that were necessary for suppressor evaluation. The suppressors consisted of steel or neoprene rubber pads having the required dimensions and were located at the geometric center of the workpiece. Excitation was near the critical frequency and was provided by an electromechanical shaker system attached to one end of the beam with several types of input signals. Beam vibration data were obtained by piezoelectric type accelerometers located on either side of the suppression device. The effectiveness of the various configurations was determined by comparing the maximum vibration levels on either side of the device. In the experimental program, a parameter of major importance was the contact length ratio (R) defined as

$$R = \ell/\lambda_{cr}$$

where ℓ = contact length of the suppressor,
λ_{cr} = bending wavelength of the beam at the critical frequency.

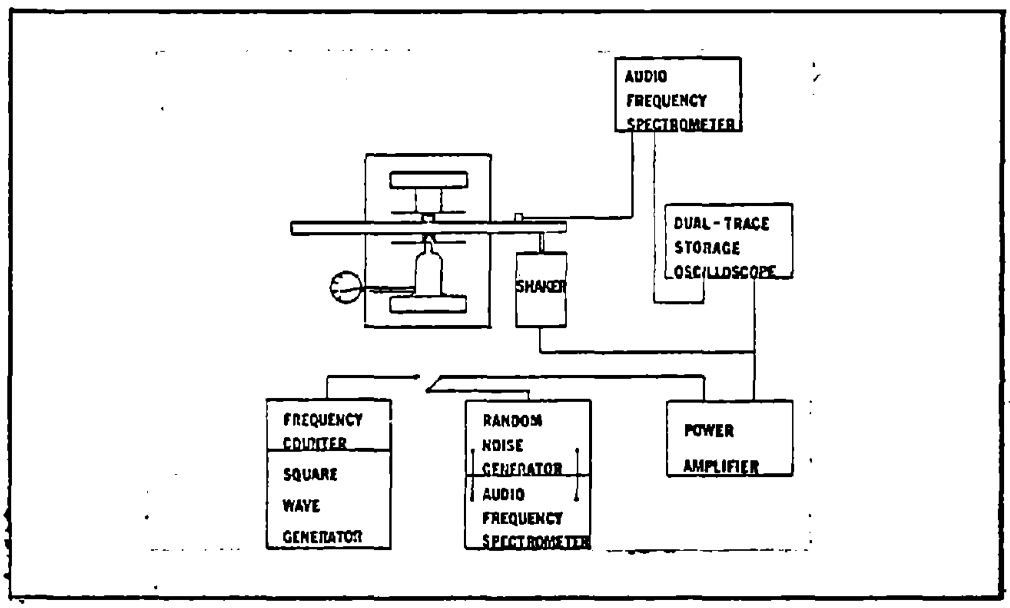

Figure 2.28 Experimental arrangement for vibration suppression tests

Values of the contact ratio (R) for the three test beams (Table 2.6) are presented in Table 2.7 for several values of the contact length.

Table 2.7 Approximate Values of R

Contact Length (in.)	R 1/8" Steel	1/2" Oak	1" Oak
1	0.30	0.10	0.05
2	0.60	0.20	0.10
3	0.90	0.30	0.15
4	1.20	0.40	0.20
5	1.50	0.50	0.25
6	1.80	0.60	0.30

Experimental results for the ½ inch oak beam subjected to a square wave excitation are shown in Figure 2.29 for a contact pressure of 80 lb/in^2. The steel suppressors are observed to effectively reduce the transmitted vibration for all contact lengths tested, while the neoprene suppressors become effective only for quite large contact lengths. Results for a 1 inch beam having a critical frequency of 638 Hz, as compared to 1277 Hz for the ½ inch beam, are shown in Figure 2.30. Again the steel system results in a more effective suppressor.

As a result of the difficulties associated with achieving contact with moving workpieces over long spans, a number of other suppressor configurations were examined. The main results of this portion of the investigation can be observed from Figure 2.31 which shows the reductions achieved for two suppressor configurations corresponding to a separation distance of 1/4 and 1/8 structural wavelength for the 1/2 inch oak beam with square wave excitation. The effectiveness of the double suppressor system was observed to be essentially independent of whether rubber or steel suppression blocks were employed, a result of major importance from an applications standpoint. The conclusions reached in reference 8 point out the necessity of achieving an impedance mismatch, as illustrated by the relative success of the steel suppression blocks as compared to the rubber suppressors. The rubber masses are capable of reflecting considerable amounts of energy but as expected are quite sensitive to contact length, exerted pressure, nature of excitation, etc., and exhibit somewhat erratic behavior. It was possible, with the separated arrangement, for flexibly attached rubber suppressors to be quite effective in vibration attenuation.

The effective use of a vibration suppression system in conjunction with a small acoustical enclosure for wood planer noise control has been reported[9]. The suppression system consisted of a rubber tire-plate arrangement and was designed to restrict vibratory motion over a relatively large area of the workpiece. The system is shown schematically in Figure 2.32. The use of foam filled tires deviated appreciably from the ideal cases described previously; however, due to inconsistencies in workpiece shape and thickness, a flexible contact device was deemed imperative. Optimization of the normal force acting on the

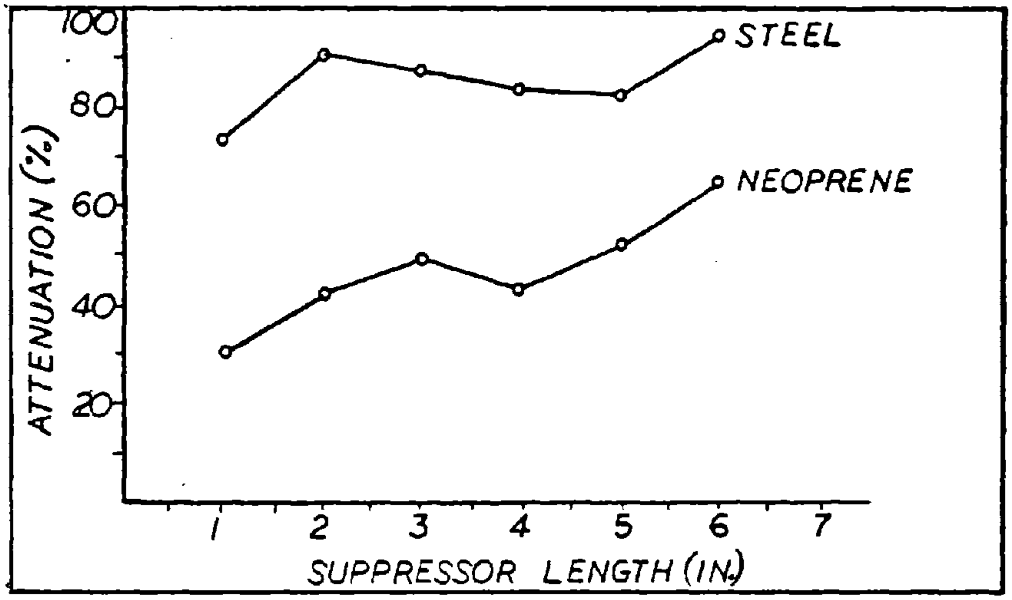

Figure 2.29 Effect of material and length on attenuation - 1 inch bar

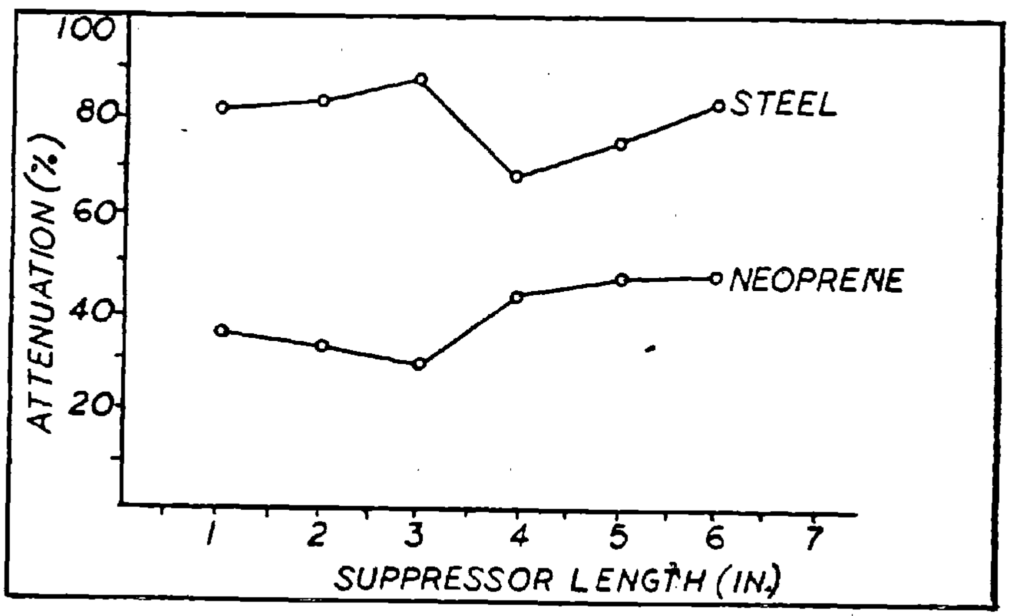

Figure 2.30 Effect of material and length on attenuation - ½ inch bar

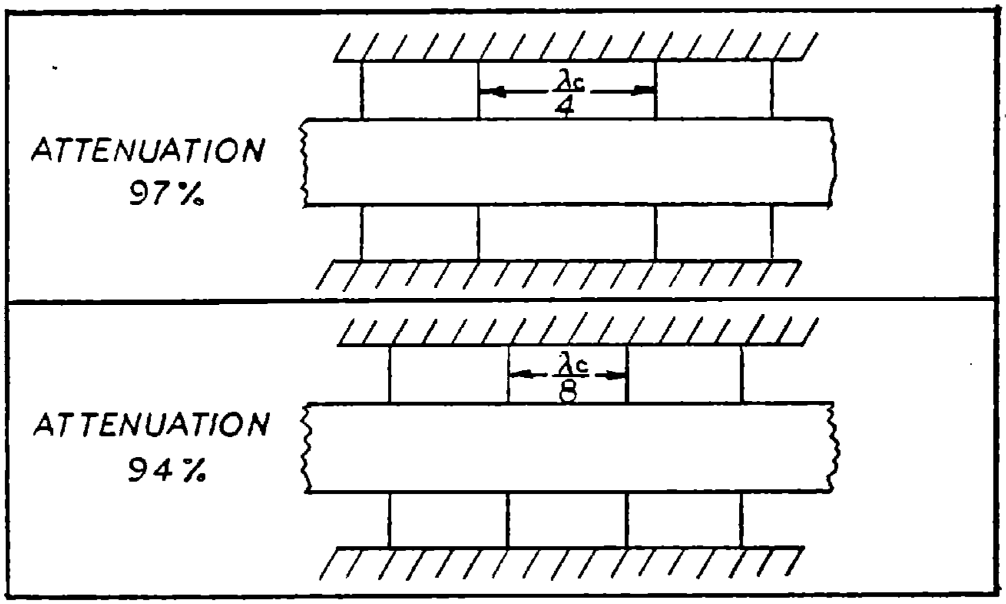

Figure 2.31 Effect of spacing on attenuation

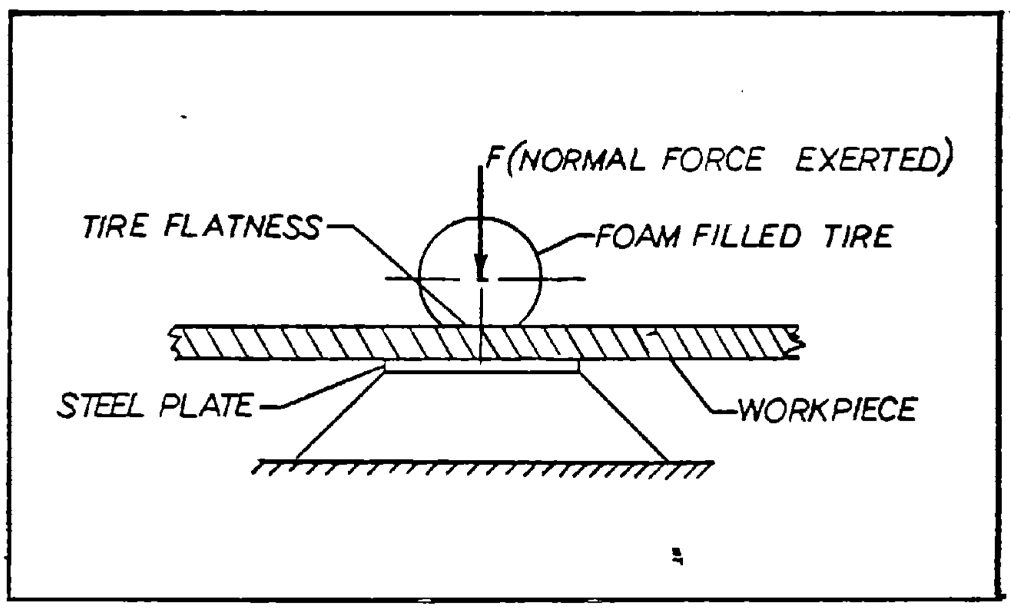

Figure 2.32 Tire-plate suppression system

workpiece, the required flat area, etc., was performed on an experimental test apparatus[9]. The suppression system was field tested on a double roughing planer equipped with two input and output sets of feed rolls powered by a direct hydraulic drive. This arrangement was altered to test the suppressors by replacing one set of infeed and outfeed rolls with the foam tire-steel plate arrangement. The tire flatness and normal force (F) were adjusted so as to provide maximum suppression without interfering with the feeding of lumber through the machine. Sound and workpiece vibration measurements taken under the conditions given in Table 2.8 are presented in Table 2.9.

Table 2.8 Operational Conditions

Cutterhead Speed = 3600 RPM
Cutterhead Diameter = 7"
Feed Speed = 250 ft/min
Board Length = 12 ft.
Board Thickness = 1/8"
Board Width = 8"
Cutting Depth = 1/8"
Wood Species - red oak

Table 2.9 Test Results

Machine Condition	Workpiece Vibration Reduction	Sound Level at Operator Position
Straight knives (no enclosure)	No suppression	118 (dB)
Straight knives (with enclosure)	No suppression	112 (dB)
Straight knives (with enclosure)	18 (dB)	97 (dB)

The overall noise level at the operator position did not reflect the 18 dB reduction expected because of limitations of the acoustical enclosure covering the area between suppressors. The enclosure is seen to be relatively ineffective (5-6 dB) for long workpieces without suppression but provides substantial noise reduction (18-20 dB) when used in conjunction with the suppressors. The suppressors do not offer noise reduction for workpiece lengths less than the length of the acoustic enclosure. Although the suppression system reduced workpiece vibration by approximately 18 dB, it acts only on the portion of the board extending beyond the suppressors. Thus, the longer the workpiece, the more evident the difference between the suppressed and unsuppressed planer equipped with a compact acoustic enclosure.

Suppression of board vibration has been demonstrated to be an effective measure in controlling planer noise when used in conjunction with a compact enclosure. A disadvantage of such systems is the additional maintenance considerations which, when compared to the available tooling alterations presented earlier, detract from the use of these systems on planers.

Special Case - Moulder

It is of interest to consider the special case of the moulder machine illustrated in Figure 2.33. The moulder is used to surface and shape material of various lengths, widths, and thicknesses and represents an important noise problem in the woodworking industry. The moulder is of special interest since it is a source of quite intense noise (often in excess of 110 dBA at the operator position) and incorporates several of the physical principles discussed in this section.

As illustrated, the workpiece is fed through the machine by a system of chain belts and rolls on the infeed and passes under a spring loaded chipbreaker section as it moves into the cutting tools. As the workpiece exits the cutters, a holddown system is employed to stabilize the workpiece. As might be expected, the dominant noise source for the moulder is structural vibrations of the workpiece. Vibration measurements have indicated that the portion of the workpiece extending beyond the feeding apparatus on the infeed does not experience appreciable vibration and from a sound radiation standpoint can often be neglected. This is due to the action of the mechanical feed works which tend to prevent the propagation of bending waves by clamping the workpiece over a large effective area. In addition, the chipbreaker mechanism acts as a vibration absorber to some degree along with several damping mechanisms which tend to attenuate transmitted vibrations by as much as 15 to 20 dB in many instances. The suppressing action of the feeding works reflects a large amount of the incident vibratory energy which then is distributed over the remaining area, resulting in increased vibration levels. On the outfeed end of the machine, the holddown pressure bar does not make ideal contact with the workpiece and rotational motion of the workpiece is not restricted appreciably. The result is effective transmission of vibration along the portion of the workpiece extending from the infeed mechanism. It is this portion of the workpiece (the length of this portion continuously changes as the workpiece is fed through) that is primarily responsible for sound generation.

Although it is possible to employ the techniques discussed for reducing workpiece response (which are present to some degree on the infeed), it has been common practice in cases where the workpiece length is not excessive to utilize an acoustical enclosure. These enclosures take advantage of the reduced workpiece vibration levels on the infeed end of the machine while allowing adequate space for the workpiece to fully disengage all cutters prior to exiting from the outfeed of the enclosure. The enclosure system is shown in Figure 2.34. The enclosure is constructed of simple building materials and crosses over the infeed of the machine so as not to interfere with feeding. The outfeed of the machine is extended to allow for complete disengagement of the workpiece from all cutterheads. Noise level reductions at the operator for such an enclosure range from 15 to 25 dBA, depending on the design.

A second approach to noise reduction in moulders is the use of helical or mill to pattern type cutterheads. While this approach is quite effective, it is often difficult to completely retool a moulder when great numbers of pattern shapes are utilized. The tooling concepts discussed previously can be applied directly to the moulder.

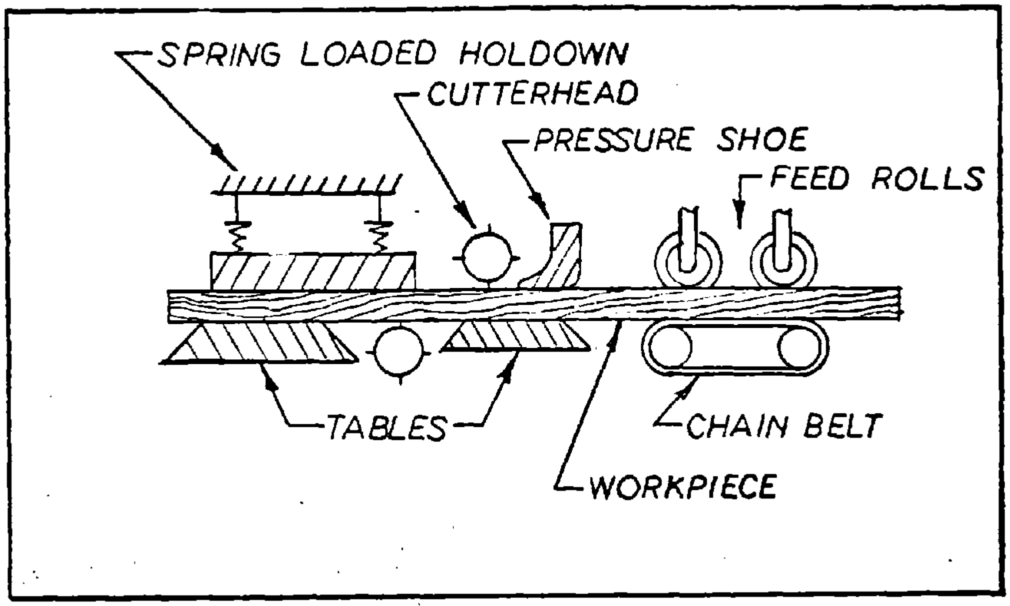

Figure 2.33 Vibration suppression in moulders

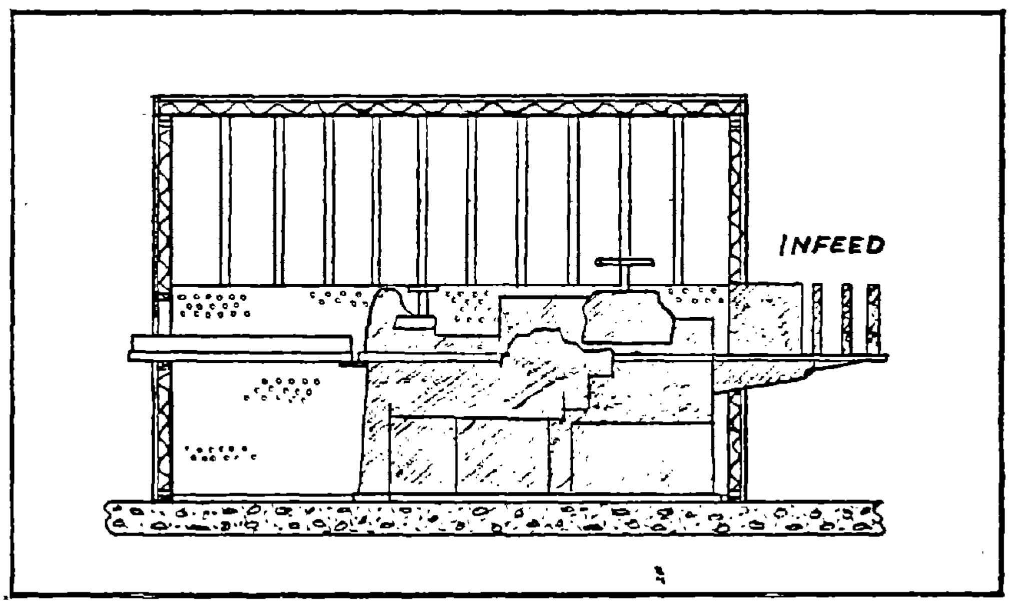

Figure 2.34 Moulder enclosure

Reduction of Workpiece Radiation

In the context of this report, reduction of workpiece radiation involves the use of acoustical techniques in the path, as opposed to the techniques for source reduction discussed previously. It is recognized that radiation characteristics can also be altered by other means such as alteration of the critical frequency, medium, etc..

The primary elements of a noise generation/receiver system are shown in Figure 2.35. In this discussion, the source of noise is the workpiece vibrating in a steady state reverberant condition. The vibration level over the entire workpiece is essentially constant so that the radiated sound is assumed to emanate from all portions of the workpiece equally. The path is the space separating the source and receiver (the machine operator). The noise observed at the operator position is directly affected by the source intensity, the distance separating the source and receiver, the room condition, and any reflective or absorptive structures placed in the normal transmission path.

Means of reducing source intensity, i.e., the vibration levels present in the workpiece, have been discussed in some detail. The effects of distance and room condition can best be discussed in view of Figure 2.36 which illustrates the direct and reflective sound fields present at the operator. The interrelationship between direct and reflective radiation, distance, and room conditions (acoustically) can be deduced from equation (2.11)[7].

$$L_p = L_w + 10\log_{10}(Q/S + 4/R) + 10.5 \text{ dB} \tag{2.11}$$

Where L_p = sound pressure level (re. .0002 μbar),
L_w = sound power level (re 10^{-12}watts),
Q = directivity factor (measure of variation in L_p with position),
S = surface area ($2\pi r$) where r is distance from the source,
R = room constant (measure of absorption).

The sound pressure level is directly proportional to the sound power output and depends on two ratios, Q/S and 4/R. The term 4/R involves the reflective field where R is a measure of room absorptivity. The term Q/S involves the direct field where S is related to the separation distance between the source and the receiver. In cases where the distance between the source and receiver is small, Q/S is greater than 4/R and the direct radiation field is of primary importance. In such cases, adding absorptive material to the room is not effective. In cases where the operator is far removed from the source (or extremely small reflective rooms) Q/S is less than 4/R and the reflective field is also of importance. In these cases the addition of absorptive material can be advantageous. For large open areas the term 4/R can often be neglected in equation (2.11), resulting in the familiar -6 dB per doubling of distance law. For extremely large open factory areas, the use of room absorption is usually of minor benefit for machinery operators, although it may have a beneficial effect on plant background noise levels (locations far removed from any particular machine). When acoustical techniques other than addition of room absorption are employed, equation (2.11) must be modified to account for changes in effective distance (barriers or shields), changes in path absorption (absorptive

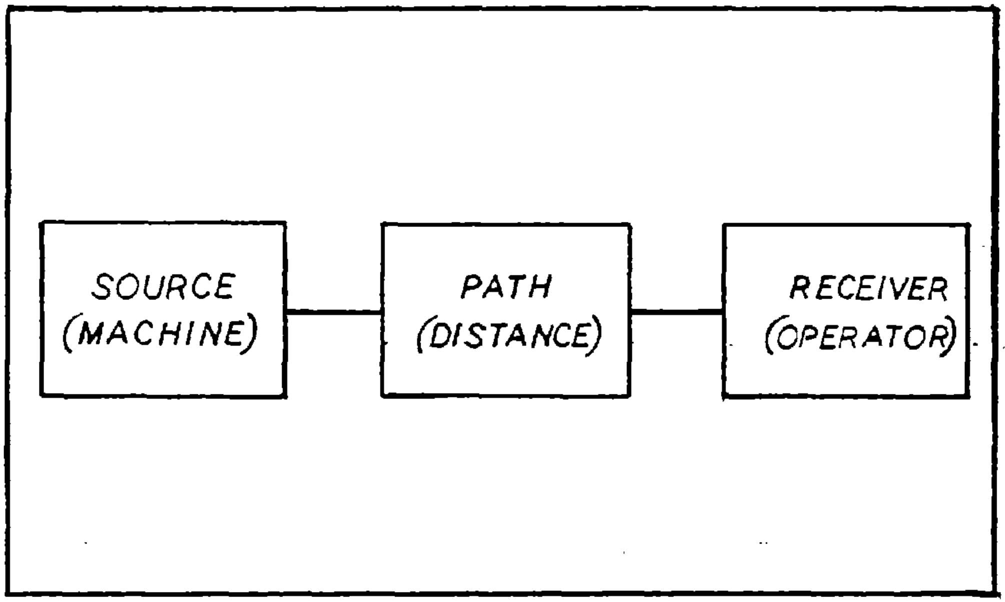

Figure 2.35 Noise control system

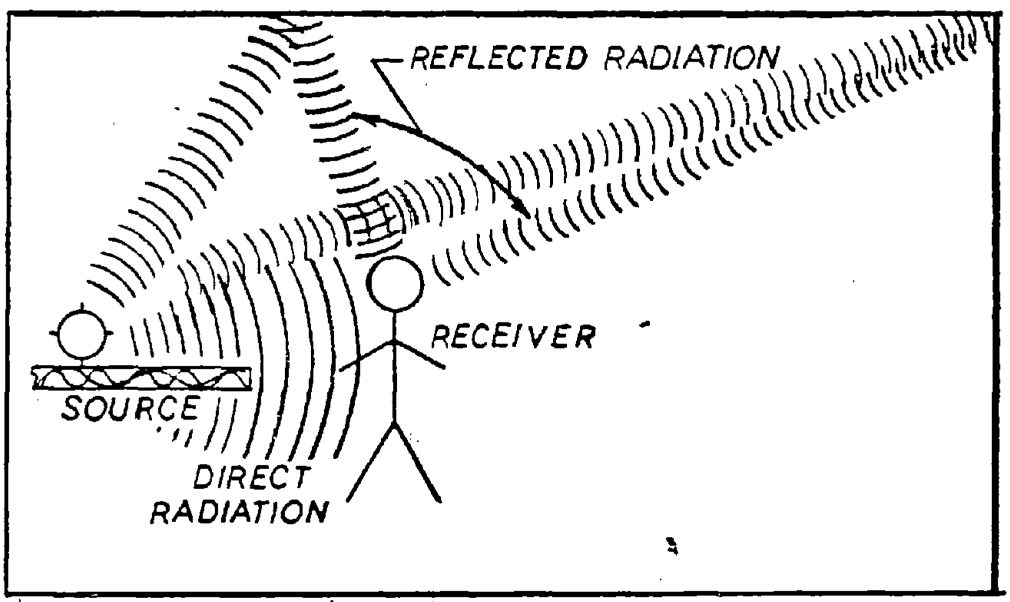

Figure 2.36 Exposure to direct and reflected sound

shields), or changes in effective source strength (enclosures). The effectiveness of several of these techniques is illustrated in Figure 2.37. In this example, the source has been characterized as emitting broad band sound (spread evenly over a wide frequency range) so that the effect of the various techniques can be easily compared at the various frequencies. The use of an acoustical shield or barrier typically results in noise level reductions of 5 to 10 dB at the higher frequencies, provided the operator is in the acoustical shadow of the shield. The effectiveness of such shields can be determined from the theory of Fresnel diffraction[7]. The attenuation provided by barriers can be increased by utilizing sound absorption material on the source side.

The use of acoustical enclosures (total or partial) is the most effective path means of reducing noise levels substantially. There are, however, several important considerations when an acoustical enclosure is to be employed which are listed in Table 2.10.

Table 2.10 Factors Influencing Total Enclosure Effectiveness

(1) Construction materials
(2) Proper construction and isolation
(3) Sealing of openings and cracks
(4) Vibrating workpiece not covered by enclosure

The effectiveness of an enclosure is affected by the transmission loss characteristics of the wall material (a measure of the wall's ability to prevent sound transmission) and the type absorption material employed. Both the transmission loss and absorptivity characteristics depend on frequency, as illustrated in Figure 2.38 for typical construction materials.

Another important consideration in enclosure effectiveness is proper construction and vibration isolation. Vibrations of a low frequency imparted to an enclosure through the floor or through direct contact with the machine can excite higher frequency vibrations in the enclosure structure. Proper construction of an enclosure therefore requires that the machines (or enclosure itself) be isolated from the floor to prevent a loss in effectiveness. It is also advisable to decouple the enclosure panels from the framework by some type of isolation system to reduce the edge stiffening that occurs when panels are rigidly fastened to a support structure.

A major problem with the use of total enclosures is the proper sealing of openings left for machine feeding purposes. The effect of untreated openings is shown as a percentage of wall area in Figure 2.39 along with the effect of the total area treated with absorption material. The importance of minimized opening area is apparent from Figure 2.39. Cases where a wall or partition has windows or openings (either treated or untreated) can be analyzed by using equation (2.12) which is valid for a uniform acoustic field of plane waves.

$$R = 10\log_{10}(1 + A/T) \tag{2.12}$$

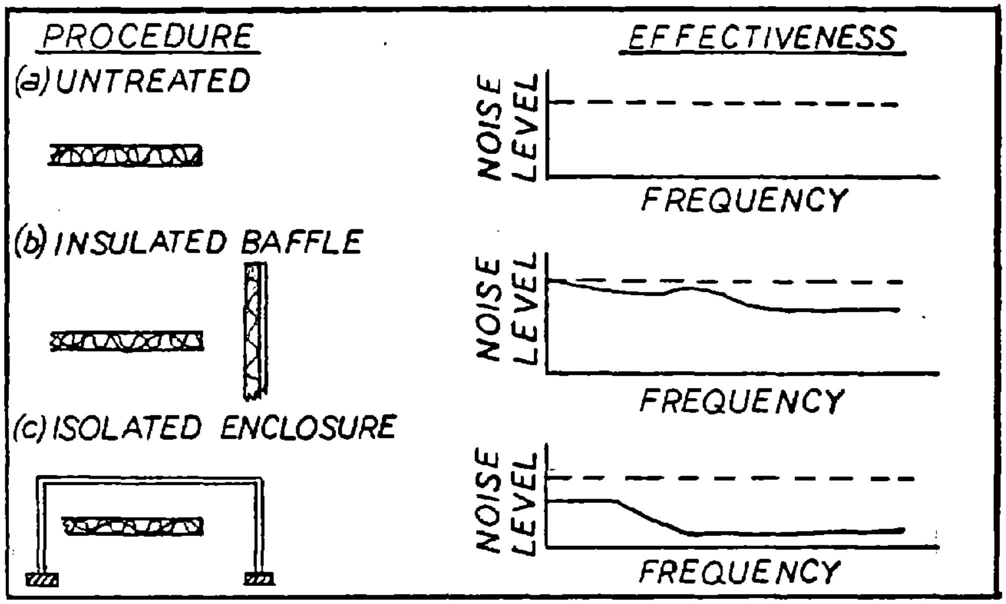

Figure 2.37 Effectiveness of operator booths with various treatments

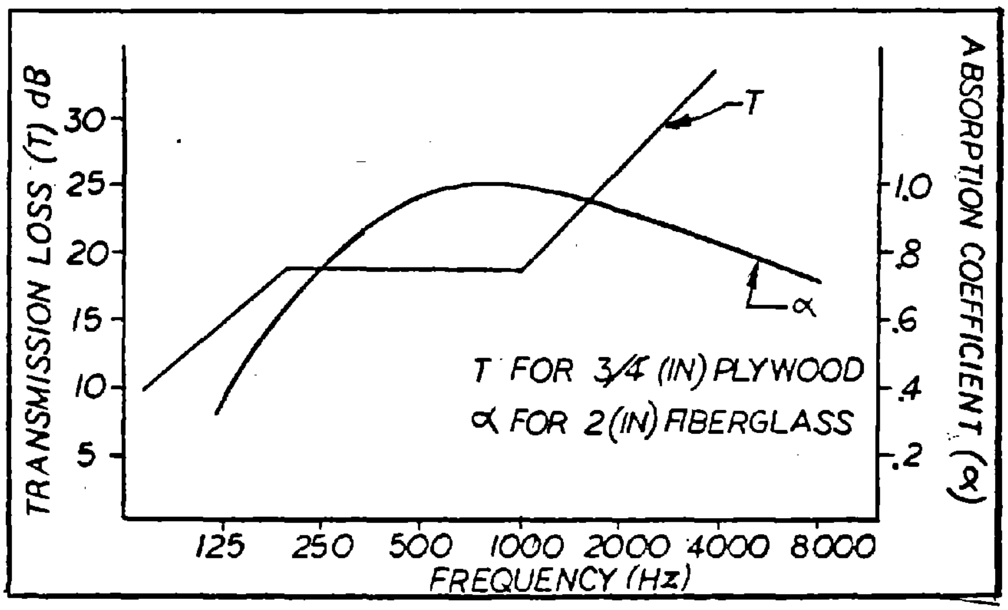

Figure 2.38 Frequency dependence of control materials

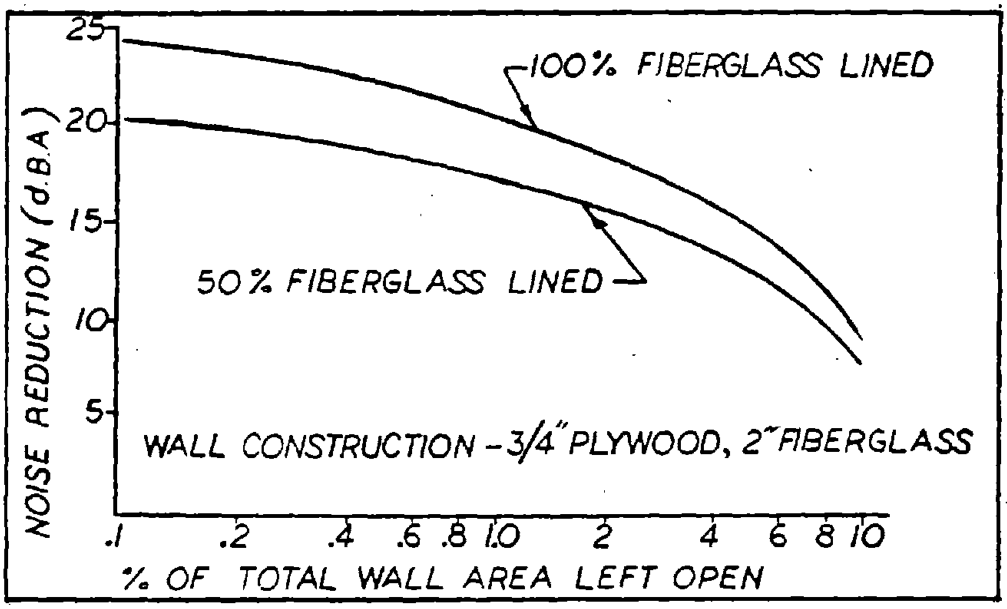

Figure 2.39 Effect of opening on noise reduction

In equation (2.12) R = noise reduction factor (dB),
α = absorption coefficient,
τ = transmission coefficient,

$$A = \sum_{j=1}^{n} \alpha_j S_j \ ,$$

$$T = \sum_{j=1}^{n} \tau_j S_j \ ,$$

S_j = area of jth surface.

It is of interest to note that when $A = 0$ (no absorption) the noise levels inside and outside the enclosure become equal. This does not occur in practice since all materials absorb sound to some degree. The case of an opening ($\tau = 1$) can be examined in terms of the percentage of total opening area from equation (2.12).

The requirement for air makeup to replenish air removed from an enclosure due to dust and chip collection is normally accomplished through ducts; however, the acoustical performance must be taken into account. Adequate noise reduction can be accomplished through the use of acoustically lined ducts having one or more elbows.

Since sound is produced along the entire length of the vibrating workpiece, it is apparent that the complete source is not enclosed when any portion of the workpiece is exposed while engaged in the cutting unit. This results in considerable difficulty when extremely long workpiece lengths are encountered and prompted studies into vibration suppression techniques. The importance of enclosing a high percentage of the workpiece becomes apparent in view of the fact that halving the size of an acoustic source results in only a 3 dB reduction in radiated acoustic power.

Control of Tool Vibration Noise

Tool vibration produced noise is characterized by Figure 2.2. Most cutterheads, bits, drills, etc. do not radiate appreciable sound via structural vibration; however, such tools do create significant aerodynamic noise as will be discussed in a later section of this report. The most important tool from a vibration standpoint is the circular saw blade (shown in Figure 2.40) which is capable of producing intense noise as a result of forced and free vibration response. Typical saw blades are constructed of alloy steel which possesses a low internal damping factor. In addition, saw blades usually range between 1/16 and 3/16 inches in thickness and are easily excited, responding with large vibrational amplitudes. In view of the importance of this tool noise generation source, only the circular saw blade will be discussed in any detail.

Means by which this noise can be reduced are apparent from Figure 2.2 and include: 1) reduction of tool excitation, 2) reduction of tool response, and 3) reduction of tool radiation. A reduction in excitation normally involves

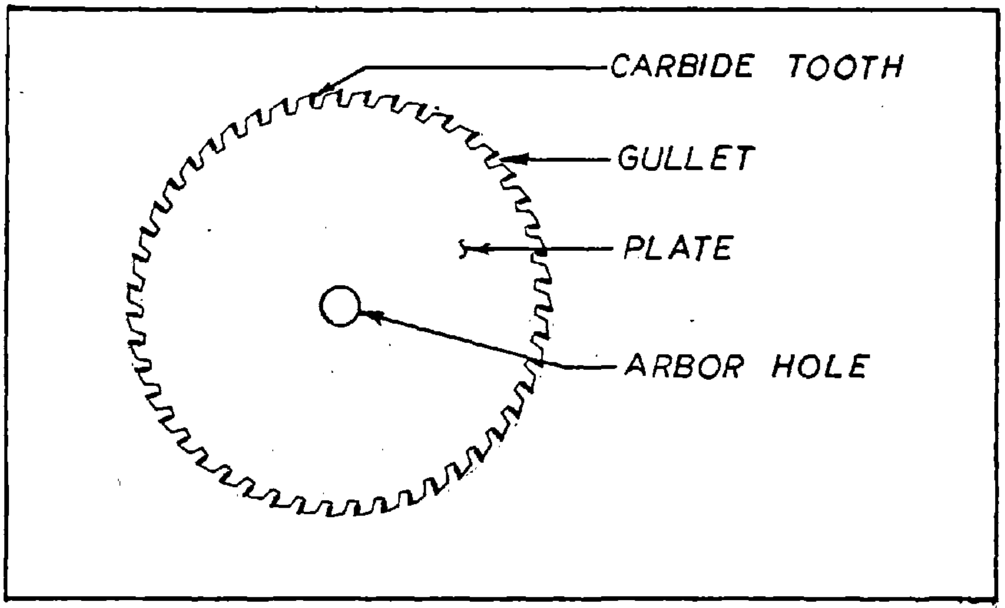

Figure 2.40 Circular saw blade

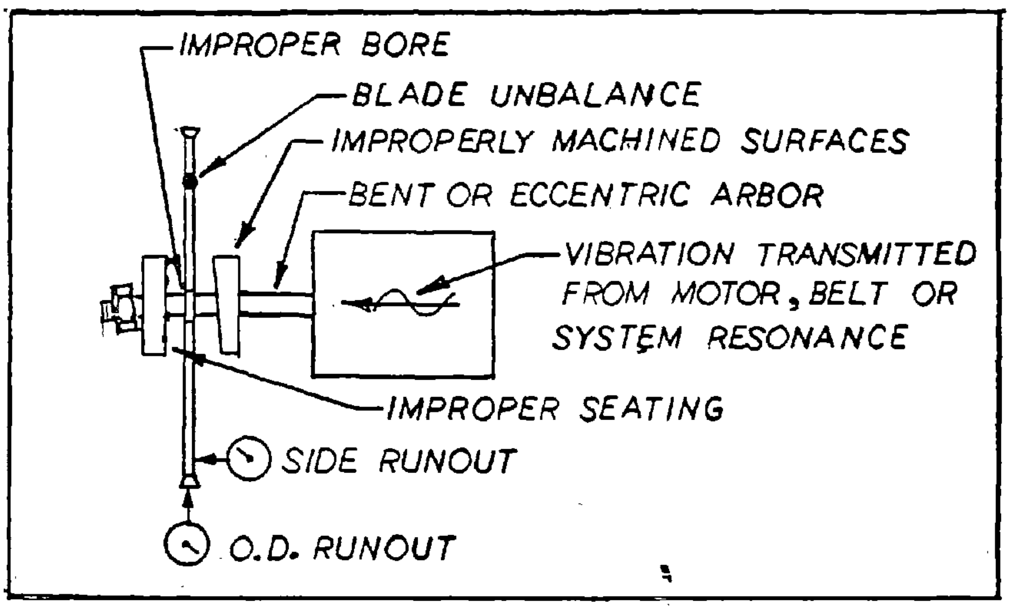

Figure 2.41 Sources of saw blade vibration

tool redesign or a change of cutting process, while a reduction of tool response could incorporate redesign, damping, or vibration suppression techniques. Reduction of tool radiation refers to the use of shields, barriers, and partial or total enclosures; although in some instances alteration of sound radiation characteristics can be accomplished.

Reduction of Tool Excitation

The excitation mechanism for circular saws is generally attributed to aerodynamic disturbances or rotational sources during idle and tooth impact sources during cutting. These sources along with several other excitation mechanisms known to exist for circular saws are listed in Table 2.11.

Table 2.11 Sources of Excitation for Circular Saws

(a)	Mechanical	balance alignment
(b)	Aerodynamic	turbulent flow stationary surfaces
(c)	Acoustic	guards reflective surfaces
(d)	Operational	tooth impact misalignment

The response characteristics of circular saws are presented in the next section; however, it is appropriate to point out at this juncture that only a very small external force at the proper frequency can produce resonant response. In view of the number of external forces acting on the saw blade, it is apparent that sufficient energy is available to excite strong vibration response over a wide frequency range. The vibration response consists primarily of forced and free resonant response. Such a classification is obviously an oversimplification of a complex and interrelated aero-elastic problem; however, it does serve as a basis from which the various sources of excitation given in Table 2.15 can be evaluated and means for reduction proposed.

Mechanical Source Reduction

Mechanical sources such as unbalance, blade wobble, and saw blade/shaft misalignment are responsible for exciting saw blade resonance in some cases and explain the apparent anomalies involving blades which resonate on some machines and do not on others. Simply removing and reinstalling a saw blade has eliminated a resonant condition in some instances. The unsteady forces that result from mechanical shortcomings can be reduced through more careful manufacturing and installation procedures. This can result in a reduction in excitation of the blade and eliminate the "screaming" phenomenon. Typical mechanical problems capable of producing unsteady forces are illustrated in Figure 2.41. The means of reducing the mechanical problems illustrated in Figure 2.41 are apparent and include a thorough check of manufacturing tolerances and component condition along with dynamic balancing of the saw blade.

Aerodynamic Source Reduction

The reduction of aerodynamic excitation is of major importance from both vibration excitation and aerodynamic noise generation standpoints. The existence of a turbulent boundary layer in the vicinity of the tooth and gullet area can produce an unsteady force field which is responsible for aerodynamic noise production as well as saw blade vibration excitation. Means of reducing aerodynamic excitation include gullet redesign, reduction in peripheral speed, etc. and will be discussed in section III.

Acoustic Source Reduction

Sound energy which is normally radiated into free space can be reflected in such a manner as to acoustically excite the saw blade. This situation can occur when a guard or highly reflective enclosure is placed around the saw blade or conceivably when an unguarded saw is operated in a highly reverberant room. In these cases, vibration can be excited by means of reflected sound waves in much the same manner as a plate excited by airborne sound[10]. In addition, the effect of radiation loading on the saw blade due to the ambient medium must be considered in some cases. From an energy standpoint, this forms a coupled system consisting of the sound field and the vibrating saw blade. The effect of a structure near the saw blade which is not acoustically treated is to change the modal mass or stiffness. The stiffness becomes large at frequencies corresponding to certain standing wave patterns (i.e., standing waves having nodes at the structural surface). For other values of the distance from the blade to the structure, the additional stiffness becomes zero at a particular frequency resulting in resonant response. The modal frequencies can be considerably different from those encountered in free space in these cases. In cases where acoustic excitation is of importance, blade resonances can be eliminated by removal of the guard or reflecting surface or the addition of absorptive materials.

Operational Source Reduction

During the cutting operation, the rotating saw blade is brought into contact with the workpiece, and both the blade and workpiece experience unsteady forces. These forces result from the impact of the saw teeth on the workpiece and the rubbing action set up between the saw blade body and the workpiece. The magnitude of these sources of excitation depends on saw blade design, operational conditions, and machine/workpiece alignment. The response during cutting is primarily forced response occurring at or near resonant frequencies where energy input is available.

A reduction in excitation of the saw blade requires a reduction in external forces developed during the cutting process which can be accomplished to some degree by alteration of saw blade design. The exact nature of the forcing function acting on the saw blade is unknown since it is quite difficult to obtain experimentally. The individual tooth impact may be considered as a simple case of point impact delivering an ideal "spike" force, F(t), or

$$F(t) = \sum_{n=1}^{\infty} F_n \cos(n\omega_s t) \qquad (2.13)$$

having a period $T = 1/f_s$

where f_s = frequency of impact $(\omega_s/2\pi)$,

$F_n = 2/T \int_0^T F(t)\cos(n\omega_s t)dt$ (the Fourier coefficients),

t = time,

n = index.

If it is assumed that the pulse duration is so short that $\cos(n\omega_s t)$ does not deviate appreciably from unity, then

$$F_n \simeq 2I/T \qquad (2.14)$$

where I is the momentum resulting from impact, and

$$F(t) = \sum_{n=1}^{\infty} (2/T) I \cos(n\omega_s t) \ . \qquad (2.15)$$

Thus, the input frequency spectra produced by this simple case consists of contributions of magnitude 2I/T occurring at frequencies f_s, $2f_s$, $3f_s$ ....

A second source of excitation is the rotational source mechanisms which include unbalance, rubbing, etc. caused by manufacturing error or improper alignment. The exact nature of this excitation is also quite difficult to characterize; however, the force pulse can be characterized in much the same manner as tooth impact through the use of Fourier's theorem. Essentially, the excitation can be broken up into an infinite number of pure tone components, which results in a frequency spectrum consisting of contributions at the rotational frequency and harmonic frequencies. No assumption is made as to the exact shape of this rotational input; however, it is normal to expect the magnitude of the harmonic frequency contribution to decrease with increasing frequency. A composite of the forcing functions discussed is presented in Figure 2.42 for the

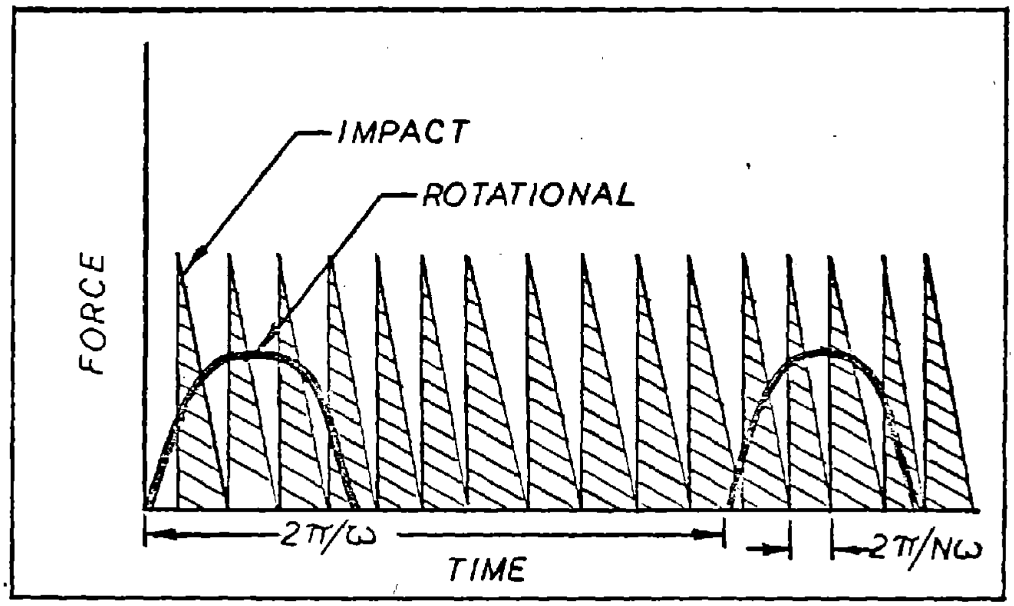

Figure 2.42 Vibration excitation force

case of ideal (delta function) impact at the tooth passage frequency (NRPM/60) along with rotational forcing in the form of a modified sine wave, representing a rubbing action occurring over one portion of the blade.

A reduction in excitation during cutting requires an alteration of both rotational disturbances as well as impact characteristics. The magnitude of the rotational excitation can be reduced through improved manufacturing tolerances, balance, etc., and in some cases increased clearance between the saw tooth and the plate (side clearance). The use of material spreaders, wiper slots, etc. are also useful in reducing contact between the saw plate and the workpiece.

The excitation provided by tooth impact is dependent on the momentum (I) and the period (T) as indicated in equation (2.15). A reduction in F_n requires an increase in T, a reduction in I, or both. From a practical standpoint, the impulse magnitude can be reduced by increasing the impact frequency and thereby reducing the work done per tooth. It is also noted that by increasing the frequency of impact, the excitation frequency may be quite large, and harmonics may lie beyond the frequency range of interest. The momentum upon impact may also be reduced by lowering tip speeds which can effectively reduce input excitation, provided the number of teeth is sufficiently large to compensate.

The response of a saw blade is extremely sensitive to both the magnitude and frequency spectrum of the excitation. As a result, the optimum case of minimum saw blade response involves a multiplicity of considerations as opposed to simply minimizing the input forces.

Reduction of Tool Response

The importance of the circular saw blade from a sound radiation standpoint has been explained, and as a consequence only the response of circular saws will be considered in this section. The response of other tools commonly found in the woodworking industry is generally of little importance; however, the basic approach followed for the circular saw blade can be used for any vibrating tool. The basic concepts outlined in the section concerning workpiece vibration response apply to any structure, including saw blades; however, in this case alterations to the saw blade (consistent with its functions) are possible.

The response of a circular saw blade to the input energy supplied by tooth impact and rotational sources fits into two categories: being resonant response during idle and forced response during cutting. As pointed out in the previous section, energy is available over a wide frequency range resulting in a broad band frequency excitation spectrum. Thin steel disks have a large number of structural resonances throughout this range and respond well to small input forces. During idle it is common for a vibrational mode to be excited due primarily to a forced frequency component closely matching a resonant frequency. This modal response increases with time until equilibrium with damping mechanisms are reached. In cases of lightly damped saw blades, this condition becomes quite serious from a noise generation standpoint resulting in the so-called "screamer." The screaming saw is quite sensitive to boundary conditions as well as the source and frequency of excitation, and represents a finely tuned dynamic system. Minor changes in mounting, mass, or stiffness (such as plugs, slots, retensioning, etc.) can shift a resonant frequency so as to reduce or prevent screaming. It is also possible for resonant vibration response to occur at one or more frequencies without proceeding to the screaming state. Such blade response has led to considerable experimental problems involving both vibration and aerodynamic saw noise sources.

During cutting, energy is also supplied by contact between the workpiece and the saw blade, resulting in energy input over a wide frequency range as well as an increase in damping. The saw blade responds well at each of its resonant frequencies, the amplitude being controlled by the damping present in each particular resonant mode and the contact with the workpiece.

The resonant frequencies of a circular disk clamped at its center have been determined[11] and consist of modal patterns involving both circumferential and radially travelling waves. Several modes of vibration are illustrated in Figure 2.43. Each mode corresponds to a particular frequency with the response in each mode depending on the modal stiffness, damping, and nature of excitation. From a sound radiation standpoint, the diametrical modes are believed to be of major importance[11]. A modal response pattern corresponding to an excitation frequency of 2000 Hz is shown in Figure 2.44. The saw blade response was forced by an electromagnetic shaker system, and the nodal lines are indicated by salt patterns formed on the blade (the salt accumulates along modal lines). In this particular case, a pattern of 8 nodal diameters and 1 nodal circle is present.

Techniques for reducing the response of circular saw blades are similar to those presented in the section on workpiece vibration reduction. It should be noted that alterations in mass and stiffness of the saw blade will affect the critical frequency which governs the efficiency of sound radiation. These considerations are discussed later in this report. Techniques for reducing response are listed in Table 2.11.

Table 2.11 Methods for Reducing Saw Blade Response

(a) Increased stiffness
(b) Increased damping
(c) Alteration of wave propagation characteristics

<u>Effect of Stiffness</u>

For a given excitation, response decreases with increased stiffness. In the case of circular saw blades, the stiffness can be increased through material changes, increases in thickness, stiffening collars, etc.. The effect of changing these factors is, however, to change other parameters which may have a detrimental effect on noise reduction. In general, increases in stiffness tend to be accompanied by a reduction in the critical frequency (the frequency at which the structural wavelength is equal to the acoustic wavelength at the

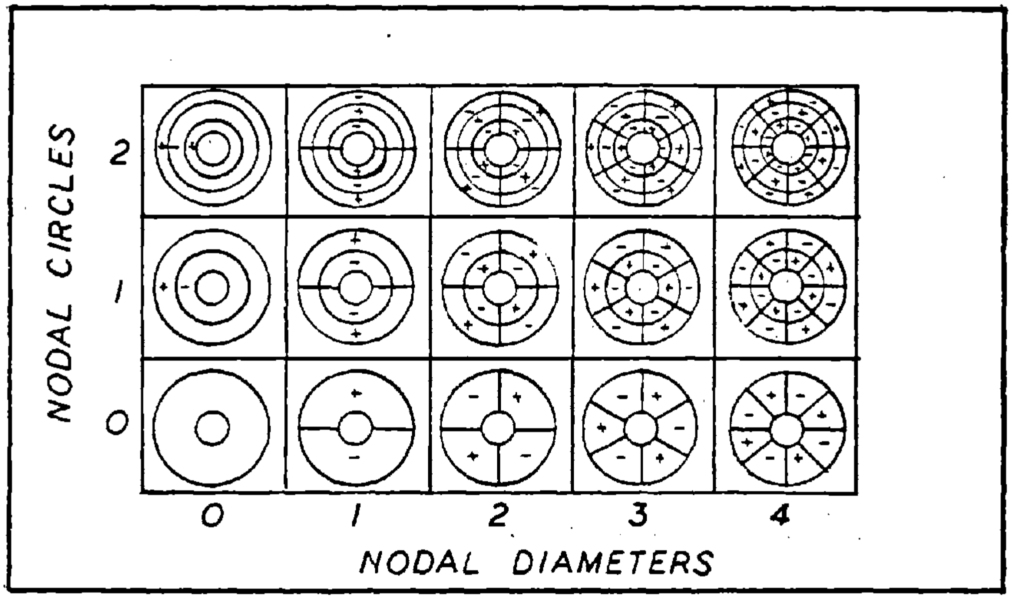

Figure 2.43 Modes of vibration of circular disk

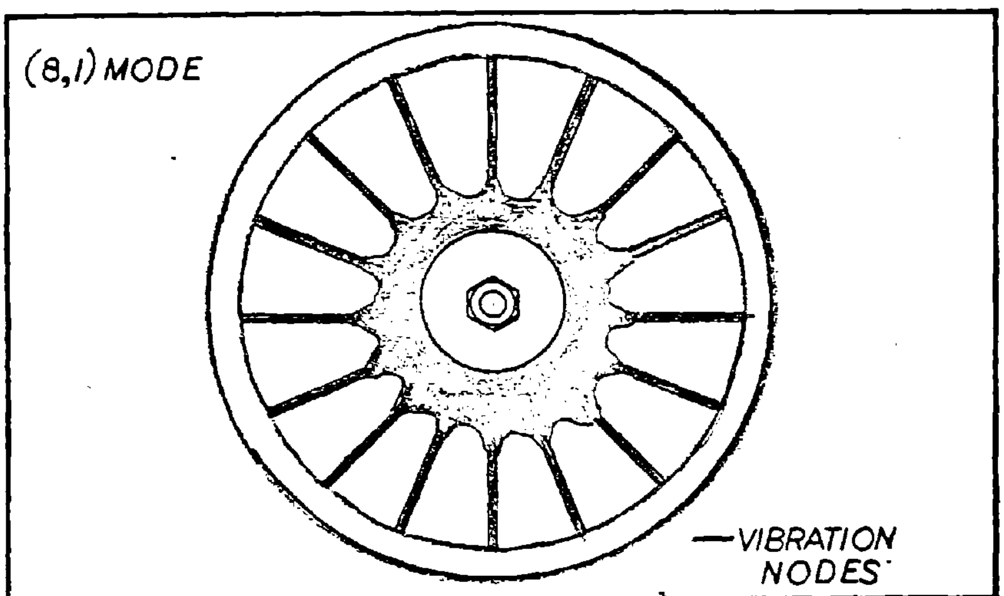

Figure 2.44 Natural mode of vibration of circular disk

same frequency). Sound generation is quite efficient above this frequency, and for noise control purposes the critical frequency is usually made as high as practically possible. This approach is discussed in the section dealing with reduction of radiation. Increases in thickness are usually accompanied by increased damping which, along with the reduced response characteristics of the thicker plates, tends to reduce resonant response. It should be noted that in-increased plate thickness involves wider saw blades (increased kerf) which results in increased power consumption, greater impact loading, and material waste.

The use of stiffening collars can appreciably alter the blade stiffness characteristics provided a high percentage of the saw is within the collar. In some cases, resonant response due to aerodynamic or rotational sources can be reduced, eliminating the idle screaming effect. Forced response during cutting can also be affected by the increased stiffness provided by collars. The main utility of such collars, however, is the additional damping introduced due to the sandwiching of the blade between the collars. This damping mechanism can be increased by the use of viscoelastic materials, reducing the blade response near resonances.

Artificial stiffening techniques include systems which maintain contact with a portion of the blade during both idling and cutting, such as blade packing techniques and clamping disks or pads in contact with the saw blade[12].

Effect of Damping

The response of a structure near resonance is controlled by damping, as discussed previously. In theory, all of the methods discussed can be applied to the circular saw blade; however, damping through attached layers and changes in material properties represent the most useful approaches and will be considered in some detail.

The internal damping of a material is affected by small changes in the material composition and treatment, and is often difficult to measure. The primary mechanism involved for metals is associated with dislocations in the crystal lattice and the resulting heat conduction between differently strained regions. Materials other than steel are available which have considerably higher loss factors, however, extensive testing is required to determine the practicality of alternative materials.

The attachment of free or constrained viscoelastic layers is of considerable practical value in reducing saw blade response during both the idle and cutting operations. Free layers have an advantage over constrained layers in that their effectiveness is essentially frequency independent. The total damping provided by a unconstrained (free) layer increases with increasing value of the layer modulus of elasticity, damping factor, and thickness. The requirement that the layer have a high modulus of elasticity as well as a high damping factor suggests the use of plastics for this application. A number of free damping layer treatments are commercially available for circular saw blades. The constrained layer approach involves the attachment of damped stiffening collars to either side of the saw blade, as shown in Figure 2.45. The effectiveness of the collars is limited by the area of the saw blade that can be covered, since this approach limits the depth of cut possible. A substantial portion of the blade must be within the collars to achieve appreciable damping due to the low amplitude motion near the center as opposed to the outer portions of the blade. In some cases the vibration patterns of the blade may be altered to provide greater damping for a given collar size. This approach is considered later in the report. The effectiveness of a constrained layer system is frequency dependent as opposed to the free layer case. The frequency at which maximum damping occurs is governed by the shear modulus and is known to decrease with increasing temperature. Fortunately, the fall off in damping on either side of the maximum occurs slowly so that a relatively wide range of frequencies can be damped effectively. The use of laminated saw blades is also recognized as a highly efficient damping system.

The use of loose rivets or plugs in the expansion slots of circular saw blades is quite common and may in some cases provide sufficient additional damping to prevent resonant response during idle. As discussed, the resonant mode is quite sensitive to any change in damping or mass, making a precise cause and effect study difficult. Such techniques are not considered to be effective in reducing forced vibration response during the cutting process.

Damping from mechanisms such as clamps or contact points are not considered in detail; however, their effect can be evaluated using the techniques described. Novel techniques such as the application of viscous foam in the area surrounding the blade and workpiece are of interest in specialized applications.

Special Case: The design of damped stiffening collars for purposes of contrained layer damping can be done by considering the system as a composite of three plates having equal thickness with elastic interlayers, as illustrated in Figure 2.46. The frequency at which maximum damping occurs for plates of equal thickness can be shown to be[6]

$$f \simeq (2/11)(G_2/E_3)(C_{L1}/d_2) \tag{2.16}$$

where G_2 = shear modulus of the damping layer (2),
E_3 = modulus of elasticity of constraining plate (3),
C_{L1} = longitudinal wave speed in the saw plate (1),
d_2 = thickness of damping layer (2).

The damping factor at this optimum frequency is

$$\eta_{optimum} = 3\eta_2/(5+4\sqrt{1+\eta_2^2}) \tag{2.17}$$

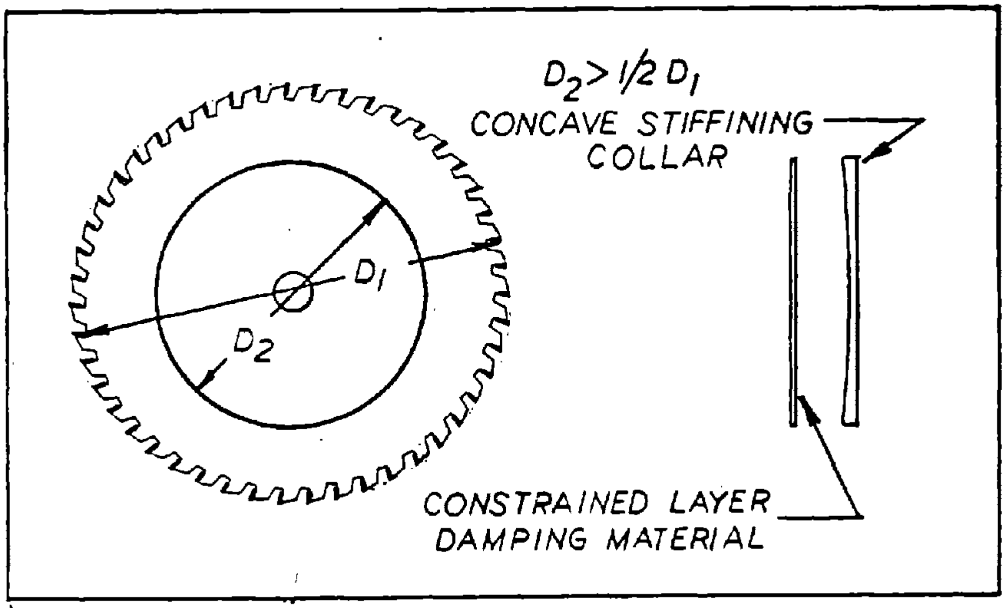

Figure 2.45 Damped clamping collars

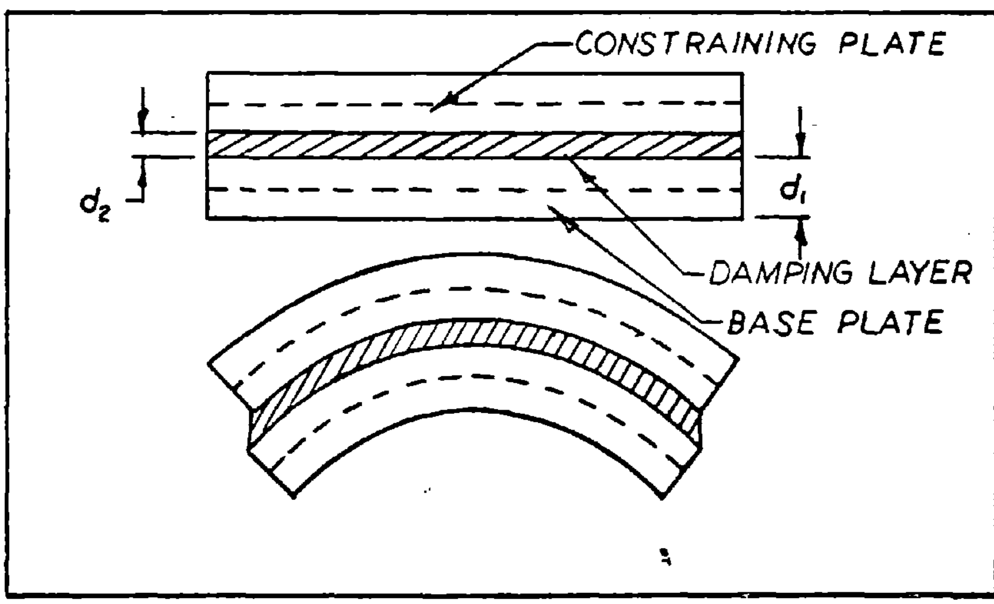

Figure 2.46 Constrained layer damping

where η_2 is the damping factor of the constrained layer at that frequency. In the case of circular saws, damping is required in the frequency range of 500 Hz to 6000 Hz so that the maximum damping should occur at about 2000 Hz. Figure 2.47 shows the damping factor for a properly designed system as a function of frequency assuming a bandwidth of damping of about five octaves. The loss factor for untreated steel plates is of the order of 2×10^{-4}.

Alteration of Wave Propagation Characteristics

The vibration response of circular saw blades, as illustrated in Figure 2.43, involves both diametrical and circumferential travelling waves. This modal response is usually excited near the periphery by aerodynamic disturbances or the actual cutting process. It is known that contributions to the radiated sound are greatest near the periphery, where vibrational amplitudes are large. Since the diametrical modes are of considerable importance in the radiation of sound, it is of interest to investigate means of altering the circumferentially travelling waves.

One such technique which can be used in conjunction with stiffening collars is radial slots cut into the blade body extending from the periphery down to a radius less than that of the stiffening collars. These slots disrupt circumferential wave motion and provide coupling of individual sections only through the hub and collar section as shown in Figure 2.48. The increase in damping is significant with such an arrangement. Alteration of sound radiation characteristics through changes in the critical frequency is discussed elsewhere in this section.

Reduction of Tool Radiation

Most woodworking tools, with the exception of circular saws and saw type cutters, are rigidly constructed, possess a high degree of damping, and are relatively small in size. As a result, noise sources in the vicinity of the cutting tool are usually aerodynamic in nature. The reduction of tool radiation through the use of shields, partial and total enclosures presented previously applies directly to this case. Another approach to the reduction of tool radiation is of interest for circular saws and involves alteration of the saw blade itself to reduce its efficiency as a sound radiator. The sound radiation from a vibrating structure can be expressed as,

$$W = \rho c A \sigma \langle \overline{V^2} \rangle \qquad (2.18)$$

where W = acoustic power,
ρ = density of air,
c = acoustic velocity,
A = surface area,
σ = radiation efficiency,
$\langle \overline{V^2} \rangle$ = transverse vibrational velocity.

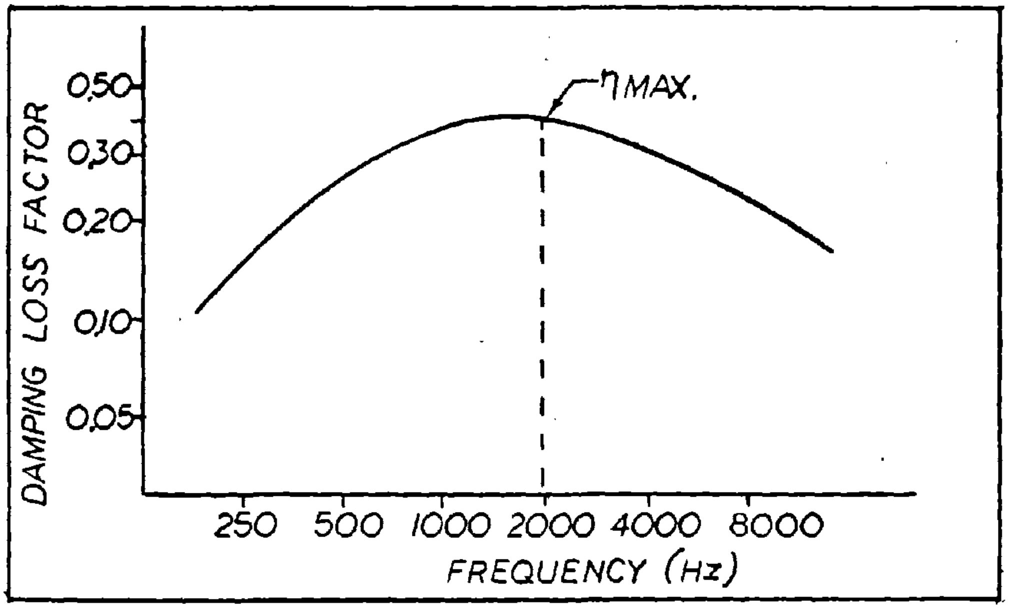

Figure 2.47 Damping factor versus frequency

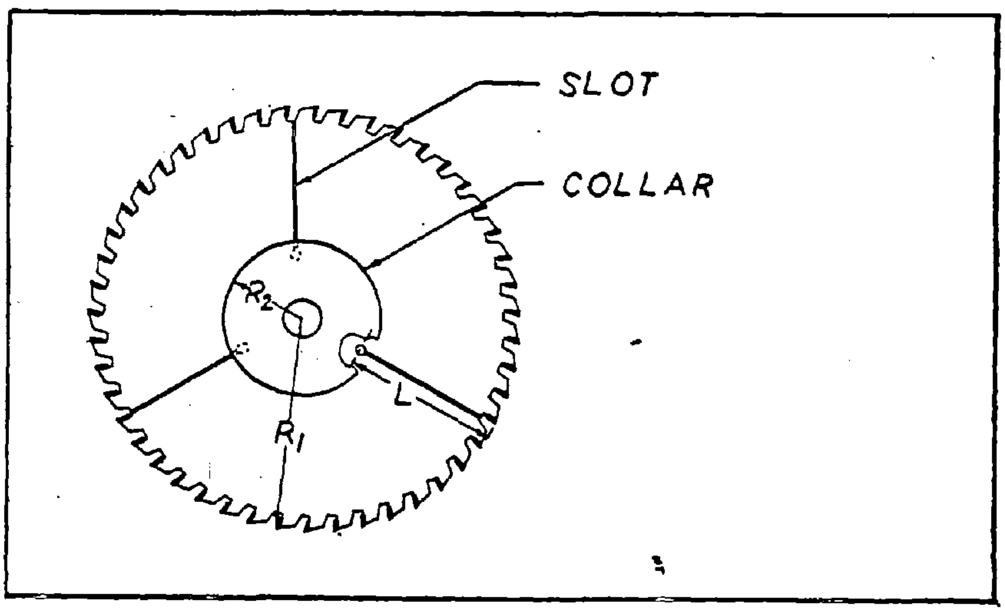

Figure 2.48 Slotted saw blade

The radiation is seen to depend on the vibrational velocity (which depends on the force/response characteristics of the system), the radiation efficiency (which depends on the critical frequency), and the source area (A). Alteration of the excitation and response characteristics has been considered; however, changes in source efficiency and radiating area are also plausible.

Alteration of the Radiation Efficiency

The radiation efficiency depends on the critical radiation frequency of the saw blade, defined as the frequency at which the structural wavelength is equal in length to the acoustic wavelength. Below the critical frequency, sound is not efficiently radiated, while near and above the critical frequency there is little interference between adjacent structural elements and radiation is usually quite efficient. By equating the bending wavelength with the acoustic wavelength, the critical frequency is determined as

$$f_c = (c^2/2\pi)\sqrt{12\rho/Et^2} \text{ (Hz)} . \tag{2.19}$$

It is noted that the critical frequency increases with increasing plate mass and decreasing plate stiffness. Thus, the critical frequency of a plate can be computed from a static measurement of the ratio of mass to bending stiffness. From a radiation reduction standpoint, it is desirable to make the critical frequency as high as possible in order to obtain a wide frequency range of low radiation efficiency. Although the general behavior is sensitive to the particular resonant mode, the variation in radiation efficiency with frequency can be assumed to be similar to that shown in Figure 2.49.

In devising methods for increasing the critical frequency, a detailed understanding of the vibrational fields set up in the plate is required. For anisotropic plates, it is necessary to define two critical frequencies, i.e., for wave propagation in the direction of maximum stiffness and in the direction of minimum stiffness. For circular plates, this corresponds to the radial and circumferential directions. The critical frequency in the direction of greatest stiffness is a lower bound for the system, i.e., waves in other directions correspond to a higher critical frequency. Accordingly, the critical frequency in the direction of the lowest stiffness represents the frequency above which the plate behaves like a homogeneous plate above its critical frequency. The most direct means of altering the critical frequency are changes in stiffness and mass.

Techniques for reducing stiffness include a reduction of plate thickness or the cutting of slots, grooves, etc. in order to reduce the flexural stiffness in the direction of wave propagation. Other techniques for increasing the critical frequency include the use of multilayer plates, using the frequency dependent properties of elastic materials to an advantage. The critical frequency of laminated plates is known to be considerably higher than for equivalent standard plates having a similar static stiffness. Reduced stiffness may also be achieved by employing saw plate materials having low modulus of elasticity; however, practical difficulties involving hardness and brazing could arise.

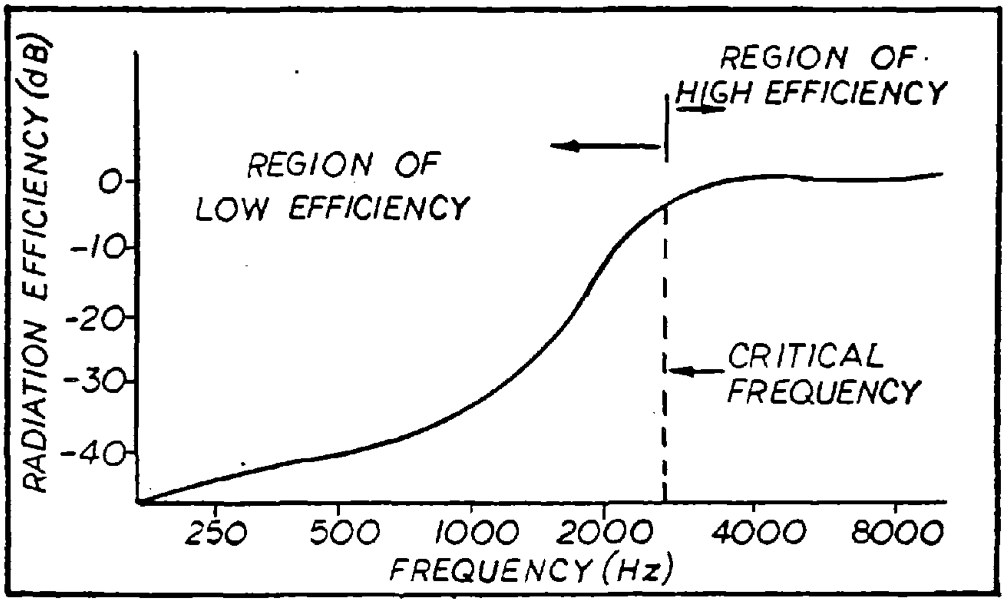

Figure 2.49 Radiation efficiency versus frequency

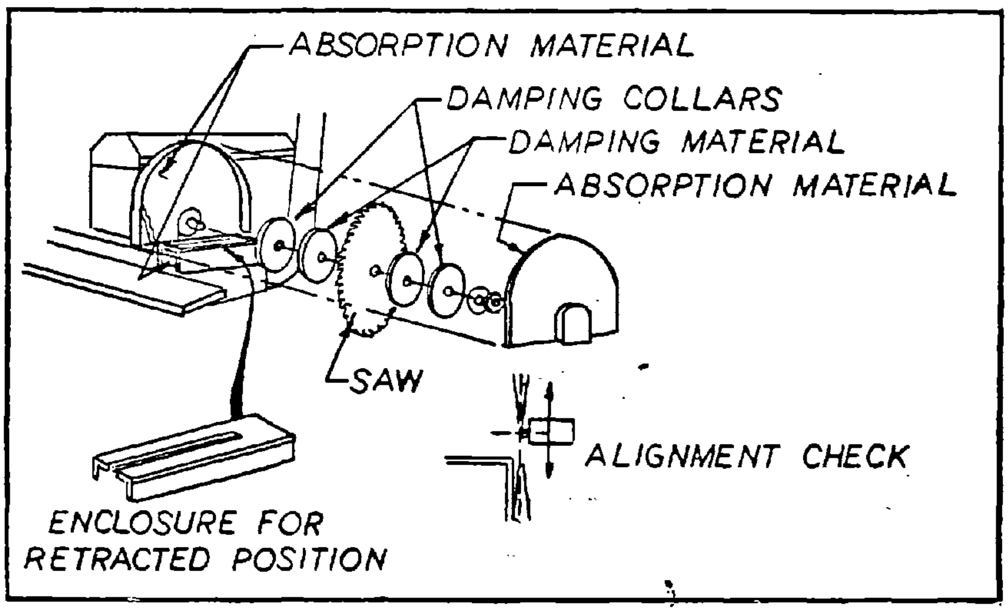

Figure 2.50 Techniques for cut-off saw blade noise control

Increased blade mass for a given bending stiffness increases the critical frequency. This may be accomplished by a change to plate materials having a higher mass or by artificially increasing the mass by attachment of weights to the existing saw plate (plugs, etc.).

A reduction of radiating area can also result in reduced radiated power, as indicated in equation (2.18). It should be noted that a reduction in area is sometimes accompanied by higher velocity levels due to the distribution of energy over a smaller area.

The noise reduction techniques for circular saws discussed in this section include:

(a) reduced excitation by

- reduced mechanical sources
- reduced aerodynamic sources
- reduced impact sources

(b) reduced response by

- increased stiffness
- increased damping
- alteration of wave propagation characteristics

(c) reduced radiation by

- acoustical shields
- alteration of the critical frequency

Noise control techniques best suited to the industrial cut-off saw are shown in Figure 2.50. The system shown typically reduces noise levels by 10 to 15 dBA and results in both idle and cutting noise levels of 90 dBA or below in most cases.

Control of Machine Vibration Noise

Numerous machines in the woodworking industry radiate noise as a result of structural vibration of the machine parts, components, and castings. This vibration is usually transmitted to the machine from a rotating tool or cutter which is acted on by unsteady forces. These forces can be of a rotational nature such as unbalance, faulty bearings, etc. or may result from tool impact on the workpiece. The response of the machine itself may be of a resonant nature or purely forced vibration response at the excitation frequency(s). Techniques for reducing input forces outlined previously are applicable in this case. Conventional techniques for vibration isolation may be used to attack such problems; however, the isolation must be inserted between the source of vibration and the remaining machine components if noise reduction is to be accomplished. Isolation of a machine from the floor is effective in reducing noise only when the floor is radiating sound. Floor vibrations are important in some cases but usually radiate low frequency sound which is of little importance. There are other advantages in machinery isolation, and it is highly recommended. Machine isolation from the floor should always be done prior to constructing acoustical enclosures since low frequency vibrations may excite higher frequency vibrations in an enclosure wall section. An alternative is to isolate the enclosure from the floor.

In some instances in the woodworking industry (routers, carvers, etc.) the necessity of isolating relatively low frequency vibrations requires an isolator that is too soft to permit machine operation. In such cases, structural damping techniques may be required. In cases where forced response is involved, the use of an acoustical enclosure may offer the only practical means of noise reduction short of machine redesign. The important aspects of machine vibration produced noise can be discussed by considering the special case of the high speed router, a machine which has been studied extensively[12]. The main results of this study are presented herein.

Reduction of Machine Excitation

The primary excitation sources for the router were found to be bearing and belt induced forces being transferred to the main machine structure. During cutting, this excitation is amplified as a result of the additional input force due to tool/workpiece interaction. Many older machines employ bearing systems that do not perform well from a vibrational standpoint when operated at high rotational speeds. In these cases, the structural vibration levels during idle are quite high, and the contribution of cutting forces does not appreciably increase noise levels.

The effects of the particular cutter design on radiated noise were also studied in reference 9. The type tool has some effect on radiated noise levels; however, the mechanism of vibration transmission from the bearing housing area was found to be of major importance. Isolation of the bearing housing from the rest of the machine, as illustrated in Figure 2.51, has been used successfully in reducing noise levels. The utility of the isolation system is dependent on the resulting deflection, illustrated in Figure 2.52.

Reduction of Machine Radiation

In cases where it is impractical to reduce input forces by bearing system alteration and the use of isolation and damping is either impractical or undesirable, a compact acoustical enclosure system can be used effectively. The system shown in Figure 2.53 for the high speed router has resulted in noise reductions during idle and cutting which are comparable to the isolation techniques. The validity of the technique is predicted on the dominance of machine vibration as the dominant noise source.

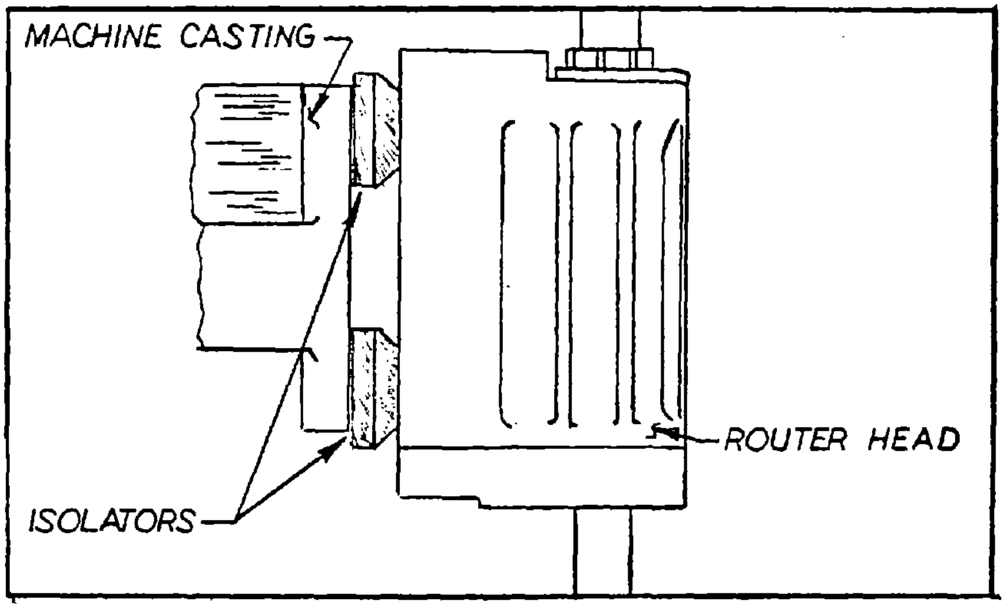

Figure 2.51 Isolation of router head

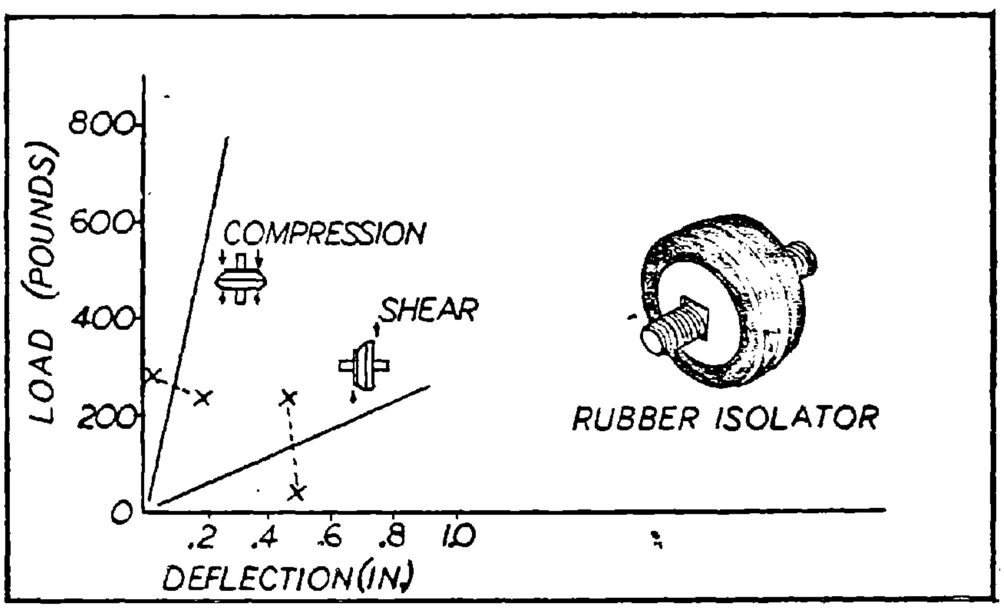

Figure 2.52 Properties of isolator

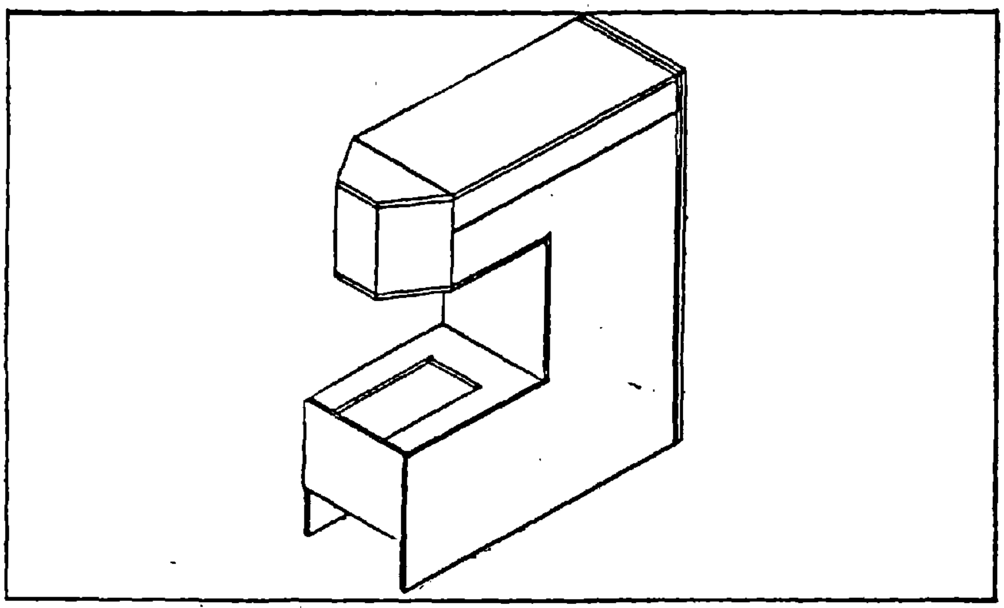

Figure 2.53 Enclosure for router

III. WOODWORKING MACHINERY NOISE PRODUCED BY AERODYNAMIC SOURCES

A detailed consideration of aerodynamic noise reduction is presented in this section. The presentation is made according to three categories of tooling: 1) long cylinders with relatively low tip speeds (around 100 ft/sec); 2) thin disks such as saw blades; and 3) intermediate size cylinders which may operate at relatively high tip speeds. The factors which have an effect on noise are treated separately where possible so that an estimate can be made of the noise reduction benefit that can be obtained from a specific change in an operational or geometric variable.

Scaling Laws for Long Cutterheads

The basic objectives of investigations into control of noise at the source are to understand the mechanism of noise production and to develop relationships among the physical system parameters which give a description of the acoustic power emitted. A result of such studies can be scaling laws which show the dependence of specific parameters on the acoustic power radiated by the source. This kind of understanding has been developed for aerodynamic noise, based primarily on studies of cutterheads of the size (diameter and length) normally used in wood planers[2,13]. The results of these works are presented here and are interpreted in terms of the reduction (or increase) in noise for a given change in a specified variable.

Noise Reduction Due to Parameter Changes

The physical system parameters that might be expected to affect noise emission are rotation speed, clearance between the table and knife tips, number of knives, knife projection above the base circle of the cylinder, length of cylinder, cutting circle diameter, and presence of induced air flow over the cylinder. Functionally, this dependence can be expressed as

$$W = f(N_b, D, N, C, L, V_a, h) \quad , \tag{3.1}$$

where W = acoustic power,
N_b = number of knives,
D = cutting circle diameter,
N = rotational speed,
C = clearance (tip/table),
L = cylinder length,
h = knife projection
V_a = induced air flow velocity.

It is noted in equation (3.1) that table lip geometry has not been included. Changes in table lip geometry can be expressed in terms of an "effective" increase in clearance and can be associated, therefore, with the parameter C.

In the following sections, the parameters included in equation (3.1) are treated individually; for example when the effect of rotation speed on acoustic radiation is considered, the other variables are held constant. The equations and graphs presented are based on analytical and experimental studies.

Effect of Changes in Rotational Speed

The parameter that has the most dramatic effect on sound power reduction is tip speed. For a given clearance, the relation for sound power level reduction is given by

$$\Delta L_w = 10\beta\log_{10}(V_i/V_f), \tag{3.2}$$

where ΔL_w = sound power level reduction, dB,
V_i = initial knife tip velocity,
V_f = final knife tip velocity,
β = clearance parameter = $4 + 0.46\log_{10}(f)$, where $f = .31(.1+C)^3$ and C is the clearance in inches.

The preceding relation, equation (3.2), shows that the slope of ΔL_w versus the speed ratio, V_i/V_f, depends on the clearance parameter β. For small values of clearance, $\beta \approx 4$ where as for larger clearances $\beta \approx 6$. The case of $\beta= 4$ is indicative of a monopole source whereas $\beta= 6$ refers to a dipole type source. A realistic range for clearances is .04 to 1.20 inches, and these values are used to produce practical upper and lower bounds for reduction in sound power level due to a reduction in speed. These curves are shown in Figure 3.1. A speed ratio reduction of up to ten (10) is used in the figure which should cover most situations encountered.

While Figure 3.1 shows the reduction in sound power level with reduction in speed, it is often desirable to increase speeds for production purposes. The preceding equation can be used to predict the increase in radiated sound power level for an increase in rotation speed,

$$\Delta L_w = 10\beta\log_{10}(V_f/V_i) \;.. \tag{3.3}$$

In this case V_f/V_i is greater than 1, and ΔL_w is positive. To avoid any confusion in using the same curve for reductions and increases, Figure 3.2 has been prepared to show the increase in sound power level for a given increase in speed.

Effect of Cutterhead/Table Clearance

The reduction in noise that can be expected by increasing the clearance can be estimated from the following relation which gives the reduction in

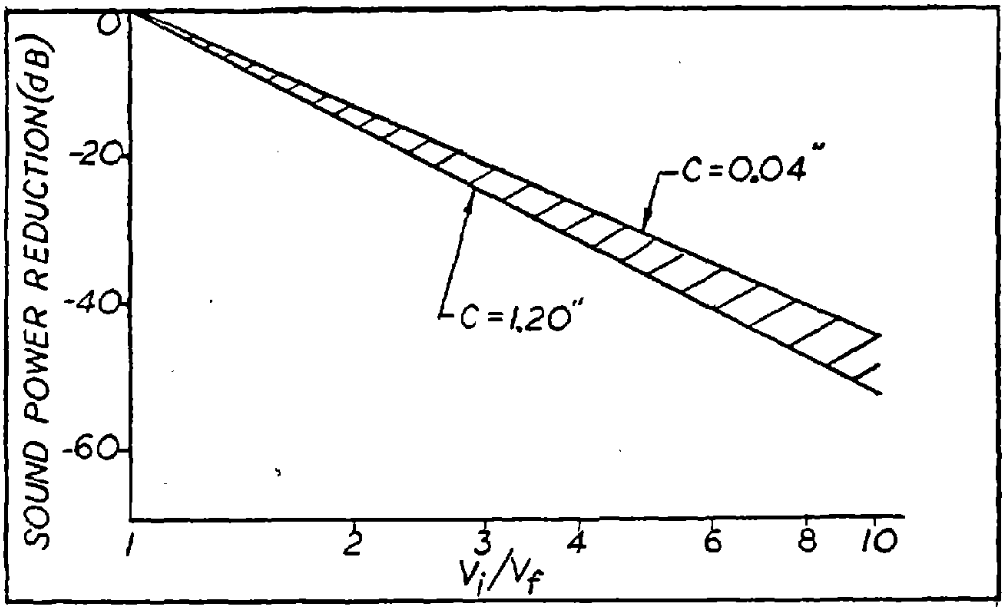

Figure 3.1 Sound power reduction versus speed ratio

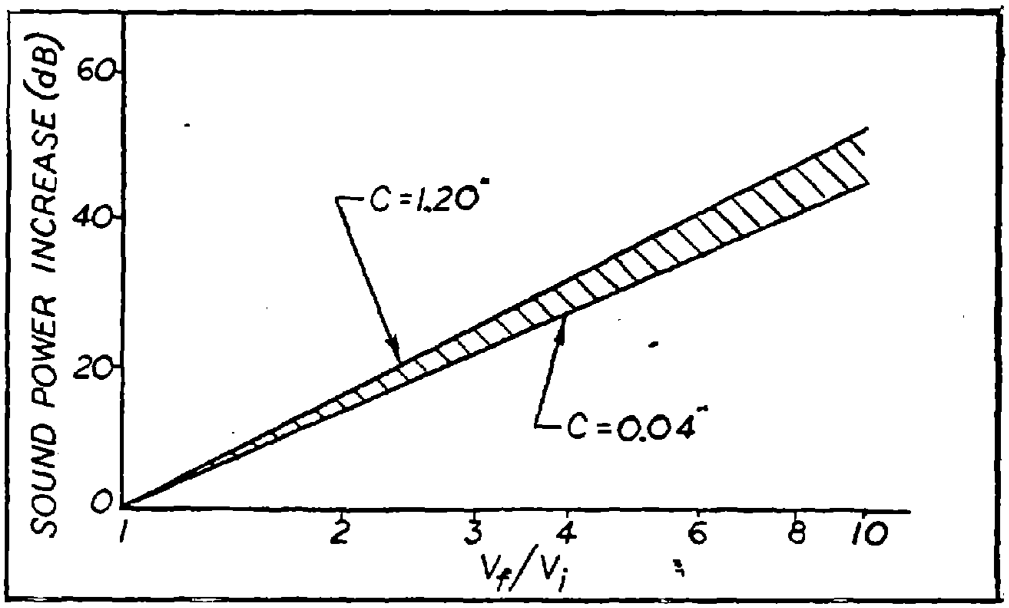

Figure 3.2 Sound power increase versus speed ratio

acoustic power level due to an increase in clearance,

$$\Delta L_w = (\Delta E) 10\log_{10}(M) \; * \qquad (3.4)$$

where ΔL_w = sound power level reduction, dB,
M = V/a (ratio of velocity of knife tip to the velocity of sound),
V = $\pi DN/720$ (knife tip velocity),
a = velocity of sound,
D = cutting circle diameter,
N = rotation speed (revolutions/minute),
ΔE = $0.46\log_{10}(f_2/f_1)$ (the clearance parameter),
$f_{1,2}$ = $5\times10^3(.11+C_{1,2})^3$, where $C_{1,2}$ represents the initial or final clearance in inches.

The preceding equation is presented in graphical form in Figure 3.3. The lower set of two curves holds for an initial clearance, C_i, of .04 inches while the upper set of curves refer to an initial clearance of .40 inches. The figure <u>cannot</u> be used to simultaneously estimate the effect of clearance changes and speed changes on sound power level reduction. The upper (7200 RPM) curve and lower (3600 RPM) curve in each set shows how speed affects the requirement on final clearance for a specified power level reduction. For example, for an initial clearance of .04 inches, a 10 dB reduction can be obtained by selecting a final clearance of .50 inches if the operating speed is constant at 3600 RPM; whereas at 7200 RPM, the final clearance must be 1 inch to give the same reduction. Further, two machines with constant operating speeds of 3600 RPM and 7200 RPM both with an initial clearance of .04 inches would exhibit reductions of about 8 dB and 6 dB, respectively, if the clearance in both machines were to be increased to .40 inches. It is seen that reductions are much less dramatic for an increase in clearance when the initial clearance is large.

Estimates of power level reduction can be obtained from Figure 3.3 for the range of the parameters of most practical interest. For applications where parameter values do not match those used in Figure 3.3, equation (3.4) given previously for power level reduction due to an increase in clearance can be employed. To illustrate the computational procedure, an example is given for a set of parameters that fall within the range of the figure. The problem is to evaluate the benefit relative to reduction in sound power level of changing the clearance from an initial value, C_i, of .40 inches to a final value, C_f, of .80 inches. The RPM is maintained at 3600, the cutting circle diameter is 6.36 inches, and 1130 ft/sec is used as the speed of sound. The computation is as follows:

$$V = \pi DN/720 = (\pi)(6.36)(3600)/720 = 99.85 \text{ ft/sec}$$

* The particular form given in equation (3.4) for the parameter, f, is restricted to straight knives at angles in the neighborhood of 18 degrees to the cutter.

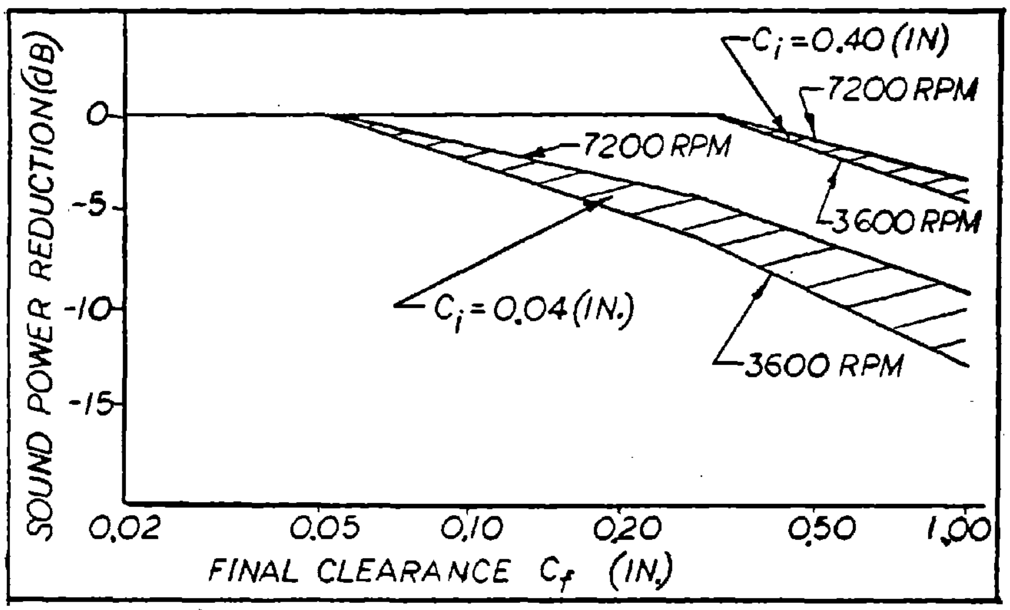

Figure 3.3 Sound power reduction versus clearance

and for the quantity M,

$$M = 99.85/1130 = 0.09$$

$$\frac{f_2}{f_1} = \frac{5\times10^3(.11 + .80)^3}{5\times10^3(.11 + .40)^3} = 5.84$$

$$\Delta E = 0.46\log_{10}(f_2/f_1) = 0.46\log_{10}(5.84) = 0.35$$

$$\Delta L_w = (\Delta E)\{10\log_{10}(M)\} = 0.35\{10\log_{10}(0.09)\}$$

$$\Delta L_w = -3.65 \text{ dB}$$

Thus, if all parameters are held constant except the clearance, which is increased from .40 to .80 inches, then a reduction in acoustic power level of about 4 dB can be expected. Of course, for this set of numerical values, the reduction can be read directly from the curves of Figure 3.3.

Effect of Cutterhead/Table Length

When speed of rotation, clearance, and cutterhead/table lip geometry are held constant, the reduction in sound power level at a particular harmonic frequency due to a change in effective cutter/table interactive length is given by[2],

$$\Delta L_w = 20\log_{10}(L_2/L_1) \;,\; f_n \leq 8600/L_1 \text{ (Hz)} \tag{3.5}$$

or

$$\Delta L_w = 10\log_{10}(L_2/L_1) \;,\; f_n > 8600/L_1 \text{ (Hz)} \tag{3.6}$$

where L_1 = initial table length,
L_2 = final table length,
f_n = harmonics of blade passage frequency, (Hz) = $n(N_b)(RPM/60)$,
N_b = number of knives,
n = integers, 1, 2, 3 ....

These predictive equations hold for values of clearance, C, in the range $.04 < C < 1.20$ inches. Figures 3.4 and 3.5 give the reduction in sound power level as a function of length ratio and precentage reduction in length. It is seen that the sound power level reduction for a given percentage reduction in length

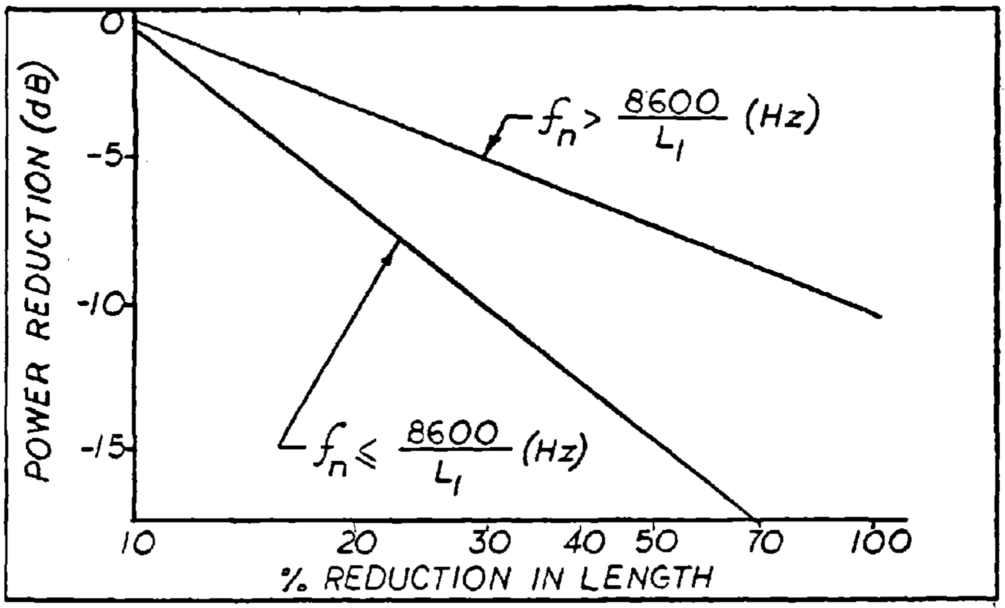

Figure 3.4 Sound power reduction versus change in length

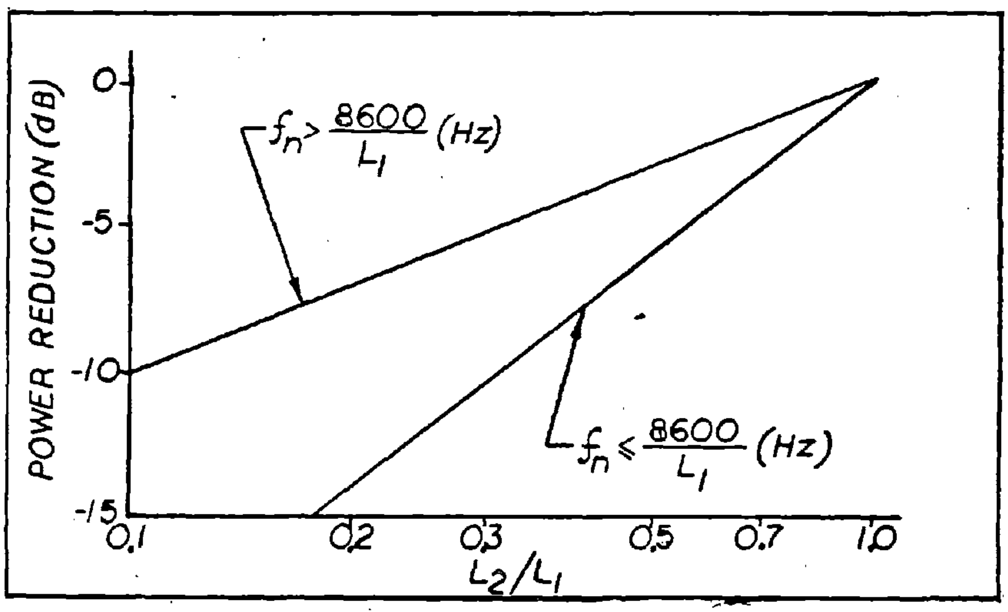

Figure 3.5 Sound power reduction versus length ratio

is greatest for the lower frequency components contained in the power spectrum. The two curves serve to give an upper bound and a lower bound for reduction in overall sound power level. For practical rotation speeds, the power spectrum is controlled by the lower harmonics so that sound power level reduction is controlled by the lower curve in Figures 3.4 and 3.5. If the attenuation characteristics of the A-weighted network are employed, then the higher harmonics become more important so that A-weighted sound power level reduction for a change in length would more likely follow the upper curve in Figures 3.4 and 3.5.

It must be emphasized that the physical situation appropriate to the preceding discussion consideras a single table lip that interacts with the rotating cutterhead. If more than one surface is present, then the usual simplified rules for sound power addition can be employed to estimate the total effect of changing the effective interaction length of all such surfaces. For example, if two such surfaces are present and their separate effects produce sound power levels of 85 dB and 88 dB, then the power level as determined from measurements due to both sources would be about 90 dB. Reducing the 88 dB source by 3 dB would produce only a 2 dB reduction in combined power level, i.e., the combined power level would be 88 dB as compared to the original value of 90 dB.

Effect of Knife Projection

It has been found that changes in sound power level due to changes in knife projection above the cylinder body are more pronounced when the cutterhead is operating in a free condition, i.e., when no nearby stationary surfaces are present. The overall power level produced by the free rotor will usually be significantly lower than that produced when a lip or surface is present. Thus, to treat knife projection as an independent variable, both categories must be considered.

Data from several sources have been analyzed and give consistent results when interpreted to compare increases in power due to relative changes in knife projection. Gib-face design and gullet geometry serve only a secondary role in establishing the level about which variations in power level occur due to changes in knife projection[2,13,14].

Knife Projection in Presence of Lip

The overall sound power level produced when a cutterhead is operating in the presence of a lip is determined primarily by the clearance between the knives and lip, tip speed, and length of cutterhead. Knife projection enters into such determinations through the computation or establishment of the tip velocity (V). If knife projection is increased without changing RPM, then the tip speed would be increased; and if no table adjustment is made, then the increase in noise would be determined by an increase in tip speed accompanied by a reduction in clearance. For example, for the same RPM, tip velocity ratios are equal to ratios of diameter. Increases in sound power follow a V^4 to V^6 power law and increasing knife projection from .08 to .40 inches would cause an increase in sound power level on the order of 3 dB even though adjustments were made to maintain constant clearance.

Taking into account the difficulties in interpretation, available data indicate that the separate effect of changes in knife projection on sound power production is to cause an increase of about 3 dB when the projection height is doubled. Such a variation is shown in Figure 3.6 in terms of the ratio of final height, h_f, to the initial height, h_i. The clearances for the experimental setting in which these data were obtained ranged from .08 to .12 inches. The data also show that the relative increase in sound power level tends to be less than the 3 dB per doubling of h for values of h exceeding .64 inches. Available information indicates that an upper limit of a 3 dB increase per doubling of knife projection can be expected. The curve given in Figure 3.6 can also be used to estimate the reduction in power level that can be expected by reducing the knife projection.

Effects of Knife Projection with No Lip

As noted previously, the effect of knife projection for a free rotor is more important than the situation in which an interacting lip is present. For this case, the variation depends on the initial projection h_i. An examination of available data shows that for projections, h, less than about .01 inches, the increase in sound power level is about 3 dB per doubling of the ratio h_i/h_f, whereas for h greater than about .01 inches the variation is 6 dB per doubling of the ratio h_i/h_f. This is illustrated in Figure 3.7 which can be used to estimate the effect of change in knife projection on sound production when no interacting lip is present.

Comparing Figures 3.6 and 3.7, it is seen that the largest reduction that can be expected due to a change in knife projection (all other variables remaining the same) would be 6 dB for a reduction in knife projection by a factor of two.

Effect of Induced Airflow

In most operations, the machines are equipped with dust and chip removal systems. The systems employ a fitted hood to collect dust and chips as air is drawn by the cutterhead. The air handling systems are a source of noise, but usually their separate effects do not contribute to the overall noise level. The interactive effect of airflow across the cutterhead required for dust and chip removal does cause an increase in noise when compared to the noise produced by an idling cutterhead without airflow.

A detailed description of air flow over the cutterhead in an actual machine is difficult due to the complexity of the geometry and the variation in air velocity in the vicinity of the rotor. A first order of magnitude estimate of airflow effects can be obtained by considering a more ideal situation, i.e., a rotating cutterhead in a uniform flow field, Figure 3.8. It has already been shown that for a free rotor, sound power follows almost a V^6 variation, where V is the tip velocity. Airflow, then, effectively causes an increase in tip velocity so that if V_a is the airflow velocity and V_t is the tip speed, the increase in power level due to induced airflow should be somewhat less than

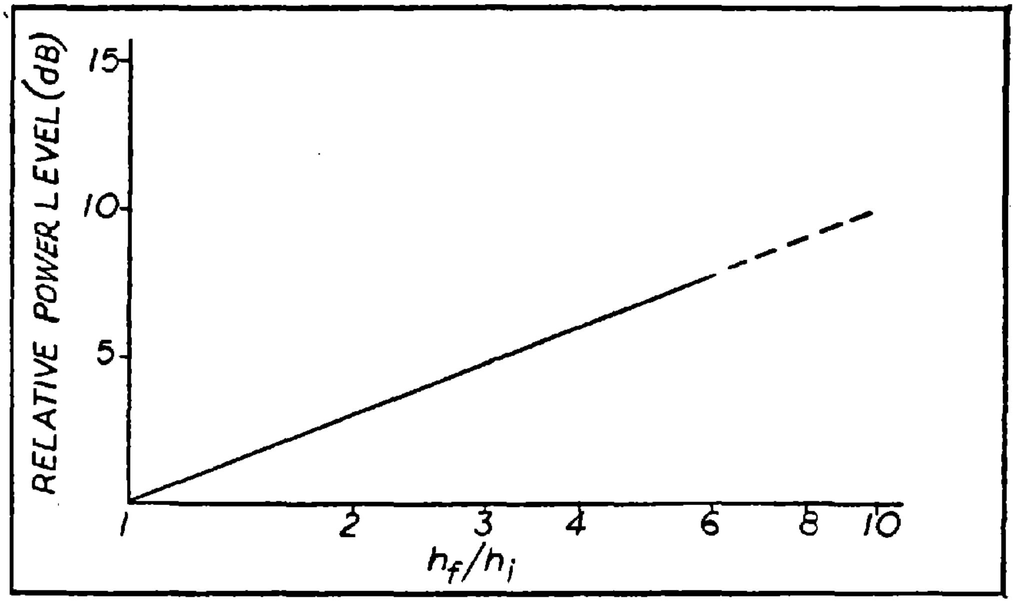

Figure 3.6 Change in noise level due to change in knife projection

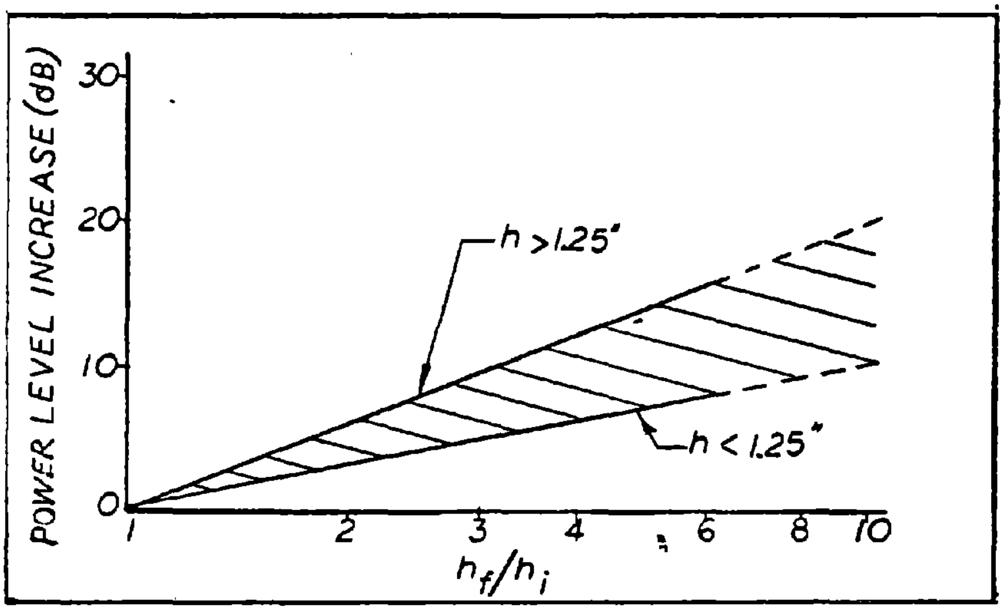

Figure 3.7 Relative noise level versus knife projection ratio

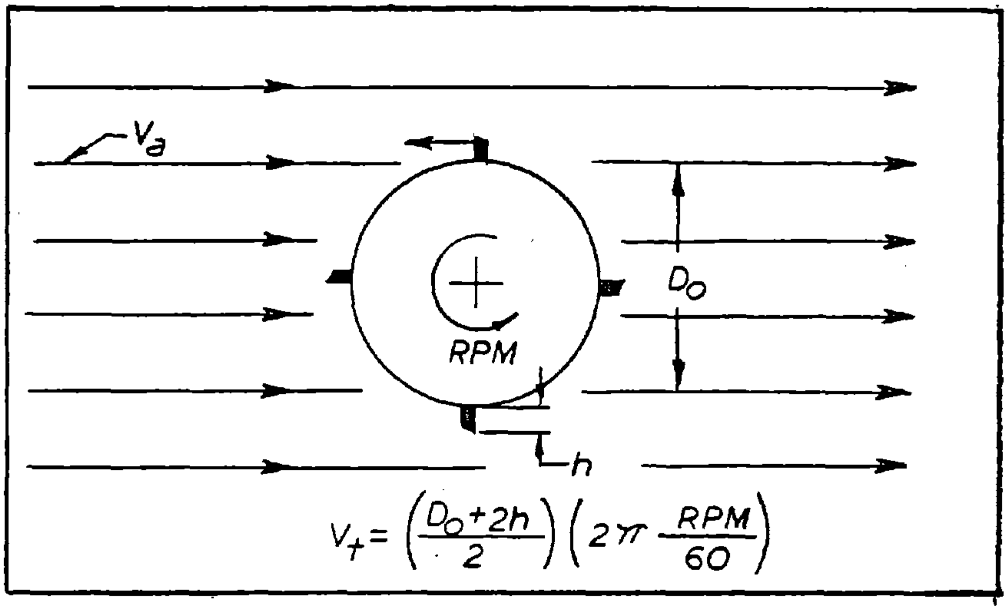

Figure 3.8 Induced airflow over rotating cutterhead

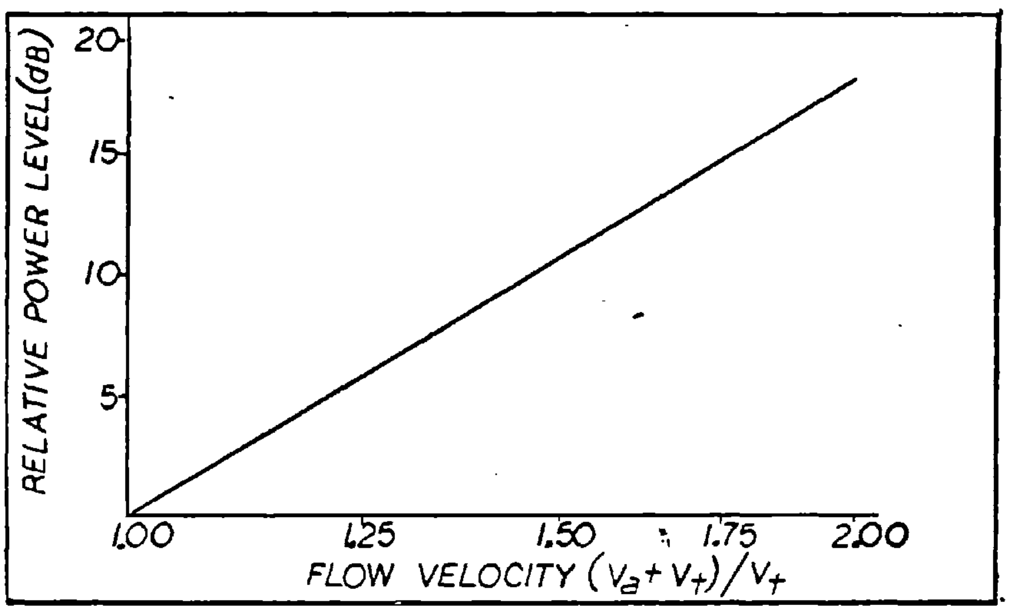

Figure 3.9 Increase in sound power due to airflow

predicted by the following equation:

$$\Delta L_w = 60\log_{10}\{V_a + V_t\}/V_t \text{ dB} \tag{3.7}$$

Several experiments conducted on conventional wood planers indicate that the preceding equation gives reasonable estimates. The difficulty in such estimates is the specification of an effective airflow velocity V_a. The pretive equation is illustrated in Figure 3.9 for what is believed to be an adequate range of the velocity ratio.

Noise Reduction Due to Alteration of Table Lip Geometry

In some cases it may be more practical to modify the table lip geometry than to change the clearance, rotation speed, or other parameters to achieve a reduction in aerodynamic noise. This approach has been studied extensively through experimental means. Brooks and Bailey[2] show that lip modifications can be included in a semiempirical prediction equation which gives consistent results by considering lip alteration to effectively increase the clearance. Two table lip modifications are shown in Figure 3.10. The illustration on the left shows a saw-tooth configuration with a base triangle dimension of about .43 inches and triangle height of about .40 inches. Material removal in the form of triangles with given dimensions to form a saw-tooth pattern along the free edge gives an effective increase in clearance of about .20 inches. The reduction in noise resulting from such modifications depends on the actual clearance between the knives and the table lip.

Figure 3.3 can be used to make estimates of power level reduction if the actual clearance and the effective increase in clearance due to lip modification are specified. For example, if the actual clearance for a solid lip (no modifications) is .04 inches, then the .20 inches effective increase in clearance for the saw-tooth configuration noted previously gives a final effective clearance of .24 inches. Figure 3.3 shows that such a change would give a sound power level reduction of about 6 dB if the speed of operation were 3600 RPM. If the actual clearance were initially .40 inches, then the modification would cause a much less dramatic reduction, about 2 dB, in sound power level.

Figure 3.10 also shows a slotted modification to the table geometry. The slots used for experimental studies were about .63 inches wide; the three shorter slots being centered about .79 inches from the free edge. This configuration gives an effective increase in clearance of about .40 inches. Using this value, an estimate of power level reduction can be obtained by using Figure 3.3 as in the previous example.

Spiral, Segmented, and Staggered Knife Geometries

Figure 3.11 shows several cutterhead designs that are commonly used to reduce noise. While the most significant aspect of these heads is the reduction of cutting noise, there is also a reduction in idling or aerodynamic noise.

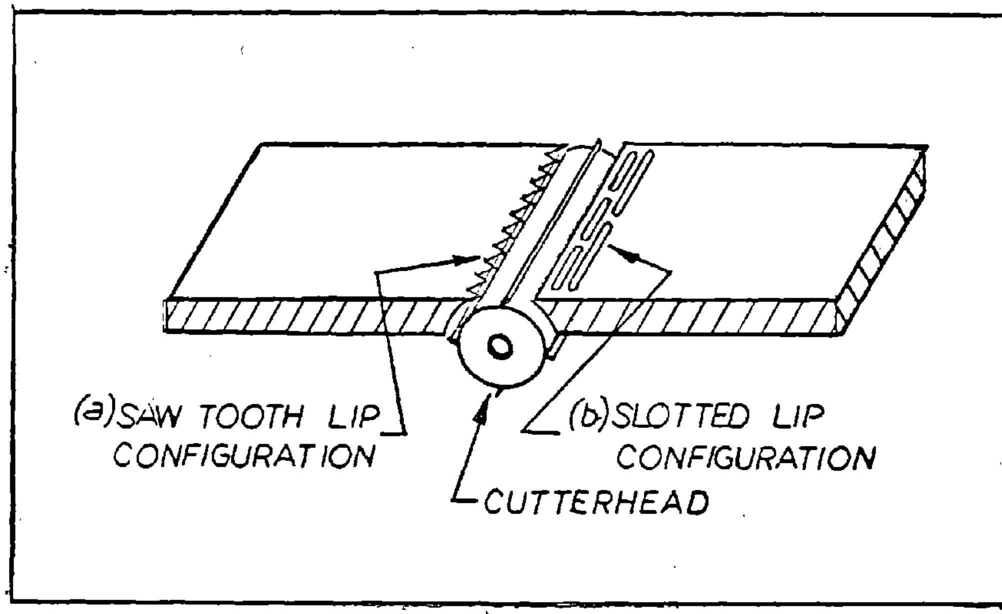

Figure 3.10 Table lip modifications

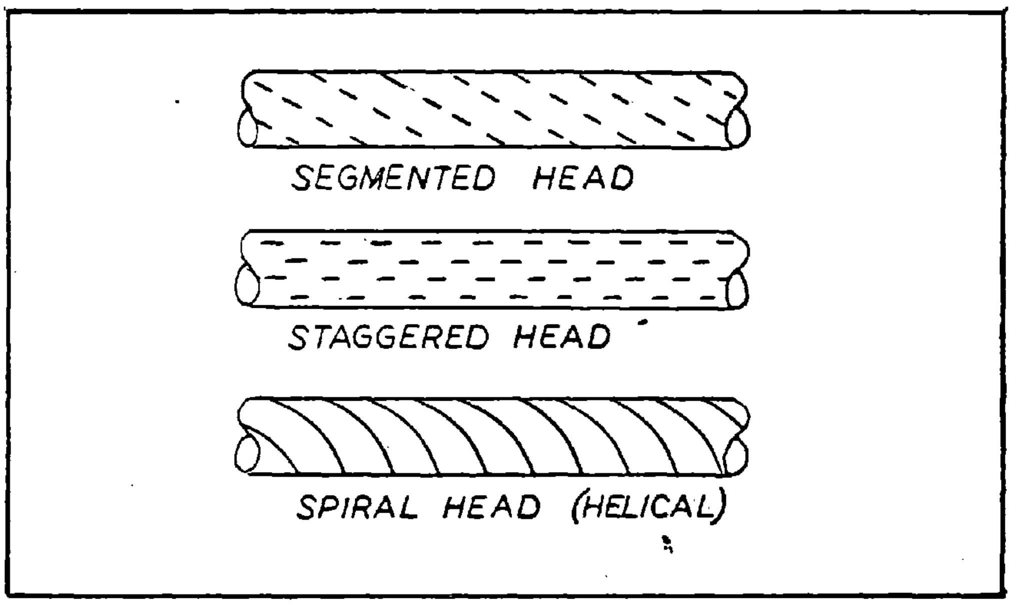

Figure 3.11 Noise reducing cutterheads

The physical process by which the segmented and staggered knife heads reduce aerodynamic noise is quite similar to the way in which the saw-tooth table lip modification works. That is, the intensity and rate of mass injection associated with monopole source mechanisms are decreased. Additionally, the rate and intensity of force fluctuation of the dipole mechanism are reduced. Only a qualitative explanation is available for the aerodynamic noise reduction observed for these geometries; specific scaling laws have not been developed.

A detailed explanation has been given for the reduction of aerodynamic noise provided by long cutterheads with helical knife geometry. This study[2] accounts for both monopole and dipole source mechanisms which have been demonstrated to simultaneously contribute to the sound power output, depending on the clearance between the knife tips and the table lip. Figure 3.12 shows the sound radiation efficiency for the monopole source mechanism (air mass injection) for a long cylinder operating in the presence of a table lip. The independent variable, K_nL, is related to frequency by

$$K_nL = 2\pi f_nL/a \qquad (3.8)$$

where f_n = harmonics of the blade passage frequency,
L = table/cutterhead length,
a = speed of sound.

It is necessary to introduce a parameter, α, to relate the helix angle, ϕ, to the cutterhead tip velocity, V,

$$\alpha = (a/V) \tan\phi \; . \qquad (3.9)$$

For the straight knife case, $\phi = 0$ so that $\alpha = a/V$. For a given speed of operation and fixed diameter, α depends only on the helix angle, ϕ. It is seen, Figure 3.12, that the radiation efficiency increases steadily with increases in K_nL for the straight knife case ($\alpha = 0$) whereas for larger values of α, indicating a nonzero helix angle, the radiation efficiency levels off and shows no variation above $K_nL \simeq 10$.

To illustrate the difference between the straight knife and helical knife cases, consider a typical situation in which the speed of rotation is 3600 RPM, the cutting diameter is 6.30 inches, the table length is 30.00 inches, and the helix angle is about 45°. Choosing a harmonic near 1000 Hz,

$$K_nL \simeq 14, \; \alpha = 0 \text{ straight knife}$$

$$K_nL \simeq 14, \; \alpha \simeq 11 \text{ helical knife.}$$

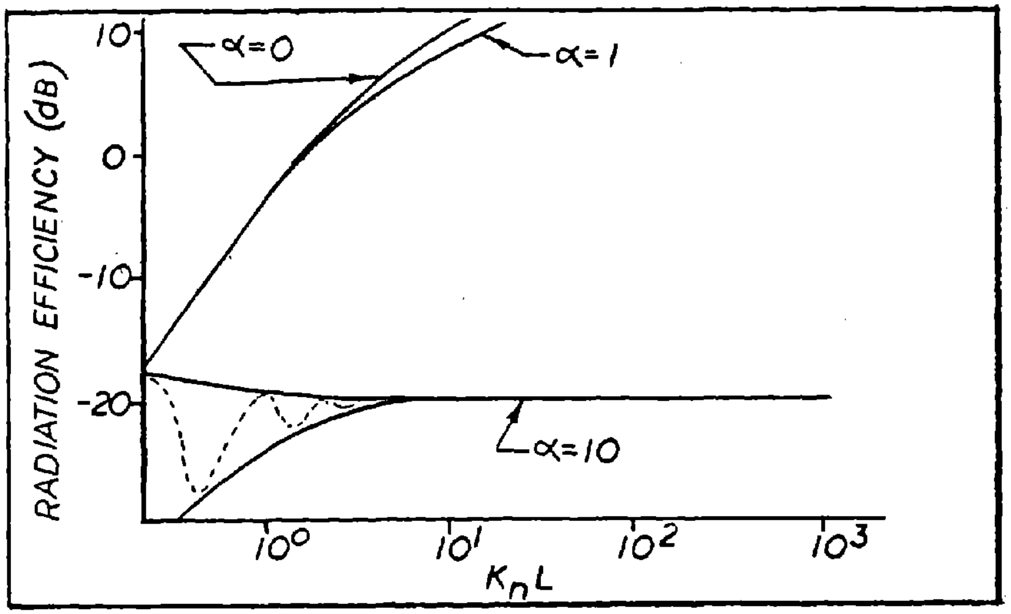

Figure 3.12 Radiation efficiency of monopole source mechanism

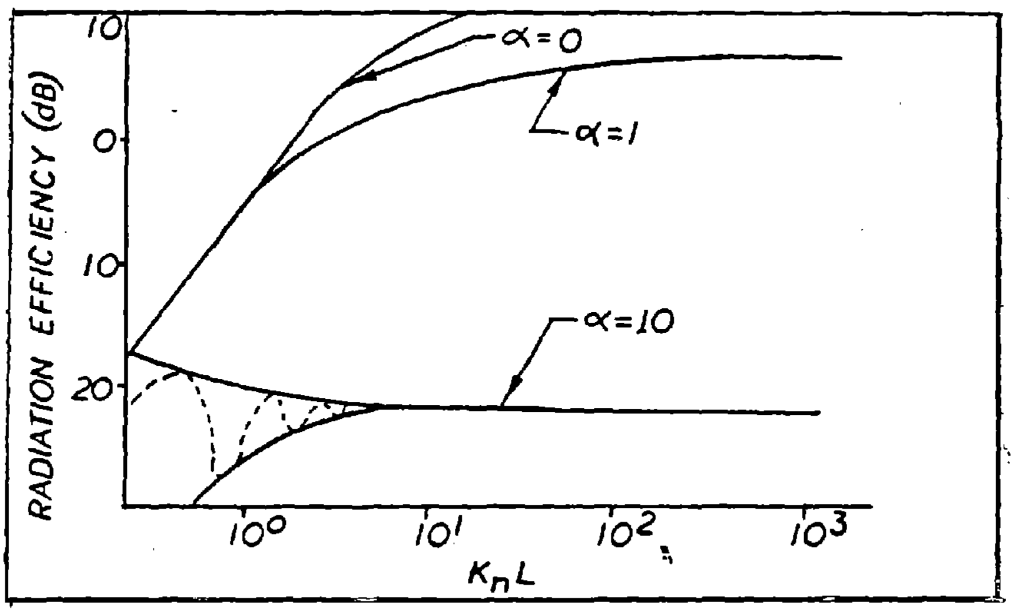

Figure 3.13 Radiation efficiency of dipole source mechanism

From the Figure 3.12 it is seen that a total difference of about minus 30 dB exists for the helical case as compared to the straight knife case. This is consistent with experimental observations that aerodynamic noise is essentially eliminated for helical knife geometries with a helix angle near 45°.

Similar trends are noted for the dipole source mechanism as illustrated in Figure 3.13. In general, increasing the helix angle while keeping other factors constant provides significant reduction in noise radiated by both dipole and monopole source mechanisms.

Scaling Laws for Circular Saw Blades

The aerodynamic noise produced by circular saw blades results from air disturbances caused by discontinuities in the saw periphery. Experimental studies have revealed a dipole like dependence on peripheral speed as well as dipole source directivity characteristics. This characterization realistically describes the increase in radiated power with increasing tip speed but does not define the effect of the various geometrical parameters known to affect noise generation. In an effort to better understand the effect of the various parameters involved in circular saw blade design, a program aimed at identifying the relationships between important parameters was undertaken. The results of this study[3] have been incorporated into simple prediction equations in order to aid in circular saw blade design and selection.

In studying circular saw blade noise, it is helpful to examine the means by which the air surrounding the rotating saw blade can be disturbed. The classical approach of source modeling in terms of monopole (mass fluctuation), dipole (force fluctuation), and quadrupole (shear stress) radiation is useful in the modeling process. The modeling process is considerably simplified if one of the above source mechanisms is clearly dominant from a noise generation standpoint. In the case of circular saw blades, a monopole source would most likely result in discrete frequency contributions at the tooth passage frequency and harmonics of the tooth passage frequency and would result in a 12 dB increase in sound power level per doubling of tip speed, indicative of a V^4 type power law. The dipole source represents fluctuating forces on the surrounding air and may be produced by drag forces, vortex shedding, and several other mechanisms. The V^6 power law associated with dipole sources leads to an 18 dB increase in radiated sound power level per doubling of tip speed. The quadrupole source mechanism results from fluid shear stress and is normally associated with turbulent flow fields. A V^8 power law is associated with quadrupole radiation and would result in a 24 dB increase in radiated power level per doubling of tip speed. Each of the three source models described contribute to the total aerodynamic noise generation from circular saw blades; however, the dipole source model is believed to best represent the dominant noise generation mechanism. That this is the case can be observed from the approximate $V^{5.6}$ power law shown in Figure 3.14 and the dipole like directivity pattern shown in Figure 3.15.

The circular saw blade can be described acoustically as a disk having a number of discontinuities spaced around the periphery which are actually gullets provided for stock removal during the sawing operation. As the blade

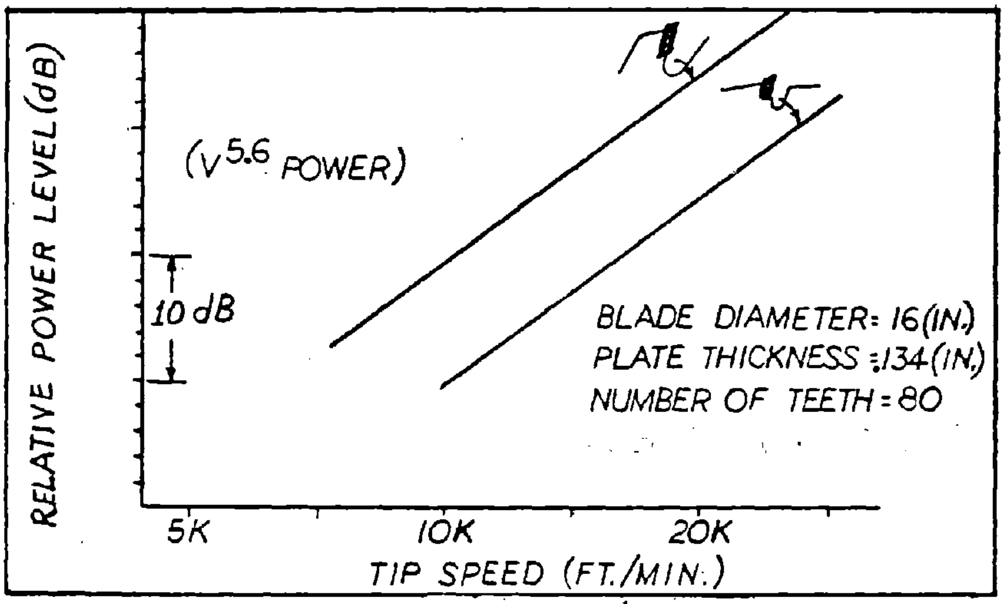

Figure 3.14 Sound power versus tip speed

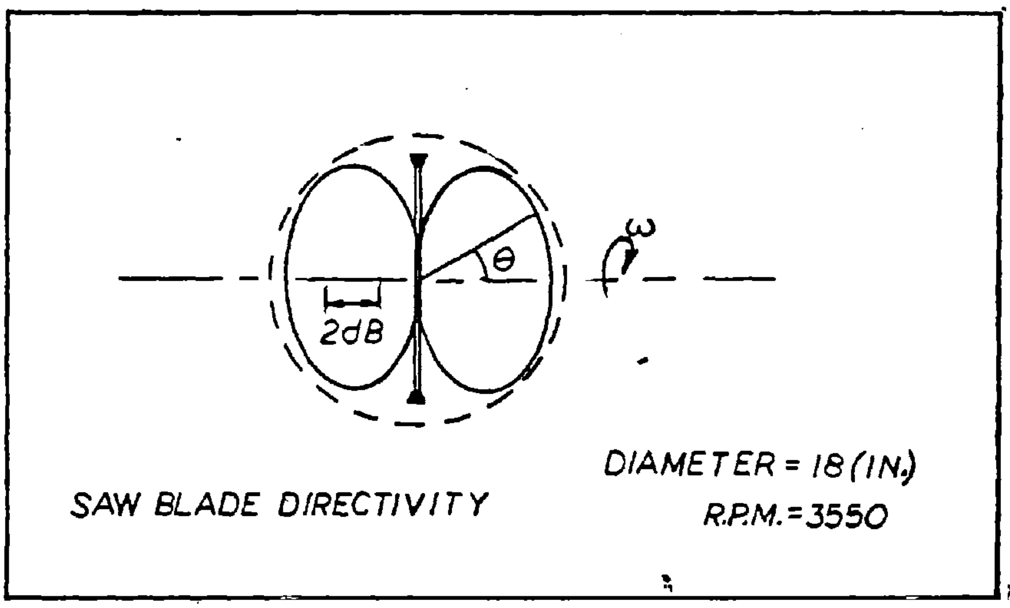

Figure 3.15 Directivity of saw blade noise radiation

rotates, a flow velocity (V) is produced near the periphery. The interaction between the flow field and the gullet area cut into the periphery causes the fluctuating force field to develop (for a nonrigid disk, this force can excite blade vibration).

General aerodynamic noise theory extended to cases where solid bodies are present allows for a dipole source distribution at the boundaries as well as the normal quadrupole sources. For the case of sound radiated from a turbulent boundary layer on a rigid flat plate at low Mach numbers, the dipole sources actually overwhelm the quadrupole sources. Surface irregularities drastically increase the boundary layer pressure fluctuation creating unsteady forces. The mechanism by which the unsteady forces occur is complex; one such mechanism is the shedding of vortices from the trailing edge of the discontinuity. The degree to which cavities influence the vortex action depends strongly on cavity size to cavity depth ratio. This ratio is known to affect the magnitude of the boundary layer disturbance, the fluctuating forces, and to facilitate the development of regular vortex patterns[15]. For the circular saw blade this ratio is taken as the ratio of gullet width (w) to plate thickness (t) as illustrated in Figure 3.16. The effect of this ratio on radiated acoustic power can be incorporated into a constant Q which is, in effect, a source strength.

The frequency spectrum is defined through the dimensionless Strouhal number which relates the peak frequency (f) to the free stream velocity (V) and a characteristic thickness (t), or

$$S_t = ft/V$$

The thickness parameter is seen to have a direct influence on both source strength (Q) and frequency content. The dipole acoustic power resulting from vortex motion can be expressed in the approximate form[10],

$$W \sim Q(Re)^{-.4} A \rho V^6 / a^3 \tag{3.10}$$

where W = acoustic power,
Q = source strength,
Re = Reynolds number,
A = disturbed area,
ρ = density of air,
V = tip velocity,
a = acoustic velocity.

From equation (3.10) it is observed that the sound power depends on the Reynolds number, the disturbed area, the source strength, and the tip speed.

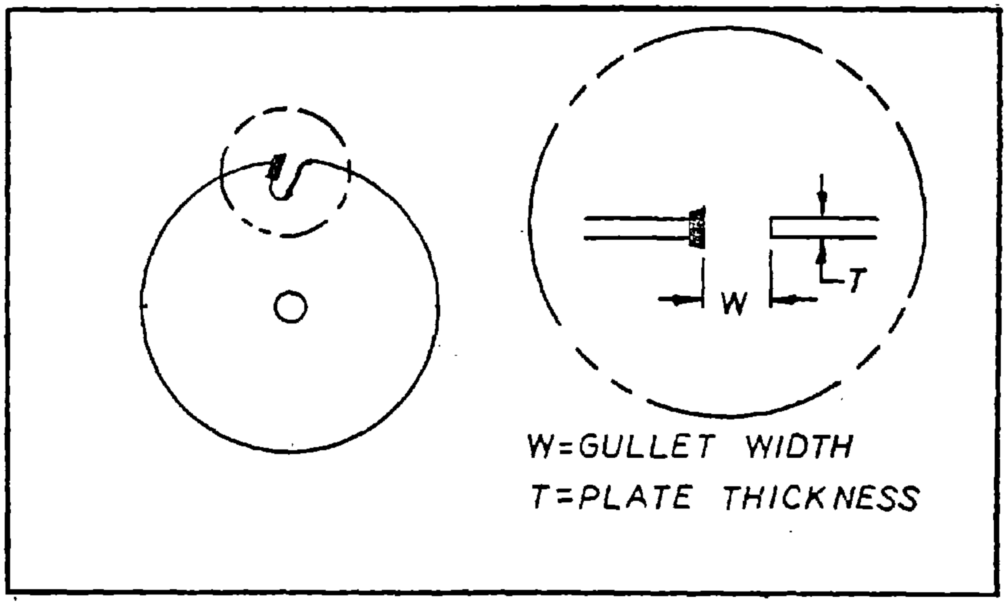

Figure 3.16 Gullet width and saw blade thickness

The Reynolds number is given by

$$Re = \rho V \ell / \mu \tag{3.11}$$

where ℓ is a characteristic dimension and μ is the viscosity of air. Equation (3.10) can be rewritten as

$$W \sim Q(\rho \ell / \mu)^{-.4} A \rho V^{5.6} / a^3 \tag{3.12}$$

indicating a variation in power with tip speed to the 5.6 power, a result that has been observed experimentally (see Figure 3.14). From a practical standpoint, variation in the quantities $(\rho \ell / \mu)^{-.4}$ and A in equation (3.12) are expected to result in relatively small differences in radiated power for a particular diameter saw blade. Since power level reductions in excess of 10 dB are needed to control aerodynamic noise of circular saws, the effect of these parameters will not receive further consideration. The effect of reduced tip speed has been shown to decrease the radiated noise by approximately 15 dB per doubling of tip speed. The remaining quantity of interest is source strength (Q), which must then be responsible for the wide variation in radiated power observed for different gullet designs (variations of 15 dB or more have been observed for different saw blades having the same tip speed). It is known that the vortex strength is dependent on the fluctuating forces in the disturbed region, which are in turn related to the particular gullet-plate geometry.

A characterization of the flow field involving the formation of regular vortex patterns in the turbulent boundary layer will not be attempted; however, the general trends discussed have led to a series of experiments designed to identify the importance of the various geometrical parameters. In these experiments, a parameter of major importance was identified as the gullet width to plate thickness ratio (w/t), which was found to govern both the radiated power and the resulting frequency spectra. A "tuned in" region was found to occur at a critical ratio, which may be characterized as an acoustic resonance. Other parameters such as gullet area, number of teeth, gullet spacing, etc. were not found to have a direct effect on sound production; however, there is a general tendency to reduce radiated power with reduced gullet size.

An illustration of the importance of these geometrical considerations with respect to radiated sound and frequency spectra is shown in Figures 3.17 and 3.18 which compare disks of equal size and gullet shape having thicknesses of 1/8 inch and 1/4 inch, respectively. It is observed from Figure 3.18 that the frequency spectra are different for the two cases and that the overall noise levels differ by nearly 15 dB . Efforts are continuing to obtain specific criteria for quieter saws based on an understanding of the source mechanisms and the effect of the various parameters on sound generation.

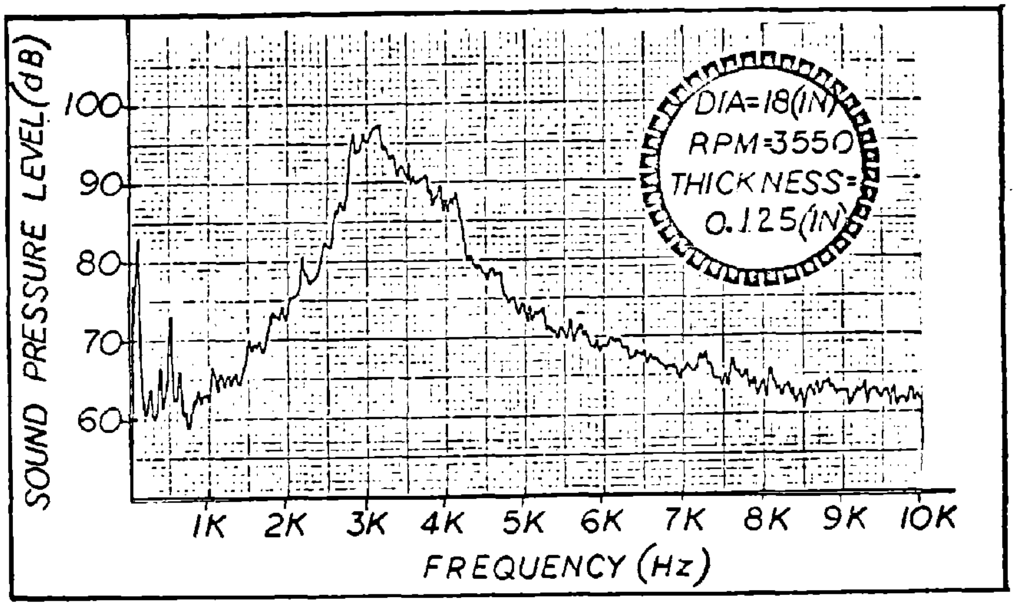

Figure 3.17 Noise spectrum for 1/8" blade thickness

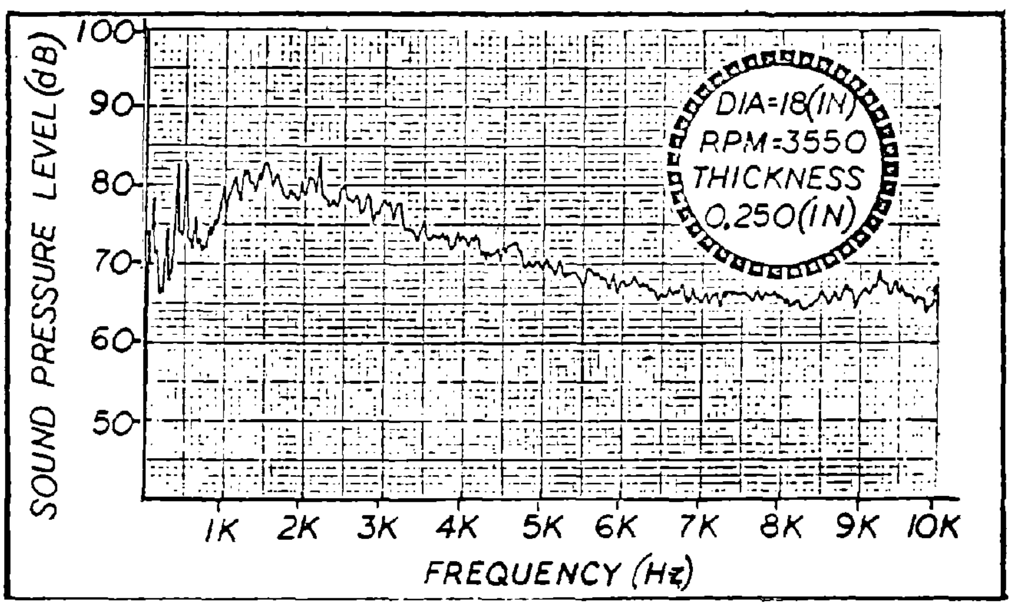

Figure 3.18 Noise spectrum for 1/4" blade thickness

Application to Intermediate Size Cutting Tools

Considerable theoretical and experimental study has been devoted to understanding noise generation mechanisms of long cutterheads. By contrast, relatively little effort has been given to intermediate length cutterheads as used in shapers, moulders, tenoners, etc.. In this section, the extent to which scaling laws developed for long cutterheads apply to intermediate size cutterheads is explored. The analysis is based on experimental studies conducted on shapers with two different types of cutterheads. Figure 3.19 shows the experimental arrangement for noise measurements taken on a shaper operating at 8810 RPM. Both cutterheads tested had three knives so that the blade passage frequency was 441 Hz. The results for three specific parameters are discussed: 1) free cutterhead versus cutterhead with fence in place, 2) height of cutterhead above table, and 3) constant diameter cutterhead versus one with variable tip diameter.

A series of measurements were taken to determine the increase in sound radiated when the fence was in place. The increase in average sound level was about 9 dB due to the presence of the fence with a clearance between knife tip and fence edge of about .20 inches. The cutterhead interacts with two edges of the fence so that if both edges contribute equal amounts to the increase, each edge is responsible for an increase of about 6 dB. Attempts made to estimate the effect of the fence from the semi-empirical scaling laws discussed previously for long cutterheads show that such extrapolations cannot be made with confidence. The scaling laws tend to overestimate the difference between the fence and no fence cases.

Figure 3.20 shows results for two cutterhead geometries and three machine setups. The average diameter of the tapered edge cutterhead was about 5 inches, as compared to a 4 inch diameter for the straight knife cutterhead. The first case given in Figure 3.20 is for the cutterheads operating in free space resulting in a difference in noise level of about 6 dB. For this case, the increase in sound power level is approximately,

$$\Delta L_w = 40\log_{10}(D_2/D_1) \qquad (3.13)$$

Equation (3.13) is based on a monopole source mechanism where D is the cutterhead diameter and the RPM is held constant. Using this expression and the average diameter of the tapered cutterhead, an increase of 5.6 dB is predicted as compared to the measured value of 6 dB. It is noted from Figure 3.20 that the relief provided by the taper effectively eliminates the effect of the fence for the variable diameter cutterhead. Furthermore, the proximity of the table is not important for the straight knife case whereas for the tapered knife there is an interactive effect.

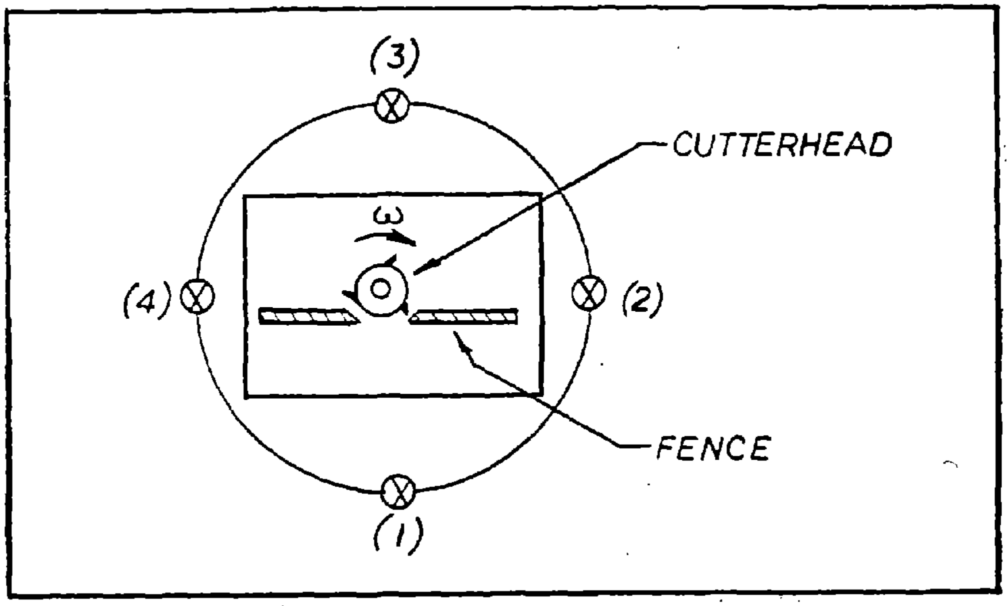

Figure 3.19 Noise measurements on shaper

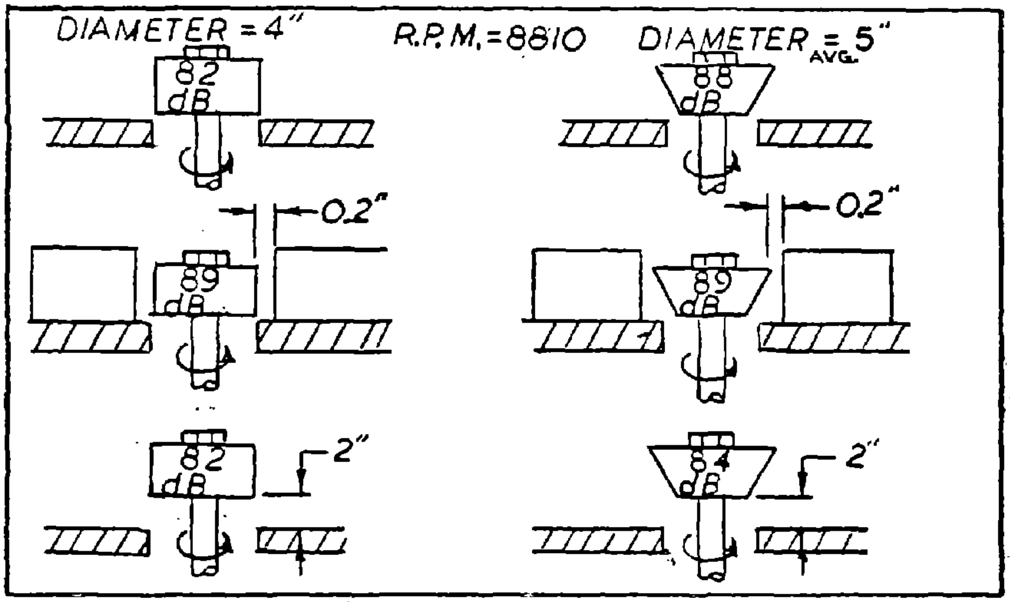

Figure 3.20 Noise levels for several cutting arrangements

IV. NOISE CONTROL GUIDE FOR WOODWORKING MACHINERY

In the preceding sections an overview of noise sources and control procedures was presented. Much of the discussion was related to vibration and aerodynamic phenomena and was not specific to particular machine classifications. A guide to noise control by machine type (as commonly used in practice) is presented in this section. This guide is given in Table 4.1. The first two columns give representative types of machinery used in the woodworking industries and the general function of each machine type. The third and fourth columns present the dominant noise sources identified with each machine category during both cutting and idling operation. These columns are arranged so that the dominant source is listed first in either the cutting or idling situation. For example, for planers workpiece vibration is usually the dominant source during cutting with aerodynamic noise being the secondary source during cutting. During idling operation, aerodynamic noise usually is the primary source while mechanical sources (drive rolls, chain drives, hydraulics, gears, etc.) are secondary. The last entry (router) is a case where the dominant source is the same for both idling and cutting.

The last four columns in Table 4.1 give a letter coded method of control for each machine classification for cutting and idling operation. In each case, at least two alternative approaches (direct and indirect) are listed. The direct approach normally applies to situations where standard tooling is employed whereas the indirect approach often involves the use of new or modified tooling.

The key to the letter guide of Table 4.1 is given in Table 4.2. Given with each letter in the table is a brief description of the modification or procedure for noise reduction and a reference to particular sections of this report where the control procedures or modifications are treated in more detail. As an example, the letter A refers to direct methodology for control of aerodynamic noise from standard straight knife cutterheads. As indicated, the parameters that dictate noise reduction are RPM, lip geometry, cylinder length, etc., and these parameters are treated in detail in Section III of this report.

Table 4.1 Noise Control Guide for Woodworking Machinery

Woodworking Machine		Dominant Noise Sources		Control Method* Guide			
Classification	Operation	Idling	Cutting	Idling		Cutting	
				Direct	Indirect	Direct	Indirect
Surfacer	Surface	*Aerodynamic +Mechanical	Workpiece Vibration Aerodynamic	A	B	C,E	D
Planer	Surface	*Aerodynamic +Mechanical	Workpiece Vibration Aerodynamic	A	B	C,E,F	D
Moulder	Surface Shape	*Aerodynamic +Electric Motor	Workpiece Vibration Aerodynamic	A	B	C,E,F	D
Facer- Jointer	Surface	*Aerodynamic +Mechanical	Workpiece Vibration Aerodynamic	A	B	C	D
Cutoff Saw	Crosscut	*Aerodynamic +Blade Resonance	Blade Vibration Aerodynamic	G	H	I	I,J
Rip Saw	Rip	*Aerodynamic +Mechanical	Blade Vibration Workpiece Vibration	G	H	I	K
Trim Saw	Crosscut	*Aerodynamic +Blade Resonance	Blade Vibration Workpiece Vibration	G	H	I	I,J
Gang Saw	Rip or Crosscut	*Aerodynamic +Blade Resonance	Workpiece Vibration Blade Vibration	G	H	C,E,F	K
Tenoner	Saw and Shape	*Aerodynamic +Blade Resonance	Workpiece Vibration Blade Vibration	G,L	B,H	I,E,F	J,D
Shaper	Shape	*Aerodynamic +Electric Motor	Workpiece Vibration Aerodynamic	G,L	B	E,F	D
Profile Shaper	Shape	*Aerodynamic +Electric Motor	Workpiece Vibration Aerodynamic	G,L	B	E,F	D
Router	Shape	*Base Vibration +Electric Motor	Base Vibration Workpiece Vibration	M	N	M	N

*Primary

Table 4.2 Key to Letter Code for Noise Control Guide

Letter Key	Technique or Procedure	Reference Section
A	Standard cutterheads - changes in RPM, clearance and lip geometry	III
B	Modified or redesigned cutterheads - spiral, segmented, or breakup geometry	III
C	Workpiece vibration reduction - stiffening, damping techniques	II
D	Cutterhead redesign - spiral or breakup geometry	II
E	Vibration suppression techniques - acoustical enclosures	II
F	Total acoustical enclosures	II
G	Treatment of guards or surfaces with absorption material	II
H	Saw plate/gullet redesign to reduce air flow noise	III
I	Blade vibration suppression systems - stiffening techniques	II
J	Blade damping techniques - blade coating, laminated plates, damping collars	II
K	Reduced saw kerf, raker teeth, strob saws, material spreaders	II
L	Standard cutterhead - changes in RPM, knife projection	III
M	Acoustical skin system	II
N	Isolation or redesign of router head assembly	II

V. CLOSURE

The information on which this final project report is based has been obtained through theoretical analysis, experimental investigations in the laboratory and in the field, and years of practical experience with woodworking mamachinery noise. In keeping with the basic objectives of the project, the report has concentrated on methods to minimize noise produced by vibrations and noise due to aerodynamic phenomena. An attempt has been made to make the report stand on its own without undue dependence on cited literature. In the opinion of the authors, it also represents the state-of-the-art in woodworking machinery noise control technology.

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