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MIL-HDBK-767(MI)
17 SEPTEMBER 1993

MILITARY HANDBOOK

DESIGN GUIDANCE FOR INTERIOR NOISE REDUCTION IN LIGHT-ARMORED TRACKED VEHICLES

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FOREWORD

1. This military handbook is approved for use by all Activities and Agencies of the Department of the Army and is available for use by all Departments and Agencies of the Department of Defense.

2. Beneficial comments (recommendations, additions. and deletions) and any pertinent data that may be of use in improving this document should be addressed to Commander. US Army Missile Command, ATTN: AMSMI-RD-SE-TD-ST, Redstone Arsenal, AL 35898-5270, by using [he self-addressed Standardization Document Improvement Proposal (DD Form 1426) appearing at the end of this document or by letter.

3. This handbook was developed under the auspices of the US Army Materiel Command's Engineering Design Handbook Program, which is under the direction of the US Army Industrial Engineering Activity.

4. Noise of tracklaying vehicles has historically been a problem that interferes with communication. produces hearing loss, and permits the vehicle [o be detected at great distances. Past noise reduction efforts in tracked vehicles were ineffective because tracked vehicle noise generation was not well-understood. The design guidelines presented in this handbook are the product of 10 years of research and development of the interior noise reduction of light-armored tracked vehicles. This research and development program was conducted by the FMC Corporation under Contract No. DAAK 11-81-C-0068. *Development of Advanced Technology for Quiet Vehicles*, under the joint sponsorship of the US Army Human Engineering Laboratory" and the Survivability! Technology Center of the US Army Tank-Automotive Command. The principal investigators at the FMC Corporation were Mr. Jerome Schmiedeberg and Mr. Karl Turner. The development of the handbook was guided by a technical working group composed of the following individuals from three US Army organizations: Chairman. Mr. Georges Garinther of the Human Research and Engineering Directorate of the US Army Research Laboratory, Dr. Edward Shalis of the Survivability Technology Center of the US Army Tank-Automotive Command. and Mr. Felix Sachs of the US Army Environmental Hygiene Agency. These three individuals from the Army activities devoted much time and energy to the preparation of this handbook,

*The name of this organization has been changed to the Human Research and Engineenng Directorate of the US Army Research Laboratory.

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LIST OF ABBREVIATIONS AND ACRONYMS

APU = auxiliary power unit
BBN = Bolt, Beranek, and Newman
CAC = chordal action control
FEA = finite element analysis
FFT = Fast Fourier Transform
MR = receiver mobility
MS = source mobility
PSD = power spectral density
RMS = root mean square
RTA = real-time analyzer
SEA = statistical energy analysis
SLM = sound-level meter
SPL = sound pressure level

CHAPTER 1
INTRODUCTION

This chapter presents the purpose, scope, and intended application of this design handbook. It introduces the reasons, purpose, and benefits of reducing tracked vechicle noise.

1-1 PURPOSE

This handbook gives proven guidelines for designing quiet tracked vehicles and reducing interior tracked vehicle noise by redesigning vehicle components. The guidelines primarily focus on track and suspension components; additional guidelines are provided for designing a quiet hull and engine enclosure.

1-2 SCOPE

Techniques available [o reduce interior noise in light-armored tracked vehicles are discussed. Emphasis is on those sources that have their major effect on interior noise. Exterior noise is addressed only as a result of having reduced interior noise.

This handbook provides a good overview of the design of quieter light-armored tracked vehicles. A more complete understanding of this subject may be obtained by consulting the references at the end of each chapter.

1-3 APPLICATION OF THE HANDBOOK

This handbook gives design guidance for interior noise reduction of light-armored tracked vehicles weighing less than 27 tonne (30 ton). These guidelines may be applicable for heavier vehicles but have been only validated for weights less than 27 tonne (30 ton). These guidelines are suitable for new vehicle designs as well as redesign of existing vehicles.

The intended audience includes

1. Designers of combat vehicles who are seeking guidance in designing inherent) quieter tracked vehicles
2. Vehicle project and product managers who are seeking an overview of the importance of interior noise reduction and how to achieve it
3. Military officers who are part of the procurement or development community and who are seeking tradeoff information on the difficulty, expense, impact, and advantages of designing quieter tracked vehicles.

1-4 PREVIOUS NOISE REDUCTION EFFORTS

Early attempts [o reduce tracked vehicle noise failed for the reasons that follow:

1. Multiple noise sources, typical for tracked vehicles. were not recognized.
2. Major noise sources were not identified.
3. Major noise sources were not ranked according to contribution and frequency.
4. Appropriate noise goals were not defined.
5. Noise treatments did not reduce the causes of noise generation; they tried only to muffle the noise being produced.

Other factors limiting the development of quieter tracked vehicles included the following:

1. Noise reduction techniques usually add weight, cost, and complexity to military vehicles.
2. Vehicle durability may be adversely affected.
3. Benefits of noise reduction were not clearly understood; thus the results of design tradeoff studies were biased.

A joint effort by the US Army Human Engineering Laboratory (HEL)* and the US Army Tank-Automotive Command (TACOM) investigated noise generation and ways to reduce internal noise in light-armored vehicles. This program used an M 113,41 as a test-bed for developing noise reduction techniques. Several techniques produced significant noise reduction. Others produced little noise reduction or were impractical and some others were too expensive. too heavy, or required too much space for proper component isolation. The program determined that [he greatest noise reduction would be achieved by modifying the suspension system. All techniques investigated, whether successful or not, are discussed.

The HEL and TACOM effort was conducted in several phases and was documented in five HEL technical memoranda (Refs. 1-5). The research work consisted of five major tasks:

1. Identifying and ranking the major noise sources by) contribution to total interior noise
2. Developing noise reduction concepts for major noise sources, i.e.. idler wheels, sprocket wheels, and road wheels
3. Verifying the interior noise reduction capability of each modified suspension component by testing full-scale, experimental suspension hardware
4. Verifying the producibility. durability, and practicality of each suspension modification by fabrication and testing of full-scale prototype hardware

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*The name of this laboratory has been changed to the Human Research and Engineering Directorate of the US Army Research Laboratory.

5. Verifying total vehicle interior noise reduction by installing and testing a full complement of modified suspension components. engine mounts, and engine compartment access panels in a demonstration vehicle.

The primary goal of these studies it as to reduce interior noise in a lightweight tracked vehicle to the limits specified in MIL-STD-1474. i.e.. Category B (100 dB(A)) at two-thirds of full vechicle speed (Ref. 6). The secondary goal was to reduce exterior noise by 6 dB. The demonstration quiet vehicle showed total vehicle noise could be significantly reduced by modifying the vibratory energy paths from the suspension components to the vehicle hull. These modifications reduced interior noise in the crew area from 114 to 105 dB(A) at 48.3 km/h (30 mph). This reduction in interior noise results in increasing the allowable exposure time by a factor of 6.7 for the crew members wearing the DH-132 Helmet. Exterior noise was reduced up to 5 dB at vehicle speeds between 8 and 48.3 km/h (5 and 30 mph): this reduction represents a maximum reduction in the area of detectability of approxlmately 68% (Ref. 5).

1-5 NEEDS FOR NOISE REDUCTION

1-5.1 COMBAT PERFORMANCE

Crew performance in tracked vehicles is seriously affected by high noise levels, which lead to hearing loss. degraded communication. misunderstandings. errors. accidents, or even failure to accomplish the mission. Although these adverse effects may be minimal in stationary or slowly moving vehicles, the armor tactics of today dictate rapid movement with constantly changing tactical situations. Thus the result could be disastrous if the commander or squad leader in a moving vehicle is unable to update his personnel on the changing tactical situation. In idling vehicles the inability of commanders to hear tactical sounds produced by enemy helicopters or armor could permit the enemy to engage first. Even after the vehicle has stopped and personnel have dismounted. hearing loss resulting from vehicle operation. even ternporary hearing loss, could make it difficult for a soldier to hear commands and detect combat sounds.

Studies conducted by the HEL using tank and armored personnel carrier simulators have quantified the effects of communication on performance. The specific measures used to evaluate performance as a function of various levels of communication fell into four categories:

1. *Mission time.* Time required to arrive at an objective; time required to identify the target; time required to complete a mission

2. *Mission completion.* Percent of time the crew correctly navigated to the objective: percent of targets correctly identified: percent of targets killed

3. *Mission errors.* Percent of communication errors: percent of times wrong target was hit; percent of time tank was killed

4. *Gunner accuracy.* Percent of times target was hit by the first round: aiming errors.

These studies showed that poor communication reduces crew performance and affects mission accomplishment. They also showed that these performance effects can be measured. For example, as communication was degraded. more navigational errors occurred, fewer enemy targets were killed, wrong targets were killed more often. and mission time increased (Ref. 7).

These reductions in performance can affect the outcome of a battle: therefore. it is evident that interior noise levels in tracked vehicles must be reduced.

1-5.2 COMMUNICATIONS AND SPEECH INTELLIGIBILITY

The most immediate effect of interior noise in tracked vehicles is its interference with communication. both person-to-person and electrically aided. MIL-STD-1472, *Human Engineering Design Criteria for Military Systems, Equipment, and Facilities.* provides speech intelligibility limits for establishing acceptable communication in various situations. Person-to-person speech intelligibility varies as a function of voice level, talker-to-listener distance. background noise. and hearing protection worn. Person- [o-person communication in tracked vehicles that requires shouting directly into a person's ear is very difficult at noise levels above 100 dB(A) (Ref. 6). Electrically aided communication varies as a function of the quality of the intercom system, the voice level at the ear, and background noise. When background noise levels are above 104 dB(A), speech intelligibility falls below an acceptable level during use of the intercom in the combat vehicle crewman’s helmet (Model DH-132) (Ref. 8).

Precise communication must be provided within moving tracked vehicles if the crew is to operate in a coordinated manner. In personnel carriers the squad leader must provide constantly changing tactical information to his squad prior to dismounting. In tanks and personnel carriers it is not desirable for commanders to stop the vehicle in order to communicate, Recent military experience has provided numerous examples of adaptations created to overcome communication difficulties: pulling a rope tied to the driver to tell him to stop. poking the gunners shoulder to indicate turret direction. and providing a light box to tell the driver in which direction to turn. These examples indicate that communication in current vehicles is extremely poor at all but the slowest speeds. Proper communication should not be left to the innovativeness of the crew; it should be designed into the vechicle.

Assuming the crewman’s helmet cannot be significantly improved. the most effective approach to improving communication is to reduce interior noise below l00 dB(A) (Ref. 6). This reduction will improve the speech-to-noise ratio, increase intelligibility, and decrease hearing loss.

1-5.3 ACOUSTIC SIGNATURE

Since tracked vehicle noise can be heard by the unaided ear at great distances, it is usually the first indication to the field soldier that armor is in the area. Tracked vehicle noise at long distances is most readily detected in the low to midrange frequencies. This fact is because noise attenuation by atmospheric absorption is negligible at low frequencies, and human hearing is most sensitive in the midrange,

Unfortunately, tracked vehicle acoustic signatures are highest in these low to midrange frequencies. The primary source of low-frequency noise in tracked vehicles is the interaction between track and suspension, whereas for midrange noise it is the power train, i.e., engine exhaust, cooling fan, and gear noise. As a general rule. a noise reduction of 6 dB will reduce detection distance 50% (based on spherical spreading propagation losses alone) and thereby will offer a great tactical battlefield advantage. For example. if a tracked vehicle were acoustically detectable at 4.0 km, a 6-dB reduction in the acoustic signature could allow it to be 2.0 km closer to the enemy observer before being acoustically detected. This difference greatly improves the ability of the vehicle to engage the enemy first and thus improves vehicle survivability.

1-5.4 HEARING HAZARD

Noise-induced hearing loss is one of the most prevalent occupational handicaps among soldiers. Over 50% of combat arms careerists develop significant hearing loss after 15 yr of service. Exposure to loud noise, even for a few minutes, can cause a temporary hearing loss. Initially, hearing is recovered a few hours after the noise stops. If exposure is repeated over time, however, recovery decreases and eventually the temporary degradations become permanent. Individual susceptibility to noise-induced hearing loss varies widely. The Department of Defense (DoD) allowable exposure of an 85-dB(A) time-weighted average over an 8-h period protects about 90% of all personnel. This criterion is factored into the design limits discussed in Chapter 3.

The ability of the crewman's helmet to provide protection against vehicle noise is limited by two factors. First, the high noise level in current tracked vehicles exceeds the rated protection ability of the helmet. As a result, noise levels at the ears of the crew exceed the allowable exposure limits, even for short daily periods. Second, the rated noise attenuation is attained only if the helmet is properly fitted, worn, and maintained—precautions often overlooked or neglected. Wearing earplugs with the helmet offers additional protection but at the expense of communication intelligibility. It can therefore be concluded that lowering the noise in tracked vehicles will reduce hearing hazards and will not reduce intelligibility.

1-5.5 VIBRATION

Vibrating surfaces are a fundamental source of noise. Thus designs that reduce vibration often reduce noise also. Depending on its frequency, vibration can harm the crew in various ways. For example. at extremely low frequencies (below 1 Hz) it can induce motion sickness. At 1 to 30 Hz it can cause general discomfort, fatigue, decreased proficiency, and, if of sufficient amplitude. injury. At higher frequencies it can cause local discomfort and in extreme cases can injure the contacting body parts.

Electronic and mechanical equipment is generally most sensitive to vibration-induced failures at 60 to 500 Hz. Equipment that benefits most from vibration reduction at these frequencies includes electronic parts, missile components, heaters, and electrical wiring. Noise control efforts that reduce vehicle vibration also improve equipment reliability and help to protect the occupants from harmful vibration effects.

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CHAPTER 2
ACOUSTIC AND VIBRATION FUNDAMENTALS

Special terminology and concepts used in acoustic and vibration work are defined and explained.

2-0 LIST OF SYMBOLS

A = maximum acceleration amplitude. m/s^2
C = actual system damping, N·s/m
C_c = critical damping for the system. N·s/m
D = maximum displacement amplitude, m
F = frequency of oscillation, Hz
F_f = forcing frequency. Hz
F_n = resonant or natural frequency, Hz
$F_x(f)$ = Fourier spectrum of the input time signal measured at Point x
$F_y(f)$ = Fourier spectrum of the output time signal measured at Point x
g = acceleration due to the gravitational field of the earth. dimensionless
$H_{xy}(f)$ = transfer function from Point x to Point y
i = each individual source
K = stiffness of' spring. N/m
m = mass of vibrating element, kg
n = number of noise sources to be summed, dimensionless
p = measured sound pressure. µPa
p_0 = reference sound pressure = 20 µPa
SPL = sound pressure level. dB
SPL_i = sound pressure level of the ith source, dB
SPL_s = sound pressure level of sum of sources, dB
V = maximum velocity amplitude. m/s
ζ = damping factor. dimensionless

2-1 1NTRODUCTION TO ACOUSTICS AND VIBRATION

The purpose of this chapter is to introduce the designer of tracked vehicles to some of the special terminology and concepts used in acoustic and vibration work.

2-2 PRINCIPLES OF ACOUSTICS AND VIBRATION

Acoustics (sound) and vibration are different interpretations of the same phenomenon: wave motions oscillating about a point of equilibrium. Acoustics usually deals with waves in a gas, such as air, whereas vibration deals with waves in solid materials. These waves are characterized by magnitude and rate of oscillation. Frequency is the number of oscillations Occurring in 1 s (cycles/s). Cycles per second are named hertz (Hz). The simplest wave motion (constant amplitude and single frequency) is a pure tone, which is the most fundamental form of periodic motion. More complex waves combine frequencies and amplitudes into patterns that may or may not be repetitive. Fig. 2-1 illustrates the time history of the extremes of wave motion from a simple pure tone (one frequency) to complex random (all frequencies). Examples of sounds that are produced by these waves are a musical note produced by a tuning fork (pure tone) and static from a radio or television that is not tuned to a station

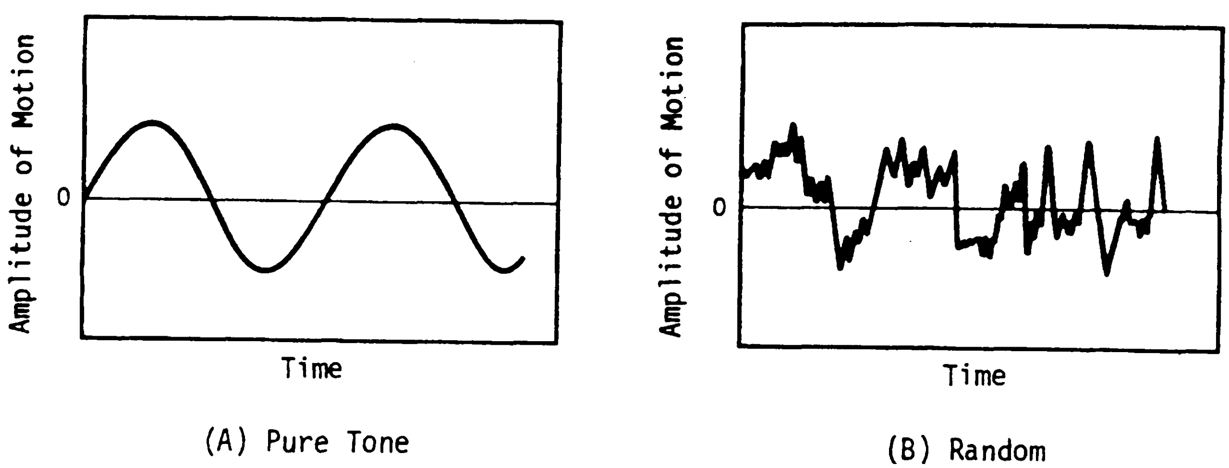

Figure 2-1. Time History of Two Extremes of Acoustic Wave Motion

Transducers measure sound or vibration by transforming the wave motion into a signal [hat can be recorded and analyzed. The ease with which electrical signals can be manipulated makes it a simple task to convert mechanical oscillations into an analogous electrical signal, which can then be displayed on a monitor screen or other signal analyzers.

The frequency content of sound or vibration, in addition to amplitude, is required to understand which designs will work best to reduce noise or vibration. The frequency information can be extracted from a time history using a method known as a Fourier. or spectral, analysis. This method decomposes a time history signal into its frequency components. Spectral analysis is used in virtually every technical field It is a very powerful too! for identifying and under-Standing the behavior and makeup of materials. components. and phenomena that comprise our physical environment. using spectral analysis in noise and vibration testing. for example. assists in identifying and localizing the sources of noise and vibration and devising methods for reducing their high levels.

Fourier. a French mathematician, showed that repetitive cyclic curves can be exactly reproduced by superimposing a sufficient number of simple harmonic tunes. as shown in Fig. 2-2 (Ref. 1). The complex waveform 1 shown on the time domain axis is the sum of the simple harmonic waveforms 2 shown parallel to the time domain axis. The height of the discrete bumps 3 on the frequency axis shows the amplitude of the individual simple harmonic waveforms; the positions of the bumps along the frequency axis show their frequency.

A simple mechanical analogy for spectral analysis is the process of screening a sand and gravel mixture into various piles of material ranging in size from the finest sand particles to the largest rocks by using a variety of screen sizes. The electronic equivalent uses a series of band-pass filters to decompose selectively a time-varying voltage signal into a number of signals varying in frequency content by the bandwidths of the filters. If adjacent filters are selected to cover an increasing section of the frequency range of interest. a plot of the output amplitude of each filter against its center frequency creates a spectral graph. (Ref. 2).

Modem spectral analysis uses a digital computer algorithm to transform a time history signal into its frequency components. Such analyzers are known as real-time analyzers (RTA) because they essential} acquire and analyze data at the same time.

2-2.1 ACOUSTICS

Pressure waves are created when the equilibrium of an elastic medium. such as air. is disturbed. In the audible range these waves are experienced as sound. The variation of this pressure. which is extremely small compared to the average atmospheric pressure at sea level above and below atmospheric pressure is called sound pressure. At a distance of 1 m from the talker, the average pressure difference for normal speech above and below atmospheric pressure is

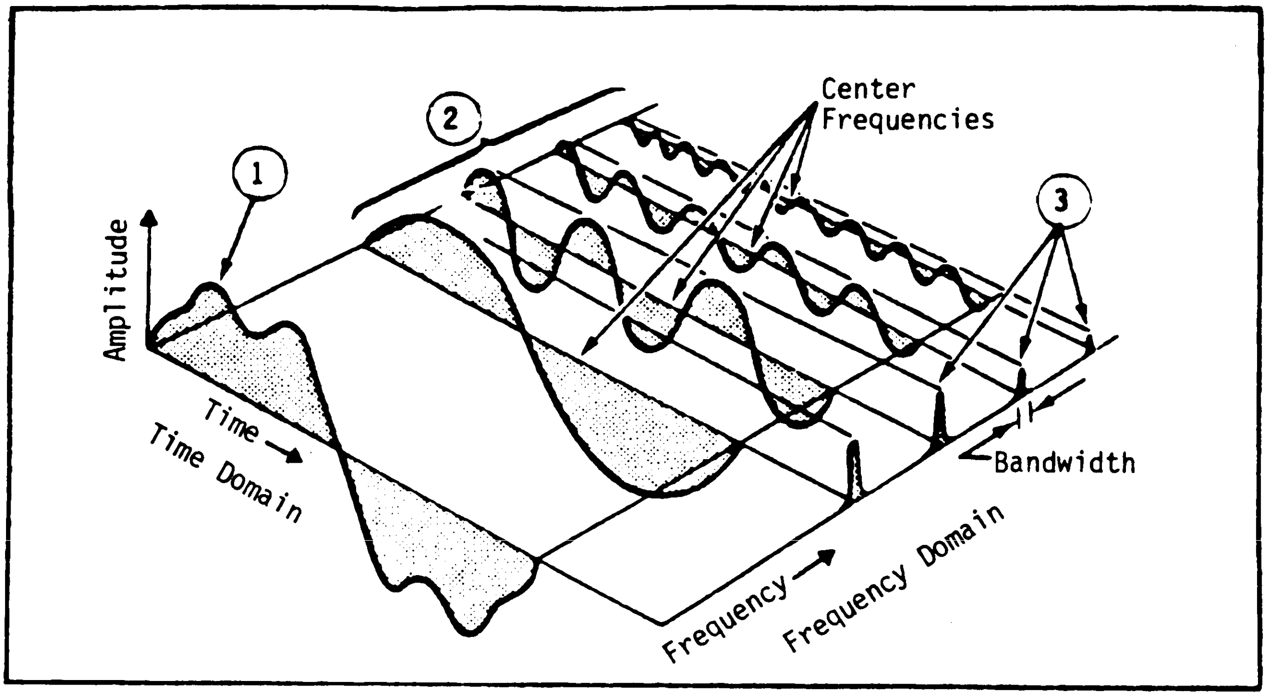

Figure 2-2. Illustration of the Time Domain and Frequency Domain Relationship (Ref. 1)

about 0.03 Pa compared to atmospheric pressure of 1.01 x 10^5 Pa (Ref. 3).

The human ear can perceive sound pressure fluctuations as small as 20 µPa to greater than 100.000 Pa—a dynamic range of about 10^{10}. Thus applying linear scales to sound pressure measurements would lead to large. unwieldy numbers. However. since the ear responds logarithmical]} to stimuli. not linearly. sound level is expressed as the logarithm of a ratio of measured sound pressure to a reference value. The bel, named after Alexander Graham Bell, is the unit used to measure sound level. although the preferred unit is the decibel (dB). which is one-tenth of a bel (Ref. 4). Sound pressure level *SPL* is calculated by

$$SPL = 10\log\left(\frac{p^2}{p_0^2}\right) = 20\log\left(\frac{p}{p_0}\right), \text{dB} \quad (2\text{-}1)$$

where

p = measured sound pressure. µPa

p_0 = reference sound pressure = 20 µPu.

The reference value p_0, is the faintest sound pressure at 1000 Hz that it can be perceived by a person with normal hearing. A 6-dB change in *SPL*— doubling or halving of sound pressure—is easily detectable by the human ear and is considered a significant change. Sound pressure levels are measured with a sound-level meter (SLM). which basically consists of a microphone. amplifier. and voltmeter. Table 2-1 shows familiar noises and their associated sound pressure levels.

Because the human ear is less sensitive to low-frequency noise, a filter is often used to "reduce” the low frequencies when measuring sound pressure levels. These “reduced” sound levels. called A-weighted levels With units dB(A). correlate well with the loudness of noise as perceived by human observers. These A-weighted levels also correlate with the susceptibility of human hearing to noise damage. The frequency response characteristics of the A-weighting network are shown In Fig. 2-3. Table 2-2 shows an octave band spectrum and its corresponding overall and A-weighted level.

Frequency, or spectral. analysis of acoustic signals is a technique frequently used to quantify sounds for noise reduction. Two of the most popular frequently-analysis techniques for acoustics use octave band and one-third octave band filters. Each octave covers a 2:1 frequency range. the width of each octave band filter is 71% of its center frequency.

Dividing each octave band into three parts gives a more detailed analysis of sound energy as a function of frequency. This one-third octave band analysis uses a filter bandwidth 23% of center frequency (Ref. 3). Fig. 2-4 shows both an octave band and a one-third octave band analysis of typical interior noise in a military tracked vehicle.

2-2.2 VIBRATION

vibration in military tracked vehicles is generated primarily by the suspension components and secondarily by the engine and power train. Vibration amplitude can be expressed in terms of displacement. velocity. or acceleration. The magnitude of each as a function of frequency is related as follows:

$$V = 2\pi FD, \text{ m/s} \quad (2\text{-}2)$$

$$A = 4\pi^2 F^2 D, \text{ m/s}^2 \quad (2\text{-}3)$$

where

A = maximum acceleration amplitude, m/s²

D = maximum displacement amplitude. m

F = frequency of oscillation, Hz

V = maximum velocity amplitude. m/s.

Transducers are available that measure vibration in any of the three amplitude units. but the unit most commonly used in tracked vehicles is accelertition. Although acceleration is often expressed in meters per second squared (m/s²). it is more conveniently expressed in the gravtational unit g. One g equals the acceleration due to the gravitutiontal field of the earth, which is 9.8 m/s² (32.2 ft./s²).

Spectral analysis is a powerful tool used to understand the behavior of materials and structures better and to identify more precisely the causes of unwanted vibration. In vibration studies narrow band or constant bandwidth frequency analysis is more commonly used than the proportional bandwidth filters used in acoustics.

A common vibration analysis technique normalizes the spectral amplitude relative to a 1-Hz bandwidth. This method of characterizing random vibration signals, known as power spectral density (PSD) analysis. helps ensure that similar results are obtained regardless of the particular analyzer used to conduct the analysis. Power spectral density is expressed in units of g^2/Hz. Fig. 2-5 shows typical acceleration power spectral density vibrations measured for a tracked vehicle operating at high speed on a paved road. The large amplitude peak at approximately 90 Hz is a typical component of tracked vehicle vibration caused by the periodic chordal action of the track. (Refer to Chapter 4 for a detailed description of tracked vehicle vibration generation.)

2-3 BASIC CONCEPTS

2-3.1 SUMATION OF THE NOISE FROM MULTIPLE SOURCES

A thorough understanding of the summation of sound levels from multiple noise sources is crucial to designing a quiet tracked vehicle. Most noise control problems involve several independent. uncorrelated noise sources. This is especially true for tracked vehicles. However, if the sound

TABLE 2-1. SOUND LEVELS OF COMMON SOUNDS

AT GIVEN DISTANCE FROM NOISE SOURCE	SPL. dB(A)	ENVIRONMENTAL
	140	
37 kW (50 hp) Siren at 30.5 m (100 ft)		
	130	
Jet takeoff at 61 m (200 ft)		
	120	
	116	Inside lightweight tracked vehicle at 48.3 km/h (30 mph)
Riveting machine (at operator's ear)	110	
Pneumatic peen hammer (at operator's ear)	100	Electric furnace area
Subway train at 6.1 m (20 ft)	90	Boiler room Printing press plant
Pneumatic drill at 15.2 m (50 ft)		
	80	Inside sports car at 80.5 km/h (50 mph)
Freight train at 30.5 m (100 ft) Vacuum cleaner at 3.1 m (10 ft) Speech at 0.3 m (1 ft)	70	Near freeway (automobile traffic)
	60	Large store Accounting office Private business office
Large transformer at 61 m (200 ft)	50	Light traffic at 30.5 m (100 ft) Average residence
	40	Minimum levels, residential areas in large city at night
Soft whisper at 1.5 m (5 ft)		
	30	Studio (speech)
	20	Studio (motion pictures)
	10	
Threshold of hearing for youths, 1000-4000 Hz	0	

level for each source is known. the overall noise level can be predicted by summing the individual sources. This summation can be done in each octave or one-third octave band to delelop an overall spectrum, or each source level can be summed to obtain an overall vehicle level.

When the sound levels from multiple noise sources are summed. the decibel levels of the individual sources are converted to relative power ratios. The power ratios are then added and the sum is converted to a corresponding decibel level:

$$SPL_s = 10\log \sum_{i=1}^{n} 10^{\left(\frac{SPL_i}{10}\right)}, \text{dB}$$

where

SPL_s = sound pressure level of sum of sources. dB
SPL_i = sound pressure level of the ith source, dB.

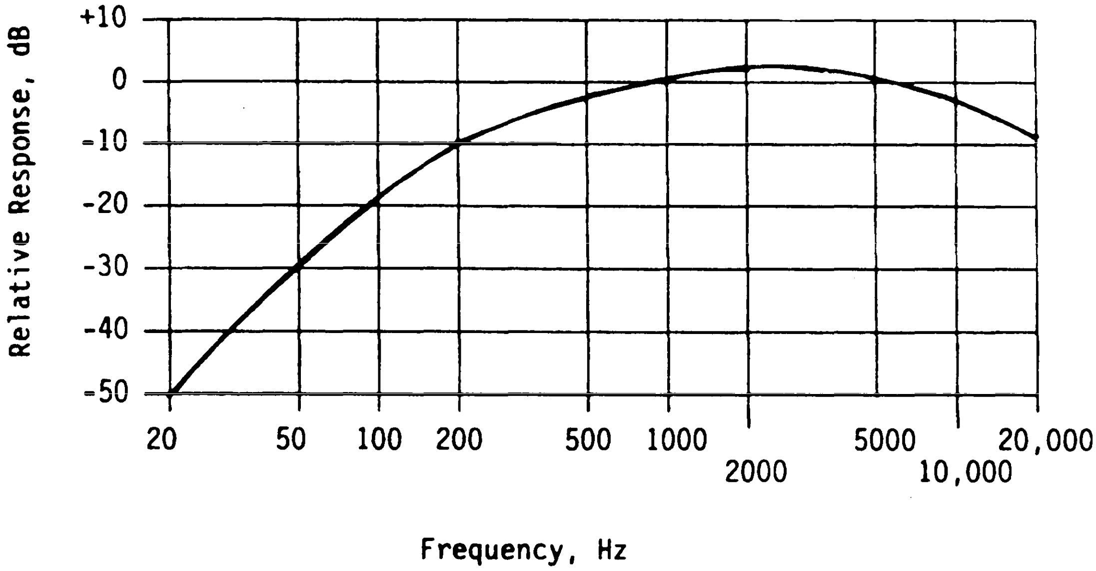

Figure 2-3. Frequency Response Characteristics of the A-Weighting Network

TABLE 2-2. EXAMPLE A-WEIGHTED, SOUND LEVEL COMPUTED FROM OCTAVE BAND LEVELS

OCTAVE BAND CENTER FREQUENCY, Hz	NOISE LEVEL, dB	LEVEL CHANGE FOR A-WEIGHTING	A-WEIGHTED NOISE LEVEL, dB (A)
63	92	−26.2	66
125	94	−16.1	78
250	97	−8.6	88
500	90	−3.2	87
1000	85	0	85
2000	82	+1.2	83
4000	77	+1.0	78
8000	71	−1.1	70
Overall Level	100 dB		93 dB (A)

Summing two noise sources. each with a noise level of 100 dB. yields a noise level of 103 dB, not 200 dB. Likewise. adding five noise sources at 80 dB each yields an overall noise level of 87 dB. Hovever, if one of the five sources is eliminated. the overall level is reduced only 1 dB. Further. when two noise sources are summed and one source is 10 dB louder than the other. the ovrerall noise level is only 0.4 dB louder than the loudest source. Table 2-3 shows additional combinations of multiple sources with the same or different noise levels. Fig. 2-6 shows an example of the summation of four noise sources to forms an overall spectrum.

The following key points about noise summation must reconsidered when reducing the noise of tracked vehicles:

1. All major noise sources must be reduced before noise levels can be significantly reduced.

2. Significantly reducing one noise source more than others will not greatly reduce overall noise.

3. Optimum overall noise reduction is obtained When the noisiest sources contribute equally to the overall noise level.

2-3.2 VIBRATION ISOLATION AND DAMPING

Free mechanical vibration occurs when a body with mass, which is connected to a spring element. is disturbed from equilibrium and released. The spring attempts to restore the body to equilibrium by releasing its stored energy: however. the inertia of the body carries it past the equilibrium position and stores energy in the spring again until the. body comes to rest. This oscillation process. once started. repeats until the disturbing energy has been dissipated.

When a body with mass is connected to a spring element, it is known as a vibrational system. Such systems can be very simple, e.g., a block suspended from a coil spring, or very complex, e.g., a structural element of a machine or

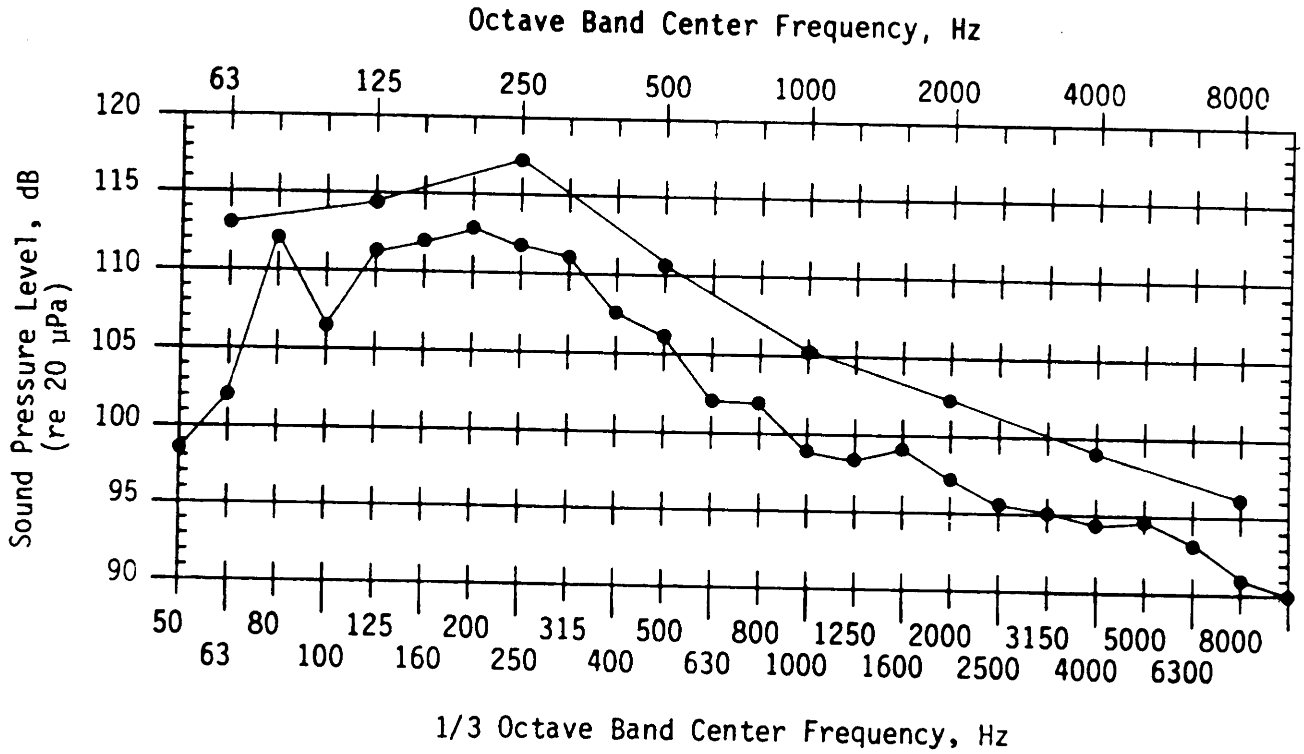

Figure 2-4. Example Octave Band and One-Third Octave Band Analysis Plots

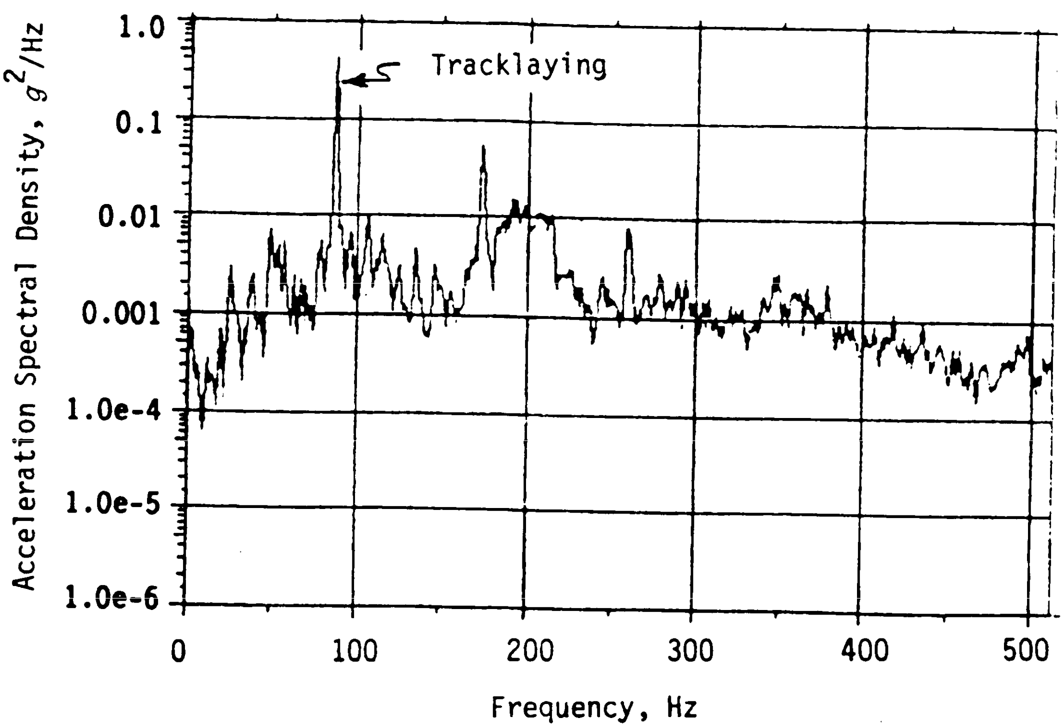

Figure 2-5. Typical Constant Bandwidth Tracked Vehicle Vibration Spectrum

TABLE 2-3. EXAMPLES OF ADDITION OF MULTIPLE NOISE SOURCES OF THE SAME AND DIFFERENT LEVELS

		Number of Sources at 80 dB Each					
		0	**1**	**2**	**3**	**4**	**5**
	0	–	80	83	84.8	86	87
	1	90	90.4	90.8	91.1	91.5	91.8
Number of Sources at 90 dB Each	**2**	93	93.2	93.4	93.6	93.8	94
	3	94.8	94.9	95.1	95.2	95.3	95.4
	4	96	96.1	96.2	96.3	96.4	96.5
	5	97	97.1	97.2	97.2	97.3	97.4

Values are sound pressure levels, dB.

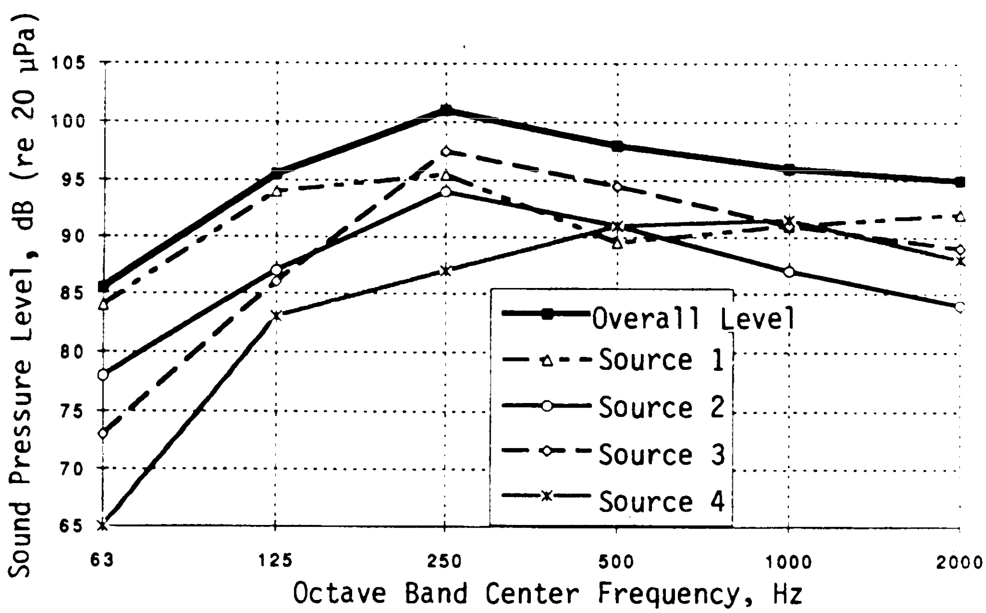

Figure 2-6. Example of Spectral Summation of Acoustic Sources

building. In structural beams or plates the mass and spring are distributed throughout the structure rather than being concentrated in individual components. Fig. 2-7 shows a simple vibrational system with one degree of freedom.

The mechanism used to dissipate disturbing energy from a vibrational system is called damping (not 'dampening'.. which deals with the application of water). Damping results from one of three frictional forces in a vibration system: dry or coulomb friction (between rigid bodies). viscous friction (between a body and the fluid through which it moves), or internal friction (between the molecules of an elastic material),

The rate at which a vibrational system freely oscillates is its natural or resonant frequency F_n. This frequency is determined by the mass of the vibrating element and the stiffness or spring rate of the spring as

(2-5)

where

F_n = resonant or natural frequent). Hz
K = stiffness of spring. N/m
m = mass of vibrating element, kg.

The damping factor of a vibration system is measured in relation to the critical damping of the system, which is defined as the damping required to allow a displaced system to return to its initial position without oscillation. If the system is underdamped, i.e.. less than critical damping, it oscillates with decreasing amplitude during each cycle until it reaches equilibrium. Fig. Q-8 illustrates the effects of various damping factors on a vibration system with one degree of freedom.

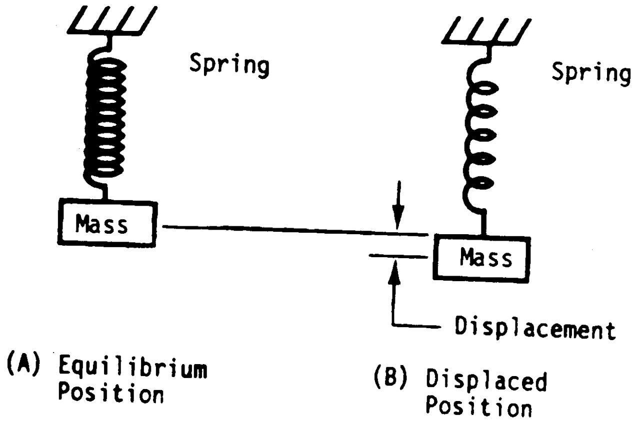

Figure 2-7. Simplified Single-Degree-of-Freedom Vibrational System

Single-degree-of-freedom systems can provide vibration isolation or reduction in vibration amplitude when their natural frequency is lower than the frequencies to be reduced. In contrast to systems that vibrate only at their natural frequency. it is possible to force a mass to vibrate at various frequencies other than its natural frequency. If the vibration system of Fig. 2-7 is supported on a driving mechanism consisting of a variable speed motor and crank that will force the spring-mass system to oscillate in the vertical direction only. as shown in Fig. 2-9. the block can be forced to vibrate at various frequencies other than its resonant frequency. This is known as forced vibration.

If the motor speed is slowly increased from zero. the displacement of the block increases until it reaches a maximum value at the resonant frequency of the system. When the forcing frequency exceeds the resonant frequency. the amplitude of the block displacement decreases. When the

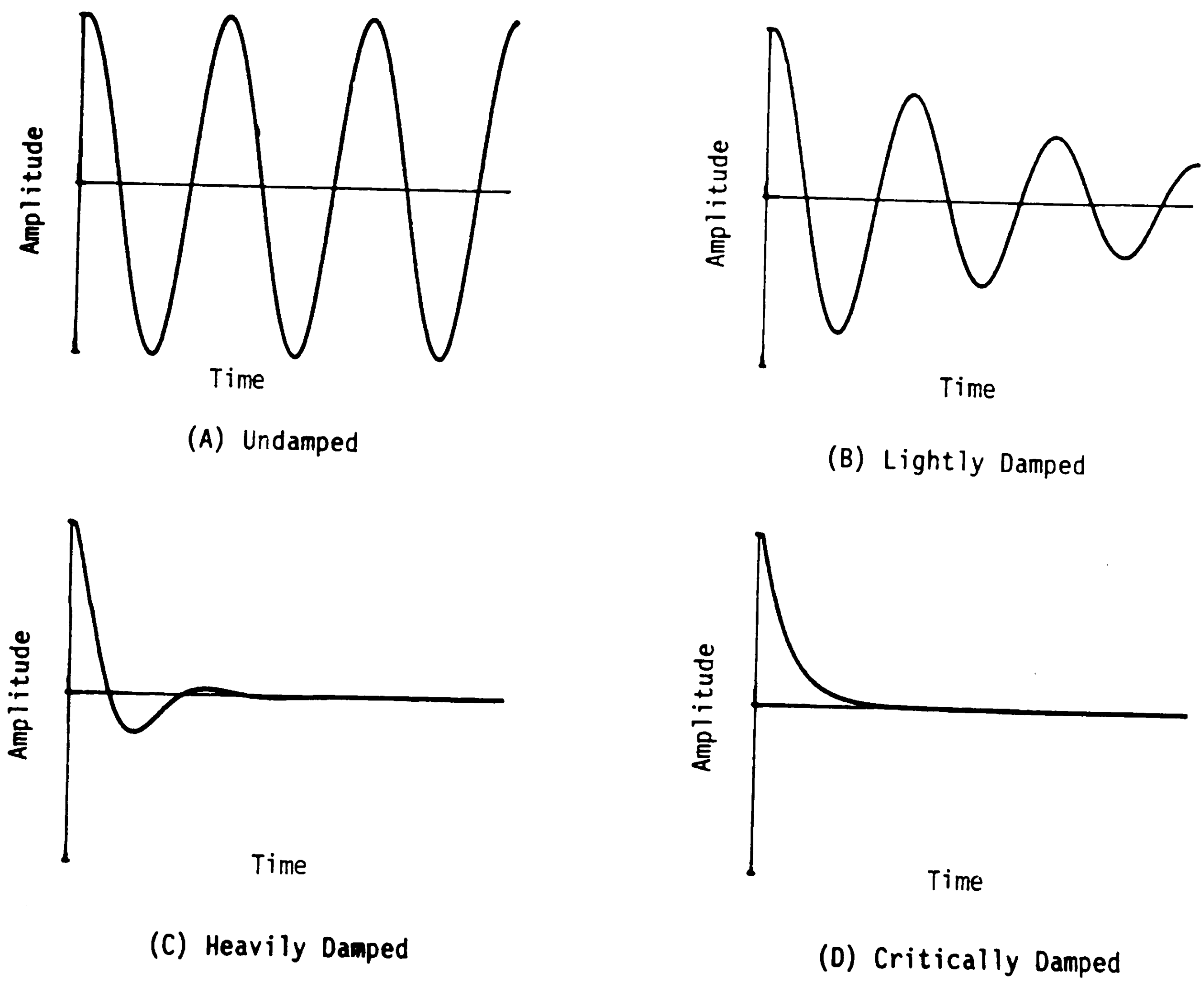

Figure 2-8. Effects of Damping on the Step Response of a Single-Degree-of-Freedom Vibrational System

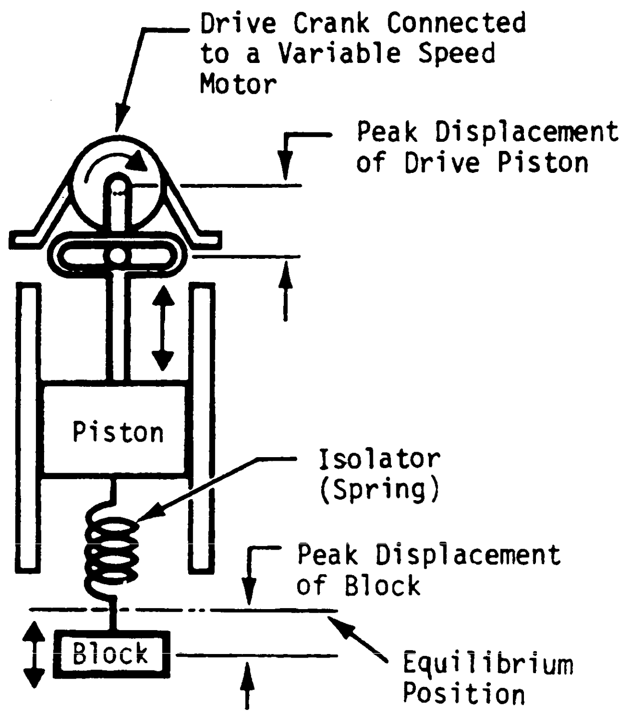

Figure 2-9. Forced Vibration in a Single-Degree-of-Freedom System

frequency is increased more than 1.4 times the natural frequency. the block displacement will be smaller than the piston displacement. This resulting vibration isolation or attenuation is the principle behind vibration isolation mounts. The ratio of block displacement to driver displacement is the transmissibility (output/input) of the vibrational system. For frequencies near or above the resonant frequency of the system, the transmissibility is dependent on the amount of damping present. The damping factor ζ is defined as

$$\zeta = \frac{C}{C_c}, \text{dimensionless} \tag{2-6}$$

where

ζ = damping factor, dimensionless
C = actual system damping. N•s/m
C_c = critical damping for the system, N•s/m.

Fig. 2-10 shows the transmissibility of a single-degree-of-freedom resonant system with various damping factors. Vibratory systems with inadequate damping can produce severe vibrations due to resonant vibration amplification.

Vibration isolation mounts are placed in the direct structural path between the source of vibration and the location of [he vibration reduction. A good example of the use of vibration isolators is the mounting of an automobile engine. The engine is not bolted directly to the frame but is attached [o compliant mounts that are bolted to the frame. Vibration

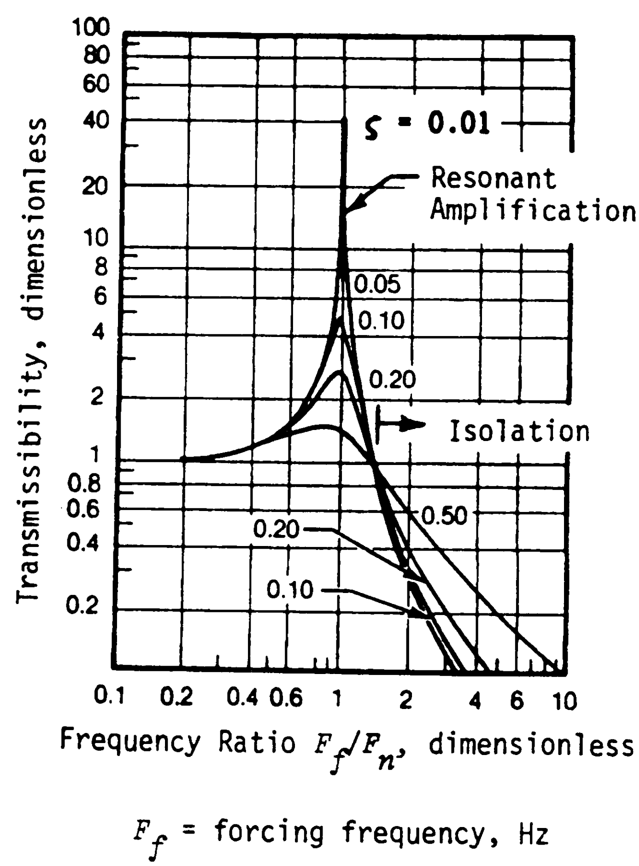

Figure 2-10. Transmissibility of a Single-Degree-of-Freedom System With Various Amounts of Damping

isolators are made of many different materials and come in various configurations. Rubber is commonly used because it is easily fabricated into mounts of various shapes that provide good isolation in a small volume. Also the composition of the rubber compound can be varied to provide the best spring rate and damping factor. Damping reduces resonant amplification in isolation mounts. but it also decreases isolation effectiveness. In addition, highly damped elastomeric materials may be permanently damaged when the heat from damping is generated faster than it can be dissipated. Thus the damping factor for a vibration isolation mount must be selected with care to obtain the best vibration isolation system. A good vibration isolator material for tracked vehicles is lightly damped natural rubber mounted in a rubber-in-shear configuration. Vibration isolators constructed of rubber and mounted to stress the rubber in shear have the advantage of low spring rates in a small volume. To achieve the same low spring rate using rubber in compression would require a much greater volume. Rubber in compression. however. is veil-suited to handle large momentary overloads. which could destroy a rubber-in-shear mount. Often commercial vibration isolators are constructed using rubber in both compression and shear to make a soft mount that can also handle overloads

2-3.3 ACOUSTIC COCPLING

If a vibrating structure comes in contact with air. it transfers vibrational energy to the air in the form of sound waves. The amount transmitted is determined by the acoustic coupling of the structure, the area. and [he vibration amplitude. Thus to reduce noise. manufacturers seek to lower the acoustic coupling of a vibrating structure by using the proper materials and fabrication techniques.

When sound radiates from a structure. its noise source is either structure-borne or airborne. An example of structure-borne noise is the noise generated by the suspension of a tracked vehicle: the suspension vibrates the hull, which in turn radiates noise. An example of airborne noise is sound waves impinging on a structural surface and causing that structure to vibrate and radiate noise. Both acoustic coupling mechanisms exist in engine noise propagation into the crew area where part of the noise comes from engine vibration that excites the hull and part from engine compartment panels that are excited by engine noise.

Radiation efficiency is a measure of how readily a structure converts vibration to noise. It is highly dependent on frequency and is controlled by structural stiffness. density. and internal damping. (See Ref. 5 for an in-depth study of radiation efficiency.)

2-3.4 TRANSFER FUNCTION

Transfer function (also called frequency response) is defined as the complex ratio of the output of a system to its input, measured as a function of frequency (Ref. 5). This function is useful for quantifying the response characteristics of a structure due to a time-varying force input.

The input [o the structure is a dynamic force created by some device. such as a vibration shaker or impact hammer. The output (response) is its amplitude of motion measured with an accelerometer, velocity pickup. displacement transducer. or the sound pressure measured with a microphone. Transfer function is defined as

$$H_{xy}(f) = \frac{F_y(f)}{F_x(f)} \qquad (2\text{-}7)$$

where

$F_x(f)$= Fourier spectrum of the input time signal measured at Point x

$F_y(f)$= Fourier spectrum of the output time signal measured at Point y

$H_{xy}(f)$= transfer function from Point x to Pointy.

For a measured transfer function to be valid. the input and output must be correlated, i.e.. the output must (for the most part) be caused by the input. Here, the transfer function can quantify how well a structure (or location on a structure) accepts vibration energy and how well it radiates noise (acoustic coupling). Some common transfer functions are defined as follows:

1. Mechanical impedance = $\frac{\text{force}}{\text{velocity}}$
2. Inertance = $\frac{\text{acceleration}}{\text{force}}$
3. Mobility = $\frac{\text{velocity}}{\text{force}}$
4. Noise-to-force $\frac{\text{acoustic noise}}{\text{force}}$
5. Transmissibility = $\frac{\text{Response* at Location. } x}{\text{Response at Location } y}$

Mobility and mechanical impedance can help determine the best location for mounting a vibration source, i.e.. where it will produce the least noise. (Refer to par. 7-2.3.) Noise-to-force transfer functions combine frequency response and radiation efficiency. In a tracked vehicle these functions are difficult to measure because the noise radiated from the hull is highly uncorrelated to hull vibration. However, the ratio of the noise and force spectra is useful to quantify the acoustic coupling of the hull structure. (Refer to par. 8-3.3.) Inertance measurements are used to determine the characteristic modes of vibration of a structure in a measurement technique known as modal analysis. This technique can be

*Note: Response can be acceleration. velocity, displacement, or force.

used to predict changes in sound pressure levels that result when a tracked vehicle hull structure is modified in a particular way. (Refer to par. 7-1.)

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CHAPTER 3
NOISE LIMITS

The noise limits imposed by MIL-STD-1474 on tracked vehicles and the methodology for transforming these limits into noise reduction goals for components and systems are presented in this chupter

3-1 INTRODUCTION

As discussed in par. 1-5. interior noise levels in all high-speed. US military tracked vehicles are loud enough to interfere with communication. induce hearing losses, and reduce combat performance (Refs. 1 and 2). MIL-STD-1474 (Ref. 3) specifies the maximum noise limits for occupied areas of operating military equipment and indicates that most tracked vehicles exceed these limits at speeds above approximately 15 km/h (9 rni/h). In the past. these limits have not been met for tracked vehicles because the technology to make them quiet was not available. Recent developments in noise reduction now make it possible to lower tracked vehicle noise to the acceptable levels stipulated in MIL-STD-1474. This technology is presented in this design handbook.

3-2 BACKGROUND

MIL-STD-1474 was drafted to provide noise limits based on preventing hearing loss and on communication requirements for materiel used by military personnel. The basic steady state noise criterion for conservation of hearing is that unprotected personnel not be exposed to noise levels of 85 dB(A) or greater (Ref. 4) This level is defined as Category D in MIL-STD-1474. and exposures above this level ‘ require the use of hearing protection, such as earplugs. earmuffs, attenuating helmets or headsets. Direct person-to-person voice communication is difficult when noise levels exceed Category D. although occasional shouted communication may be possible at a distance of 0.6 m (2 ft). When hearing protection is worn. the permissible noise level is the sum of the attenuation provided by the hearing protectors and 85 dB(A). The two categories of MIL-STD-1474 applying to tracked vehicles are Categories A and B. Definitions of these categories follow.

3-2.1 CATEGORY B NOISE

Category B noise limit, 100 dB(.A). is the design limit for systems that require electrically aided communication with attenuating helmets or headsets This category was computed by adding 85 dB(A) to the 15-dB attenuation provided by the DH -132 combat vehicle crewman’s helmet. However. this level was modified by limiting the octave band equivalent of the frequencies from 1000 Hz and above to 100 dB. This moditictition was imposed because higher noise levels reduced speech intelligbility over electrically aided communication systems to unacceptable levels (Refs. 5 and 6). This category is appropriate for the crew area of a tracked vehicle. When wearing the DH-132 helmet in a vehicle meeting the limits of Category B. the crew is provided with 75% speech intelligibility vhen tested according to the Monosyllabic Word Intelligibility} Test. ANSI S3.2 (Ref. 7).

3-2.2 CATEGORY A NOISE

Category A noise limit. 108 dB(A). is the maximum design limit for Army materiel. Personnel occupying an area that has noise levels between 85 and 108 dB(A) must be provided hearing protection and will not be able to conduct direct person-to-person voice communication effectively. DA PAM 40-501 (Ref. 4) requires double hearing protection. i.e.. earmuffs or attenuating headsets or helmets in addition to earplugs. when noise levels exceed 108 dB(A). Category A was computed by adding 85 dB(A) to the 23-dB attenuation of the V-51R earplug, the predominant hearing protector in use when MIL-STD- 1474 was as originally written. This category is appropriate for the passenger area in tracked vehicles.

3-3 DESIGN LIMITS

Table 3-1 shows the six steady state noise categories and limits defined in MIL-STD- 1474. Of the six categories the two that apply to tracked vehicles are Categories A and B. Category A allows a noise limit of 108 dB(A) for vehicle passengers wearing hearing protectors and not requiring person-to-person communication. Category B allows 100 dB(A) for the crew wearing noise-attenuating devices and using electrically aided communication. However. in most tracked combat vehicles it is impractical to differentiate between Category A and B areas within the same vehicle, mainly because areas requiring communication and areas requiring none are so close to each other. Thus Category B limits should apply to all occupied areas in tracked vehicles.

To verify compliance with the noise limits of MIL-STD-1474, noise is measured at each operator or crew position while the vehicle travels at a constant two-thirds of rated vehicle speed on a level. paved road. In addition to the two-thirds of rated vehicle speed measurement. which is made for compliance with MIL-STD-1474. measurements shall also be made at 8- or 16-km/h (5- or 10-mi/h) increments for information. In general. all windows and access open-

TABLE 3-1. STEADY STATE NOISE CATEGORIES AND LIMITS FROM MIL-STD-1474 (Ref. 3)

CATEGORY	dB (A) LIMIT	SYSTEM REQUIREMENTS
A*	108	No direct person-to-person voice communication required; maximum design limit; hearing protection required
B*	100	Electrically aided communication via attenuating helmet or headset required; noise levels are hazardous to unprotected ears.
C	90	No frequent. direct person-to-person voice communication required; occasional shouted communication may be possible at a distance of 0.30 m (1 ft); hearing protection required.
D	85	No frequent person-to-person voice communication required; occasional shouted communication may be possible at a distance of 0.60 m (2 ft); levels in excess of Category D require hearing protection.
E	75	Occasional telephone or radio use or occasional communication at distances up to 1.5 m (5 ft) required
F	65	Frequent telephone or radio use or frequent communication at distances up to 1.5 m

*These noise categories are applicable to tracked vehicles.

ings shall be in the normal operating position, auxiliary equipment normally in use shall be operated. and vehicles shall be operated at two-thirds of the normal load. In addition to the A-weighted sound measurement required for compliance, C-weighted and octave-band measurements shall be collected.

MIL-STD-1474A and MIL-STD-1474B contained octave band noise limits in addition to the A-weighted noise limits. Although not in MIL-STD-1474C, these band limits are presented here as a guide to developing octave band noise goals that would permit a vehicle to meet the 100-dB(A) Category B limit. Fig. 3-1 shows the octave band limits for tracked vehicles. 1(should be noted that the overall A-weighted level of this limit spectrum is 108 dB(A). That is, if the noise in a vehicle had precisely this spectrum, the overall level would be 108 dB(A) and the vehicle would not meet Category B. However. the actual spectrum usually touches the limit at only one or two frequencies; thus the overall level is 100 dB(A).

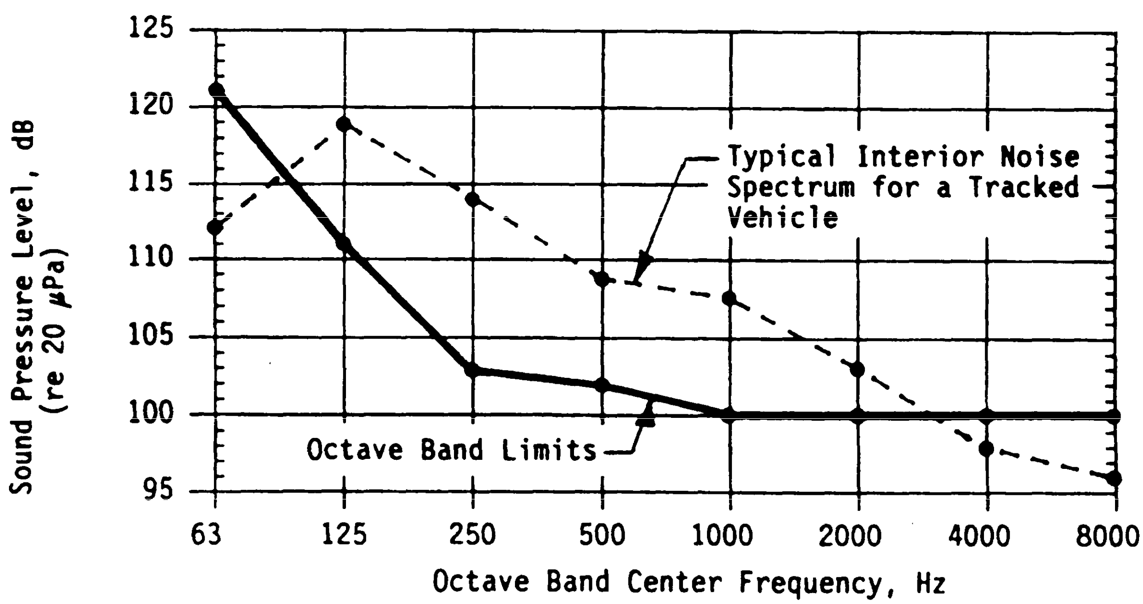

Figure 3-1. Octave Band Noise Limits for Tracked Vehicles to Comply With 100-dB(A) Noise Limit of Category B, MIL-STD-1474

3-4 EXISTING VEHICLE NOISE LEVELS

Interior noise levels of tracked vehicles vary in amplitude from vehicle to vehicle and within the same vehicle, depending on factors such as vehicle speed. atmospheric temperature and humidity, terrain, mechanical condition of the vehicle, strand condition and tension of track, load. and hatch position (open or closed). However. the shape of the interior noise spectrum of a tracked vehicle is somewhat consistent. In general, tracked vehicle noise spectra are highest in the 125- or 250-Hz octave band and decrease at a rate of 3 to 4 dB per octave above 250 Hz as shown in Fig. 3-1. The interior noise in the tracked vehicles listed in Fig. 3-2 ranges from 105 to 117 dB(A). which significantly exceeds the 100 dB(A) limit. For most tracked vehicles the 250-Hz band requires the greatest noise reduction in order to meet the MIL-STD-1474 limit shown in Fig. 3-3.

3-5 LIMITS FOR VEHICLE NOISE SOURCES

Reduced noise levels in tracked vehicles would be difficult to obtain if the system limit of 100 dB(A) were the only criterion available to guide the design effort. In the same

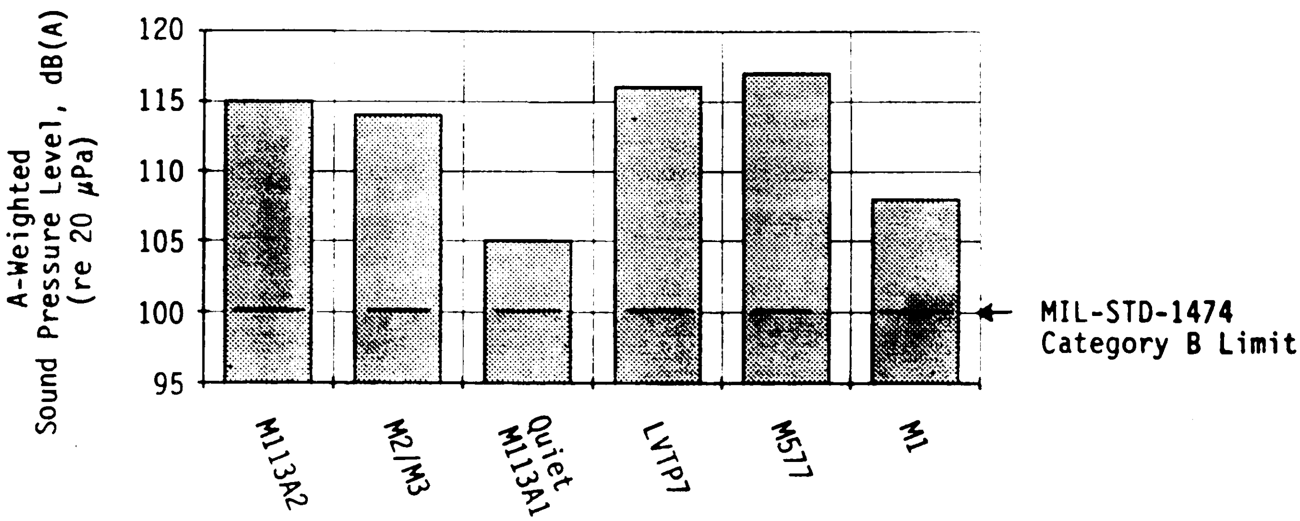

Figure 3-2. Comparison of A-Weighted Interior Noise for Various Tracked Vehicles Measured in the Crew Compartment With the Vehicle Moving at 48.3 km/h (30 mi/h)

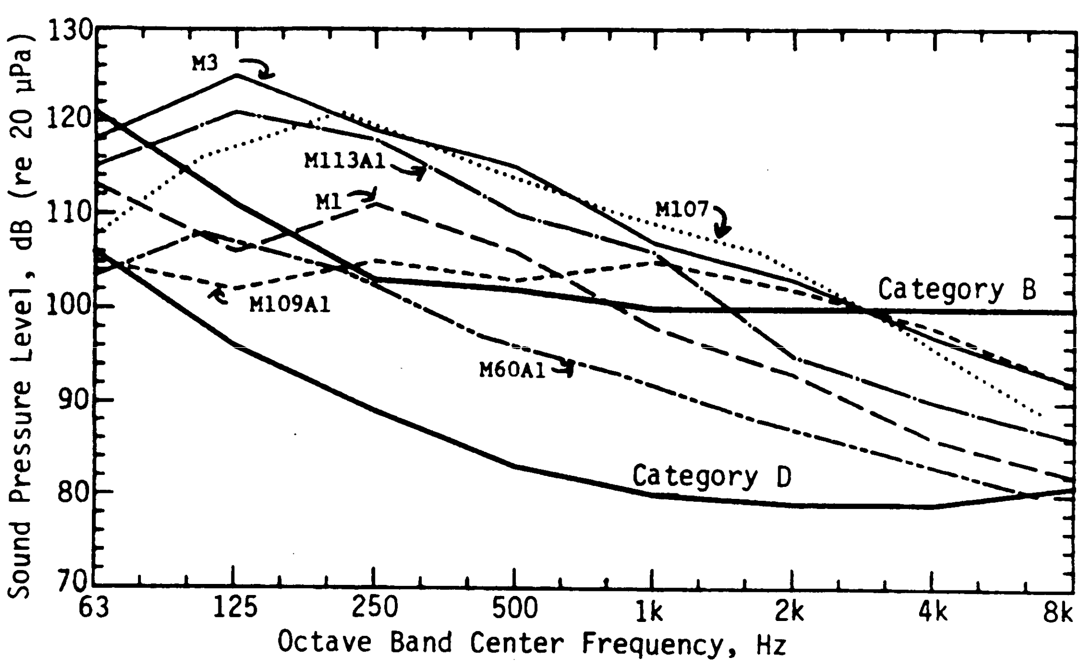

Figure 3-3. Comparison of Octave Band Interior Noise Levels for Various Tracked Vehicles Measured at the Driver's Position With the Vehicle Moving at 2/3 Maximum Speed

sense that vehicle performance requirements must be translated into component design specifications. vehicle noise limits must also be translated into noise limits and reduction goals for each contributing source. The simplest noise limit criterion is an A-weighted limit for each source. If only three noise sources contribute to the total noise, the noise goal for each source would be 5 dB less than the system limit. If the vehicle has five predominate noise sources, the noise goal for each source would be 7 dB less than the system limit. (See par. 2-3.1 for an explanation of summation of noise from multiple sources.) Spectral (octave or one-third octave band) noise limits are preferred when specifying individual noise limits because the noise reduction solution is highly dependent on the frequencies to be reduced. Fig. 3-4 shows [he major noise sources of an M113 vehicle and the noise goal for each source as derived from the limits of MIL-STD-1474.

The costs associated with noise reduction in a tracked vehicle are not linear with respect (o the amount of noise attenuation required. For example. minor changes and minimal cost may be required to reduce a noise source by one or two dB. whereas 10-dB reduction from another source may require extensive design changes and considerable cost. Using a balanced noise reduction program where the noisiest sources are reduced the most allows the least noise reduction. and hence the least cost. for each component or subsystem while still meeting the total vehicle noise limit of 100 dB(A).

Translating noise limits into component designs is somewhat easier on existing rather than new vehicles because noise reduction goals can be determined quite easily. These goals. defined as the difference between actual noise levels and limits established by MIL-STD-1474. are more difficult to define in new vehicles. In new vehicles noise contributions from each component are not known until prototypes have been tested or extensive analytical modeling has been done. Thus noise reduction goals for a new design are initially based on noise data from existing vehicles of a similar design. These noise data are collected through a source noise investigation as described in par. 8-3. Interior noise source data for the M113 vehicle are available in Ref. 8.

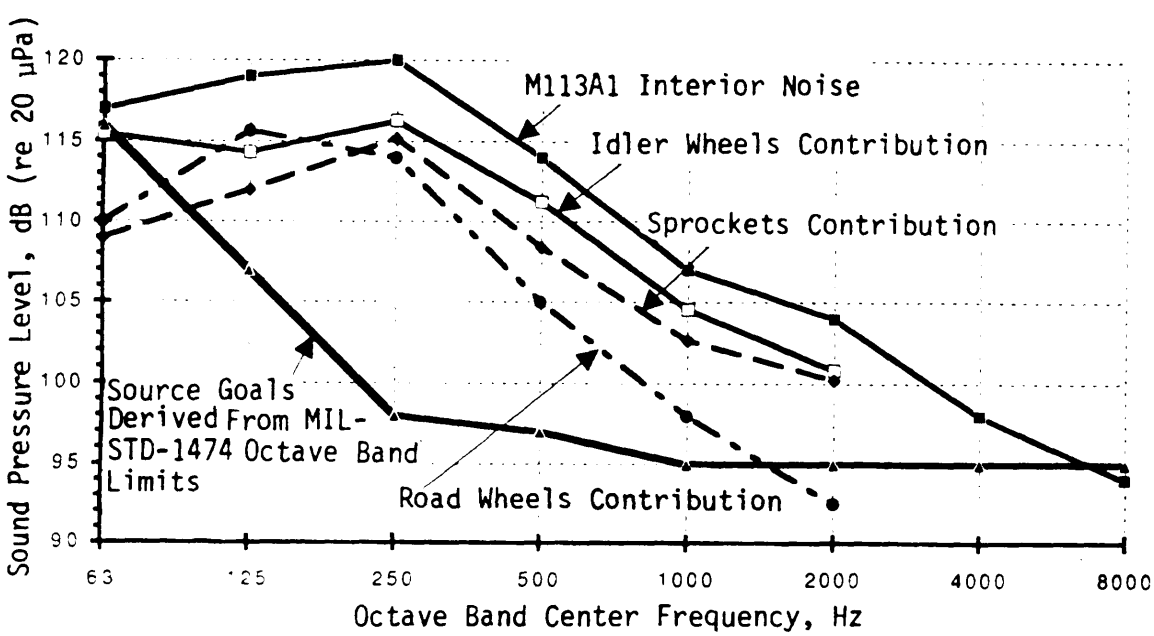

Figure 3-4. Example of Vehicle Noise Sources and an Octave Band Spectrum Goal. Which Would Allow the Sum of All the Sources (Total Interior Noise) not to Exceed the MIL-STD-1474, Category B Limit, 100 dB(A)

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CHAPTER 4
NOISE AND VIBRATION SOURCES

The major sources of interior noise and vibration in tracked vehicles are identified and discussed. Vibration generation mechanisms, chordal action, and rolling wheel action of suspension systems are explained. The noise generation mechanism of the vehicle hull is presented.

4-0 LIST OF SYMBOLS

F = original impact force, N
F_e = equivalent force of impact, N
F_{eb} = equivalent impact force before modification, N
F_{em} = equivalent impact force after modification, N
F_f = engine-firing frequency, Hz
F_m = impact force after modification, N
F_t = tracklaying frequency. Hz
K = spring rate of wheel/shoe interface, N/m
K_m = spring rate of wheel/shoe interface after modification, N/m
L = track shoe length, cm (in.)
m = mass of track shoe. kg
m_m = mass of track shoe after modification, kg
n = number of cylinders, dimensionless
p = number of power strokes per cylinder per revolution, dimensionless
R_1 = distance from wheel center to shoe pin center, m
s = vehicle speed, km/h (mi/h)
s_e = engine speed, rev/rein
v = track shoe velocity, m/s
v_l = vertical velocity at shoe impact, m/s
v_{lm} = velocity of shoe impact after modification, m/s
ΔL_n = change in sound pressure level, dB
ΔL_p = change of sound pressure level in tracked vehicle, dB
ϕ = angle subtended by 1/2 of track shoe length, rad
ω = angular wheel velocity. rad/s

4-1 TOTAL VEHICLE NOISE SOURCES

4-1.1 MAJOR NOISE SOURCES IN TRACKED VEHICLES

At vehicle speeds above 8 to 16 km/h (5 to 10 mi/h), the primary source of noise in tracked vehicles is hull vibration, which is generated by the interaction of track and suspension components, i.e., sprockets. idler wheels, road wheels, and support rollers. At speeds below 8 km/h (5 mi/h) or during silent watch, when the vehicle is stationary, the major noise source is the engine. Other noise sources, such as ventilation fans, auxiliary power unit (APU), and turret drives, may also be significant during silent watch, but at speeds above 8 km/h (5 mi/h) they are less important (Ref. 1).

4-1.2 RANKING NOISE SOURCES

Ranking the noise levels by source enables the designer to calculate how much he must reduce the noise of each source in order to lower the overall noise level of the vehicle. Noise sources can be rank ordered in different ways, depending on the type of vehicle operation in which reduced noise is most important. Fig. 4-1, for example, shows three rankings of M113A 1 noise sources by vehicle speed. A much different ranking would be obtained for stationary vehicle operation for which the suspension is no longer a noise source and the engine or APU would be the loudest source. As shown in Fig. 4-1, to reduce overall vehicle noise from 116 dB(A) to 100 dB(A) at speeds of 40.2 km/h (25 mi/h), the designer must reduce each noise source—idlers, sprocket, road wheels, power train-to 94 dB(A). (Refer to par. 2-3.1.) Subtracting 94 dB from each source yields a noise reduction goal of 18 dB for the idler wheels, 17 dB for the sprocket wheels, 12 dB for the road wheels, and 5 dB for the power train.

As stated earlier, to reduce overall noise significantly, all major noise sources must be reduced. If in the previous example the designer reduced idler-generated noise by 15 dB, total noise would be reduced only from 115 to 113 dB, and the new source ranking would be as shown in Fig. 4-2.

4-1.3 CATEGORY D OR E NOISE LEVELS

The noise of components, such as APUs, fans, and air-conditioning equipment, is usually not a significant contributor to interior noise. During silent watch operations in certain vehicles, however, these sources may be just as important to the successful completion of the mission as primary noise sources are during moving vehicle operations. Thus under silent watch, noise should meet the limits of Category D or E of MIL-STD-1474, whereas a moving vehicle evaluation should meet the limits of Category B.

4-2 SUSPENSION

The five major noise-generating components of a high-speed, tracklaying military vehicle are the track, idler wheels, drive sprockets, road wheels, and support rollers. These components are illustrated in Fig. 4-3. Interaction of the track with the four other components, through chordal action and

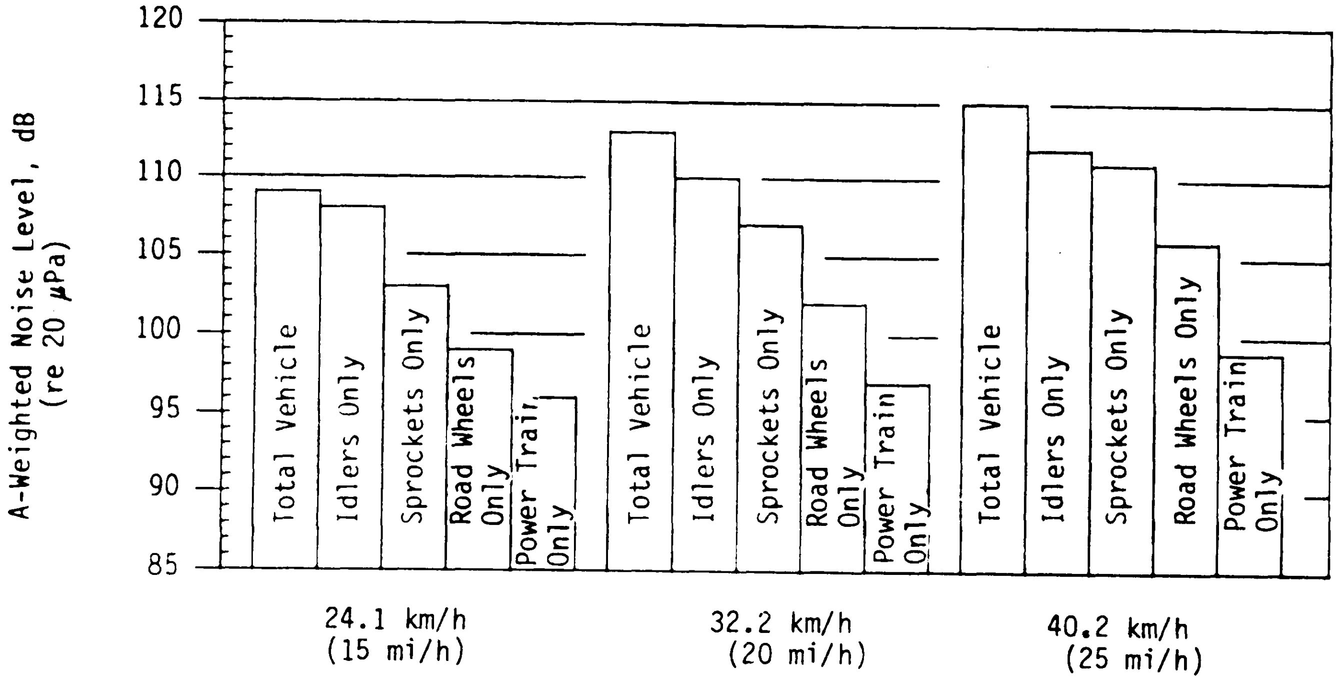

Figure 4-1. M113A1 Crew Area Noise Sources Rank Ordered for Various Speeds (Ref. 1)

rolling wheel action. generates large oscillatory forces and moments in the vehicle hull. which cause it to vibrate. A description of the five key suspension components follows:

1. *Track.* The track is composed of a number of track shoes connected in a continuous loop, The top surface of track shoes forms a running surface for the road wheels. Generally. this surface has a rubber-filled cavity-inner track pad or backing rubber-and a track guide. The bottom (roadside) of the track shoes has a replaceable rubber pad to increase traction and protect paved roads. Adjacent shoes are interlocked with single or double pins. as shown in Fig. 4-4.

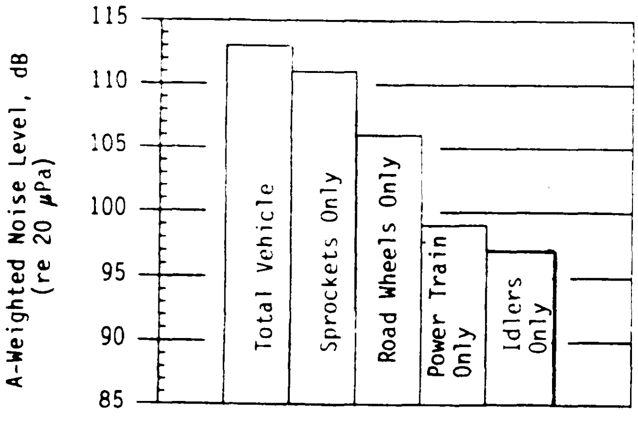

Figure 4-2. Example of M113A1 Crew Area Noise Source Ranking for 40.2 km/h (25 mi/h) After Idler Wheel Noise Reduction

2. *Idler Wheels.* Idler wheels typically provide two functions in the track system: They serve as the track return roller. and they allow adjustment of track tension, Noise is generated when the trach interacts with the wheel in a movement called chordal action. which is described in par. 4-2.1.

3. *Drive Sprocket.* Drive sprockets are toothed rings that supply driving force to the tracks. They are attached to the sprocket carrier. which is a wheel attached to the final drive that supplies the primary driving torque to the drive sprockets. The noise generation mechanism is the same as for the idler wheel. i.e., chordal action. In addition. when sprocket teeth hit the track shoes. they cause a clatter or metallic clanging sound, which contributes to the external noise signature but has little effect on interior noise.

4. *Road Wheels.* Road wheels provide two main functions: They support and distribute the vehicle weight. and they provide track guidance for track retention. Road wheels noise is principally caused when the wheels roll over the track on the ground. A secondary mechanism is chordal action at the first and last road wheels. Track guides rubbing on the inside ring of the road wheels may cause a metallic ringing sound, which contributes to external noise signature but does not affect interior noise.

5. *Support Rollers.* Support rollers are used in some track systems, especially in high-speed and high-mobility vehicles, to keep the top track strand from contacting the road wheels. These rollers generate interior noise due to the rolling action and track bounce of the supported track.

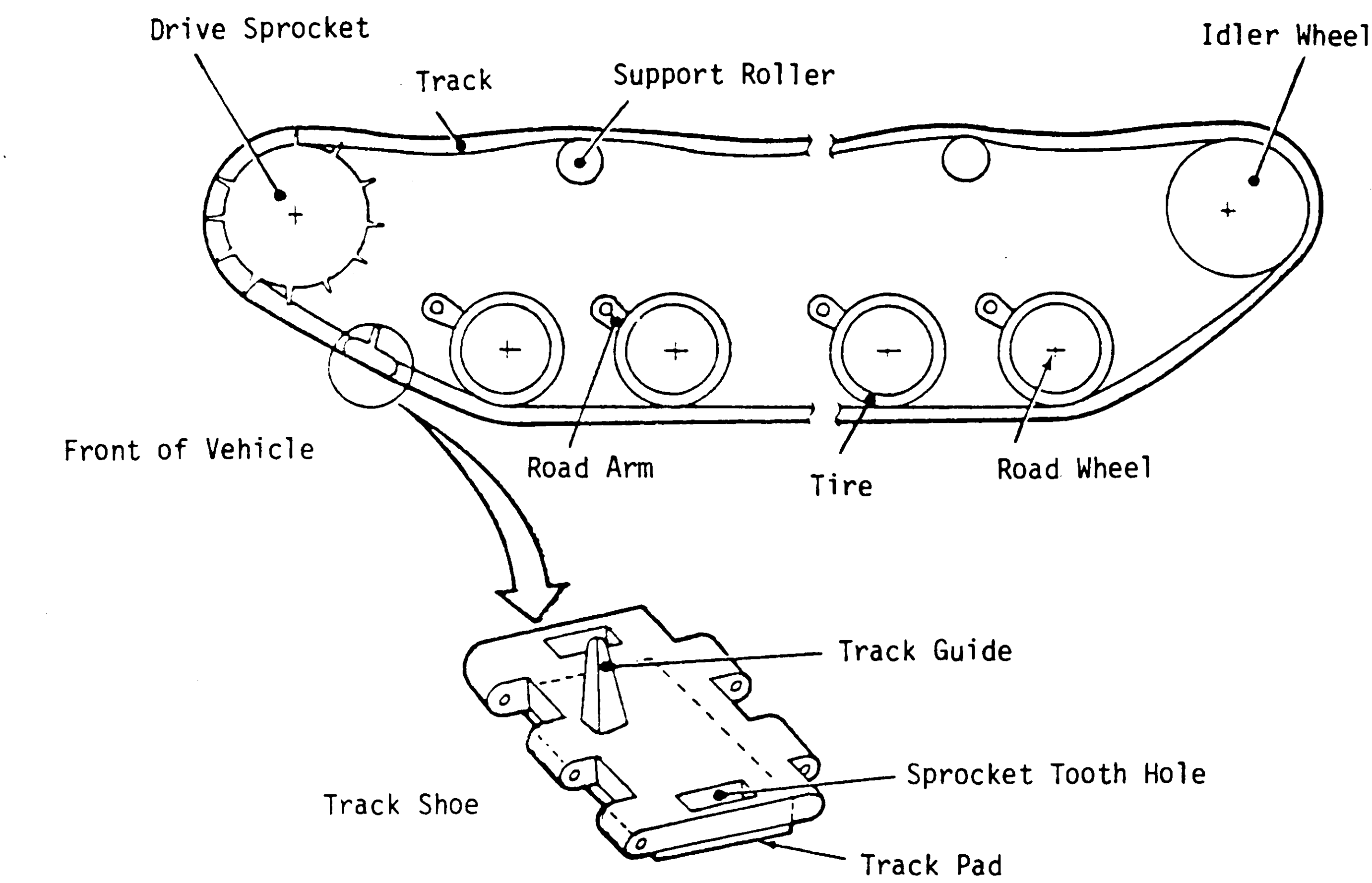

Figure 4-3. Typical Suspension System of Tracked Combat Vehicles

Vibration from the suspension components is transferred to all parts of the vehicle through the hull. where it generates broadband random noise. As shown in the typical interior noise spectra of Fig. 4-5. the majority of the sound energy from suspension components occurs in the low frequencies.

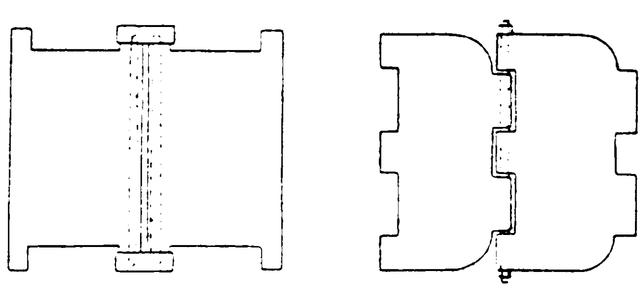

Figure 4-4. Illustration of Single-Pin and Double-Pin Track Shoes

4-2.1 CHORDAL ACTION

Chordal action occurs as each track shoe begins to revolve around the idler or sprocket wheel. Because the track is made up of a series of rigid, flat shoes, it moves around a wheel in a series of chords of a circle, hence the name "chordal action"'. If the track were very flexible, as in a continuous belt. chordal action would be eliminated. Most of the vibration in chordal action occurs when a track shoe impacts the idler or sprocket wheel. As shown in Fig. 4-6, track shoes do not contact the wheel until the pin of the leading shoe [ravels one-half the shoe length past the wheel tangent point.

The incoming shoe prior to impact is horizontal and has a velocity equal to

$$v = \omega R_1, \text{ m/s} \tag{4-1}$$

where

v = track shoe velocity,. m/s
ω = angular wheel velocity. rad/s
R_1 = distance from wheel center to shoe pin center, m.

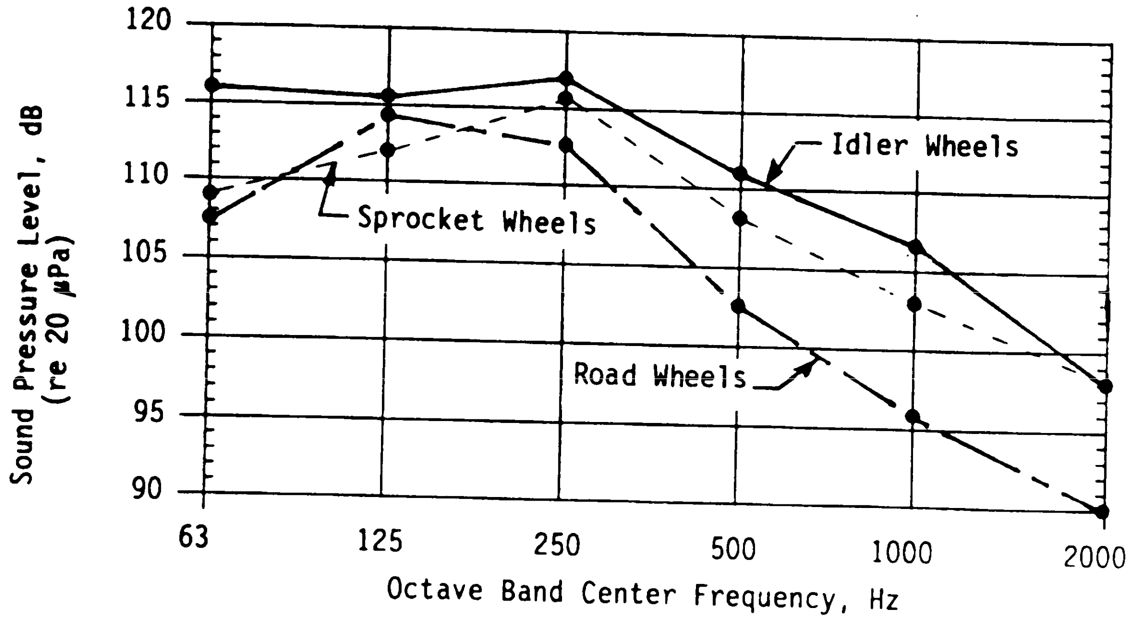

Figure 4-5. Spectra of Suspension Noise Sources in the Crew Area of an M113A1 at 40.2 km/h (25 mi/h)

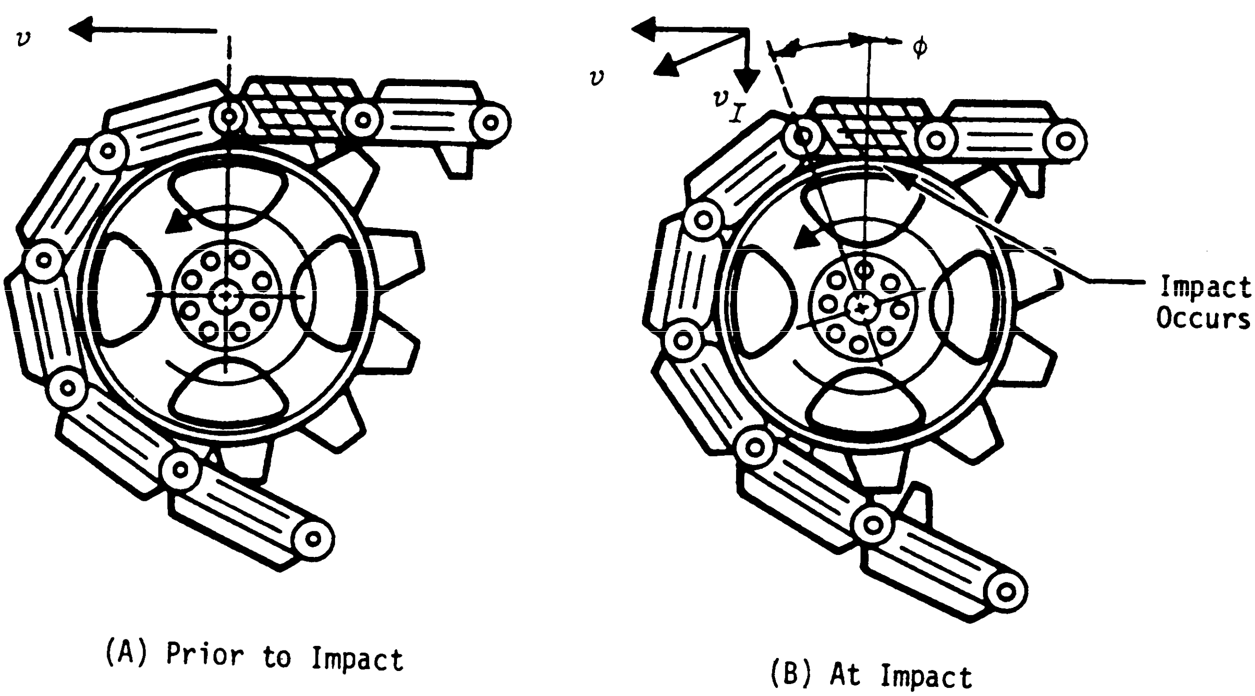

Figure 4-6. Chordal Action, Showing Track Shoe Impact

As shown in Fig. 4-6(B), the shoe has a vertical velocity component at the time of contact v, equal to

$$v_I = v\sin\phi,\ \text{m/s} \qquad (4\text{-}2)$$

where

v_I = vertical velocity at shoe impact, m/s
ϕ = angle subtended by 1/2 of track shoe length, rad.

The vertical velocity v, of the track shoe goes to zero when the shoe meets the wheel. This sudden change in velocity decreases track shoe kinetic energy, which is transferred as stored strain energy at the interface of the shoe and wheel. Additionally, the horizontal track shoe velocity variations resulting from chordal action cause track tension variations, which impose additional horizontal forces on the sprocket and idler wheels. Thus chordal action is responsible for the two major sources of tracked vehicle vibrational energy. The equivalent force of chordal impact F_e can be calculated as (Ref. 2)

$$F_e = \sqrt{mKv_I^2},\ \text{N} \qquad (4\text{-}3)$$

where

m = mass of track shoe, kg
F_e = equivalent force of impact, N
K = spring rate of wheel/shoe interface, N/m.

These impact-related forces, containing energy at many frequencies, cause broadband or random vibration of the vehicle hull.

Interior noise is directly related to the magnitude of impact. A study (Ref. 3) investigating the effects in buildings of impact noise from a standard tapping machine defined this relationship as

$$\Delta L_n = 20\log\left(\frac{F}{Fm}\right),\ \text{dB} \qquad (4\text{-}4)$$

where

ΔL_n = change in sound pressure level, dB
F = original impact force, N
F_m = impact force after modification, N.

This theory can be extended to the interior of a tracked vehicle where the track shoes supply the impact force to the idler wheels and sprocket wheels; thus the change in sound level is defined as

$$\Delta L_p = 20\log\left(\frac{F_{eb}}{F_{em}}\right) = 20\log\left(\frac{mKv_I^2}{m_m K_m v_{Im}^2}\right)^{\frac{1}{2}}$$

$$= 10\log\frac{mKv_I^2}{m_m K_m v_{Im}^2},\ \text{dB} \qquad (4\text{-}5)$$

where

ΔL_p = change of sound pressure level in tracked vehicle, dB
F_{eb} = equivalent impact force before modification, N
F_{em} = equivalent impact force after modification, N
m_m = mass of track shoe after modification, kg
K_m = spring rate of wheel/shoe interface after modification, N/m
v_{Im} = velocity of shoe impact after modification, m/s.

As seen in Eq 4-5, force, hence noise, produced by chordal action can be reduced by reducing the track shoe mass, reducing the spring rate of the track shoe and/or wheel, and reducing track shoe impact velocity.

Chordal impact occurs at the tracklaying frequency F_t, which is determined by

$$F_t = 27.78\left(\frac{s}{L}\right),\ \text{Hz} \qquad (4\text{-}6a)$$

or in the English system of units

$$F_t = 17.6\left(\frac{s}{L}\right),\ \text{Hz} \qquad (4\text{-}6b)$$

where

F_t = tracklaying frequency, Hz
s = vehicle speed, km/h (mi/h)
L = track shoe length, cm (in.).

For a shoe length of 152.4 mm (6 in.), the tracklaying frequency is approximately equal to 1.8 times the vehicle speed in kilometers per hour (3.0 times speed in miles per hour). Tracked vehicle hull vibration typically is composed of broadband or random vibration with a large periodic component at the tracklaying frequency, as illustrated in the spectrum of Fig. 2-4.

4-2.2 ROLLING ACTION

Road wheels and support rollers generate vibrational forces through the rolling action of the wheels on the track. The top surface of the bottom track strand (over which the road wheels run) is not smooth due to the necessary clearance between track shoes, which allows them to rotate around the return wheels. This "rough" surface becomes even rougher when the track shoes tip due to road wheel loading at the hinge points, as in Fig. 4-7(B) (Ref. 4).

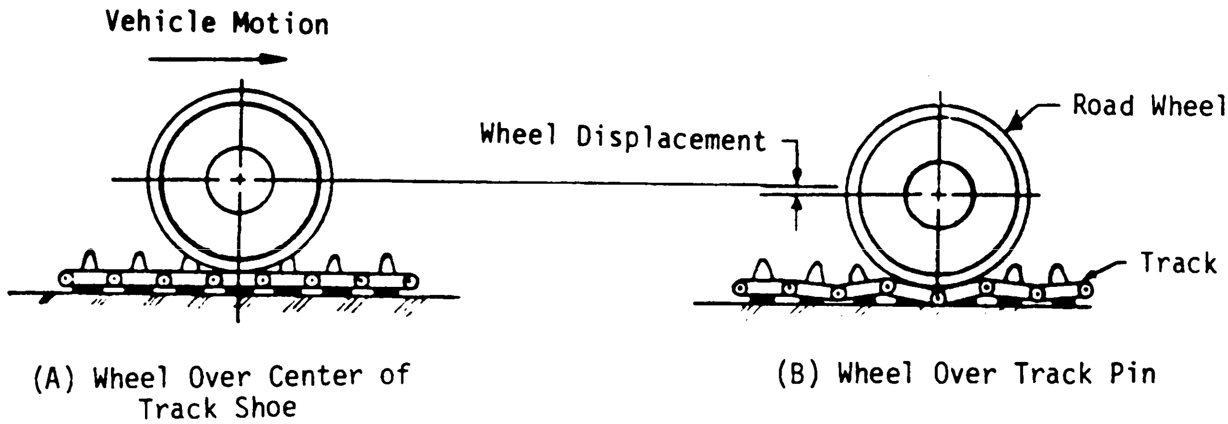

Figure 4-7. Road Wheel Vibration Generation

When road wheels roll over this rough surface, they vibrate much like a car on a rough road. This vibration occurs primarily at the tracklaying frequency, which is a function of track shoe pitch and vehicle speed. The road wheels have a rubber tire on the periphery. and the inside surface of each track shoe typically has a rubber pad. Both of these attenuate road wheel vibration, The compliance of these "vibration isolators", however. is quite low due to the requirements for large load capacity and durability in the harsh environments in which the vehicle operates. This low compliance (high stiffness) allows road wheel vibration to be readily transmitted to the road arms, where it is then coupled into the hull structure through the road arm trunnions.

The effects of track shoe tipping (Fig. 4-7(B)) are influenced by ground conditions and vehicle speed. A hard surface allows the shoes to tip more than a soft surface because the edges of the track pad act as fulcrum points for shoe rotation on a hard surface. but the) tend to sink into soft ground and move the effective rotation fulcrum points out from the pins toward the center of the shoe. Less track shoe rotation means the road wheels have a smoother surface to ride on when the vehicle is operating on soft soil or sand, Road-wheel-generated noise is therefore lower for a vehicle operating on a soft surface by as much as 1 to 3 dB. However, because road-wheel-generated noise is less severe than sprocket- and idler-generated noise, a 3-dB reduction in road-wheel noise reduces total vehicle noise only about 1 dB.

The noise produced by the track shoes impacting the ground (track slap) contributes to the external noise signature but has no effect on interior noise.

If a tracked vehicle uses support rollers on the upper track strand. they will induce vibratory forces into the hull due to the rolling action of the track. Support roller forces are less severe than road-wheel forces, but the noise generated may be as significant because the rollers are generally mounted at a location with lower hull stiffness. Thus less force is required to generate the same hull vibration levels.

4-3 POWER TRAIN

4-3.1 ENGINE

Diesel engine noise generation is a complex process involving multiple noise sources and radiation paths. Engine noise varies with type of engine (four cycle, two cycle. rotary), displacement, and operating conditions (load and speed). Combustion in each cylinder during (he power stroke produces high forces on the pistons. These forces, along with residual unbalances in rotating parts. produce vibration. which is transformed into noise through two acoustic coupling paths: structure-borne and airborne. To reduce engine noise effectively in the vehicle interior, both paths must be attenuated. Structure-borne noise is transferred to the vehicle interior through the engine-to-hull mounts: the amount depends on the isolation efficiency of the mounts. Airborne noise radiates from the engine and impinges on the engine compartment enclosure. The attenuating capacity of the enclosure determines how much noise radiates into the crew area.

Fig, 4-8 shows a typical noise spectrum in the crew area of a stationary M113A 1 with only the diesel engine operating.

Noise from reciprocating and rotary internal combustion engines is dominated by large periodic components at the engine-firing frequency and multiples of that frequency. The engine-firing frequency is determined from

$$F_f = \frac{s_e n p}{60}, \text{ Hz} \tag{4-7}$$

where

F_f = engine-firing frequent). Hz
s_e = engine speed, rev/rein
n = number of cylinders, dimensionless
p = number of power strokes per cylinder per revolution. dimensionless.

Note: p = 1 for 2-cycle engine and p = 0.5 for 4-cycle engine.

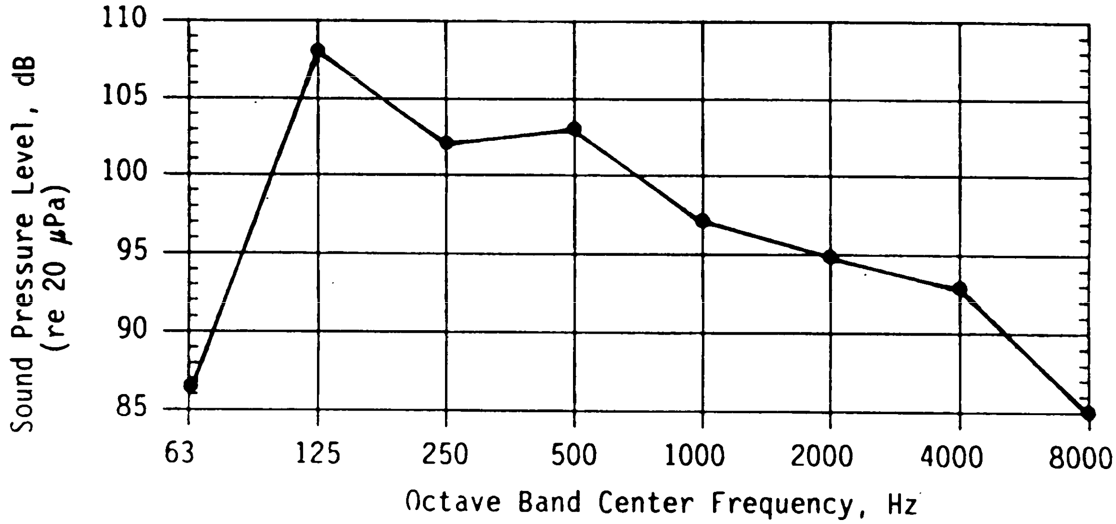

Figure 4-8. Engine Noise Spectrum of an M113A1 at 2600 rev/rein

Engine noise generally increases with engine speed and is the major noise source when vehicle speed is low and engine speed is high as when climbing a steep hill.

4-3.2 TRANSMISSION AND FINAL DRIVES

The transmission and final drive gearboxes are sources of noise and vibration. The final drive gearboxes transmit torque from the transmission or steering differential to the drive sprockets. The vibration generated by the final drives is periodic and harmonically related to the input shaft speed by the ratios of the final drive gear teeth. The final drive gearboxes are usually mounted in a stiff area of the hull to minimize vibration.

Transmission-generated noise and vibration result from gear meshing, shaft and bearing imbalances, hydraulic pressure fluctuations, and drive torque oscillations. Just as is the case with engine noise, transmission noise is both structure-bome and airborne. In many tracked vehicle designs the transmission is hard coupled to the engine. i e., engine vibration is also input into the transmission housing, which then radiates noise. Transmission noise reduction first requires identification of the relative importance of the noise generation mechanism, i.e., gear noise, hydraulic noise, etc. Transmission design changes should then be incorporated that will attenuate the noise generation mechanism. Possible fixes may include higher precision gearing, quieter hydraulic pumps, or improvements in shaft balancing. If transmission design changes alone cannot reduce noise sufficiently, the noise radiation paths would require interruption through vibration isolation mounts or improvements in the noise attenuation of the engine compartment.

4-3.3 OTHER POWER TRAIN COMPONENTS

Other secondary noise sources associated with the power train include cooling fan, engine exhaust, engine air intake, and auxiliary equipment. A discussion of each follows:

1. *Cooling Fan. The* engine cooling fan generates noise by two mechanisms: airflow turbulence around the blades (airborne noise) and rotor imbalance (structure-borne noise). Although a significant noise generator, the cooling fan is generally considered a secondary interior noise source for the reasons that follow:

   a. Usually the fan is located in the engine compartment, which is separated from the crew area. An engine compartment designed to reduce engine noise will also attenuate fan noise.

   b. Acoustically the inlet and outlet are open to the atmosphere; therefore, the fan noise is more of an exterior than an interior noise source.

   c. Mechanically fans are generally well-balanced, which significantly reduces their vibration input to the vehicle hull and thus reduces their structure-borne noise contribution. However, a driver or crew member operating with an open hatch may experience the high external noise levels generated by the fan.

2. *Engine Exhaust. The* engine exhaust components, i.e., turbocharger, piping, and muffler, are considered secondary interior noise sources for the reasons that follow:

   a. The exhaust outlet, which is directed through the hull structure to the outside of the vehicle, contributes mainly to the exterior noise signature of the vehicle. Some exhaust noise. however, can intrude if the hatch covers are open.

b. The vibration levels of the exhaust piping and muffler shell are relatively low. If fastened to the engine or major hull plates, these components are not significant sources of structure-borne noise. However, if fastened to thin bulkhead panels, their structure-borne noise could be significant.

3. *Air Intake. The* engine air intake is not considered a significant noise source for the reasons that follow:

a. The air intake is through an air cleaner, which is usually mounted in the engine compartment and thus interrupts the acoustic path to the crew area.

b. The engine air intake contributes no significant vibration because the air cleaner is usually mounted at a stiff location and is connected to the engine through a flexible tube.

4. *Auxiliary Equipment. This* equipment, such as ventilation fans, personnel heaters, and electronic components, can produce annoying levels of noise during silent watch but is generally not loud enough to contribute to potential hearing damage.

4-4 HULL DYNAMICS

4-4.1 VIBRATION

As seen in Fig. 4-9, the main generator of airborne noise is the hull, which accepts vibration energy from the primary noise sources, i.e., suspension and power train components, and radiates noise from all hull surfaces. Hull vibration amplitude is determined by the magnitude of the vibrating force at the source and the mechanical impedance of the hull structure at the attachment location of each source. For example, a road arm attached at a very stiff hull location produces less vibration than one attached at a more compliant location. The frequencies of the vibrational forces from the suspension or power train may or may not coincide with the natural vibration frequencies of the hull plates. In either case, the hull structure will vibrate and produce noise. If the hull structure is being excited at one or more of its natural vibration frequencies, the structure will go into resonance at those frequencies, which will greatly amplify the structural motion or velocity. More noise will be produced because of the resonant condition. This is known as the resonant response of the structure. At input forcing frequencies that do not correspond to structural resonances, the resultant hull structure vibration motion is termed its forced response.

The stiffness and structural damping of the hull determine its resonant response. Increasing hull stiffness tends to reduce its forced response to vibrational input forces. Increasing damping helps reduce the amplitude of the resonant vibration. In a research project conducted on an M113 vehicle on which the bottom plate, top plate, and upper “side plates were treated with constrained layer damping material, a 6-dB reduction in hull plate vibration was measured and A-weighted interior noise was reduced by 3 to 4 dB at speeds above 30 km/h (19 mi/h) (Ref. 5). Constrained layer damping is a surface treatment method of increasing the structural damping of a plate by sandwiching a thin layer of viscoelastic material between the plate and a layer of sheet metal. The practical limit for interior noise reduction through increased structural damping is approximately 4 to 5 dB and is highly dependent on the ratio of forced and resonant responses of the hull structure.

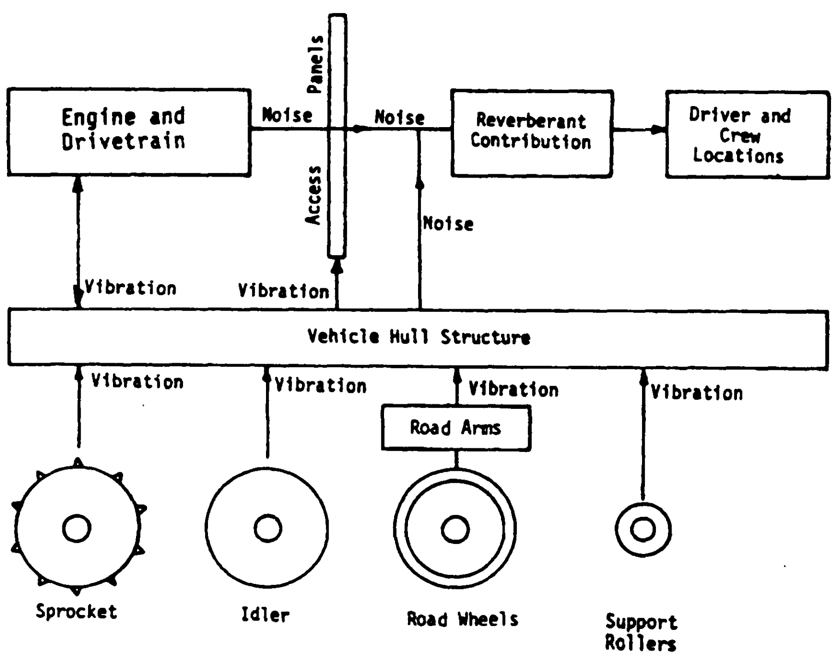

Figure 4-9. Schematic Diagram of Sources and Propagation Paths for Noise and Vibration in Tracked Vehicles

4-4.2 ACOUSTIC COUPLING

Interior noise generated by the hull structure of a tracked vehicle is determined by its acoustic coupling and radiation efficiency, as discussed in par. 2-3.3. The sound field inside a tracked vehicle is very complex and difficult to analyze. Classical acoustic theory usually deals with individual noise sources in contrast to the hull structure, which is a distributed sound source that radiates noise from virtually every surface. Interior surfaces of a tracked vehicle are for the most part low-absorbing surfaces, which reflect sound waves; thus the sound waves reverberate continuously from the walls, floor, and roof until absorbed. Although not a noise source, reverberation increases the sound level throughout the vehicle in a nonuniform manner due to standing waves or cavity resonances. Standing waves *are* generated whenever the reflection path is an exact multiple" of the wavelength of the reflected sound. As the reflected and direct waves interact. they either add to or cancel one another. The sound pressure level varies from maximum to minimum depending on the distance from the reflecting surfaces. Acoustic or cavity resonances can be reduced by reducing the reverberation levels in the vehicle interior. Reverberation is attenuated by adding absorptive materials to the interior surfaces of the hull structure; however, this treatment is difficult to achieve in a military combat vehicle due to space constraints. Because of the predominantly low-frequency content of tracked vehicle noise, very thick noise-absorptive materials are required to obtain any real reduction in the reverberant sound field. Interior space is always at a premium inside military tracked vehicles; thus the addition of 100- to 125-mm (4- to 5-in.) thick noise-absorbing material is not a practical solution to interior reverberant noise reduction. Tests have shown that 0.5 in. of absorptive material will reduce noise by 1dB at 500 Hz, by 2 to 3 dB between 500 and 2000 Hz, and by 5 to 10 dB at frequencies above 2000 Hz.

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CHAPTER 5
DESIGN OF LOW-NOISE SUSPENSION COMPONENTS

Guidelines to design tracked vehicle suspension systems tha[produce less noise are presented along with examples of previ - ous reduced-noise suspension component designs.

5-0 LIST OF SYMBOLS

A = shear area of rubber, cm^2
a = shear area of one rubber ring, cm^2
E = isolator efficiency, dimensionless
i = imaginary number = $\sqrt{-1}$
K_i = isolator spring rate, N/m
K_o = original idler wheel spring rate, N/m
K_r = radial spring rate of the idler wheel, N/cm
K_T = torsional spring rate, N·cm/rad
k = isolator spring rate, N/m
M_I = mobility transfer function of the resilient element ("isolator"), m/(s·N)
M_R = mobility transfer function of the idler spindle ("receiver"), m/(s·N)
M_S = mobility transfer function of the track ("source"), m/(s·N)
R = estimated noise reduction, dB
R_i = inner radius of rubber ring, cm
R_n = ratio of modified to original idler wheel spring rate, dimensionless
R_o = outer radius of rubber ring, cm
S = shear modulus of the rubber, N/cm^2
t = thickness of the rubber ring, cm
ω = radian frequency, rad/s

5-1. DESIGN APPROACH

The vehicle designer can reduce noise in tracked suspension systems in one of three ways:

1. Modify the source of vibrations (generate less vibration)
2. Modify the path of propagation (block vibration from entering the hull)
3. Modify both the source and path.

As discussed in par. 4-2. when the track interacts with the various suspension components (either through chordal action or rolling wheel action), it creates vibrational forces that enter the hull. If these forces can be controlled through source modification, this can be done best by modifying track shoe parameters. such as mass. stiffness. and geometry. If the path of vibration is to be modified. vibration-isolation systems for the sprocket wheels. idler wheels, road wheels, and support rollers should be designed.

The process used to modify the suspension components vibration propagation path to reduce hull-radiated noise consists of the six steps that follow:

1. Determine the noise reduction goals.
2. Predict the component spring rate required to meet the noise reduction goals.
3. Determine the design parameters necessary to achieve the desired spring rate and to maintain the load-carrying ability and durability).
4. Generate layouts and detail designs.
5. Build and test prototype hardware.
6. Revise the design as necessary to achieve noise reduction and durability goals.

Track modification is the best method to use to reduce hull-generated suspension noise. When the track is modified to generate less vibration because of a reduction in chordal action, less noise is produced by the idler wheels. sprocket wheels, and the road wheels. If the track modifications alone are not enough to meet the total suspension noise reduction goals, then vibration propagation path modification designs can be used for additional noise reduction. The amount of noise reduction required for each suspension component would then be the difference between its original noise reduction goal and the reduction achieved through track modification.

Reduced-noise suspension components must be robust and durable in addition to providing the required noise reduction. Suspension component designs [hat use elastomeric elements as vibration isolators must incorporate features to protect them from physical damage caused by ingestion of debris into the track system or excessive stresses developed during high-load vehicle maneuvers. Material becomes an important selection factor. The materials selected must allow the noise reduction features to operate over a broad range of temperatures and terrains that include snow, mud. rocks. and sand. Another major consideration in noise-reduced components is a fail-safe design that allows a damaged vehicle to continue its mission or return for repairs. To meet all these goals, extensive materials and prototype testing is required.

5-2 NOISE PREDICTION AND MODELING

To design low-noise suspension components. the noise contribution of each source must be quantified, For existing vehicles the noise sources can be measured. For new designs. however, the contribution of each source must be predicted. probably by computer models. By using such models, the anticipated noise level for each suspension component is obtained when predicted dynamic forces from each suspension component are multiplied by the measured (or predicted) noise-to-force transfer functions of the hull. One dynamic force model. original]} designed to predict suspension components forces in an M 113-sized vehicle. is called TRAX1ON (Refs. 1 and 2) and was developed by Bolt. Beranek, and Newman, Inc., (BBN) Cambridge, MA. Later modifications allowed the model to be used on M1-sized vehicles. Table 5-1 lists the required input parameters to TRAXION.

TRAXION assumes several simplifications, such as rigid suspension, flat ground profile, frequency-independent parameters, and a two-dimensional analysis. For a given speed TRAXION creates a time history of the vertical and longitudinal forces generated at the sprocket, idler, and road wheel attachments. When subjected to a spectral analysis. this time history yields a one-third octave band spectrum. Multiplying this spectrum by [he noise-to-force transfer function at the hull attachment location gives the predicted interior noise for each suspension component.

How to measure noise-to-force transfer functions for existing vehicles is described in subpar. 8-2.3: for new vehicles, however, these functions must be predicted. NOISE. a computer program developed by FMC Corporation, San Jose, CA, predicts these functions using normal modes analysis from a finite element model of the hull to predict the radiation efficiency of the hull plates and the noise produced by a unit force at a specified location on the hull (Ref. 3). Normal modes analysis of the hull structure can be performed using commercially available finite element analysis programs, such as ANSYS or NASTRAN. Fig. 5-1 illustrates the steps involved in predicting intenor noise levels.

5-3 TRACK DESIGN

As described in par, 5-1, the source of suspension system noise (vibrational forces) can be reduced by appropriately modifying the track. The elements of track shoe design that influence noise generation are

1. Shoe mass
2. Pitch (length of shoe)
3. Shoe flexibility
4. Compliance of inner track surface
5. “Flatness”’ of road wheel running surface.

Track system variables that influence interior noise levels include

1. Vehicle (track) speed
2. Track tension
3. Track age.

The paragraphs that follow describe each of the track shoe and track system parameters.

5-3.1 TRACK SHOE MASS

It is shown in par. 4-2.1 that chordal action forces on the sprocket and idler wheel are proportional to the mass of the track shoe. Thus, as shown in Eq. 4-5, reducing track shoe mass 50% should lower interior noise about 3 dB. However. when aluminum track (25% lighter than steel) was used on an XM800T scout vehicle. interior noise was lowered 3 to 8 dB over speeds of 8 to 64 km/h (5 to 40 mi/h) (Ref. 4)-a reduction much greater than that predicted by chordal action theory. One reason for this discrepancy may be that Eq. 4-5 assumed that the strain energy at the shoe/wheel interface was stored as a linear spring. When a track shoe has a rubber pad, however, the rubber is in compression when it contacts the wheel, which results in a nonlinear, increasing spring rate. Consequently, the smaller deflection of the rubber pad on the lighter track shoe produces smaller impact forces, which result in less noise than predicted.

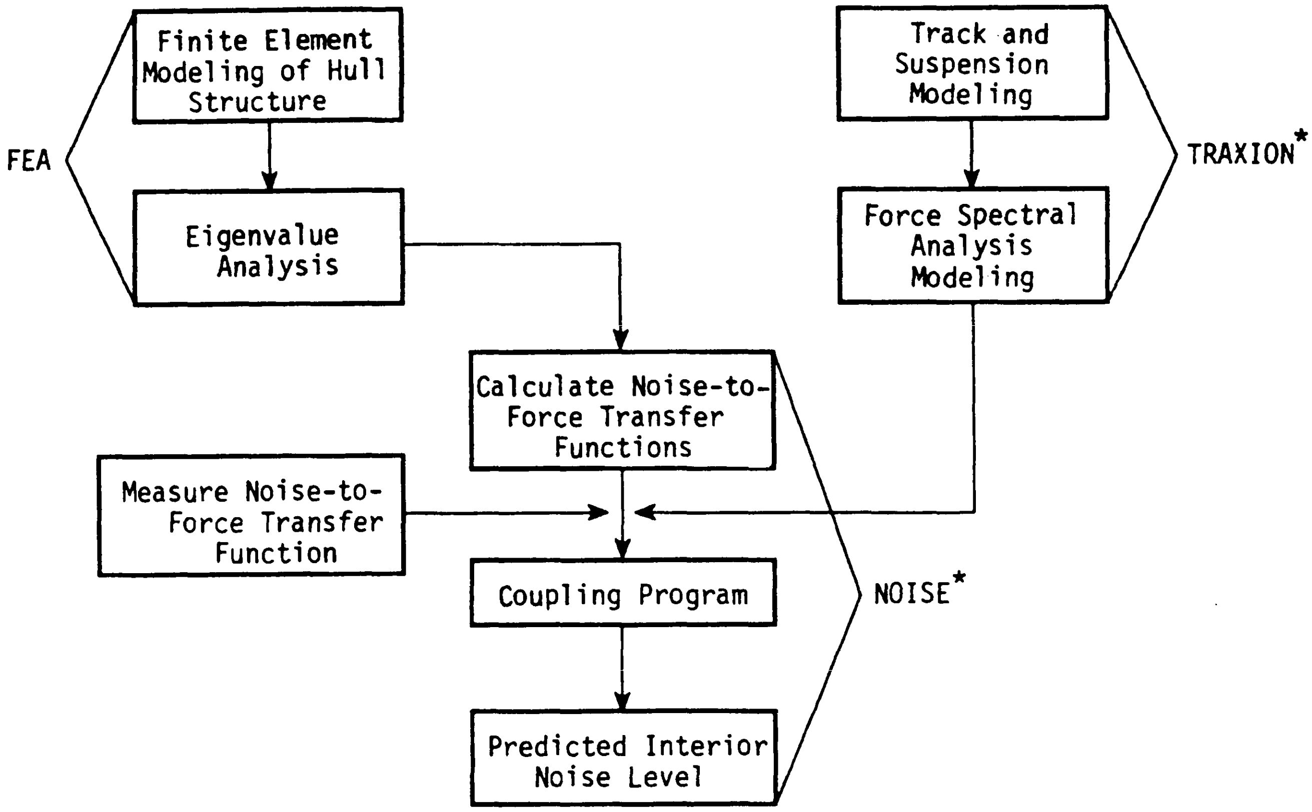
5-3.2 TRACK SHOE LENGTH (PITCH)

Decreasing the track shoe length provides at least three key benefits:

1. It lowers the impact velocity of chordal action.
2. It raises the tracklaying frequency, as shown in Eq. 4-6, beneficially increasing impedance mismatch between suspension components and hull structure, thus lowering the amount of hull vibration.

TABLE 5-1. INPUT PARAMETERS FOR TRAXION SUSPENSION MODELING PROGRAM

WHEELS (SPROCKET, IDLER, ROAD WHEELS)	TRACK	
	SHOES	BUSHINGS
x, y position	Length (pitch)	Stiffness
Wheel radius	Mass	Damping
Radial stiffness	Moment of inertia	Torsional stiffness
Tangential stiffness	Number of shoes	Torsional damping
Tangential damping	Track tension	
Friction coefficient		
Moment of inertia		



*TRAXION and NOISE are public domain software packages available from TACOM, Countermeasures Group, Warren, MI.

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Figure 5-1. Flowchart of Analytical Method of Tracked Vehicle lnterior Noise Reduction (Ref. 3)

3. It lowers the mass of the track shoe.

However, the presence of three interrelated factors makes it difficult to quantify the noise reduction achieved by the shorter pitched track alone. For example, Fig. 5-2 shows an actual A-weighted intenor noise reduction of 2 to 13 dB for an M113 vehicle tested with reduced pitch track (114 mm (4.5 in.)) compared to the same vehicle tested with standard track (152 mm (6 in.)). However, according to Eq. 4-5, the predicted noise reduction should have been only 3.6 dB, i.e., 2.3 dB for the shorter track and l.3 dB for the lower mass.

5-3.3 TRACK SHOE FLEXIBILITY

Making the track shoe more flexible allows it to bend as it goes around the return wheels at the sprocket and idler. This

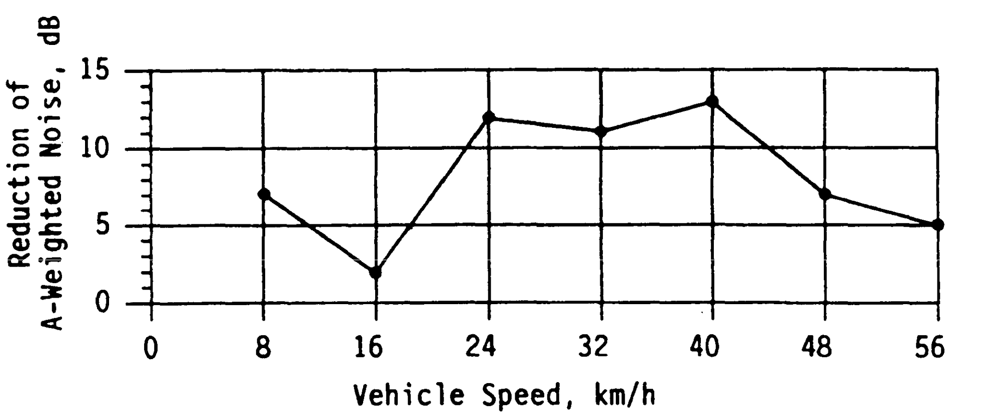

Figure 5-2. Noise Reduction Measured on an M113 Vehicle With Reduced Fitch Track

decreases shoe impact velocity and lowers track noise. If [he shoes could be made flexible enough, the track would act as a continuous band, which has no chordal action. To date. several attempts have been made to develop such a band track.

The Marine Corps, for instance, used an M113A2 vehicle to test a very flexible, wire-link track consisting of rows of rubber-encased. interconnecting links similar to bicycle chain. In this test the track reduced the low-frequency noise by 5 to 20 dB but appeared to raise high-frequency noise in the 2000- and 4000-Hz bands. However, the data may have been in error because these octave bands remained constant throughout all vehicle speeds, which indicates the presence of a high-frequency source other than the suspension-generated noise.

Lockheed Corporation, Huntsville, AL, developed another flexible track called a “loopwheel suspension” consisting of continuous, high-strength bands of composite fiber. In this design the loopwheel provided the required resilience. which eliminated the need for road wheels and torsion bars. A prototype version of this design proved to have limited durability and tended to throw the track at higher vehicle speeds.

To date. none of the experimental. highly flexible tracks have been successfully fielded due to durability and survivability problems. However. these systems may hate potential for future noise reduction applications as new high-strength materials are developed.

5-3.4 COMPLIANCE OF INNER TRACK SURFACE

Another method of decreasing chordal action is to increase the compliance of the track shoe surface that contacts the suspension wheels. As shown in Fig. 5-3, the track shoe typically has a recessed rubber pad. which is loaded in compression as the shoe impacts the suspension wheels. Eq. 4-5 predicts a 3-dB reduction for a 50% decrease in spring rate.

Durability of the inner track pad is very important in maintaining the noise reduction ability of the rubber. Low of rubber pad sections (chunking) or embedding of rocks or other debris increase suspension-generated noise.

5-3.5 “FLATNESS” OF ROAD WHEEL RUNNING SURFACE

Road-wheel-generated interior noise depends almost entirely on the “flatness” of the top of the track. which is in contact with the ground. Flatness is determined by three Factors:

1. Profile of ground surface under the track
2. Transition of running surface from one track shoe to the next
3. Flatness of inner surface on each track shoe.

Track shoe length principally determines how well the track conforms to the ground profile. Tracks having short pitch allow small ground bumps and dips to influence road wheel motion, which probably increases road-wheel-generated noise.

Transition from one track shoe to the next is probably the greatest factor in determining the noise generated by the rolling action of road wheels and support rollers. Design features such as a single-pin track, which produces less rolling action than the double-pin design, help prevent track shoe tipping as the road wheel crosses from one shoe to the next. This in turn reduces road wheel noise,

Flatness of the inner pad of the track shoe is important, but not critical. to reducing road wheel noise. Common manufacturing practices in producing precision-molded rubber parts are sufficient to produce a flat surface on the inner track pad.

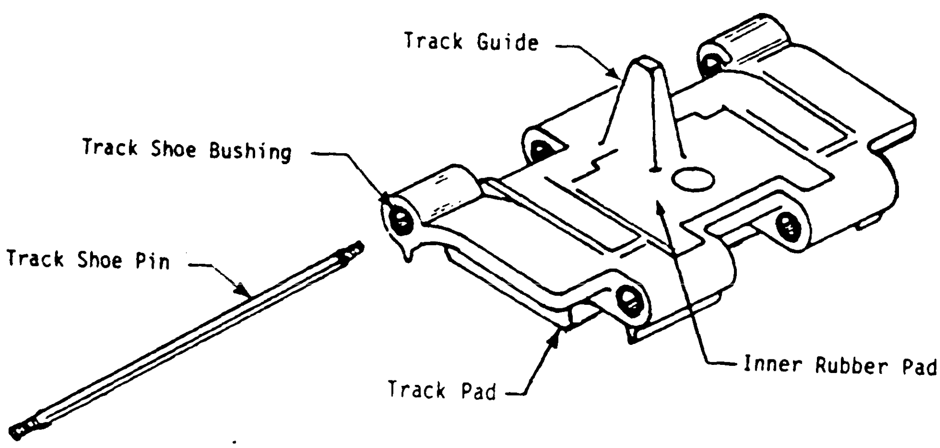

Figure 5-3. Components of Typical Track Shoe

5-3.6 TRACK SYSTEM VARIABLES THAT INFLUENCE INTERIOR NOISE

Three track system variables that affect interior noise are vehicle speed, track tension, and track age. A discussion of each follows:

1. *Vehicle Speed.* Interior noise generally increases with vehicle speed, as showrn in Eq. 4-5. which predicts a 6-dB increase per doubling of velocity. Fig. 5-4 shows the relationship of noise to speed for several tracked vehicles.

2. *Track Tension.* Although lowering track tension should generally decrease interior noise. tests on an M113A1 vehicle showed less than 2-dB reduction when track tension was decreased from 13.3 to 8.9 KN (3000 to 2000 lb) (Ref. 5). Therefore, practical reductions in track tension will not significantly reduce noise.

3. *Track Age.* Track shoe inner rubber pads (road wheel rolling surface) stiffen with age and thus tend to increase Interior noise by 1 to 2 dB. These rubber pads tend to become rougher with extended track usage. which also increases road-wheel-generated noise.

5-4 IDLER WHEEL DESIGN

5-4.1 IDLER WHEEL DESIGN APPROACH

To reduce idler-wheel-generated noise, chordal action forces (or vibrations caused by those forces) must be attenuated before they enter the hull. This can be done in either one or both of two ways: reduce track shoe impact velocity and/or isolate the structural path from the idler vheel to the hull.

5-4.2 PREVIOUS IDLER WHEEL DESIGN EXPERIENCE

Previous attempts to reduce idler-wheel-generated noise have met with varying degrees of success (Ref. 6). A discussion of some of these designs follows. All noise reductions cited are for idler-wheel-generated noise only, not total vehicle noise:

1. *Pendulous Idler.* The pendulous idler concept added a short. free-swinging arm to the idler spindle to allow the idler wheel to arc about the spindle axis. Test results showed noise reductions of 8 to 10 dB for vehicle speeds of 16.1 to 48.3 km/h (10 to 30 mi/h) (Ref. 7).

2. *Chordal Action Control Wheel.* The chordal action control (CAC) wheel reduced track shoe impact velocity by controlling the track shoe path around the idler. It was slightly larger in diameter than the standard idler wheel to which it was mounted, In tests this CAC wheel reduced noise 0 to 6 dB, depending on vehicle speed (Ref. 6).

3. *Increased Diameter Idler Wheel.* In this design the diameter of an idler wheel was enlarged from the standard 438 mm (17.25 in.) to 508 mm (20 in.). When tested on an M113A1 vehicle, the larger diameter idler wheel decreased track shoe impact velocity and reduced vibration and noise by 4 dB (Ref. 6). In a similar test, increasing the diameter to 610 mm (24 in.) reduced noise by 5 dB (Ref. 6). In both tests the reduced spring rate at the track shoe/idler wheel interface also contributed to the noise reduction. Test results correlated very well with the results predicted by Eq. 4-5.

4. *Crowned idler Wheel.* In this design the outside rim profile of the idler wheel was rounded, as shown in Fig. 5-5

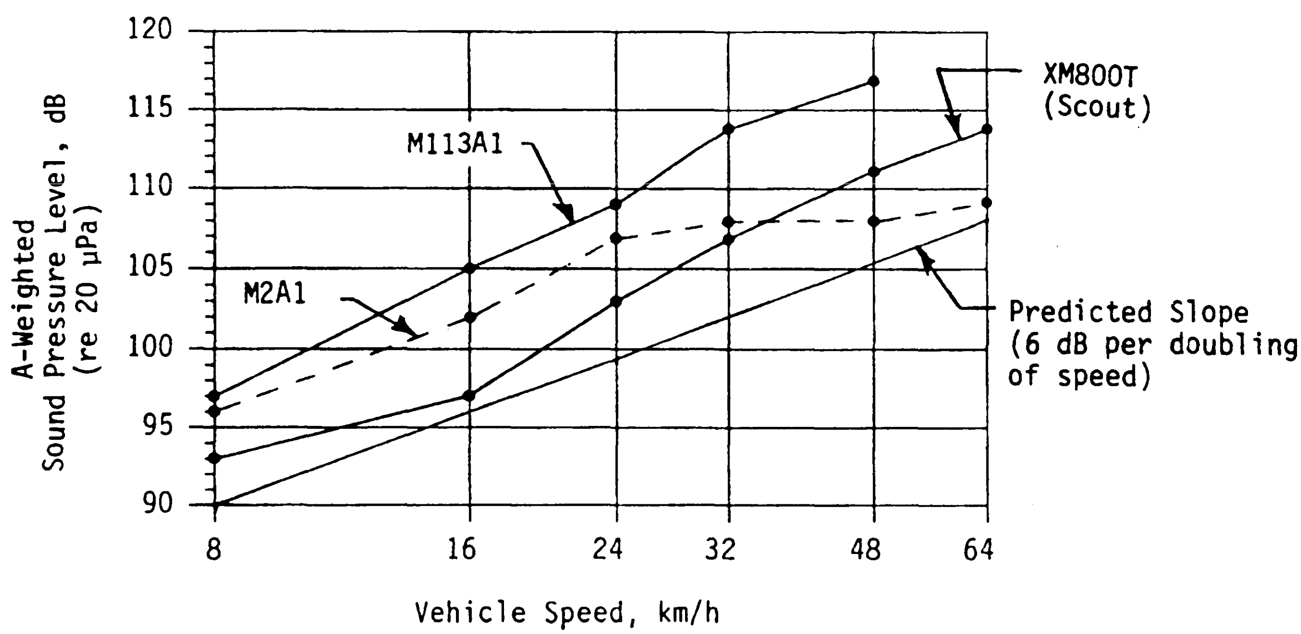

Figure 5-4. Interior Noise in Various Tracked Vehicles as a Function of Vehicle Speed

(Ref. 6), to increase the compliance of the inner rubber pad of the track shoes. (The standard idler wheel profile on the right side of Fig. 5-5 is shown for comparison purposes only.) When tested on an M113A1 vehicle, this modified idler wheel reduced noise an average of 5 dB.

5. *Damped idler Wheel. The* goal of this design was to reduce the amplitude of vibration at resonance by adding a highly damped metal alloy ring to the outside diameter of a standard M113A2 idler wheel (Ref. 8). However, test results showed no reduction in noise.

5-4.3 IDLER WHEEL DESIGN PARAMETERS

Noise reduction requirements for the idler wheel determine the type of design modifications needed. In genera], these requirements break down to two groups: reductions up to 6 dB and reductions greater than 6 dB. For noise reductions up to 6 dB, the most cost-effective way of decreasing idler wheel noise is to enlarge the idler wheel diameter and reduce the spring rate of the idler wheel/track shoe by using the relationships expressed in Eq. 4-5. Noise reductions greater than 6 dB require additional modifications, such as adding compliance between the idler rim and hub to isolate the idler hub vibration and reduce the vibrational forces entering the hull. The spring rate for this added isolator can be estimated using vibration isolation theory and measured transfer functions as follows: (Ref. 5)

$$E = 1 + \frac{M_I}{M_S + M_R}, \text{ dimensionless} \tag{5-1}$$

where

E = isolator efficiency, dimensionless
M_I = mobility transfer function of the resilient element ("isolator"), m/(sN)
M_R = mobility transfer function of the idler spindle ("receiver"'), m/(s.N)
M_S = mobility transfer function of the track ("source"), m(sN).

Isolator mobility is estimated as the mobility of an undamped, massless spring as (Ref. 5)

$$M_I = \frac{i\omega}{k}, \text{ m/(s·N)} \tag{5-2}$$

where

i = imaginary number = $\sqrt{-1}$
k = isolator spring rate, N/m
ω = angular frequency, rad/s.

Idler spindle mobility and track mobility are determined from measurements on an existing vehicle or estimated using finite element analysis. For lightweight tracked vehicles the most important frequency range for determining the required isolator spring rate is 125 to 1000 Hz. Noise reduction of the isolated idler wheel is estimated as

$$R = 20\log E, \text{ dB} \tag{5-3}$$

where

R = estimated noise reduction, dB.

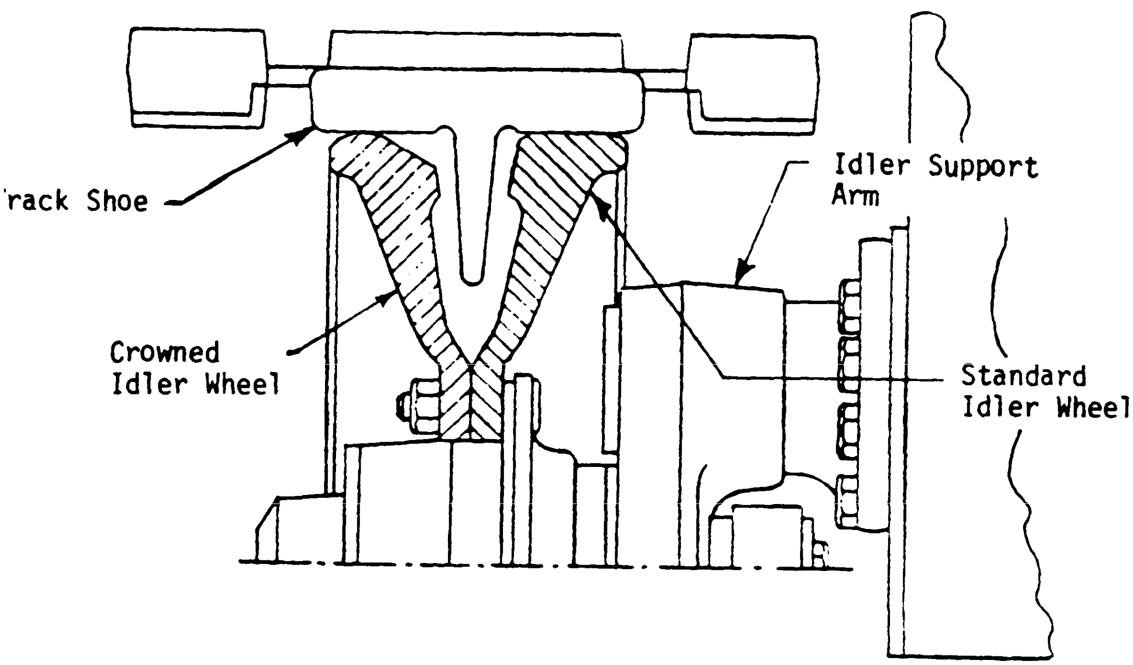

Figure 5-5. Crowned Idler Wheel Design

One way to determine the optimum isolator spring rate is to analyze various isolator spring rates and compare the required noise reduction to the estimated noise reductions in Eq. 5-3. A simpler method, however, is to estimate the required isolator spring rate by using empirical results from isolated idler wheel tests on an M113A1 vehicle. In these tests noise was reduced approximately 4.5 dB per halving of idler wheel spring rate. By using this empirical value the required spring rate is

$$K_i = \frac{K_o}{\exp\left(\frac{\log 2 R_n}{4.5}\right)} = \frac{K_o}{\exp\,(0.067 R_n)}, \text{ N/m} \tag{5-4}$$

where

K_i = isolator spring rate, N/m
K_o = original idler wheel spring rate, N/m
R_n = ratio of modified to original idler wheel spring rate, dimensionless.

The required compliance between the idler wheel rim and hub can be provided in a number of ways. e.g.. steel coil springs. rubber compression springs, or helical wire cables. Although these materials have not previously been successfully incorporated into compliant idler wheel designs (due to problems in design complexity, short fatigue life. and manufacturing difficulties), one design that has proven successful uses a rubber resilient element mounted in a shear configuration. When thus configured and built of the proper materials, this design is easy to manufacture and offers good fatigue life. When using vibration isolators with very soft spring rates, the designer must constrain isolator deflection with mechanical stops that can carry loads exceeding the design limits of the isolator. Otherwise, occasional high track tension loading will cause the isolators to fail.

Isolated idler wheels must also meet other design requirements, such as

1. Retaining the track
2. Withstanding maximum track loads (track throw)
3. Providing fail-safe capabilities
4. Offering a minimum durability of 3220 km (2000 mi)
5. Limiting weight gain
6. Withstanding tracked vehicle environment (temperature extremes, water and mud, debris ingestion, etc.).

5-4.4 M113 PROTOTYPE QUIET IDLER WHEEL

Figs. 5-6 and 5-7 show a successful idler design that meets the noise reduction requirements of MIL-STD-1474, Category B, for idler-dependent noise. Fig. 5-8 shows the interior noise reduction achieved with this idler.

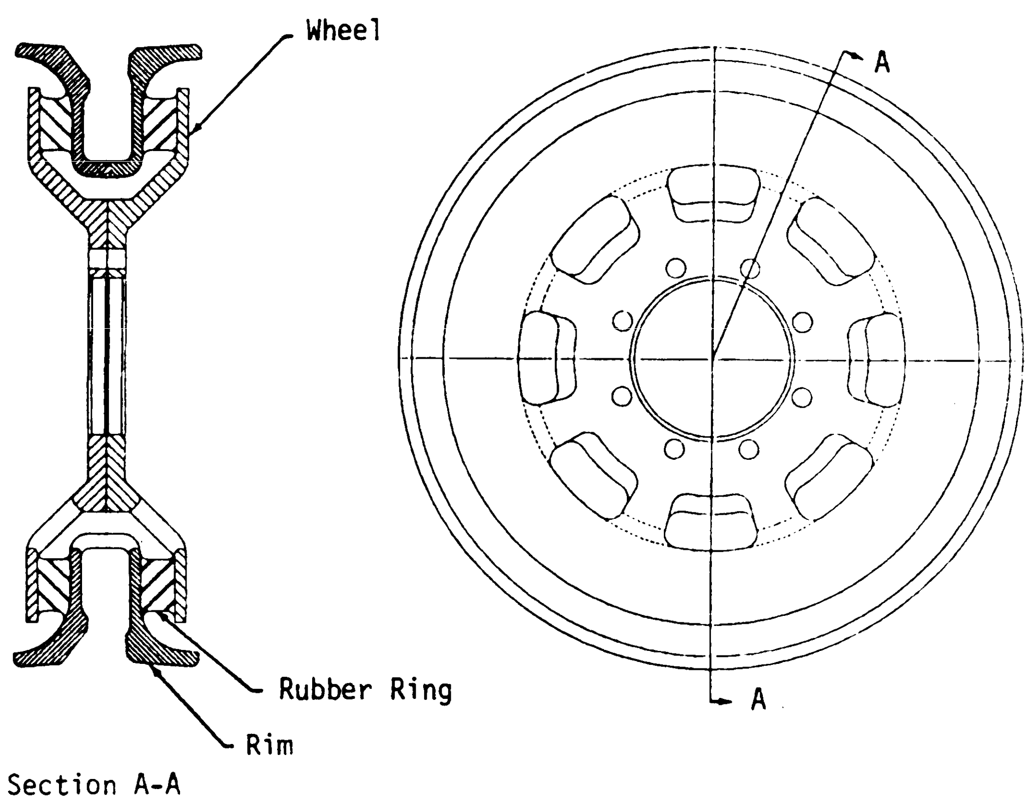

Figure 5-6. Prototype Isolated Idler Wheel Developed for Vehicles in the M113 Weight Class

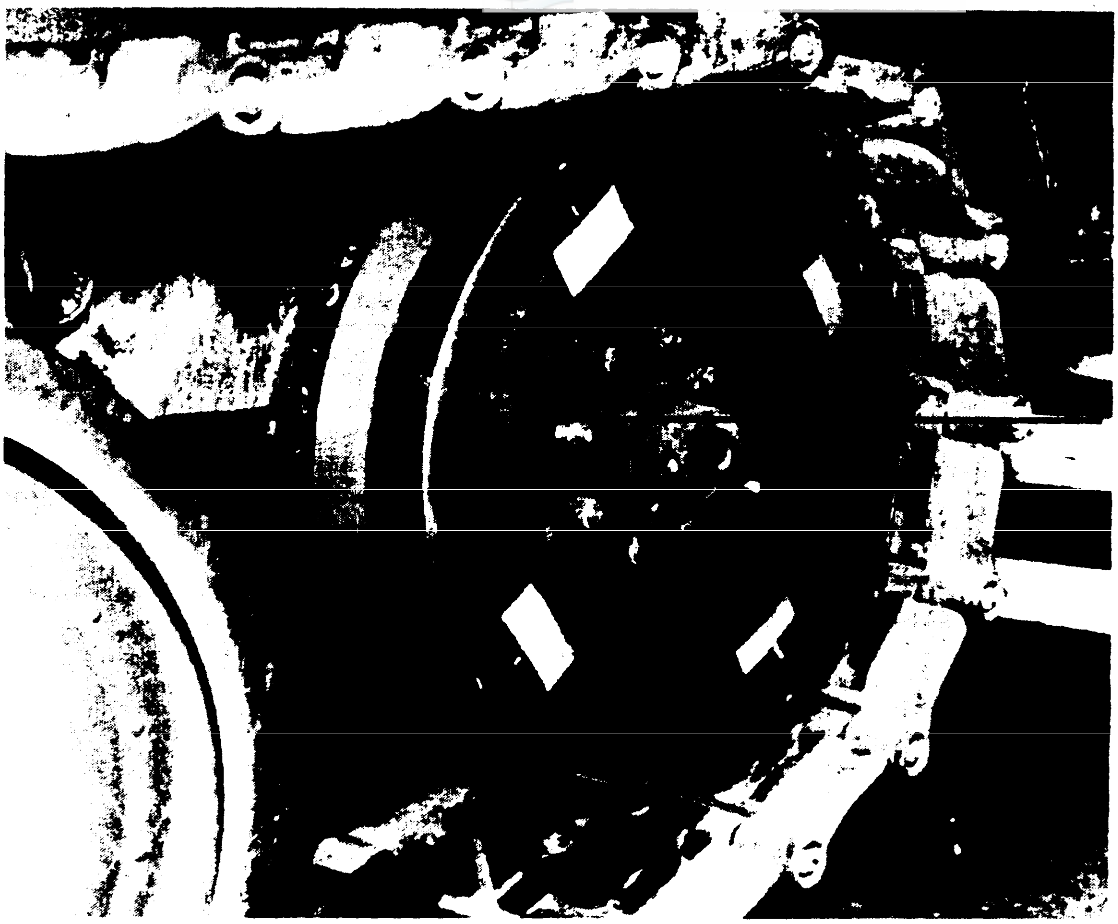

Figure 5-7. Quiet Idler Wheel Installed on an M113A1 Vehicle

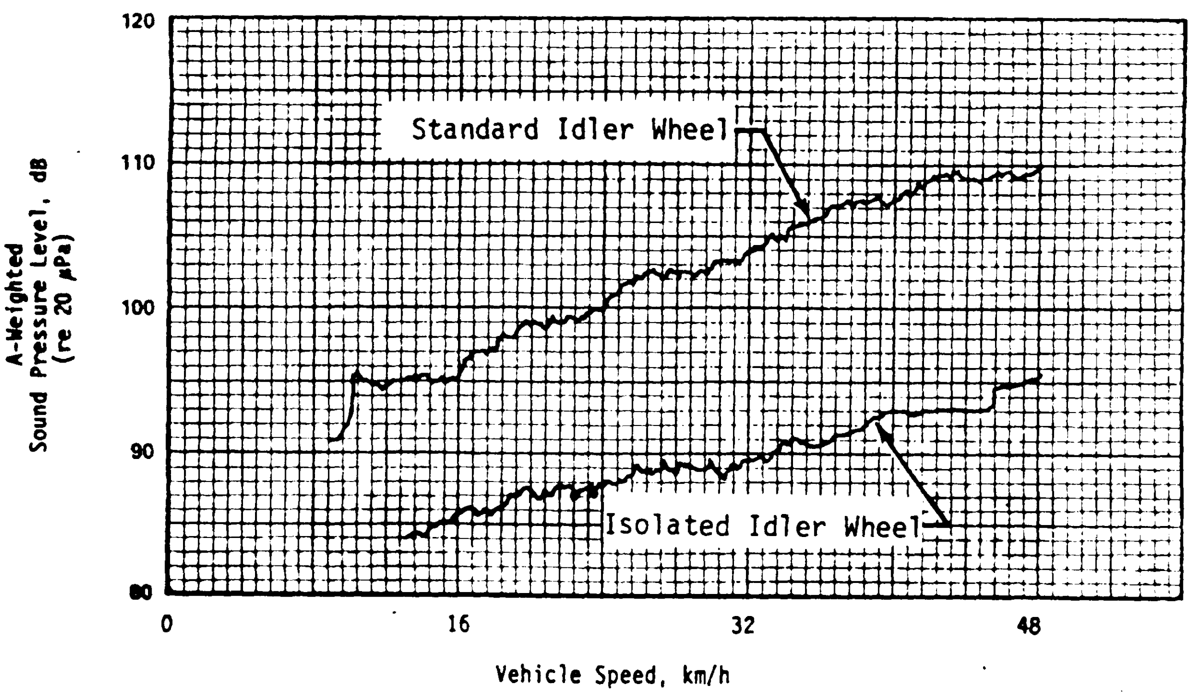

Figure 5-8. Interior Crew Area Noise Comparison of a Standard and Isolated Idler Wheel

This idler wheel meets all vehicle design requirements and has a predicted service life greater than 3220 km (2000 mi). The limiting Factor in durability is rubber fatigue. Because track tension applies a continuous static load in the same direction. the rubber rings experience a full stress reversal cycle for each revolution of the idler wheel. The peak value of this stress is 165 kPa (24 lb/in.^{2}) nominally. but it may go as high as 248 kPa (36 lb/in.^{2}) when the idler rim deflects m the limits of the overload stops. The prototype idler wheel used a low-loss natural rubber to minimize heat generation. The damping in the track is sufficient to control resonant vibration amplification.

The discussion that follows outlines the process used to design the prototype isolated idler wheel:

1. Noise *Reduction.* The required noise reduction for the prototype idler wheel was estimated at 18 dB.

2. *Radial Spring Rate.* The radial spring rate required to achieve this noise reduction was estimated at 1477 N/mm (8427 lb/in.)

3. *Radial Deflection.* The maximum radial deflection of 19 mm *(0.75* in.) was selected at 1.4 times the deflection due to nominal track tension. This fully isolated the vibration through nominal track tension fluctuations jet protected the rubber from excessive stress during high loads.

4. *Maximum Load.* The maximum load at full isolator deflection was calculated by multiplying the radial spring rate by the maximum deflection

$$1477 \text{ N/mm} \times 19 \text{ mm} = 28.063 \text{ N } (6309 \text{ lb}).$$

5. *Shear Area.* The shear area of the rubber isolators was calculated by dividing the maximum load by the maximum shear stress of 248,000 N/m^2 (36 lb/in.^{2})

$$28,063 \text{ N} \div 248,000 \text{ N/m}^2 = 0.113 \text{ m}^2 (175.4 \text{ in.}^2)$$

This maximum shear stress was selected to ensure long rubber fatigue life

6. *Rubber Ring Thickness.* The thickness of the rubber rings was selected at 31.5 mm (1.24 in.) to limit maximum shear strain to 0.6. (Keeping shear strain as low as possible limits edge stress concentrations. which increases rubber fatigue life.)

7. *Shear Modulas.* Rubber shear modulus S was calculated by (Ref. 9)

$$S = \frac{K_r t}{A}, \text{ N/cm}^2 \tag{5-5}$$

where

A = shear area of rubber. cm^2
K_r = radial spring rate of the Idler wheel. N/cm
S = shear modulus of the rubber. N/cm
t = thickness of the rubber ring. cm.

8. *Material Selection.* A rubber material having a hardness of 40 Durometer and a static shear modulus of 38 N/cm^2 (55 lb/in.^{2}) at a shear strain of 0.4 was selected.

9. *Ring Size.* Ring size was calculated by selecting an inner radius that fit the available space and solving for the outer radius R_o by

$$R_o = \sqrt{\frac{a}{\pi} + R_i^2}, \text{ cm} \tag{5-6}$$

where

a = shear area of one rubber ring. cm^2
R_i = inner radius of rubber ring. cm
R_o = outer radius of rubber ring. cm.

Because the prototype design had two rubber rings. the shear area used to calculate ring size was one-half of the total shear area.

10. *Fabrication and Subsequent Design Modification.* The rim and wheel halves of the prototype compliant idler wheel were machined from a high-strength steel alloy block. In production quantities the design is well-suited for casting or forging. The rubber rings. fabricated using a compression molding process. were attached to the rim and carrier with cyanoacrylate adhesive (Super Glue ™). A later design modification, shown in Fig. 5-9. bonded the rubber rings to thin. steel rings. This simplified bonding. improved bond strength (100% bond area). and made the rubber rings a replaceable item.

5-5 SPROCKET WHEEL DESIGN

5-5.1 SPROCKET WHEEL DESIGN APPROACH

The process for reducing sprocket-wheel-generated noise is essentially the same as that for the idler wheel. which was discussed in par. 5-4.1. Unlike the idler. however, a reduced-noise sprocket design must transmit high-torque loads in addition to reducing chordal action impact forces.

5-5.2 PREVIOUS SPROCKET WHEEL DESIGN EXPERIENCE

The discussions that follow briefly describe some previous designs for reducing sprocket-generated noise:

1. *Isolated Final Drive.* This design attempted to attenuate sprocket wheel vibration before it entered the hull b) providing vibration isolation mounts for the final drive (Ref. 10). However. when tested. total vehicle interior noise was reduced only 2 to 3 dB, leading to the conclusion that this approach was not very effective in reducing noise. More recent analysis of the data shows that actual sproket noise reduction may have been as great as 10 to 15 dB. but because of the other noise sources, the overall level was reduced only the 2 to 3 dB.

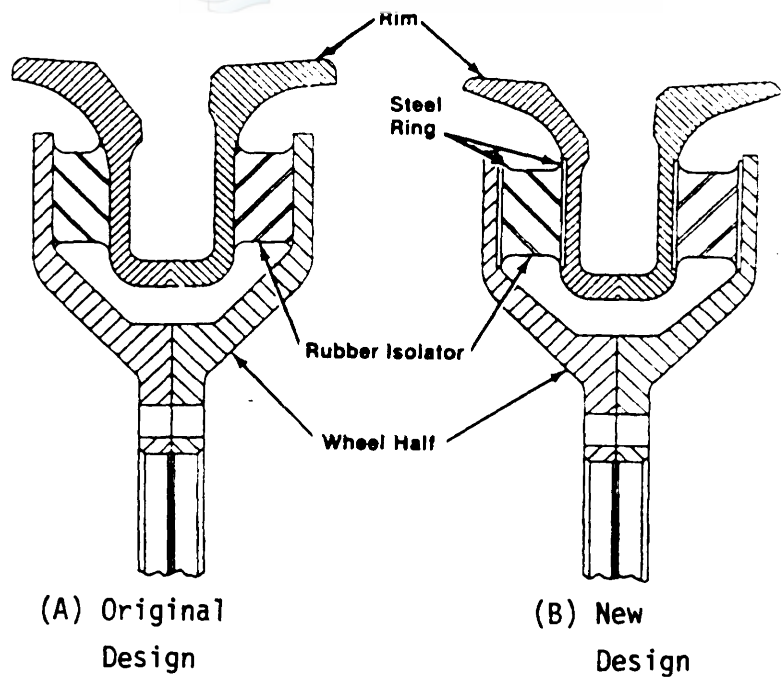

Figure 5-9. Isolator Design Modification for Prototype Quiet Idler Wheel

2. *Chordal Action Corrector Wheel.* This design sought to control the path of each track shoe as it approached the sprocket wheel by using a pendulous arm to attach a standard M113 road wheel to a standard M113A1 sprocket wheel as shown in Fig. 5-10 (Ref. 11). This lowered chordal action impact velocity and reduced interior noise 1 to 7 dB at speeds of 16 to 48 km/h (10 to 30 mi/h). However, the chordal action corrector wheel provided less track engagement than a standard sprocket wheel. which caused the track to slip during certain turning maneuvers and movement in reverse.

3. *Toothless Sprocket Wheel.* In this design the sprocket teeth were removed from the sprocket carrier of an Ml 13 vehicle. Friction between the sprocket cushion and inner rubber pad on the track shoes supplied the driving torque, Subsequent interior noise measurements with the toothless sprocket were identical to those of a standard sprocket, which showed that the sprocket teeth were not a major noise-generating mechanism inside the vehicle (Ref. 12).

5-5.3 SPROCKET WHEEL DESIGN PARAMETERS

Design parameters for a reduced-noise sprocket wheel are very similar to those for a quiet ‘idler wheel (par. 5-4.3). but the geometry is more restrictive. Wheel diameter is determined by track pitch and number of sprocket teeth. Enlarging the sprocket wheel diameter to decrease chordal action impact velocity also requires additional sprocket teeth unless the track pitch is changed. The added teeth require a change in gear ratio of the transmission or final drives to maintain proper sprocket torque and speed ratios.

Increasing the compliance at the track shoe/sprocket wheel interface may accelerate sprocket tooth wear and raise the exterior noise level because the teeth are continuously hitting the track shoes. When the track shoe interface is softened, any significant change in track tension changes the effective pitch diameter of the sprocket carrier so that it no longer matches the sprocket teeth pitch diameter. In this situation the sprocket teeth would be in constant contact with the track shoe (on either the forward or rear side of the tooth) and experience increased wear rates and greater exterior noise generation (clatter).

Sprocket wheel noise can be reduced by inserting a vibration isolator in the structural path from the sprocket to the hull. The isolator can be inserted between the sprocket wheel rim and hub (as for a compliant idler wheel). or the final drive can be vibration isolated from the hull. In either case Eq. 5-3 can estimate the required spring rate for the isolator as long as the mobility transfer functions are measured at the appropriate locations.

On compliant sprocket wheels, receiver mobility (MR) is measured at the sprocket wheel attachment to the final drive output spindle, whereas source mobility (MS) is measured at the track shoe. On isolated final drives MR is measured on the hull at the final drive attachment points, and MS is measured on the final drive at the points of attachment to the hull.

When making mobility measurements, the designer should measure the structures in their free states (unattached

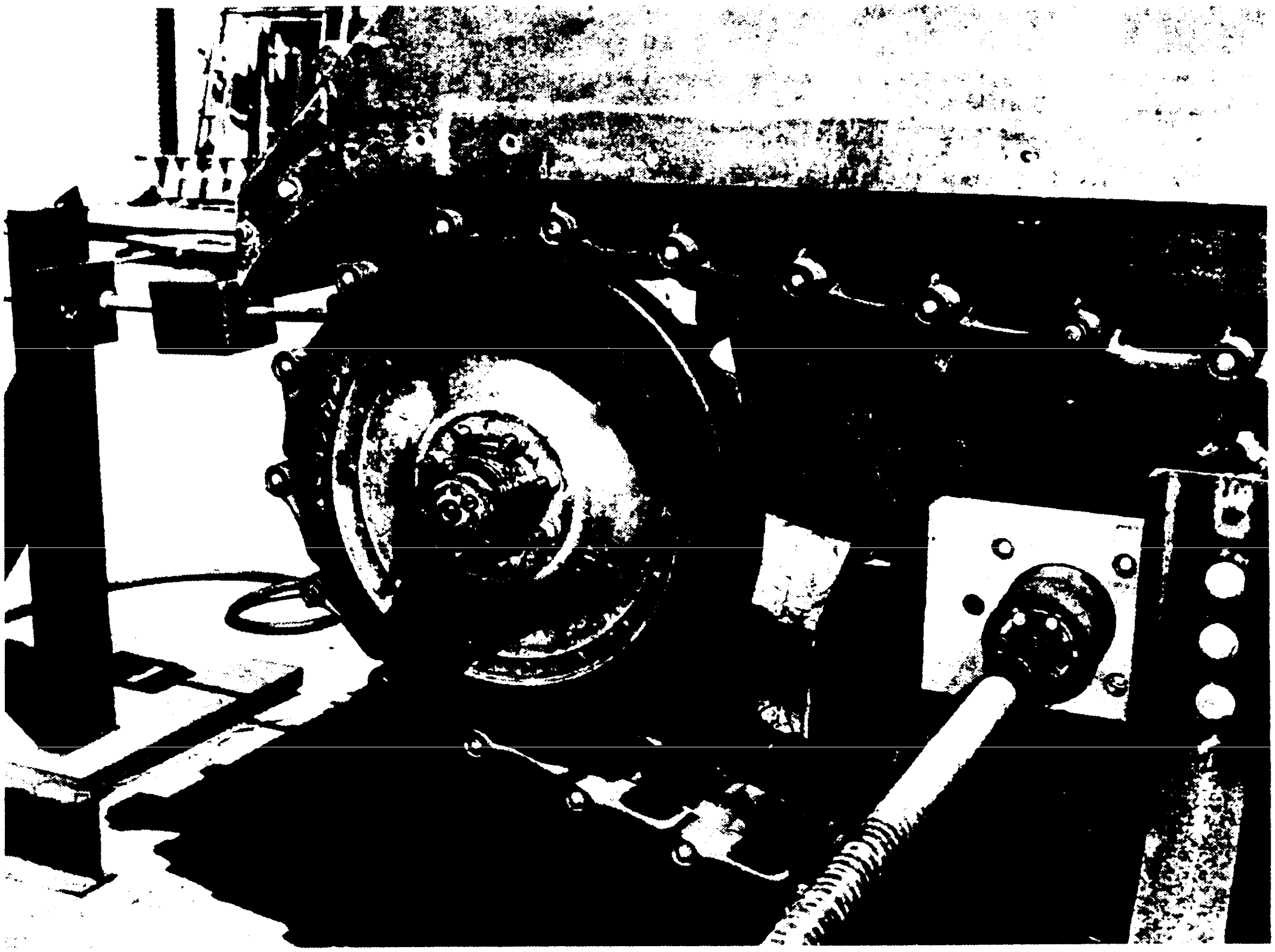

Figure 5-10. Chordal Action Corrector Wheel

to other structures) or, alternatively, estimate the required spring rates using Eq. 5-4. The 4.5-dB constant has been verified for an isolator between the sprocket wheel rim and hub; however, for an isolated final drive the noise reduction per halving of stiffness maybe slightly different.

Sprocket wheels designed to reduce noise must also withstand high torsional loads from severe vehicle maneuvers. In the M113A1 vehicle the highest torque loads occur during panic stops, which have been measured as high as 15,321 N•m (11,300 ft-lb) (Ref. 13). Soft elastomeric isolators cannot withstand such high torque loads and must be protected from them, as well as from the high radial loads caused by excessive track tension.

Other design requirements for an isolated sprocket wheel or isolated final drive (besides noise reduction) are the same as those listed for an isolated idler wheel in par. 5-4.3.

5-5.4 M113 PROTOTYPE QUIET SPROCKET WHEEL

A prototype sprocket wheel. based on the successful prototype idler wheel design. was fabricated and tested using rubber-in-shear compliant elements with torsional overload stops. Fig. 5-11 shows the prototype sprocket wheel designed for use on the M113A1 vehicle.

The discussion that follows outlines the process used to design the prototype compliant sprocket wheel:

1. Noise *Reduction.* The required noise reduction for the prototype sprocket wheel was estimated at 17 dB.

2. *Radial Deflection,* Maximum radial deflection of 16 mm (0.625 in.) was selected to fit the available space between the sprocket rim and final drive output flange.

3. *Radial Spring Rate.* The estimated radial spring rate for achieving the required noise reduction was 766.2 N/mm (4375 lb/in.). Approximately 30 mm (1.2 in.) of radial movement would be required between the hub and isolated sprocket rim to allow normal operation with a track load of 15,569 N (3500 lb), The maximum radial clearance available was only 15.9 mm (0.625 in.). To operate successfully within this small clearance, the design spring was increased to 1926.4 N/mm (11.000 lb/in.), which would not meet the noise reduction goal. However, it would reduce noise approximately 11 dB without requiring major power train

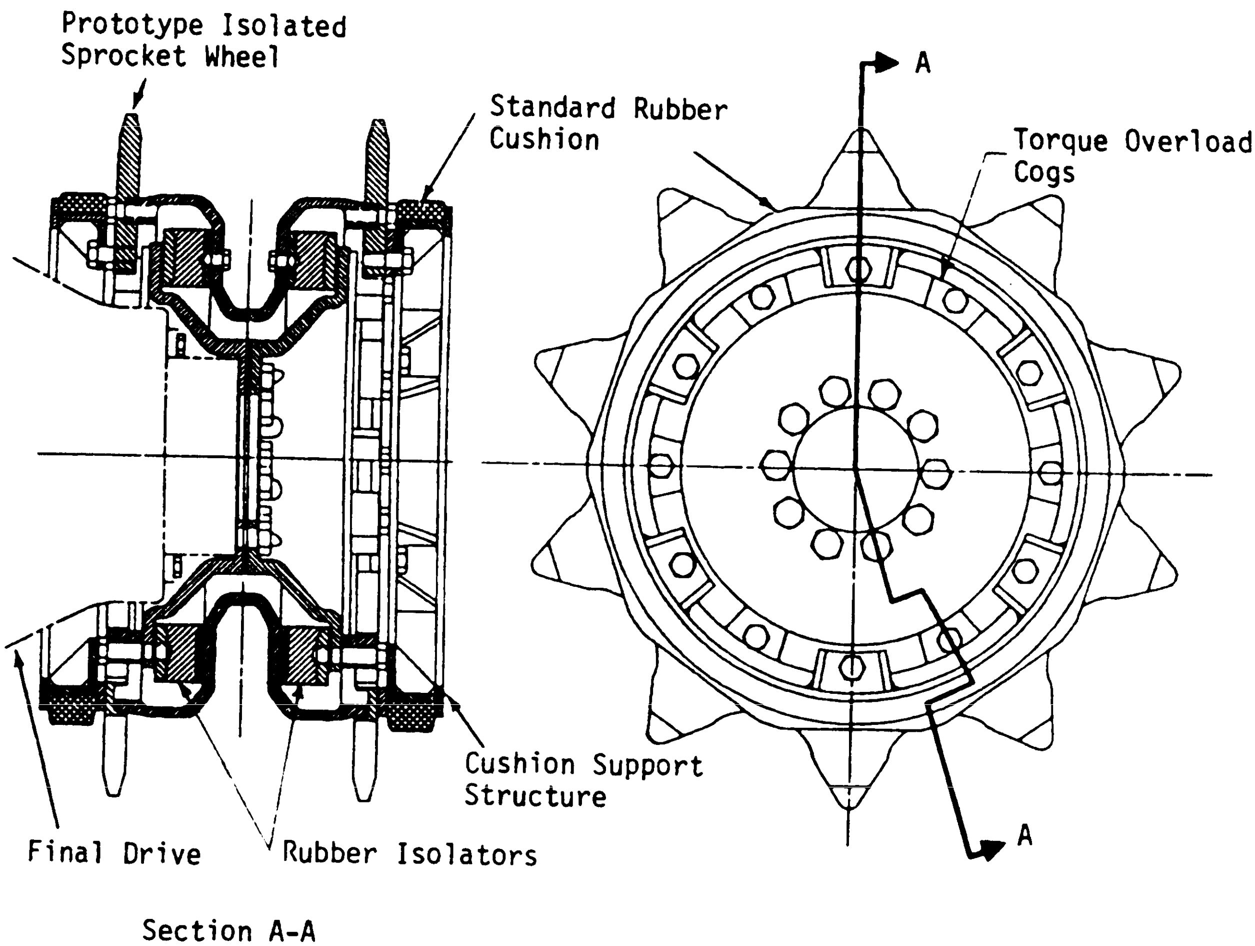

Figure 5-11. Prototype Isolated Sprocket Wheel Developed for M113 Weight Vehicles

changes to accommodate a larger diameter sprocket wheel. It was thereford selected as the best compromise sprocket isolator.

4. *Maximum Loud.* Maximum load at full isolator deflection was calculated by multiplying the radial spring rate by the maximum deflection:

1926.4 N/mm x 15.9 mm = 30.630 N (6886 lb).

5. *Shear Area.* The shear area of the rubber isolators was calculated by dividing the maximum load by the maximum shear stress of 0.303 N/mm ^{2}(44lb/in. ^{2}):

30.630 N ÷ 0.303 N/mm ^{2}

= 101,089 mm^2(156.7 in.^{2}).

Maximum shear stress was selected as low as possible to ensure long fatigue life of the rubber

6. *Rubber Ring Thickness.* The thickness of the rubber rings was selected at 25.4 mm (1.0in.) to limit maximum shear strain to 0.6.

7. *Shear Modulus* By using Eq 5-5. the rubber shear modulus is

1926.4 N/mm x 25.4 mm ÷ 101.089 mm^2

= 0.48 N/mm^2(70 lb/in.^{2}).

8. *Material Selection. A* rubber material having a hardness of 45 Durometer was selected to provide the required shear modulus at a shear strain of 0.4.

9. *Ring Size.* The size of the rings was calculated by selecting an inner radius. which fits in the available space, then using Eq. 5-6 to find the outer radius.

10. *Torsional Spring Rate.* Torsional spring rate K_T is calculated by (Ref. 9)

$$K_T = 1.5699 \frac{S}{t} (R_o^4 - R_i^4), \text{ N·cm/rad} \qquad (5\text{-}7)$$

where

K, = torsional spring rate. N cm/rad
1.5699 = torsional constant. rad^{-1}

$$K_T = [(1.5699 \times 48) \div 2.54] \times [(18.94)^4 - (13.974)^4] = 2{,}686{,}419 \text{ N·cm/rad.}$$

Note: If the torsional spring rate is not high enough to cam nominal torque loads without excessive deflection. the designer should increase the inner and outer ring radii while maintaining the required shear area and geometry constraints. To protect against large torque loads, such as occur during hard stops. the design must also include overload protection to limit the maximum tangential deflection of the rubber to a shear strain of approximately 0.8. The prototype sprocket wheel used external steel cogs that carried high-torque loads exceeding the tangential deflection of 0.105 rad (6 deg).

11. *Fabrication.* The carrier rim and decagonal cushion supports were fabricated from low -carbon steel: the sprochet and overload stops from high strength alloy steel. Compliant elements were molded from a high-grade. low-loss. natural-base rubber compound and vulcanize bonded to steel rings to create a replaceable isolator. Fig. 5-12 shows a cross section of the rubber rings. A large radius on the edges of the steel rings reduced edge stress concentrations, which would have degraded rubber fatigue life.

12. *Test Results At speeds* from 8 to 48.3 km/h (5 [o 30 mi/h. tests showed an average A-weighted reduction of noise in crew area of 10 dB.

5-6 ROAD WHEEL DESIGN

5-6.1 ROAD WHHEEL DESIGN APPROACH

To reduce road-generated noise, the designer must either reduce rolling wheel action or attenuate the vibration caused by motion , e.g., by increasing road wheel or road arm compliance, before it enters the hull

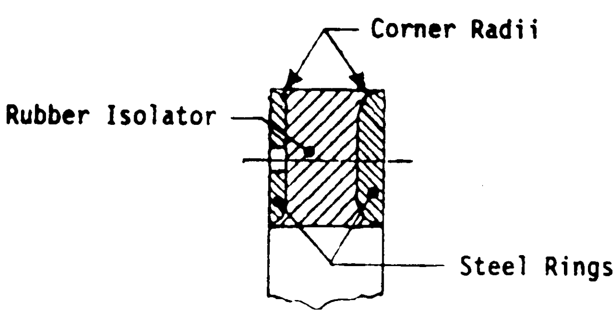

Figure 5-12. M1113 Prototype Quiet Sprocket Wheel Isolator Design

5-6.2 PREVIOUS ROAD WHEEL DESIGN EXPERIENCE

The few attempts at reducing road-wheel-genera[ed noise have not been well-documented due to the kick of a test method that could prevent masking of road wheel noise by other sources, such as idler. sprocket, and engine. A discussion of one of these attempts follows:

Damped Road Wheel. To reduce the amplitude of vibration at resonance, this design added a cure-in-place damping material to a standard M113 road wheel hub (Ref. 1). Shaker test results shoved a decrease in noise-to-force transfer function at 250 Hz. which indicates a potential for noise reduction. However. because road wheel noise is not dominated by noise at 250 Hz, any reduction due to added road wheel damping Would likely be no more than 1 or 2 dB.

5-6.3 ROAD WHEEL DESIGN PARAMETERS

There are two ways to reduce road-wheel-generated noise: increase the compliance of the road wheel tire or wheel or add vibration isolators to the road arm attachment locations. In either approach the designer can use Eq. 5-3 to predict the necessary static spring rates. Mobility transfer functions are measured (or analyzed at the following locations:

1. *Compliant Road Wheel* Source mobility is measured at the track shoe: receiver mobility is measured at the rim of the road wheel (metal structure. not road wheel tire).
2. *Isolated Road Arm.* MS is meusured at the road arm trunnion (with the road wheel attached to the road arm and the road arm not attached to the hull): MR IS measured at the road arm attachment location on the hull.

In general, for the same noise reduction. compliant road wheels require lower spring rates than isolated road arms. Compliant road wheels can use very soft tires or rubber shear rings similar to those described in pars. 5-4.4 and 5-5.4.

Reduced-noise road wheels must also conform to such design requirements as

1. Retaining the track
2. Withstanding maximum road wheel loads (large bumps)
3. Providing fail-safe “get home”” capabilities
4. Offering a minimum durability of 3220 km (2000 mi)
5. Limiting weight gain
6. Withstanding tracked vehicle environment (temperature extremes. water. rocks. mud. and debris).

5-6.4 M1113 PROTOTYPE QUIET ROAD WHEEL

Because of the harsh environmental requirements at the road wheels. the decision was made for the M113 noise reduction program to develop an isolated rod arm design in which the isolator would be in a protected location inside the hull. Unknown durability for a soft tire road wheel was also a strong factor in the selection of an isolated road arm rather than a compliant road wheel tire. Fig. 5-13 illustrates the isolated road arm design developed to reduce road wheel noise for the M113A1 vehicle. The vibration isolators, positioned between the road arms and the hull, consisted of a series of rubber pads loaded in a shear configuration.

The discussion that follows outlines the design process used to develop the prototype isolated road arms:

1. *Noise Reduction.* The required noise reduction for the isolated road arms was estimated at 12 dB.

2. *Vertical Spring Rate.* The vertical spring rate required to achieve this noise reduction was estimated at 1751.3 N/mm (10,000 lb/in.).

3. *Isolator Deflection.* To protect the rubber from excessive stresses during high road wheel load conditions. overload capability was designed to limit maximum isolator deflection to 25.4 mm (1.0 in.).

4. *Shear Area. The* rubber shear area was calculated from Eq. 5-5 by using a rubber thickness of 25.4 mm (1.0 in.) and a shear modulus of 42.8 N/cm^2 (62 lb/in.^{2}):

$$17{,}513 \times 2.54 \div 42.8 = 1039 \text{ cm}^2 \text{ (161 in.}^2\text{)}.$$

5. *Geometrical Constraints.* To fit inside the box beam of the M113A1 hull, the isolators were limited to a maximum size of 95 x 140 mm (3.75 x 5.5 in.). Consequently, eight pads were used to provide the total area,

6. *Test Results,* When a single isolated road arm was tested using a laboratory shaker, it showed a significant noise reduction, approximate] y 11 dB. However, when the vehicle was tested with a complete set of isolated road arms. total vehicle noise reduction was less than anticipated due to the hull modifications that had been required to accept the isolators and due to resonance of the isolated road arms at slower speeds (Ref. 14). These modifications increased the hull resonance and resultant noise,

Better road wheel noise reduction may have been achieved through isolation at the road wheel. It was concluded that a compliant road wheel design (similar to the compliant idler) might have produced greater noise reduction. However, developing a robust compliant road wheel is a formidable design challange.

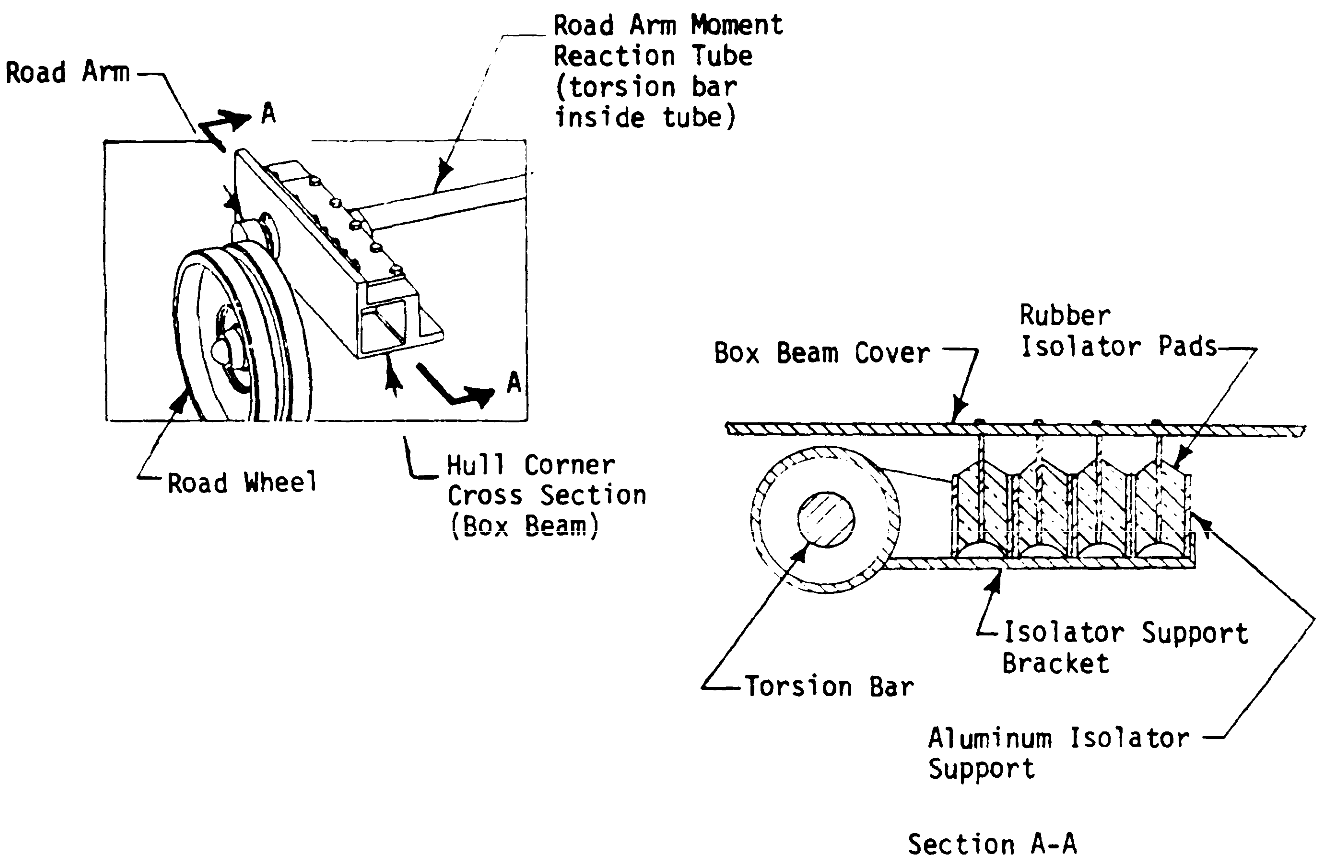

Figure 5-13. Isolated Road Arm Design for an Ml 13 Weight Vehicle

5-7 TRACK SUPPORT ROLLER DESIGN

The processor designing a reduced-noise track support roller is very similar to that for a reduced-noise road wheel because the noise-generation mechanisms and methods of increasing compliance are the same. If anything. it is somewhat easier to design a rugged, reliable, and compliant system for the support rollers because the static and dynamic loads are lower.

Eq. 5-3 is used to determine the isolator spring rate needed to reduce support roller noise to the required limit. Vibration isolators can be designed to add compliance between the rim and hub of the support roller or to isolate the support roller and its support structure from the hull. To develop a compliant wheel, the designer must determine mobility transfer functions at the track (source) and at the support roller wheel rim (receiver). For an isolated support roller these functions must be determined at the support attachment flange (source) and support attachment location (receiver). An advantage of the isolated support roller over the compliant wheel is that the isolated roller is not subjected to full reverse shear loads; thus isolator fatigue life is improved.

Reduced-noise support rollers must also meet design requirements such as

1. Providing proper track support and track retention
2. Withstanding maximum track loads (track throw and large bumps)
3. Providing fail-safe capabilities
4. Providing a minimum durability of 3220 km (2000 mi)
5, Limiting weight gain
6. Withstanding tracked vehicle environment. e.g., temperature extremes, water and mud, and debris ingestion.

5-8 OTHER SUSPENSION COMPONENTS

The preceding paragraphs in this chapter discussed the major components of a traditional suspension system for a tracked vehicle. A discussion of three additional suspension components—torsion bars, shock absorbers, and hydropneumatic struts-follows:

1. *Torsion Bars.* Torsion bars provide a high-energy spring system. which allows large-wheel travel in a tracked vehicle. When vibration from the road wheels is transferred to the vehicle hull through the road arm mounting location, a small proportion of this energy is also transferred through the torsion bar, which is a parallel path for vibration transmission. However, if the road arm is vibration isolated from the hull. the torsion bar then becomes a significant structurebome vibration path and must be isolated (Ref. 15). To block this path, both ends of the torsion bar must be isolated from the hull.

2. *Shock Absorbers.* Shock absorbers are attached between the road arm and hull to damp the suspension system and prevent large amplitude “bounces” of the vehicle. Test results of vibration input directly to an M113A1 shock absorber showed it was not a significant path for transmitting energy at acoustic frequencies to the hull (Ref. 16). Removal of the shock absorbers would not result in any significant road-arm-generated noise reduction. ,

3. *Hydropneumatic Struts.* Hydropneumatic struts combine the functions of the road arm, torsion bar, and shock absorber. The strut transfers vibration energy from the road wheels to the hull in the same way as a road arm. Road wheel noise-reduction designs for a vehicle with hydropneumatic suspension would be very similar to those for the torsion-bar suspension systems described in par. 5-4.3.

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CHAPTER 6
REDUCTION OF NOISE FROM POWER TRAIN COMPONENTS

The design approach used to reduce power-train-generated noise in tracked vehicles is discussed. Design guidelines are presented for power train component vibration isolation systems and noise enclosures.

6-0 LIST OF SYMBOLS

d = air gap between walls, cm
f = frequency at which transmission loss is calculated, Hz
f_d = double-wall resonant frequency, Hz
f_n = vertical linear resonant frequency, Hz
f_1 = lower plateau frequency. Hz
f_2 = upper plateau frequency, Hz
g = gravitational constant = 980 cm/s^2 (386 in./s^2)
k = combined spring rate of isolators, N/cm (lb/in.)
P_h = plateau height, dB
TL = transmission loss. dB
t = panel thickness, cm
w = weight of engine, N (lb)
w_1 = surface weight of first wall, g/cm^2
w_2 = surface weight of second wall, gdcm^2
δ = static deflection, cm (in.)
ρ = panel density, g/cm^3

6-1 DESIGN APPROACH

Noise from the power train components, i.e., engine. transmission, and associated equipment, is reduced using either of two general methods: source reduction or propagation path reduction. Source reduction is achieved by altering the power train components to produce less noise and vibration. Blocking power train noise from entering the crew areas is an example of propagation path reduction. For either of these noise reduction methods. knowing the relative contribution of airborne and structure-borne noise is critical because the design approach for reducing airborne noise is much different than for reducing structure-borne noise.

As discussed in par. 3-5, noise reduction goals for power train components are determined from the overall vehicle noise goals and the number of major noise sources. In general. noise from power train components is less than suspension-generated noise. Hence noise reduction goals are usually lower.

In addition to achieving the required noise reduction, designs must be durable and safe. The life expectancy of drivetrain noise reduction components should be at least as long as the life of the vehicle between rebuilds. Temperatures in the engine compartment can exceed 121°C (250°F). Such high temperatures have an adverse affect on various materials such as rubber and acoustic foam. Fuel and lubricating oil are likely to be spilled and splashed in the engine compartment, and materials that would absorb these flammable liquids should be avoided to prevent fire hazards. Drivetrain noise reduction designs must incorporate fail-safe features to prevent loss of motive power in case of a component failure.

6-1.1 SOURCE MODIFICATION

Source modification is the most direct method of power train noise reduction and should be used whenever possible. Engine modifications to decrease noise may include techniques such as using highly damped sheet metal in the fabrcation of the oil pan and valve covers, inherently quieter engine block and crankcase, smoother running internal engine components. and noise barriers to block sound from various engine surfaces (Ref. 1). Designing a reduced-noise transmission may require a stiffer housing, high-precision gears, or use of special low-noise gears, such as helical or hypoid. The design details of inherently quiet engines or transmissions are highly dependent on their size and type and are beyond the scope of this design handbook. Engine and transmission noise reduction programs are best done in conjunction with the engine and transmission manufacturers. Items that are mounted on the power train components, particularly the engine, i.e., brackets and auxiliary equipment, should be designed for maximum stiffness and have small surface areas to reduce their radiation efficiency.

6-1.2 PATH MODIFICATION

Preventing propagation of power-train-generated noise into the crew areas is accomplished by blocking both the airborne and structure-borne noise as mentioned in par. 4-3.1. Vibration isolation mounts installed between sources of vibration, such as between the engine or transmission and the hull, attenuate structure-borne noise, whereas the engine compartment panels attenuate airborne noise.

6-2 ENGINE NOISE ATTENUATION

6-2.1 STRUCTURE-BORNE NOISE

The mounting configuration of the engine determines the amount of structure-borne noise entering the vehicle interior. Compliant engine mounts should be placed at the stiffest hull locations possible. If an existing or similar vehicle is available, point impedance measurements, as described in par. 8-2.3, can be used to determine the best locations for the isolation mounts. Computer modeling of a new or existing hull structure can also be used to determine optimum locations for engine isolation mounts,

The spring rate selected for compliant engine mounts should be as low as possible in order to obtain maximum vibration isolation. The allowable static or dynamic deflection is the determining factor in selecting engine mounts.

The static deflection 8 is determined as

$$\delta = \frac{w}{k}, \text{ cm (in.)} \qquad (6\text{-}1)$$

where

δ = static deflection, cm (in.)
w = weight of engine, N (lb)
k = combined spring rate of isolators, N/cm (lb/in.).

Resonant frequency of the vertical bounce mode of the engine is

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}, \text{ Hz} \qquad (6\text{-}2)$$

where

f_n = vertical linear resonant frequency, Hz
g = gravitational constant = 980 cm/s^2 (386 in./s^2).

The whole-body rotational natural frequencies will. in general, be lower than the vertical vibrational mode. Snubbers and/or stabilizer mounts may be required to limit engine movement and prevent damage to the isolators or equipment mounted near the engine when large shock loads are imposed on the isolated engine. e.g.. vehicle experiences a severe bump.

With soft isolators the whole-body natural frequencies of the engine-engine bounce and roll—are in the range of 5 to $5 Hz. These natural frequencies are excited for only a very short time as the engine is starting or stopping and should not require large amounts of external damping to limit resonant vibration amplification. In general. a low-noise engine mount system has a high degree of flexibility and allows considerable motion of the engine to occur. To prevent flanking. i.e., vibration entering the hull through a path other than the isolation mounts, flexible hoses and pipes must be used for all connections to the engine. Engine exhaust noise is not usually a problem in the vehicle crew areas. However, if the muffler is mounted on a thin panel, it may be necessary to isolate the vibration of the muffler to reduce its structure-borne noise. Likewise, if the engine cooling fan is an interior noise source, it should be attenuated with vibration isolators or noise barriers, depending on whether the dominant path is airborne or structure-borne.

The parameters that must be considered in the design or selection of engine isolation mounts are

1. The amount of noise reduction required
2. The structural stiffness of the hull at the mounting locations
3. The operating requirements for the vehicle, i.e., temperature. load, duty cycle. life expectancy. etc.
4. The allowable motion of the engine within the engine compartment.

6-2.2 AIRBORNE PATH

In general, the hull armor of the vehicle. which usually surrounds the engine on all but two sides. has a significantly higher transmission loss than the thinner, lighter bulkhead panels. The airborne path is not significant with the hull armor plates as long as there are no openings into the crew area through the hull plates. The bulkhead panels (including access covers) must be designed to have sufficient stiffness. damping, and attenuation to provide the required noise reduction.

In general. noise attenuation or transmission loss of an enclosure wall is proportional to its surface weight (material weight density divided by thickness). For example, a 1.6-mm thick steel panel will attenuate a 500-Hz sound 9 dB more than an aluminum panel of the same thickness. This relationship is nonlinear with respect to frequency. The frequency at which the wavelength of sound in the panel equals the wavelength in air is known as the critical coincidence frequency (Ref. 2). At this frequency the transmission loss may become very small depending on the internal damping of the panel material. Additional panel damping will be very beneficial in increasing the transmission loss in the critical frequency range. Constrained layer damping that consists of a layer of viscoelastic material sandwiched between two layers of barrier material. such as aluminum or steel. produces a very effective increase in panel damping. Another method of increasing panel damping is to spot weld two metal sheets together at regular intervals to produce a composite panel, which does not resonate well.

Transmission loss of an engine compartment panel can be estimated using the generalized loss curve in Fig. 6-1.

Section cutoff frequencies f_1 and f_2 *are* determined as

$$f_1 = \exp\left(\frac{P_h + 33}{20}\right), \text{ Hz} \qquad (6\text{-}3)$$

$$f_2 = 8f_1, \text{ Hz} \qquad (6\text{-}4)$$

where

f_1 = lower plateau frequency, Hz
f_2 = upper plateau frequency, Hz
P_h = plateau height, dB (from Table 6-1).

Transmission loss *TL* values are calculated separately for each of the three sections shown in Fig. 6-1 as follows:

1. Between 50 Hz and f_1

$$TL = 20\log(f\rho t) - 26.8, \text{ dB} \qquad (6\text{-}5)$$

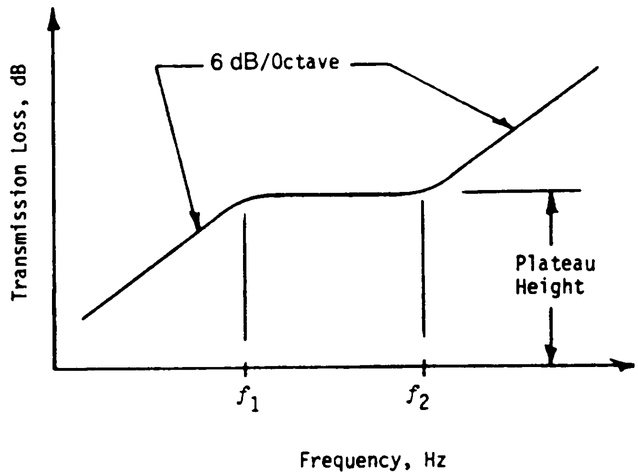

Figure 6-1. Generalized Airborne Noise Transmission Loss Curve Used for Transmission Loss Estimate (Ref. 2)

where

TL = transmission loss, dB
f = frequency at which transmission loss is calculated, Hz
ρ = panel density, g/cm^3
t = panel thickness, cm.

2. From f_1 to f_2

$$TL = P_h, \text{ dB} \tag{6-6}$$

3. Above f_2

$$TL = 20\log\left(\frac{f}{f_2}\right) + P_h, \text{ dB.} \tag{6-7}$$

An effective method of increasing the transmission loss of an enclosure panel is to change a single-wall panel into a double wall. A double wall is an acoustical barrier constructed of two panels separated by an air space (Ref. 2). The distance between the two panels is critical for two reasons: (1) the double-wall resonant frequency is determined by the air gap and (2) the additional transmission loss of a double wall versus a single wall increases with wider air spaces. The double-wall resonance occurs when two two walls respond as two masses coupled by a spring (Air is compressed and rebounds.). Double-wall resonance f_d is determined by

$$f_d = \frac{190}{\sqrt{d\left(\frac{w_1 w_2}{w_1 + w_2}\right)}}, \text{ Hz} \tag{6-8}$$

where

f_d = double-wall resonant frequency, Hz
d = air gap between walls, cm
w_1 = surface weight of first wall, g/cm^2
w_2 = surface weight of second wall, g/cm^2.

At this resonance the transmission loss will be less than provided by a single wall and may provide very little noise attenuation. Above the double-wall resonance the transmission loss is approximately equal to the transmission loss of a single wall of equal weight plus extra attenuation that is frequency- and air-space-dependent, as shown in Fig. 6-2.

The transmission loss of a double wall with an air space greater than 305 mm (12 in.) will be approximately equal to the decibel sum of the transmission loss of each wall individually. Sound-absorbing material, such as fiberglass batting, placed between the two walls increases the transmission loss of a double-wall panel.

Openings between the crew area and the engine compartment degrade the airborne attenuation of the engine compartment enclosure. Generally, sound will propagate wherever air can pass. Suitable gaskets, such as rubber strips or closed-cell foam, should be used on engine compartment access doors to prevent noise leakage.

6-3 TRANSMISSION NOISE REDUCTION

Replacement of a traditional mechanical or hydrostatic transmission with an electric generator and motor provides the greatest transmission noise reduction. Electric drive sys-

TABLE 6-1. SURFACE DENSITY AND PLATEAU HEIGHT VALUES FOR VARIOUS MATERIALS (Ref. 2)

MATERIAL TYPE	DENSITY ρ, kg/m^3	PLATEAU HEIGHT P_h, dB
Aluminum	2720	29
Glass	2500	27
Lead	11,210	56
Steel	7690	40

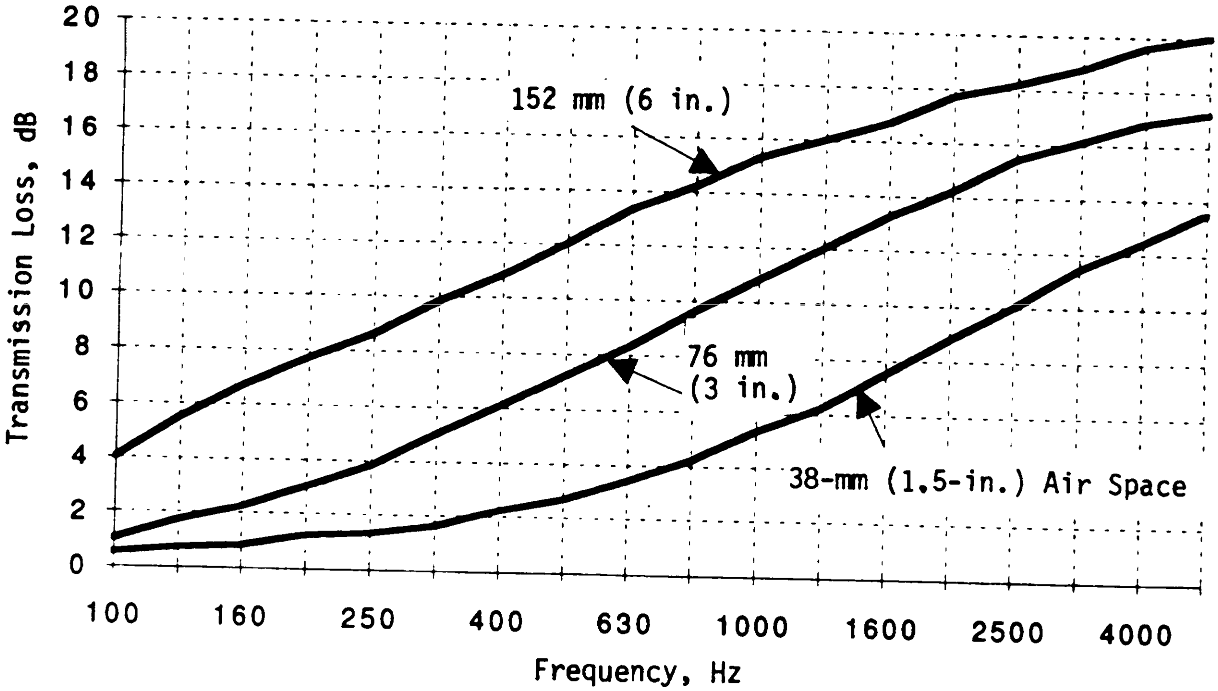

Figure 6-2. Increase in Transmission Loss of a Double Wall Above a Single Wall Due to Air Space (Ref. 2)

terns are inherently smooth running and quiet because the entire transmission consists of an electric motor and a planetary gear set. Both of these components can be designed to generate very little noise. Additionally, electric drive transmissions can be located where vibration isolation and noise attenuation will be most effective.

For tracked vehicles with conventional transmissions, noise reduction is best accomplished by designing the housing to be as stiff as possible and the gear train to be as smooth running as possible. High-precision spur gears are quieter than common spur gears. Helical, herringbone, and hypoid gears provide the quietest operation for gearbox designs. The disadvantages to these quieter gears are their much higher costs and greater design complexity.

Vibration isolation is an effective method for reducing structure-borne noise from the transmission. The frequencies to be isolated typically are somewhat higher than the frequencies associated with the engine. Therefore, transmission isolation mounts can be stiffer than engine isolation mounts and still provide significant noise reduction. However, the mount must be sufficiently flexible so that all the natural frequencies of the mount are much lower than the gear vibration frequencies. Many tracked vehicle power train designs have the engine and transmission connected together. In this configuration, if vibration isolation mounts are selected to provide the required engine isolation, the transmission will be adequately isolated also. Selection of isolation mounts for the transmission is very similar to engine mount isolator design discussed in par. 6-2.

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CHAPTER 7
HULL DYNAMICS AND ACOUSTIC COUPLING CONSIDERATIONS FOR REDUCING NOISE

Guidelines for computer modeling of hull structures and hull modifications for reduced noise are presented.

7-0 LIST OF SYMBOLS

C_b = speed of maximum frequency bending wave in plate, cm/s
c = speed of sound in air = 344 m/s
E = plate modulus of elasticity, kg/cm ^{2}
f = maximum frequency required, Hz
f_c = plate critical (coincident) frequency, Hz
f_1 = first breakpoint frequency for radiation efficiency calculation. Hz
f_2 = second breakpoint frequency for radiation efficiency calculation, Hz
g = acceleration due to gravity, cm/s^2
H = product of surface density and critical frequency, Hzg/cm ^{2}
P = perimeter of plate, m
S = surface area of plate, m^2
t = plate thickness, cm
w = surface density, g/cm ^{2}
ΔL = maximum distance between node points, cm
λ_c = wavelength in plate or air at critical frequency, m
ρ = plate material density, kg/cm ^{3}
σ_c = radiation ratio of simply supported plate at critical frequency, dimensionless
σ_{lo} = radiation efficiency of simply supported plate at low frequencies, dimensionless

7-1 NOISE PREDICTION AND HULL MODELING

The hull structure is the primary radiator of acoustic energy into the vehicle interior. By understanding hull response to the various structure-borne noise sources, the hull structure can be modified to accept and radiate less noise and thereby reduce interir noise levels. Any noise reduction achieved through hull modifications makes it easier to achieve the noise limits of MIL-STD-1474 (Ref. 1), because less reduction is required from suspension or power train modifications.

The ability to predict interior noise analytically while a vehicle is in the concept exploration phase is critical to the ability to make significant noise reductions through hull design. Studies have shown that if small changes, which can realistically be accomplished, are made to an existing hull structure, they will have a small impact (2- to 3-dB reduction) on the interior noise levels (Ref. 2). The costs of fabricating and testing even one hull configuration would easily pay for many analytical iterations, which would greatly enhance the chances of creating an optimum, reduced-noise hull structure.

As discussed in par. 5-2, predictions of tracked vehicle interor noise can be made using available computer tools. Finite element analysis (FEA) is the analytical tool used to predict hull response due to dynamic force input. Using FEA a structure is divided into a number of small parts or elements each having its share of the mass, stiffness, and damping of the structure. The elements are connected at locations known as nodal points, or nodes. Fig. 7-1 illustrates a finite element model of the left side of an M113 hull structure. Forces applied to the structure at nodal points cause the individual elements and adjoining elements to deflect according to the laws of motion. Matrix expressions are used to keep track of the motion of each element and to determine when equilibrium force and displacement conditions exist.

Structural transfer functions are calculated as the ratio of nodal displacement to dynamic force input. Extending the transfer function analysis to all nodes of the structure yields an analysis of the modes of vibration or normal modes analysis. To predict interior noise for a particular hull configuration. the modal response properties of the structure are combined with a prediction of hull plate radiation efficiencies and predicted dynamic input forces. Refer to par. 5-2 for additional information on available computer programs to predict suspension forces and hull radiation efficiencies. Computer time (and costs) required to analyze complex structures such as tracked vehicle hulls are proportional to the number of model elements. The frequency range of an FEA modal analysis is proportional to the grid density or spacing between model elements. Optimum selection of the FEA grid size is obtained by using the largest elements that will still give the required frequency response. Maximum spacing between node points AL can be estimated by

$$\Delta L = \frac{C_b}{4f}, \text{cm} \tag{7-1}$$

where

ΔL = maximum distance between node points, cm
C_b = speed of maximum frequency bending wave in plate, cm/s
f = maximum frequency required, Hz

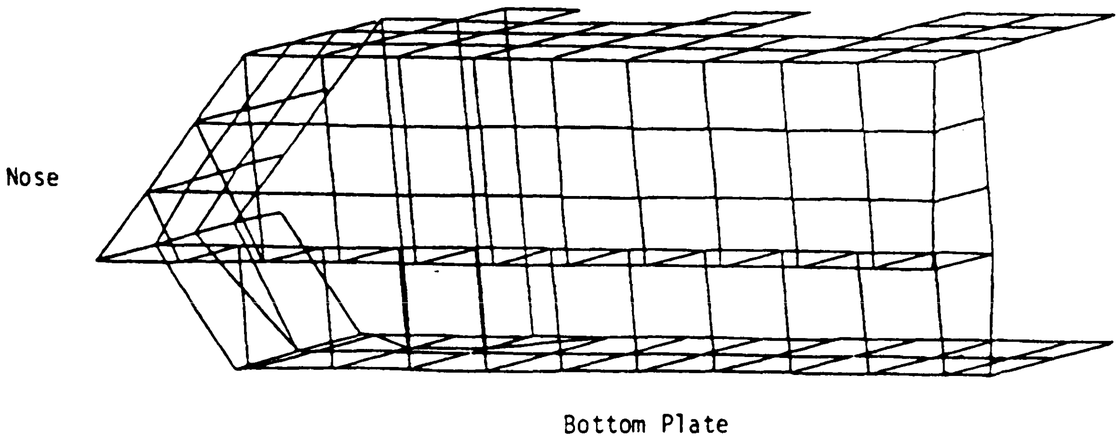

Figure 7-1. Finite Element Grid for the Left Side of an M113 Hull

and

$$C_b = \left(\frac{(2\pi f)^2 E t^2 g}{12\rho} \right)^{\frac{1}{4}}, \text{ cm/s}$$

where

E = plate modulus of elasticity, kg/cm^2
t = plate thickness, cm
g = acceleration due to gravity, cm/s^2
ρ = plate material density, kg/cm^3.

Accurate modeling of the suspension attachment locations is very important in order to obtain accurate noise-to-force predictions for suspension-generated noise. If the finite element model at the suspension attachment locations has insufficient detail, the local flexibility of the structure will not be fully accounted for, and this will lead to reduced accuracy of the predicted hull energy acceptance.

7-2 HULL CONFIGURATION

7-2.1 HULL MATERIAL AND EXTERNALLY APPLIED DAMPING

The noise radiated from a structural hull plate is determined by the excitation applied to the plate, the frequency response characteristics of the plate, and its sound radiation efficiency. The following observations of an M113 hull are typical of hull structures constructed of flat. homogeneous steel or aluminum plates. At low frequencies, plate response is much more sensitive to stiffness changes than mass or damping. In the mid frequencies the individual vibration modes of the plate control the noise produced. At high frequencies a high degree of modal overlap occurs, such that individual resonances are no longer identifiable (Ref. 3).

As explained in par. 2-3.2, damping reduces the maximum vibration amplitude at resonance. Common hull structures have a large number of plate resonances in the audio frequency range. Tracked vehicle hull materials, such as steel and aluminum, have low values of inherent damping, or loss factors, which are typically in the range of 0.0001 to 0.01. Because of these low damping values, the sound radiation of a typical hull plate is dominated by its resonant response. Hull-plate-radiated noise that is controlled by the mass and stiffness of the structure is known as its forced response. Fig. 7-2 shows predicted interior noise-to-force transfer functions for an M113 hull due to vertical dynamic forces at the left idler position. The total noise-to-force ratio shown in Fig, 7-2 is the sum of the resonant and forced responses. Because the total noise-to-force transfer function is mostly dominated by the resonant response, an increase in structural damping reduces that response: hence interior noise is reduced.

Constrained layer damping is one of the most effective methods of increasing damping in a plate or panel. This technique creates a composite plate by the addition of a thin layer of viscoelastic material to the plate surface sandwiched between an outer constraining layer of material, usually sheet metal. A plate with constrained layer damping is illustrated in Fig. 7-3.

Additional damping is provided when the plate bends at resonance and the viscoelastic layer undergoes shear strain. Design of a constrained layer damping system is fairly straightforward. The viscoehstic material is selected based on the frequency and temperature range where additional damping is needed. The constraining layer should be as stiff as possible. If the constraining layer material has the same modulus of elasticity as the parent plate, the optimum configuration will have the constraining layer the same thick-

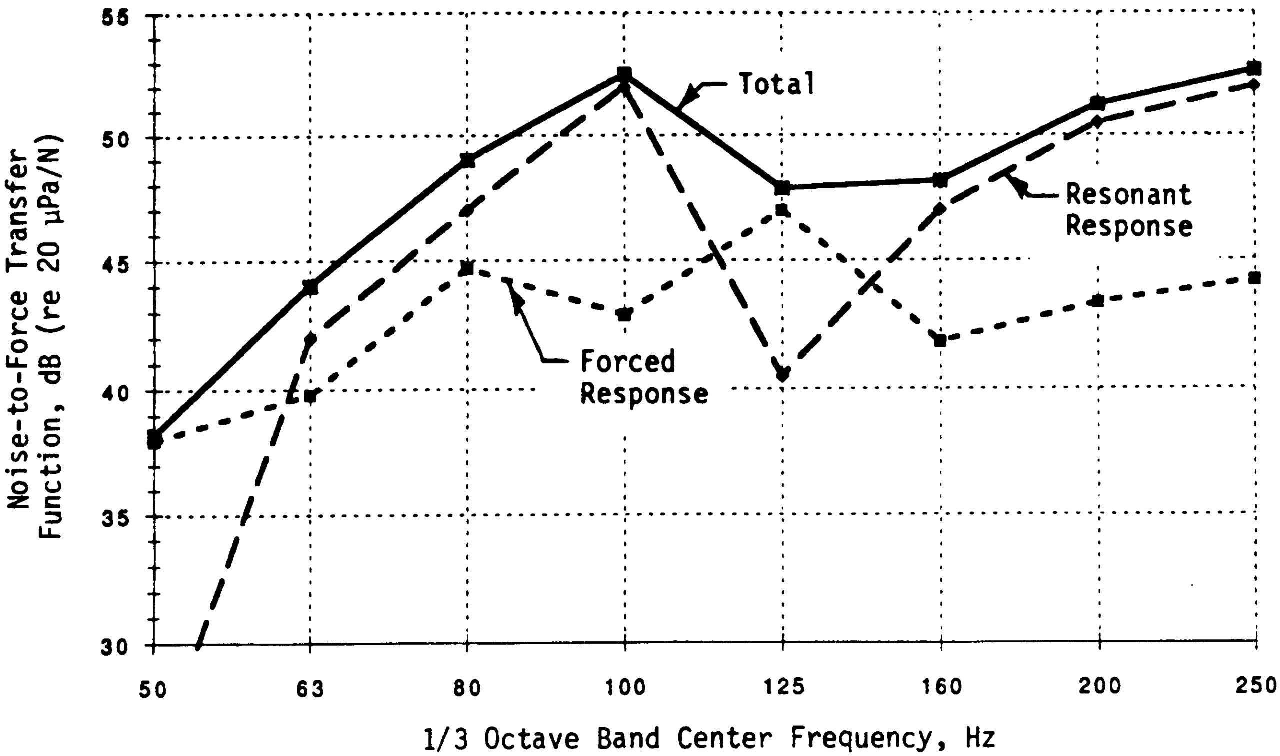

Figure 7-2. Predicted M113 Noise-to-Force Transfer Functions due to Force Inputs at the Left Idler Mount for Vertical Direction

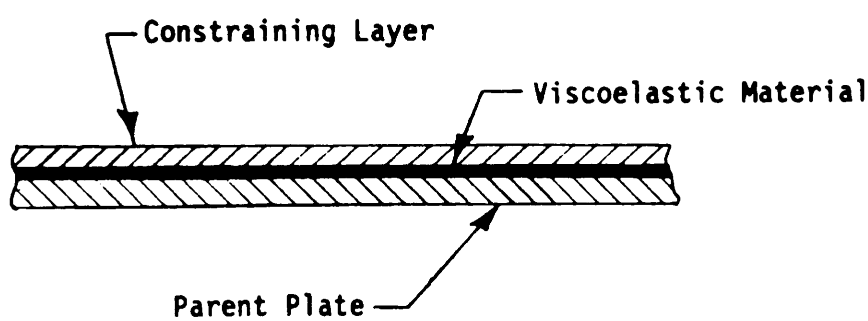

Figure 7-3. Section of Plate With Constrained Layer Damping Treatment

ness as the parent plate (Ref. 3). If the constraining layer has a greater modulus of elasticity than the base plate, optimum constrained layer thickness will be somewhat less than the base plate thickness. If geometry constraints limit the added plate to less than optimum thickness, significant damping can be obtained with even thin constraining layers to reduce the resonant response of a steel or aluminum plate, which has little intrinsic damping.

Results of experimental constrained layer damping tests on an M113A1 hull showed a 3- to 4-dB reduction in A-weighted interior noise when the vehicle was moving at speeds above 35 km/h (22 mi/h). No noise reduction was measured below 30 km/h (19 mi/h), and an average 1-dB reduction was noted at speeds between 30 and 35 km/h (19 and 22 mi/h) (Ref. 3). The lack of significant noise reduction at the lower speeds with added plate damping is a result of the suspension input frequency (tracklaying frequency) being in the stiffness-controlled region of the hull plate response where a decrease in resonant response does not exhibit a corresponding noise reduction.

7-2.2 HULL SHAPE AND ACOUSTIC COUPLING

The shape and construction details of a hull structure influence interior noise levels. Once the hull plates have been excited into vibration, the amount of noise they produce is dependent on how well they are coupled to the surrounding air. As explained in par. 2-3.3. this acoustic coupling is determined by the vibration amplitude of the plate and its radiation efficiency. Radiation efficiency, which is the ratio of radiated noise relative to the noise that would be radiated by a rigid piston oscillating with the same velocity, is basically a function of the size and thickness of the radiating element. Sound does not radiate well if the wavelength of the plate vibration is less than the corresponding wavelength of sound in air. The frequency where the two wavelengths are equal is the critical frequency of the plate. At this frequency the radiation efficiency of the plate is greater than 1.0, Large, thick plates have a low critical frequency and are therefore good radiators of low-frequency noise. Above their critical frequency. all plates are good acoustic radiators. i.e.. radiation efficiency equals 1.0. The critical frequency of a plate is calculated as

$$f_c = \frac{H}{w}, \text{ Hz} \tag{7-3}$$

where

f_c = plate critical (coincident) frequency, Hz
H = product of surface density and critical frequency. H•g/cm^2 (from Table 7-1)
w = surface density. g/cm^2.

The radiation efficiency σ_c at the critical frequency is

$$\sigma_c = \sqrt{\frac{P}{2\lambda_c}}, \text{ dimensionless} \tag{7-4}$$

where

σ_c = radiation ratio of simply supported plate at critical frequency, dimensionless
P = perimeter of plate. m
λ_c = wavelength in plate or air at critical frequency, m.

Note that

$$\lambda_c = \frac{c}{f_c}, \text{ m} \tag{7-5}$$

where

c = speed of sound in air = 344 m/s.

At frequencies below the critical frequency of the plate, the radiation efficiency decreases nonlinearly in relation to frequency. An approximation of [he radiation efficiency for frequencies below the critical frequency can be obtained by calculating the radiation efficiency in frequency regions as shown in Fig. 7-4. Radiation efficiency for the frequencies between f_1 and f_2 is

$$\sigma_{lo} = \frac{\lambda_c^2}{S}, \text{ dimensionless} \tag{7-6}$$

where

σ_{lo} = radiation efficiency of simply supported plate at low frequencies, dimensionless
S = surface area of plate. m^2.

Frequencies f_1 and f_2 are calculated as

$$f_1 = \frac{c^2}{2Sf_c}\left(\frac{P^2}{8S} - 1\right), \text{ Hz} \tag{7-7}$$

$$f_2 = \frac{3c}{P}, \text{ Hz} \tag{7-8}$$

TABLE 7-1. PRODUCT OF SURFACE DENSITY AND CRITICAL FREQUENCY FOR VARIOUS MATERIALS (Ref. 4)

MATERIAL TYPE	PRODUCT OF SURFACE DENSITY AND CRITICAL FREQUENCY H, Hz·g/cm^2
Aluminum	3470
Glass	3800
Lead	60,500
Steel	9750

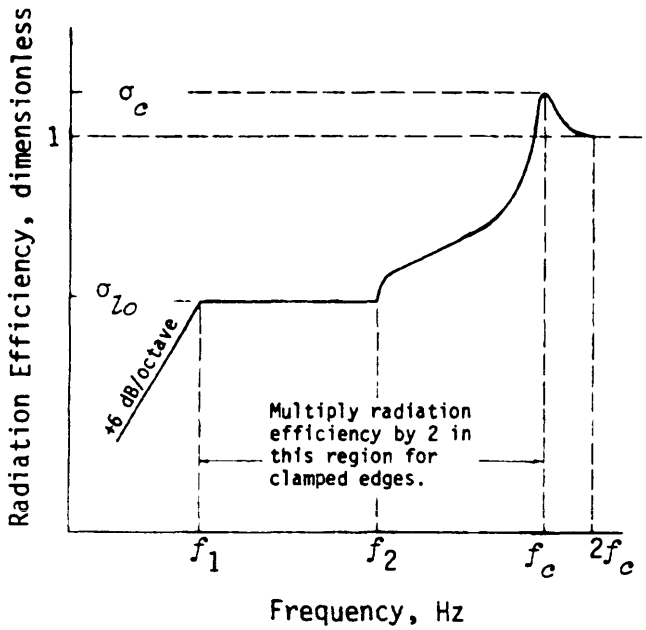

Figure 7-4. Design Curve for Approximating the Acoustic Radiation Efficiency of a Finite Panel With Simply Supported or Clamped Edges

where

f_1 = first breakpoint frequency for radiation efficiency calculation, Hz
f_2 = second breakpoint frequency for radiation efficiency calculation. Hz.

The example that follows is presented to help the designer better understand the relationship between plate size and radiation efficiency. Consider a 38-mm (1.5-in.) thick aluminum hull plate that is 3.5 m (11.5 ft) long and 1 m (3.3 ft) wide. The frequencies of the three frequency break points f_1, f_2 and f_c shown on Fig. 7-4 are calculated by using Eqs. 7-3, 7-4, and 7-8 as

$$f_c = \frac{3470}{10.34} = 336 \text{ Hz}$$

$$f_1 = \frac{344^2}{2 \times 3.5 \times 336}\left(\frac{9^2}{8 \times 3.5} - 1\right) = 95 \text{ Hz}$$

$$f_2 = \frac{3 \times 344}{9} = 115 \text{ Hz}.$$

By using Eq. 7-4, the radiation efficiency σ_c at the critical frequency is

$$\sigma_c = \sqrt{\frac{9}{2 \times 1.02}} = 2.1 \text{ dimensionless}$$

where

$$\lambda_c = \frac{344}{336} = 1.02 \text{ m}.$$

By using Eq. 7-6. the radiation efficiency for the rang between f_1 and f_2 is

$$\sigma_{lo} = \left(\frac{1.02^2}{3.5}\right) = 0.3 \text{ dimensionless}.$$

Plate radiation efficiency for frequencies above the critical frequency approaches 1.0 for frequencies greater than two times the critical frequency or 672 Hz in this example.

For a plate with edge conditions other than simply supported, the radiation efficiency in the frequency range between f_1 and f_2 will be higher than calculated. (See Fig. 7-4.) Typical real hull plates welded to adjacent plates will have a radiation efficiency somewhere in between that of a plate with simply supported edges and one with clamped edges. The radiation efficiency for a single plate can be approximated quite accurately using the information in Fig. 7-4. More complex structures require a more sophisticated approach, such as statistical energy analysis (SEA). Statistical energy analysis permits energy flow calculations between connected resonant structures, such as plates and beams, and between plates and the surrounding air (Ref. 4).

Plate radiation efficiency decreases with thinner, smaller area plates; however, thinner plates have lower stiffness and greater levels of vibration. Because of this oppositional relationship between plate geometry and radiated noise, it is difficult to design hull structures that produce less noise based on intuition or rules of thumb. A good finite element model coupled with a noise predictor as discussed in par. 7-1 is required to optimize hull plate changes that will decrease interior noise.

7-2.3 ATTACHMENT POINT IMPEDANCE

In general, analysis has shown that the key to making a quiet hull is to reduce its structural response due to excitation at the suspension attachment locations (Ref. 2). Point impedance is the transfer function obtained by exciting a structure with a known dynamic force and measuring the structural response at that same point. Hull impedance—force/acceleration—at the suspension attachment locations is a measure of the energy-accepting ability of the hull at those locations. The greater the hull impedance at locations where vibrational energy is inserted, the less the structure will vibrate and thereby generating less noise. Increasing hull stiffness and/or mass increases attachment point impedance.

A research project was conducted using a finite element model of an M113 hull structure to determine the effects that major hull changes would have on interior noise. The following is a description of some of the modifications that were analyzed:

1. Rib stiffeners, 100 mm high x 6.4 mm thick (4 in. x 0.25 in.) attached perpendicular]} to the hull plates and placed circumferentially and longitudinally around the hull. spaced approximately 300 to 460 mm (12 to 18 in,) apart
2. Double bottom plate consisting of two 28-mm (1.12-in.) thick aluminum plates separated by a 150-mm (6-in.) air space and connected with rib stiffeners
3. Chamfered upper side plates to eliminate the square joint at the top of the hull
4. All plate material stiffness increased by a factor of ten (10 times Young's modulus of elasticity)
5. Hull material damping increased by a factor of two
6. Hull material density decreased by a factor of two

This analysis was conducted for the frequency range 50 to 250 Hz. The results of the analysis for each hull configuration modification are summarized in Table 7-2.

Note that the predicted noise reduction for the double damping hull configuration is in close agreement with the experimental results obtained with constrained layer damping, as discussed in par. 7-2.1.

This analysis indicates that a quiet hull design must be very stiff and well damped. Also very drastic changes in materials or configuration produce only moderate noise reduction.

7-3 INTERIOR ACOUSTICS

The interior noise field in a tracked vehicle is highly reverberant due to a small interior volume and hard surfaces. Noise control theory suggests that the addition of sound-absorbing material should provide significant noise reduction. An experiment was conducted on the interor of an M113 vehicle in which virtually all of the interior surfaces were covered with 51 -mm (2-in.) thick sound-absorbing foam/barrier/foam material. Test results showed a 13-dB reduction in A-weighted sound levels with the full treatment and a 6- to 8-dB reduction for partial coverage (Ref. 5). Due to extensive use of interor surfaces for equipment stowage. large amounts of acoustic absorptive material are difficult to install in a military tracked vehicle. For safety, sound-absorbtive materials need to be durable, nontoxic. and non-flammable in case of fire.

Another classical approach to noise reduction is to add sound barriers between noise sources and observers. Since all hull surfaces radiate noise into the hull cavity. curtains or panels spaced several centimeters from the hull surfaces would be effective only if virtually all the hull surfaces were covered. In this configuration the predicted noise reduction would be that obtained from a double-wall enclosure. as discussed in par. 6-2. The added interior panels would need to be heavy and spaced as far as possible from the hull plates to obtain significant noise reduction in the critical 250-Hz frequency range. The added interor panels would also need to be structurally isolated from the hull to prevent vibration from entering the barrier panels and causing them to radiate noise.

The increased weight and lost interior volume penalties associated with an isolated barrier wall are large and may not be acceptable for the noise reductions achieved. Partial coverage of interior surfaces with barrier material. such as span liners. has been shown to provide very little noise reduction at frequencies below 1000 Hz (1 to 3 dB) and approximately 5 to 10 dB at frequencies above 1000 Hz.

To date, practical solutions to reduce tracked vehicle interior noise through absorptive or barrier treatments have produced limited noise reduction.

TABLE 7-2. PREDICTED INTERIOR NOISE REDUCTION IN AN M113 HULL DUE TO MAJOR STRUCTURAL CHANGES

HULL CONFIGURATION	EFFECT ON INTERIOR NOISE-TO-FORCE TRANSFER FUNCTION
Rib Stiffeners	2- to 3-dB reduction
Double Bottom Plate	2- to 3-dB reduction
Chamfered Upper Side Plates	1- to 2-dB reduction
Ten Times Young's Modulus	10-dB reduction
Double Damping	3-dB reduction
Half Material Density	2- to 5-dB increase

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CHAPTER 8
ACOUSTIC TESTING

Basic acoustic test procedures and data analysis are presented. Measurement techniques useful for identification and quantification of tracked vehicle noise sources are discussed.

8-0 LIST OF SYMBOLS

N_P = power train noise, dB
N_S = suspension noise (towed), dB
N_T = total vehicle noise (self-powered), dB

8-1 INTRODUCTION

The ability to measure noise and vibration accurately is critical to the success of a tracked vehicle noise reduction effort. There are numerous factors involved in the noise generation and measurement process, such that a certain variability is always inherent in any noise measurement. The task of acoustic test personnel is to minimize the variability caused by measurement equipment and test procedures to a level where the residual variability is manageable and test results provide meaningful information. The two types of noise testing typically used during a noise reduction effort are proof of compliance and diagnostic testing. Proof of compliance testing is described in detail in MIL-STD-1474 (Ref. 1) and is used to verify whether a tracked vehicle meets the noise requirements of the standard. Diagnostic testing consists of many types of tests to determine noise sources, vibration sources, attachment point impedances, propagation paths, attenuation characteristics, reverberation characteristics, and other parameters critical to a noise reduction effort.

This chapter provides testing guidelines for personnel involved in tracked vehicle noise reduction work. An understanding of proven test procedures is helpful to avoid pitfalls that are commonly encountered in noise testing and measurement.

8-2 TESTING CONSIDERATIONS

8-2.1 SPACE-TIME AVERAGING

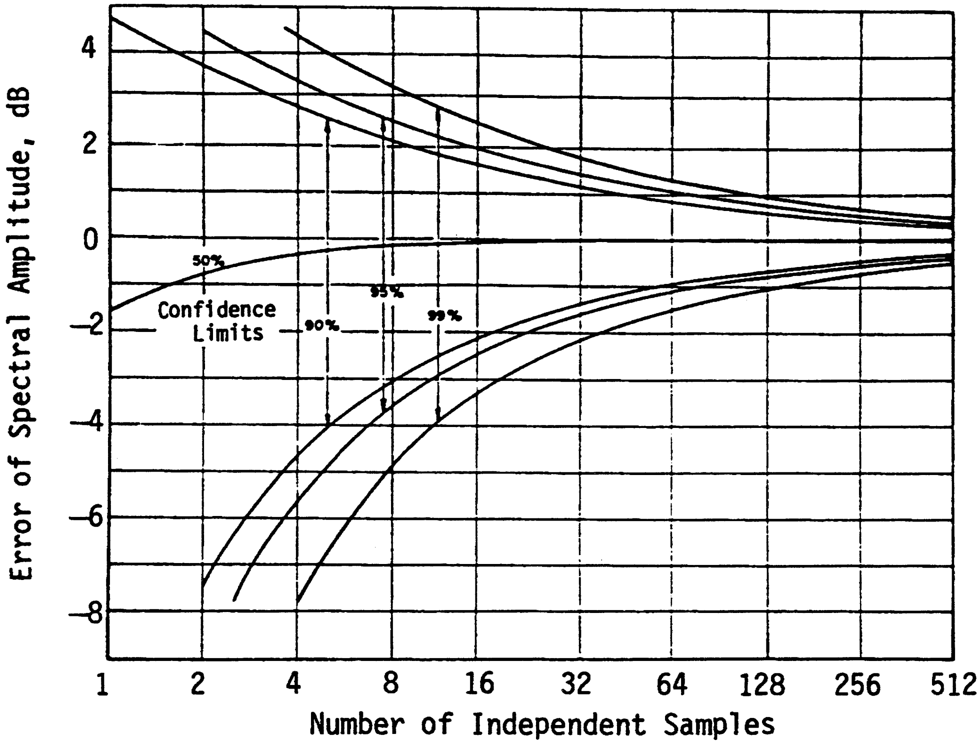
Noise produced by a tracked vehicle is predominantly random noise. As explained in par. 2-2, a random signal cannot be characterized simply by a magnitude and frequency. A spectrum or spectral sum is required to characterize average features of the signal, which is constantly changing with time. Noise measurements are conventionally made using a hand-held sound-level meter (SLM), which provides a time-averaged root-mean-squared (RMS) sound level. A visual average of the extremes of the meter needle movement gives a rough indication of the average sound pressure level (SPL). Although this method of noise measurement is acceptable for general noise surveys, it is not sufficiently repeatable for precision noise analysis, especially if both the noise level and frequency are varying rapidly.

The confidence level of measuring a spectrum that is equal to the true long-term (infinite sample time) spectrum increases with each independent spectrum added to the average. As seen in Fig. 8-1, which shows the upper and lower error bands as a function of the number of independent sample averages, it is important to average as many samples as possible.

The assumptions in this error analysis are that the noise is predominantly white—equal energy per unit frequency—and stationary—frequency and amplitude distribution do not change with time. For some noise measurements, such as a passing vehicle (exterior noise), the time sample available to analyze spectral content is very short. For this type of measurement it is important to understand the statistical errors in the data analysis and to repeat the measurements a number of times to increase the confidence level.

When measuring noise in the interior of a tracked vehicle, microphone location is critical to the production of accurate and repeatable measurements. As explained in par. 4-4.2, acoustic resonances or standing waves exist in a reverberant field. Space averaging is required to make accurate measurements in this type of acoustic environment. Moving the microphone in a circle or ellipse in the area to be measured has been found useful in reducing errors associated with acoustic standing waves. Statistical analysis of measurements using several methods of microphone placement in an M113A1 vehicle has shown that a small circular motion of approximately 30 cm (12 in.) diameter produced the most consistent noise levels (Ref. 3). The speed of rotation is not critical; one revolution per 5 s has been used with good results. If the microphone motion is produced by a human observer, maintaining a constant microphone speed and being careful not to produce extra noise, such as by sliding the microphone through the fingers, are important testing considerations.

An alternative approach to space averaging is to place a stationary microphone in one corner of the interior cavity. Measurements at this location will be approximately 9 dB higher than central cavity measurements because the corner is a node for all cavity acoustic resonances. This technique works best for verifying the noise reduction provided by



QuadTech, Inc., Bolton, MA 01740, Manufacturer of Impedance Measuring Instruments and Standards

Figure 8-1. Error Analysis for Frequency Averaged Random Noise Signals (Ref. 2)

modifications rather than for measuring the noise to which the crew is subjected.

8-2.2 INSTRUMENTATION

Three types of precision microphones—electret-condenser, ceramic, and condenser—are typically used for instrumented noise measurements. All of these microphone types have good frequency response and accuracy. For interior tracked vehicle measurements a reverberant field exists in which sound is coming at the microphone from all angles; therefore, a microphone with a flat random incidence response is preferred. Incidence is the angle of the sound wave with respect to the microphone diaphragm.

Microphone diaphragm size determines the frequency range and sensitivity of a microphone. In general, a 25.4-mm (1-in.) diameter microphone is 6 to 8 dB more sensitive than a 12.7-mm (0.5-in.) diameter microphone. However, the flat frequency response of a 12.7-mm microphone (20,000 Hz) is twice the range of a 25.4-mm microphone (10,000 Hz) (Ref. 2). Either of these two common sizes of microphones is acceptable for general tracked vehicle noise measurements.

Structural vibration is measured using transducers, such as accelerometers, velocity pickups, and displacement transducers. For acceleration measurements not requiring very low-frequency response (<4 Hz), a rugged piezoelectric accelerometer is ideal. If very low-frequency measurements, such as for ride quality, are required, a servo accelerometer or a strain-gage (piezoresistive) accelerometer is required. Velocity and displacement transducers are seldom used for structure-borne noise investigation work. When mobility transfer functions are required, a velocity pickup can be used directly, or frequency domain integration of an impedance transfer function acquired with an accelerometer will provide the same information.

The electrical signal from the microphone or vibration transducer is typically input to a signal analyzer to convert the raw signal into usable engineering units, conduct frequency analysis, and generate graphs used to interpret test results. Often the transducer signal is recorded on a magnetic tape at the test site, and the recorded data are later input to a signal analyzer in the laboratory.

System calibration prior to and following the start of a test is very important to assure that test results are accurate. In a system calibration a known signal—sound or vibration—is supplied to the transducer using a calibrated source generator (calibrator). The resultant electrical signal as modified by any cables, amplifiers, or recorder is used as a reference to determine the transducer sensitivity for subsequent measurements. The calibrator must be checked on a regular basis to ensure that its calibration signal is within the accuracy tolerances prescribed for the type of transducer being calibrated. This check is usually done at an independent laboratory, that maintains calibration equipment traceable to the National Institute of Standards and Technology reference sources.

Assuming that quality instrumentation is used, there are two common sources of measurement error: errors caused by background noise and errors caused by improper test, calibration, or documentation procedures. A discussion of each error follows.

1. *Errors Caused by Background Noise*. If the background noise is not at least 10 dB lower than the noise being measured in all frequency bands of interest, a significant error is introduced into the measurement. Adjustment (reduction) of the measured noise levels using Fig. 8-2 can compensate for the error. However, reduction of the background level during the measurement is the best method of avoiding signal-to-noise ratio problems.

2. *Errors Caused by Improper Test Calibration or Documentation Procedures*. Many hours of measurements may become useless if key test parameters are not recorded. These parameters include background noise level; transducer positions; vehicle operational parameters, such as speed and direction; calibration values; and amplifier or recorder gain levels. For example, if a recorder or sound level meter gain sensitivity is changed during the course of a test to prevent signal overload or to improve signal-to-noise ratio, this change must be recorded before testing continues so that the range change will be taken into account during data reduction.

8-2.3 OTHER TESTING CONSIDERATIONS

An instrumented test that is useful in noise diagnostic work is the force/response test. In this type of testing force is applied to a structure, and the resultant response (vibration or noise) is measured. Experimental modal analysis is a common application of a force/response test. Measured modal analysis of an existing hull can be used to verify the validity and accuracy of a finite element hull model, as described in par. 7-1, by comparing modal parameters of the predicted and measured modes. In a hull modal analysis the hull forcing function is supplied by electric or hydraulic actuators. Force is measured using a load cell at the point of force application, and response is measured using accelerometers at the grid locations, that correspond to nodal points in the analytical model. Fig. 8-3 shows a modal analysis being conducted on the hull of an M2 vehicle. The oscillating force is being applied by three 445 N (100 lb) (rated force) electrodynamic shakers at the side, rear, and front of the hull structure.

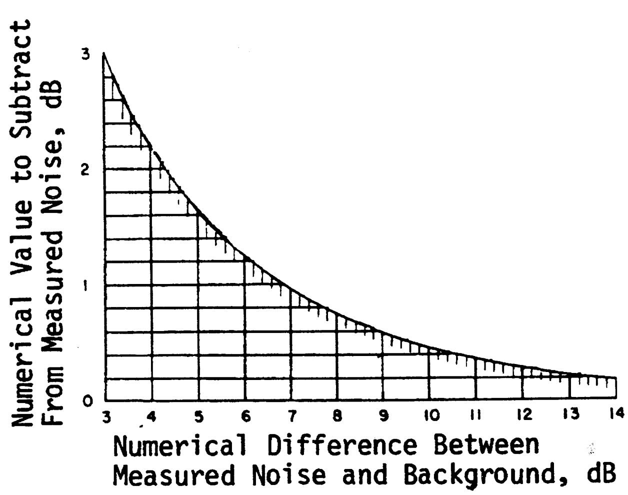


Figure 8-2. Correction for Background Noise

An alternate method of exciting a structure into vibration uses an instrumented hammer. The hammer has a force transducer built into its head so that upon impact the force pulse transmitted to the structure is measured. This technique is fast to set up and works well with small structures that have a small number of measurement locations.

As discussed in par. 5-6.3, mobility measurements can be very helpful in assessing the isolator spring rates required to achieve a given noise reduction. A mobility measurement consists of the resultant velocity-to-force transfer function of a structure when excited by an oscillating force. For example, to measure the mobility of the road arm mounting location, a shaker or impact hammer would be used to vibrate the structure at the road arm attachment point. A velocity pickup or accelerometer attached to the hull directly adjacent to the force input location would measure the resultant structural response. A Fast Fourier Transform (FFT) signal analyzer would be used to calculate the mobility transfer function. Using the same measurement signals, a structural impedance measurement could be obtained from the acceleration-to-force transfer function.

8-3 NOISE SOURCE IDENTIFICATION TECHNIQUES

Window analysis is a proven methodology for isolating the individual sources of noise within a tracked vehicle. In a window analysis each source is measured separately to quantify its noise independently of other noise producers. That is, a "window" is opened on the source of interest, and all other noise sources are eliminated.

8-3.1 TEST STAND

For tracked vehicle suspensions a test stand, such as shown in Fig. 8-4, is an invaluable tool for conducting a suspension window analysis. A suspension test stand provides normal track operation with only one of the suspension components actually attached to the vehicle. An idler wheel window analysis, for example, consists of the idler wheel attached to the test vehicle with all other suspension wheels attached to the test stand. The track is driven by a variable speed electric motor through the sprocket wheel on the test stand. Noise inside the vehicle is solely the result of chordal action forces as the track travels around the idler

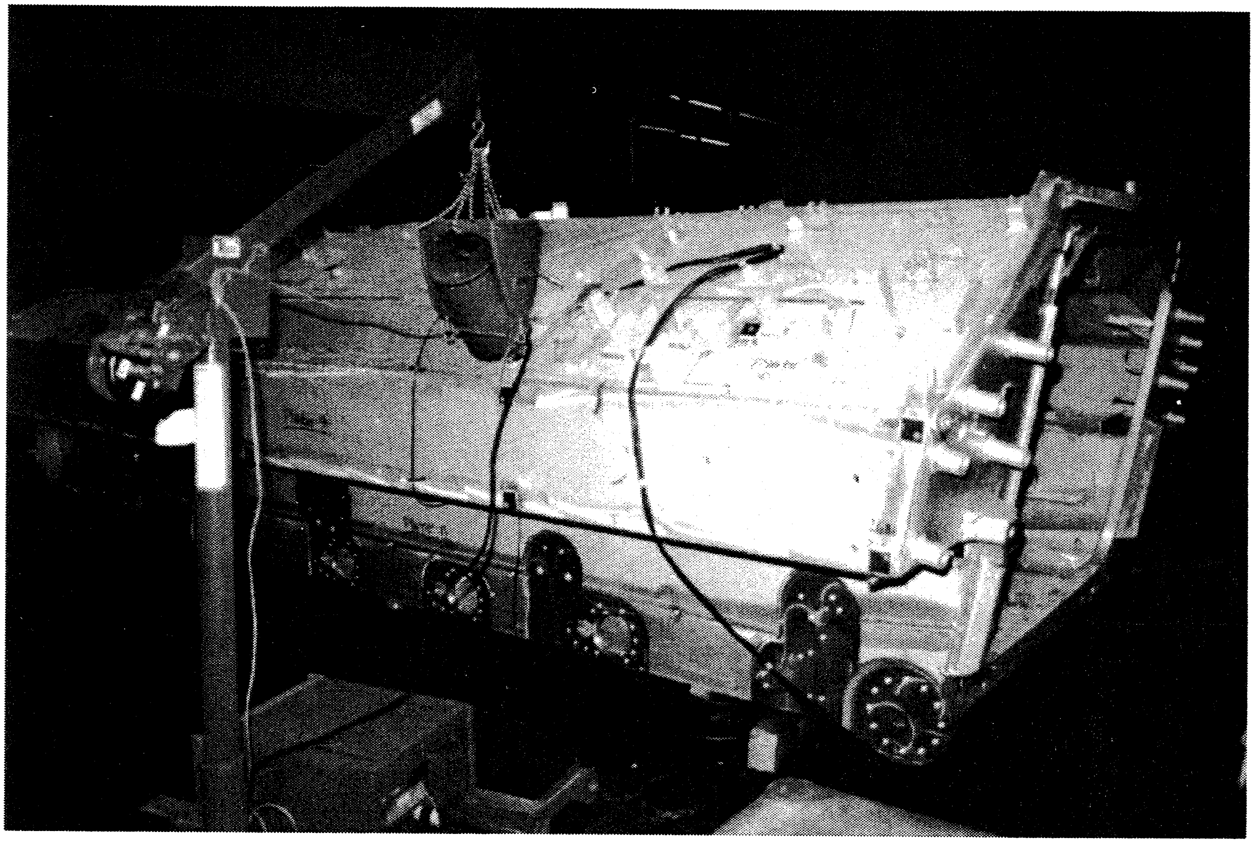


Figure 8-3. Modal Analysis Testing of an M2 Hull Structure

wheel. By using this test methodology, the noise of one suspension component can be measured at various track (vehicle) speeds.

The most important consideration in designing a suspension test stand is to ensure that the noise generated by the stand is at least 10 dB below the levels expected to be measured. This is a relatively simple task for measuring standard tracked vehicle components; however, when compliant suspension components are installed, the self-noise of the stand may be significant. No hard connections should exist from the vehicle support structure to the suspension mounting portion of the test stand. Heavy-duty vibration isolators are effective for providing structure-borne noise attenuation from the test stand to the vehicle. Airborne noise from the stand can be minimized by using heavy, stiff plates and beams to construct the stand. The rule to follow for a low-noise stand is maximum stiffness and minimum surface area for the suspension support structures. An enclosure around the electric motor may be required if motor noise is a problem. A special low-noise track return wheel used in the M2/M3 suspension window analysis is shown in Fig. 8-5. This special wheel consisted of two small automobile tires mounted on an adapter plate, which then bolted onto the idler or sprocket hub. The soft rubber tires provided noise reduction because they reduced the chordal action forces transmitted to the test stand.

Test stands used for the M113 noise reduction work as well as the suspension window analysis on the M2/M3 vehicle were able to measure only sprocket-, idler-, and track-support-roller-generated noise because no provisions were included to simulate ground forces for road wheel noise measurements. A test stand designed to measure road wheel noise would require a low-friction mechanism to supply upward force to the bottom of the track. One possible method would have a smooth, flat plate that is covered with a soapy water lubricant under the track. The purpose of this lubricant is to lower the friction between the plate and the track pads to allow the limited torque from the electric motor to rotate the track. Another possibility would use heavy-duty conveyor-belt-type material over rollers to provide a low-friction method of supplying realistic ground simulation forces.

8-3.2 TOWING TESTS

Towing the test vehicle while measuring interior noise is a suitable technique for some tests. The interior noise of a towed vehicle is generated entirely by the suspension system when the sprocket input drive shafts are disconnected from the transmission or steering differential gearbox. With engine, transmission, cooling fan, and exhaust noise eliminated, total suspension noise is measured directly. Total

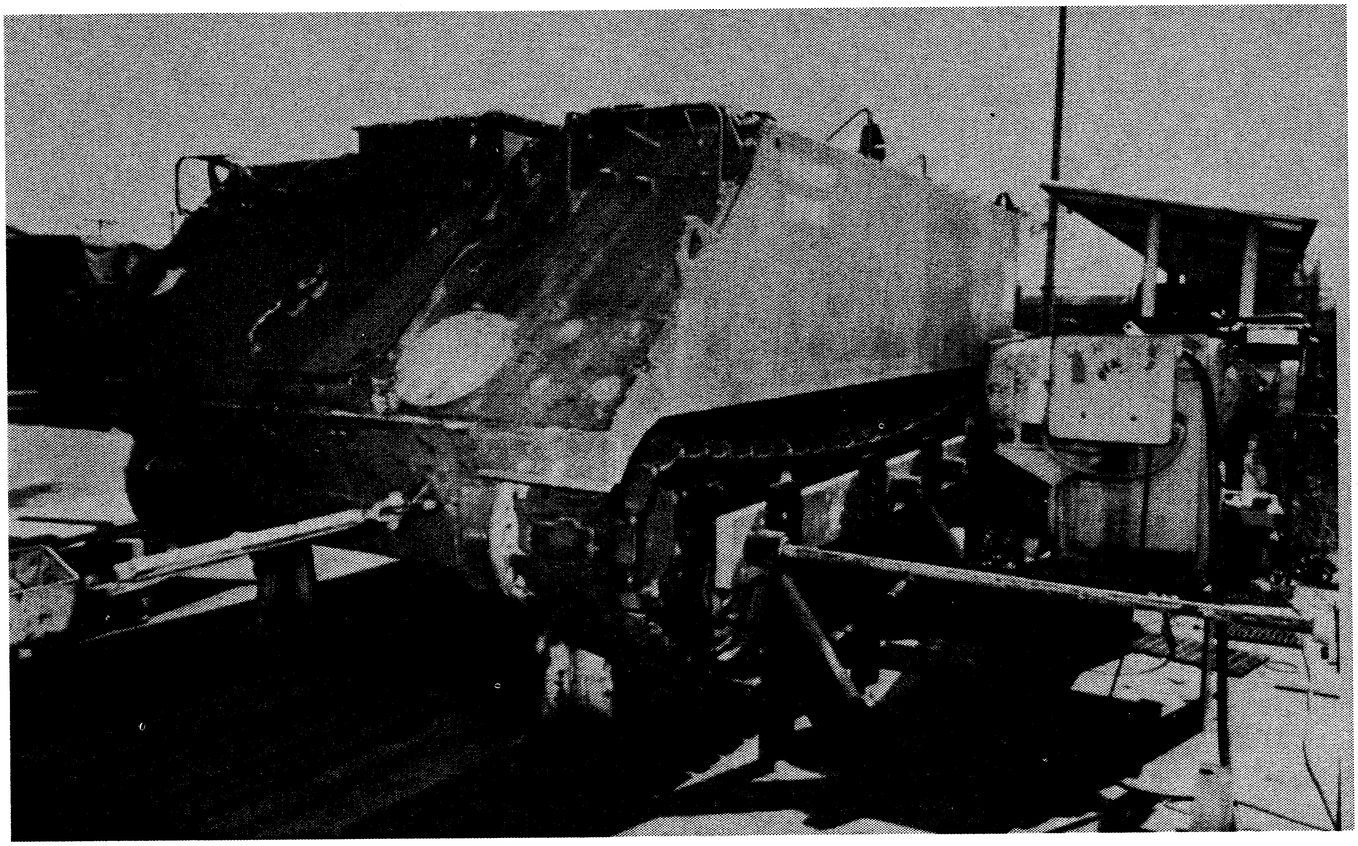


Figure 8-4. M113 Vehicle Suspension Window Analysis Test Stand

power train noise can be estimated by subtracting suspension noise from self-powered noise measured at a given vehicle speed as

$$N_P = 10\log\left[10^{\left(\frac{N_T}{10}\right)} - 10^{\left(\frac{N_S}{10}\right)}\right], \text{dB} \qquad (8\text{-}1)$$

where

N_P = power train noise, dB
N_S = suspension noise (towed), dB
N_T = total vehicle noise (self-powered), dB.

Another type of test that can be conducted with a towed vehicle is a road wheel noise test. Interior noise generated by the road wheels alone can be measured as the vehicle is towed if the sprocket and idler wheels are removed and the track is configured as shown in Fig. 8-6. In this configuration the track is only wrapped around the road wheels, so special road arm configurations are required to maintain normal road arm motion and track tension. For the M113 vehicle configuration shown in Fig. 8-6, the first road arm was inverted from a trailing to a leading arm. This configuration enabled the track to be wrapped around the road wheels with the weight of the vehicle providing some track tension. The road wheel noise levels in this type of towed test are probably somewhat higher than occur in a vehicle with normal track configuration due to the extra chordal action at the first and last road wheels.

8-3.3 FORCE INPUT TESTING

Several examples of force input testing for measuring transfer functions and modal analysis were presented in par. 8-2.3. Other types of testing that can be done with dynamic force input devices (vibration shakers) include insertion loss measurements, structure-borne path investigations, and simulation of vehicle operation. The road wheel noise reduction verification tests on the M113 vehicle are examples of insertion loss measurements using a vibration shaker. To maintain low costs through the development phase of the isolated road arm design, only two of the 10 road arms were experimentally modified. Therefore, neither the test stand nor a towing test could be used to evaluate the results of the modifications. The insertion loss of the isolated road arms was calculated as the difference between noise-to-force transfer function measurements made using a shaker under the road wheel before and after the road arm modifications (Ref. 4).

8-3.4 DYNAMOMETER TESTING

Accurate assessment of power train contribution to interior noise requires a dynamometer facility, such as shown in

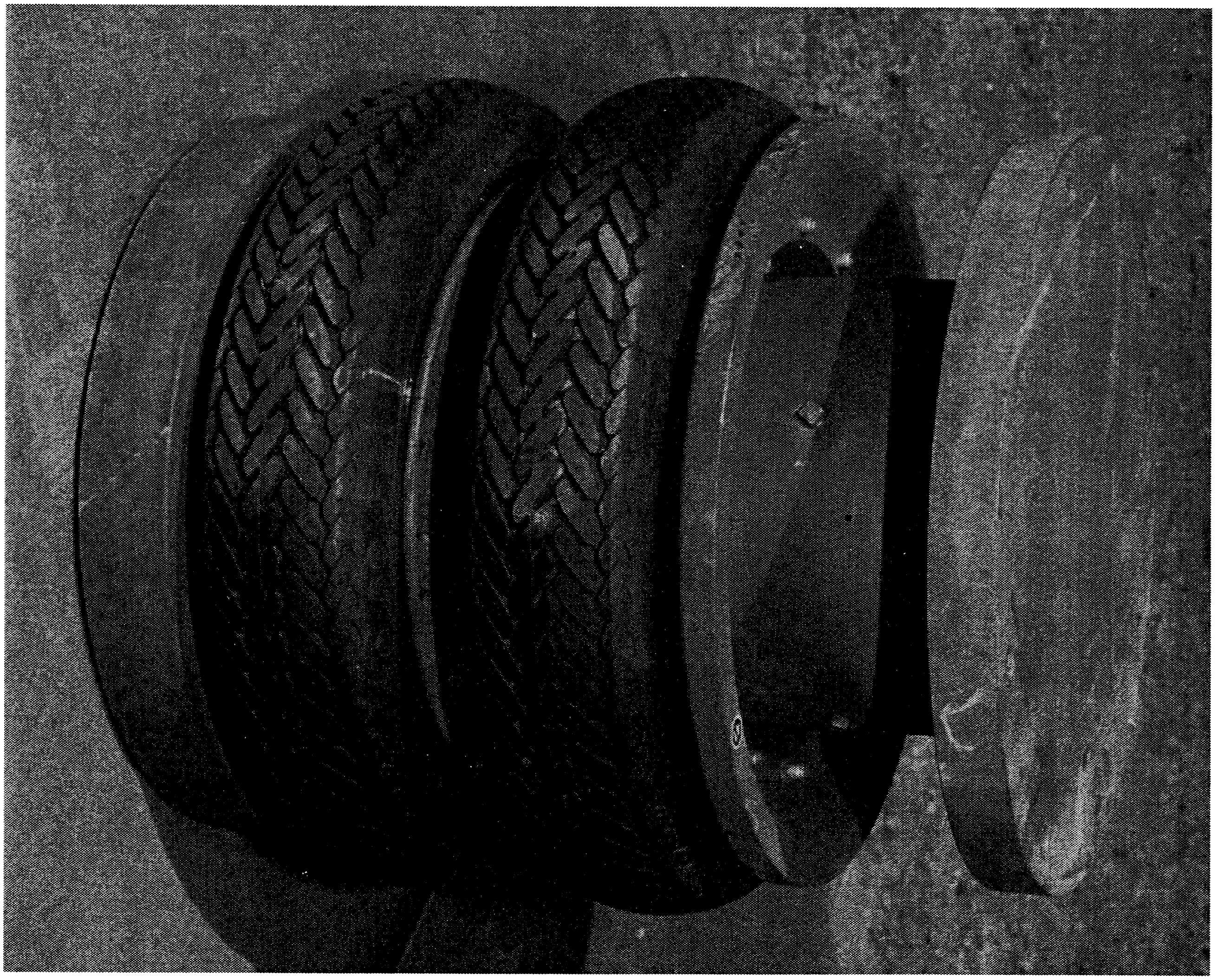


Figure 8-5. Special Low-Noise Return Idler Wheel for the M2/M3 Suspension Test Stand

Fig. 8-7, where speeds and engine loads can be controlled to simulate actual vehicle operation without any noise contribution from the suspension components. Interior noise measured during this simulated vehicle operation will be generated only by the engine and power train. For most tracked vehicle designs it is very difficult to conduct a window analysis to determine the noise of the transmission or final drive gearbox separately. However, using the vehicle towing technique described in subpar. 8-3.2, estimates of transmission noise can be made by subtracting engine-only noise, i.e., transmission in neutral, from total noise with the transmission engaged. Some error will be introduced because the noise from the engine varies slightly when it is operating under load from when it is just operating against its own inertia. However, this measurement and analysis technique will determine whether the transmission is a larger, equal, or smaller noise source than the engine.

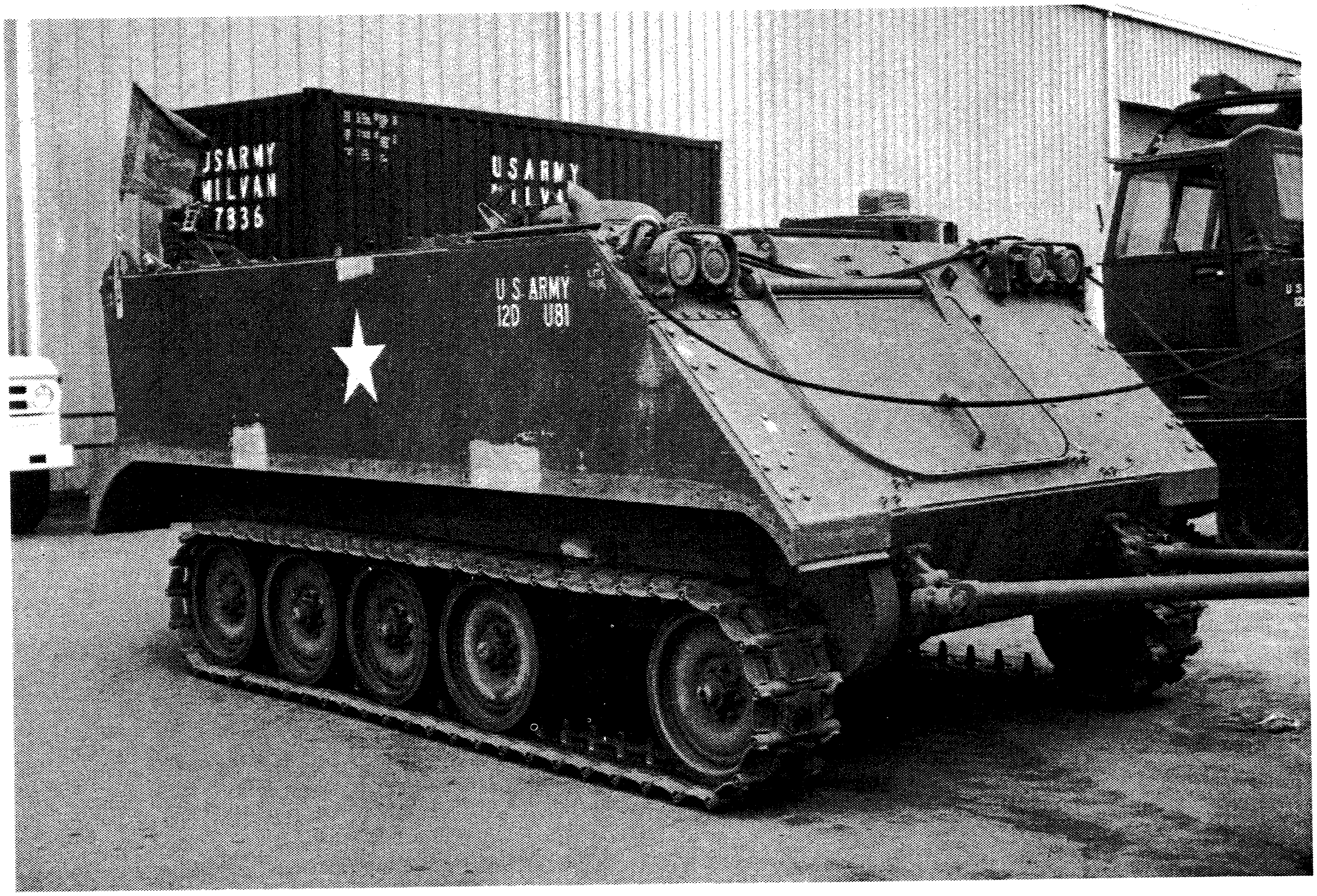


Figure 8-6. **Road Wheel Noise Measurement on an M113A1 Vehicle With Modified Track Configuration**

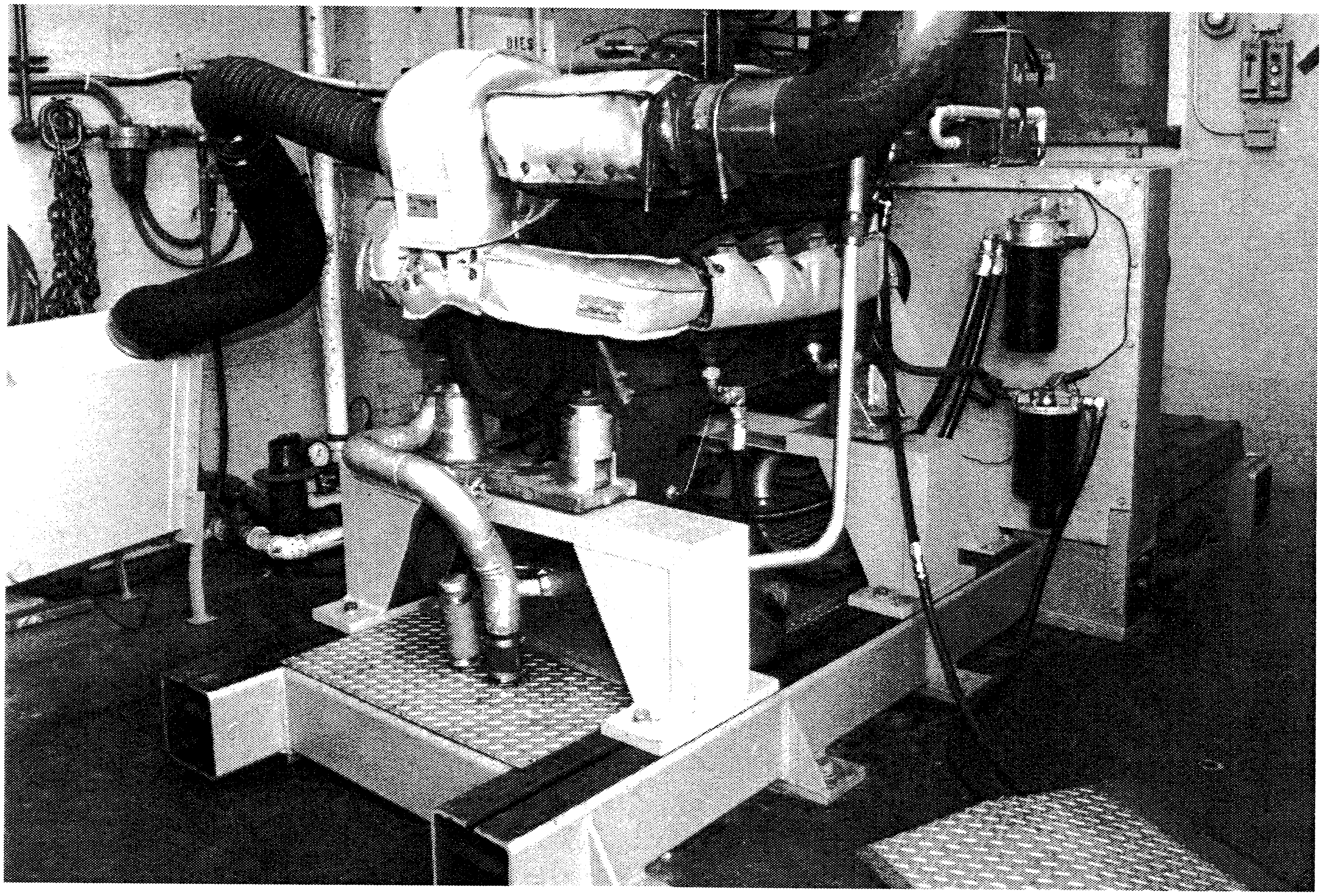


Figure 8-7. Dynamometer Test Facility

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GLOSSARY

A

Acoustic Absorption. The property possessed by materials and structures to convert sound energy to heat by either propagation in the medium or dissipation when sound strikes a surface.

Aural Detectability. The ability to hear or measure a sound that has propagated over a distance from a source such that its amplitude is approximately equal to the background noise level.

C

Compliance. A transfer function obtained as the complex ratio of displacement (response) to force (excitation) for a mechanical structure.

D

Damping. The dissipation of vibrational energy with time or distance usually by conversion to heat energy.

Decibel. A unit of amplitude level that denotes the ratio between two quantities which are proportional to power; the number of decibels corresponding to this ratio is 10 times the logarithm (to the base 10) of this ratio. Sound pressure levels are typically noted as 20 times the logarithm of the ratio of sound pressure to a reference pressure.

E

Excitation Force. An external oscillating (shaker) or transient (hammer) force applied to a structure to cause the structure to vibrate.

F

Frequency Domain. Analysis of time variant signals as to their respective frequencies and amplitudes.

I

Impedance. A transfer function obtained as the complex ratio of acceleration (response) to force (excitation) for a mechanical structure.

M

Masking. The inability to detect a particular sound because of the presence of another sound of approximately equal or greater amplitude.

Modes. Deformation of a structure due to resonant vibration at its natural frequencies. A fundamental or first mode of the structure corresponds to its lowest natural frequency; succeeding natural frequencies are designated as 2nd mode, 3rd mode, etc.

Mobility. A transfer function obtained as the complex ratio of velocity (response) to force (excitation) for a mechanical structure.

N

Noise ***Floor.*** The lowest amplitude of a sound or vibration signal capable of being measured. The noise floor may be determined by background noise, self-generated transducer noise, signal conditioner or amplifier noise, or the dynamic range limitations of a tape recorder or signal analyzer.

O

One-Third Octave Band Spectra. Frequency analysis in which the filter bandwidth is 23.1% of the center frequency. Center frequencies commonly used in acoustics covering a decade frequency span are 10, 12.5, 16, 20, 25, 31.5, 40, 50, 63, and 80 Hz. Additional decade one-third octave band center frequencies are obtained by multiplication of these basic frequencies by the appropriate power of 10.

P

Propagation Paths. The transmitting medium whereby oscillations (vibration or sound) get from the source location to the listener.

R

Radiation Efficiency. The ability of a vibrating structure to excite the surrounding air particles into oscillation and thus generate sound. Radiation efficiency is the ratio of radiated noise from a vibrating plate relative to the noise that would be radiated by a rigid piston oscillating with the same velocity.

Random Noise Signal. A noise signal for which the amplitude and frequency at any given time cannot be predicted based on past signal history. A random noise signal is nonrepetitive but is usually characterized by its statistically measurable attributes of frequency and amplitude shown graphically as mean square amplitude on the ordinate axis and frequency along the abscissa.

Resonance. A situation of a system in forced oscillation that exists when any change, however small, in excitation frequency causes a decrease in system response.

Reverberation. The sound that persists in an enclosed space as a result of reflection and/or scattering after the source of sound has stopped.

T

Transfer Function. The complex ratio of system response to excitation input. For a linear system a sinusoidal input at a frequency *f* will produce a sinusoidal output at the same frequency *f*. However, the amplitude of the output will generally be different from the input amplitude, and the output will generally be shifted in phase from the input. Transfer functions are often shown as Bode plots consisting of a graph of magnitude ratio vs frequency and a graph of phase shift vs frequency.

Transmission Loss. A measure of the sound or vibration attenuation in a system due to modification of the original propagation path.

V

Vibration. Oscillations of the particles of an elastic body or medium in alternately opposite directions from the position of equilibrium when that equilibrium has been disturbed.

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SUBJECT TERM (KEY WORD) LISTING

Acoustics
Auditory detection
Chordal action
Damping
Drive sprocket
Engine
Final drive
Hearing hazard
Hull
Idler wheel
Interior acoustics
Measurement errors
Microphone
Modal analysis
Power train
Radiation efficiency
Road wheel
Speech intelligibility
Sprocket wheel
Support roller
Track
Vibration
Window analysis

Custodian:
Army-MI

Review activities:
Army-AT, MD

Preparing activity:
Army-MI

(Project HFAC-A024)

STANDARDIZATION DOCUMENT IMPROVEMENT PROPOSAL

INSTRUCTIONS

1. The preparing activity must complete blocks 1, 2, 3, and 8. In block 1, both the document number and revision letter should be given.
2. The submitter of this form must complete blocks 4, 5, 6, and 7.
3. The preparing activity must provide a reply within 30 days from receipt of the form.

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I RECOMMEND A CHANGE	1. DOCUMENT NUMBER MIL-HDBK-767(MI)	2. DOCUMENT DATE *(YYMMDD)* 930917

3. DOCUMENT TITLE
Design Guidance for Interior Noise Reduction in Light-Armored Tracked Vehicles

4. NATURE OF CHANGE *(Identify paragraph number and include proposed rewrite, if possible. Attach extra sheets as needed.)*

5. REASON FOR RECOMMENDATION

6. SUBMITTER

a. NAME *(Last, First, Middle Initial)*	b. ORGANIZATION	
c. ADDRESS *(Include Zip Code)*	d. TELEPHONE *(Include Area Code)*	7. DATE SUBMITTED *(YYMMDD)*

8. PREPARING ACTIVITY

a. NAME US Army Missile Command	b. TELEPHONE *(Include Area Code)* (1) Commercial 205-876-6980 (2) DSN 746-6980
c. ADDRESS *(Include Zip Code)* ATTN: AMSMI-RD-SE-TD-ST Redstone Arsenal, AL 35898-5270	IF YOU DO NOT RECEIVE A REPLY WITHIN 45 DAYS, CONTACT: Defense Quality and Standardization Office 5203 Leesburg Pike, Suite 1403, Falls Church, VA 22041-3466 Telephone (703) 756-2340 AUTOVON 289-2340