NOISE CONTROL HANDBOOK FOR DIESEL-POWERED VEHICLES

R. J. Damkevala
J. E. Manning
R. H. Lyon

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7. Author(s) R.J. Damkevala, J.E. Manning, R.H. Lyon

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16. Abstract

This handbook has been prepared with the intention of assisting the truck fleet operator and the independent truck owner/operator in understanding and diagnosing noise problems and in selecting retrofittable components to lower truck exterior and interior noise levels. The handbook includes procedures for identifying and evaluating major truck noise sources, considerations for selection of acoustic materials, procedures for minimizing exhaust, intake and cooling fan noise, and methods for the minimization of in-cab noise levels. Appendixes give standard noise measurement procedures, muffler and intake filter selection data, cooling system design considerations and a list of known manufacturers of acoustic materials.

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Noise, Motor Vehicle Noise, Truck Noise, Engine Noise, Exhaust Noise, Intake Noise Mufflers, Silencers, Cooling System, Fan Noise

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PREFACE

In preparing this handbook, the authors have received extensive assistance from several Cambridge Collaborative staff members. In particular, Mr. J. Masiak prepared many of the graphs and figures appearing in Section 4, and Mr. J.D. Brito compiled the tables for mufflers and intake filter selection appearing in Appendix IV. The staff of the following muffler, engine and truck manufacturers were also of invaluable service:

- Stemco Manufacturing Co., Inc., Longview, Texas
- Donaldson Co., Inc., Minneapolis, Minnesota
- Cummins Engine Co., Columbus, Indiana
- Detroit Diesel Allison Div., Detroit, Michigan
- International Harvester Co., Fort Wayne, Indiana

We are also grateful for the many practical comments and suggestions as to the format and contents of this handbook that were received from the technical monitor of this contract, Mr. Robert L. Mason, and from other Department of Transportation staff, especially Mr. W.H. Close.

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by city and state governments, in general specify maximum sound levels as measured in decibels on the A-weighted scale, during different operations. Most of these regulations specify noise levels at 50 ft. from the vehicle. Several are based on an acceleration test procedure described in SAE Recommended Practice J366a (or its recent revision, J366b). Figure 1.1 summarizes the more restrictive noise laws that have been adopted (as of January 1973) pertaining to the sales of new highway vehicles of over 6000 GVW.

With the passage of the Noise Control Act of 1972, the Congress has authorized the Environmental Protection Agency (EPA) to set Federal Noise limits. In addition to protecting public health, the regulations will reflect the degreee of noise reduction achievable through the application of best available technology, taking into consideration the cost of compliance. Table 1-1 summarizes the EPA proposed regulation which is currently under final consideration.

TABLE 1-1

EPA PROPOSED NOISE LIMITS FOR VEHICLES 10,000 LBS. GVW AND OVER [1.1]*

Operation Condition	Noise Limit+
Less than or equal to 35 mph	86 dB(A)
Above 35 mph	90 dB(A)
Stationary run up test	88 dB(A)

Subpart B of this regulation [1.1] which details the Interstate Motor Carrier Operations Standards is reproduced in Appendix I.

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*Numbers in brackets [ ] refer to the list of references at the end of each section.

+Measured at 50 feet from centerline of vehicle travel lane.

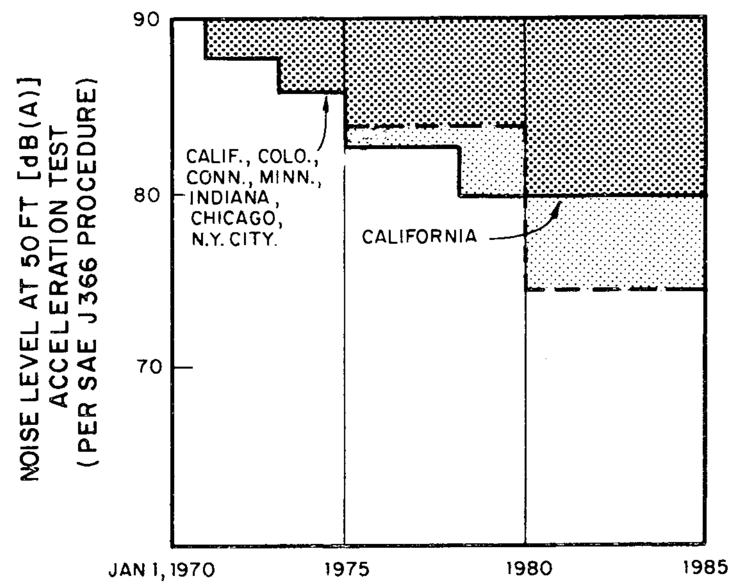

FIG. 1.1 SUMMARY OF PRESENT STATE AND LOCAL NOISE LAWS FOR THE SALE OF NEW TRUCKS 6,000 lb GVW AND OVER

portation. Three major truck manufacturers are conducting the program in an attempt to reduce the overall noise levels of new trucks. The goal is a noise level of about 75 dB(A) measured according to the SAE J366a standard test method. Results indicate that this goal can be attained using an engine enclosure, a modified and thermostatically controlled fan system and an improved muffler. The increase in production cost is somewhat compensated for by decreased fuel consumption due to improved efficiencies of the cooling system. Test vehicles are presently undergoing fleet service trials.

It may be noted that buses are generally more quiet than trucks mainly due to an enclosed engine compartment and the use of larger mufflers.

The in-cab noise levels are Federally regulated from the viewpoint of occupational noise exposure criteria. The Bureau of Motor Carrier Safety has prescribed that the interior sound level at the driver's seating position must not exceed 90 dB(A) with the vehicle stationary and the engine running at its maximum governed speed [1.2]. This regulation applies to all motor vehicles manufactured on and after October 1, 1974; on and after April 1, 1975 the regulation applies to all motor vehicles, regardless of date of manufacture. The SAE Recommended Practice J336 for in-cab noise levels specifies a more complicated procedure requiring noise measurements in octave bands while the vehicle is accelerating. Much effort in correlating different test procedures is presently under way [1.3] and it is hoped that future noise regulations will be based on simplified stationary vehicle tests. Therefore, whenever total vehicle or component noise measurements are being made, it will be advantageous to obtain noise readings using both stationary and accelerating vehicle test procedures. The accumulated data will save much labor in the future and give industry and government correlation information between the two test procedures.

relevant to heavy truck noise emissions are reproduced to Appendices IIA, IIB, and IIC.

REFERENCES

1.1. "Proposed EPA Noise Emission Standards for Interstate Motor Carriers", Federal Register, Vol. 38, No. 144, July 27, 1973.

1.2. "Motor Carrier Safety Regulations--Vehicle Interior Noise Levels", Federal Register, Vol. 38, No. 215, November 8, 1973.

1.3. "Correlation of Truck Cab Interior Noise to Currently Legislated (Walsh-Healy) Limits", Report prepared for the Motor Vehicle Manufacturers Association, Inc., by Wyle Laboratories, El Segundo, Calif., June 16, 1972.

2. DESCRIBING AND MEASURING TRUCK NOISE

This chapter presents some basic ideas about noise--what it is physically, how it is described in engineering terms, and how one goes about measuring it. We limit the discussion to the needs of this handbook, but more detailed material may be found in textbooks on sound [2.1] and in handbooks on noise control [2.2], [2.3].

2.1. What Sound Is

Sound is any pressure fluctuation that can be sensed by the human ear. Noise is sound, but there is an additional connotation that the sound is unwanted or undesirable. Therefore, one man's noise may be another man's symphony.

Sound also has a subjective dimension. Thus, the old saw about whether there is sound when a tree falls and no one hears it, is challenging our use of the same word for both the physical disturbance and the psychological sensation. We cannot change this dual usage, but we have to be aware of it.

From the viewpoint of truck noise work, we must be concerned with the amplitude of the pressure fluctuation and the rate of fluctuation. Measuring instruments used in noise work display the root-mean-square (rms) value of the amplitude of the pressure wave at the microphone on a logarithmic scale. This scale has the units of decibels (dB). Mathematically, the amplitude of the pressure fluctuation is expressed by a "sound pressure level" (SPL) given by

$$SPL = 20 \log p/p_{ref}. \quad \text{(units are dB)}$$

The pressure p is the rms value of the fluctuation averaged over a short period of time T. The quantity p_{ref} is a reference pressure.

of atmospheric pressure. To avoid such small numbers in calculations, a unit of pressure, the micro pascal (μP) is used which is a hundred-billionth of an atmosphere.* Thus p_{ref} = 20 μP. The averaging time T for an SLM corresponding to ANSI S1.4-1971 is about 50 milliseconds (m sec) for the "fast" reading of an SLM and about 300 m sec for the "slow" reading.

The rate of pressure variation also affects how audible the sound will be. Generally, the higher frequencies, from 1000 to 4000 hertz (cycles per second) are more audible (or louder) than frequencies below 1000 hertz (abbreviated Hz). For this reason, a SLM also has built-in selectable filters; A, B, and C, whose pass-characteristics are as shown in Fig. 2.1. Filter (weighting network) A was designed to simulate the sensitivity of the human ear to sounds at different frequencies. It is much more sensitive to frequencies above 200 Hz than below, as is the ear. Filter C weights all frequencies more or less equally. Thus a difference in the A and C weighted sound levels can be used to indicate the relative amount of low frequency energy in the noise.

The "fast" reading of the SLM simulates the rate at which the ear can hear changes in sound level. Since sound levels do change quickly in many noise situations - a factory in which hammering or other transient operations are prevalent, for example - one must be concerned with this effect. On the other hand, if one has an engine turning at constant rpm, the small fluctuations that would occur with the "fast" setting may be averaged out by switching to the "slow" reading, for a simpler and more accurate meter reading of the sound level.

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*This unit of pressure is to be preferred over the older units: dyne/cm^2 or Newton/m^2. 1 μP = 10^{-5} dyne/cm^2 = 10^{-6} N/m^2 so that 20 μP = .0002 dyne/cm^2.

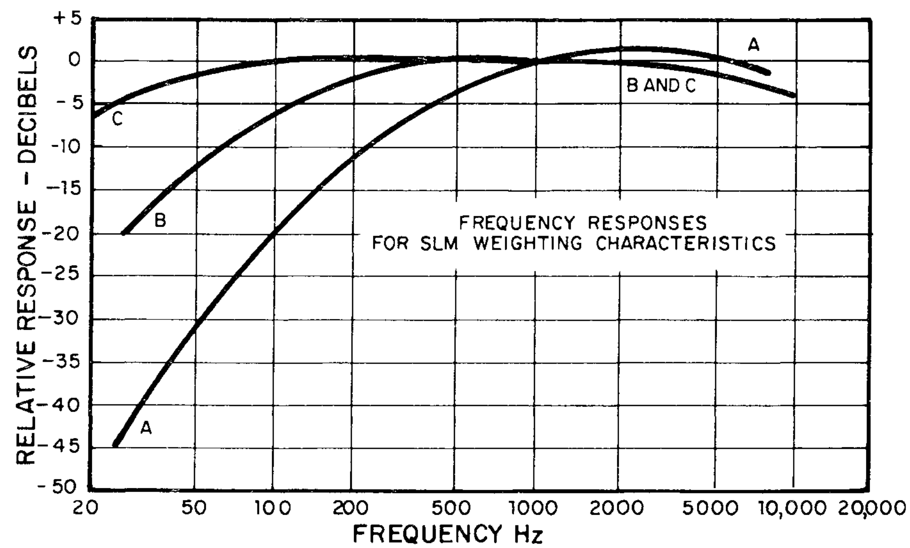

FIG 2.1 FREQUENCY-RESPONSE CHARACTISTICS IN THE AMERICAN NATIONAL STANDARD SPECIFICATION FOR SOUND-LEVEL METERS ANSI-S1.4-1971

2.2 How Sound is Produced

Sound is produced by the unsteady injection of heat and fluid mass into an otherwise quiescent atmosphere, the vibration of surfaces and the flow of air over airfoils and bluff obstacles. All of these effects are important in truck noise. The exhaust flow injects fluid and heat into the surroundings, and since the flow is not steady, noise is produced. The purpose of the muffler is to reduce the pulsations in flow and therefore, the noise. Muffler performance is discussed in Chapter 4.

Noise is also produced when vibrating surfaces cyclically compress the adjacent air. Such surfaces are plentiful in a truck; they are the casing of the engine, the walls of the muffler, exhaust pipe and tailpipe, the casings for the differential and transmission, and the sheet metal structure of the vehicle itself. Generally, larger surfaces radiate more than smaller surfaces, given equal vibration amplitudes. Treatment for reducing the noise usually consists of reducing the vibration level of the structure, or isolating the vibration from the air by a barrier layer. Vibration reduction is accomplished by applying dissipative (damping) material to the structure. Isolation of vibrations from the air by a barrier layer is accomplished by applying a combination of a spongy, resilient layer next to the structure, covered by a "massy" layer of metal or heavy canvas or plastic. Descriptions of these methods of noise reduction are contained in Section 3.2.

The most important air flow noise source in a truck is the fan. The noise is produced by the flow of air over the fan blades. The level of noise depends on fan configuration, speed, and on the general flow conditions external to the fan-radiator combination. The time varying blade forces during rotation of the fan and the impact of air on the engine block and other obstacles can be directly related to sound radiation. The way to reduce this component of noise is to reduce the fluctuating forces by reducing the fan speed. Fan noise is discussed in Chapter 6.

The sound level measured at any particular location depends on the nature of the noise source and the source-receiver geometry. A great deal of technical work has been expended on ways of predicting noise at a distance from knowledge of sound levels very close to the source or from the sound power emitted by the source. The factors that affect the sound level at a distance are

(1) The directivity of the sound source, determined by the physical nature of the source and the number and shape of nearby reflecting surfaces. The inherent directivity of fan noise for example is very strongly fore and aft, but the presence of the engine and the engine compartment will alter this directivity and may reduce radiation to the rear.

(2) The distance from the source to the microphone. This is typically 50 ft. for truck noise measurements, but may be less for measurements of noise at roadside or more for community noise measurements. When the measurement distance is greater than the size of the source (we may take this to be the tractor section of the truck), the rule is very simple: reduce the sound level by 6 dB for every doubling of distance from the source. A truck that produces a peak noise level of 80 dB at 25 ft. can be expected to produce a peak level of 74 dB at 50 ft.

(3) All other factors that reduce the sound level are lumped together into "excess attenuation", and include absorption of sound in the air, the effects of ground reflection, which reduce sound levels at low frequencies and augment them slightly at higher frequencies, and the scattering and obstructing effects of land profile, trees, houses, and the like.

2.3 How People Hear Noise

The study of the response of humans to sound is called psychoacoustics. One part of this field is concerned with the relation between the physical measure of sound amplitude (sound level) and the subjective impression of loudness. From many studies, it has been found that the ear does not find all frequencies equally loud for equal sound pressure levels. For fairly quiet sound levels (residential neighborhood) the ear sensitivity curve is about that shown as "A" in Fig. 2.1. At quite high levels, the level of a noisy factory, for example, the sensitivity curve flattens and approaches the "C" curve of Fig. 2.1.

When the sound level meter was first developed, it was thought that three filter characteristics, A, B, and C would be needed to approximate hearing sensitivities at different levels of intensity. However, a series of studies have shown that hearing damage potential, annoyance, and speech interference are better measured by the "A" filter reading alone. Consequently, most environmental noise measurements including the measurements of truck noise, are expressed in terms of A-weighted sound levels.

In addition to the question of relative loudness of different frequency components, there is also the question of how the subjective impression of loudness increases as the sound level increases. Experimental studies of the change in loudness with sound pressure level show that at sound levels of interest in truck noise problems, the loudness doubles when the sound pressure level increases by 10 dB. [If we are measuring A-weighted sound pressure levels, this increase would be 10 dB(A).] Because of the logarithmic nature of the decibel, two equal sources increase the sound level by 3 dB. To double the subjective impression of loudness, ten noise sources of equal level would be required.

2.5 Taking Noise Data

Truck noise data is taken to find the noise emitted by the various components of the truck, and in addition, to assess the overall sound radiated by all components to determine if the noise level is in accord with some standard. A sound level meter (which includes a microphone) is the usual instrument used to obtain noise data. This instrument has been discussed previously. Some important considerations for its use are given in the following paragraphs.

If it is desirable to preserve the noise data for more detailed analysis later on, a magnetic tape recording may be made. Recording of data has may pitfalls, however, that must be avoided by attending to details of background noise, dynamic range, and calibration. Several excellent books on noise measurement exist and can be consulted [2.4, 2.5].

2.5.1 Background Noise

As a practical matter, noise measurements are never carried out in the complete absence of noise from sources other than the one being measured. If the background A-weighted sound level with the source turned off is 70 dB(A) for example, and the level rises to 76 dB(A) when the source is turned on, we know that the actual level of noise due to the source alone is somewhat less than 76 dB(A) - but how much?

The answer to this question is found from the "background correction graph", shown in Fig. 2.2. To use it, we first note the difference between the measured level and the background (6 dB in the example above) and enter the graph along the horizontal axis. Then moving vertically, we find the amount that the measurement has been "contaminated" by the background noise (1.4 dB in our example). Thus, the true value of the sound level is the measured value less the correction (76-1.4 = 74.6 dB in the example).

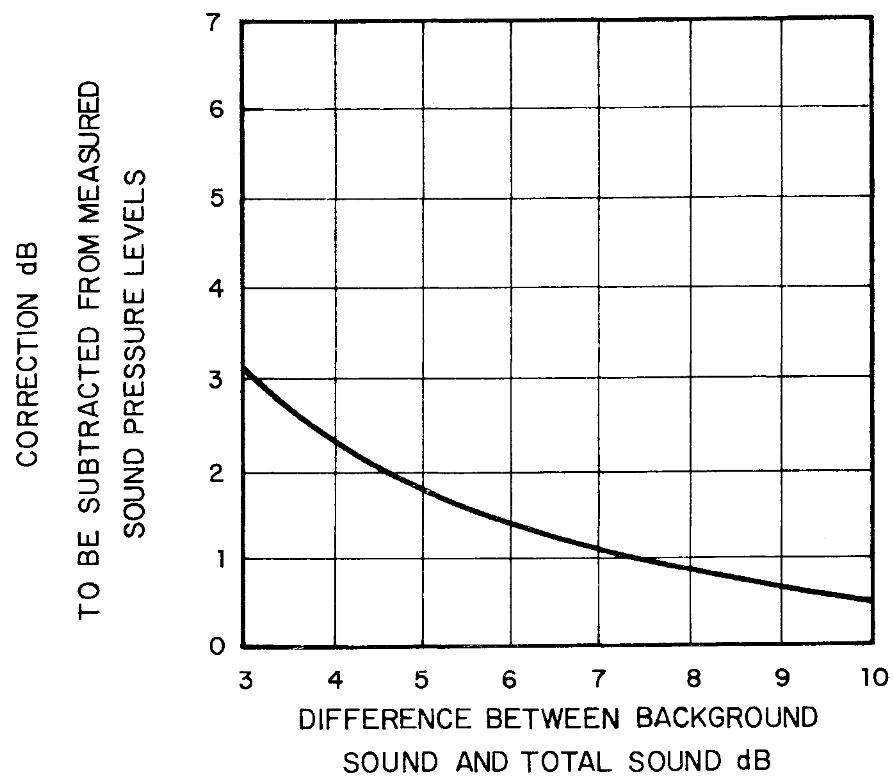

FIG.2.2 CORRECTION FOR BACKGROUND SOUND

Note from Fig. 2.2 that if the background noise is 10 dB below the measured level, there is still 0.5 dB of background contamination of the measurement. Most text books on noise suggest trying to achieve a difference between overall and background levels of at least 10 dB before data is taken. This is a desirable situation that is not always achievable.

When background noise is less than 3 dB below the measured level, a reliable correction is no longer possible. The reason for this is that the correction needed will increase rapidly with decreasing difference, and small errors in measurement will give much larger errors in the background correction. This point is illustrated later in Section 3.6 and Figure 3.5.

2.5.2 Dynamic Range

Dynamic range of a measurement refers to the difference between the maximum and minimum sound levels to be measured in a given situation. The lowest background levels that are likely to be encountered in such a situation are 20 to 30 dB, while the measured sound levels very near the noise sources may be as much as 100 dB. Thus, the equipment must operate without distortion over a range of about 80 dB.

Generally, the microphone has sufficient dynamic range to encompass the range of expected sound levels. The electronic amplifiers and processing circuits have a more limited range. In addition, in order to keep from distorting the measurement, the maximum expected level should be kept 10 dB below the upper limit of the instrumentation; which reduces the effective dynamic range even more. The effect of the limited dynamic range is to raise the effective background to the "noise floor" of the instrumentation. Sound level meters get around this problem by the use of variable gain circuits and attenuators. A tape recorder is unlikely to have a useful dynamic range of more than 40 dB. Hence, the signal level into the tape recorder also has to be varied to avoid this problem.

2.5.3. Calibration

The term "calibration" is used in two ways in noise work. The first relates to the accuracy and interpretation of the reading of our instrument. If a sound pressure level of 80 dB at the microphone produces a reading of 80.5 dB on the face of the sound level meter, the SLM is said to be 0.5 dB out of calibration. Microphones, accelerometers and other transducers are "calibrated" when their output voltage for a prescribed pressure, acceleration, etc. is known.

The second use of "calibration" relates to an entire instrumentation set-up, from transducer to recorder to read out. A known sound level is introduced at the microphone prior to and at the end of each data run. Then, when the data is examined, this known sound level can be related to the rest of the data taken during the run. Similarly, if the read-out of the calibration signal for the two ends of the run is different, then either the instrumentation is faulty, or changes in attenuator settings have not been kept track of properly. Clearly, the individual elements of the instrumentation chain do not have to be "calibrated" in the sense of the preceding paragraph in order that the system calibration being discussed here be valid.

2.6 Processing Noise Data

The processing of noise data means applying and observing the output of various instruments to obtain information about the noise signal. In Section 2.1, we discussed the frequency weighting operation of the SLM, the rms and averaging operation, and meter display. These are the elements of most noise data processing. The things that change from one situation to another are the filter characteristics, detection (i.e., taking a true rms or some approximation), averaging times, and manner of display. In the following paragraph we discuss these things, but we must leave many of the details to text books and to the literature from the instrument manufacturers.

2.6.1. Filtering Noise Data

We have already discussed the A, B, and C-weighting filter networks of the SLM. For most measurements concerned with compliance to noise standards, the simple A-weighting filter network may be used. Often, however, we may want to know more about the way that the noise energy is distributed in frequency. For such work, we may use one of the standard octave or third octave filter sets, or possibly, an even narrower band system.

Octave and third octave band filters are "constant percentage" bandwidth; i.e., their frequency limit is a constant percentage above their lower frequency limit. For octave bands, the upper limit is twice (100% more than) the lower band limit. For third octave bands, the upper limit is 26% more than the lower limit. Center frequencies of these bands are the geometric mean of the upper and lower limits and are shown in the diagram of Figure 2.3. An example of truck noise data filtered in octave bands is shown in Figure 2.4.

A typical noise "spectrum", presented in octave bands, will involve 8 numbers (the band levels from 63 to 8000 Hz). A third octave spectrum over the same frequency range will require 24 numbers. Thus, the price one pays for the more detailed spectral information is more time spent in collecting and interpreting the information gathered. Unless one is useing the spectral information specifically to identify some noise mechanism, the increased effort in handling all of this information is usually not justified.

The noise in each different frequency band contributes independently to the overall mean square pressure at the microphone. The octave (or third octave) band levels can be combined to give the overall level using the chart shown in Figure 2.5. The levels are combined, two at a time, until all the levels have been combined, smallest values first, and selecting values as close to each other as possible.

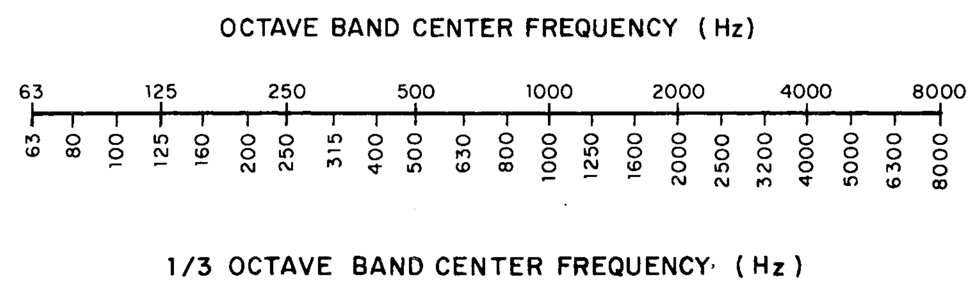

FIG 2.3 CENTER FREQUENCIES OF OCTAVE AND ONE - THIRD OCTAVE BANDS

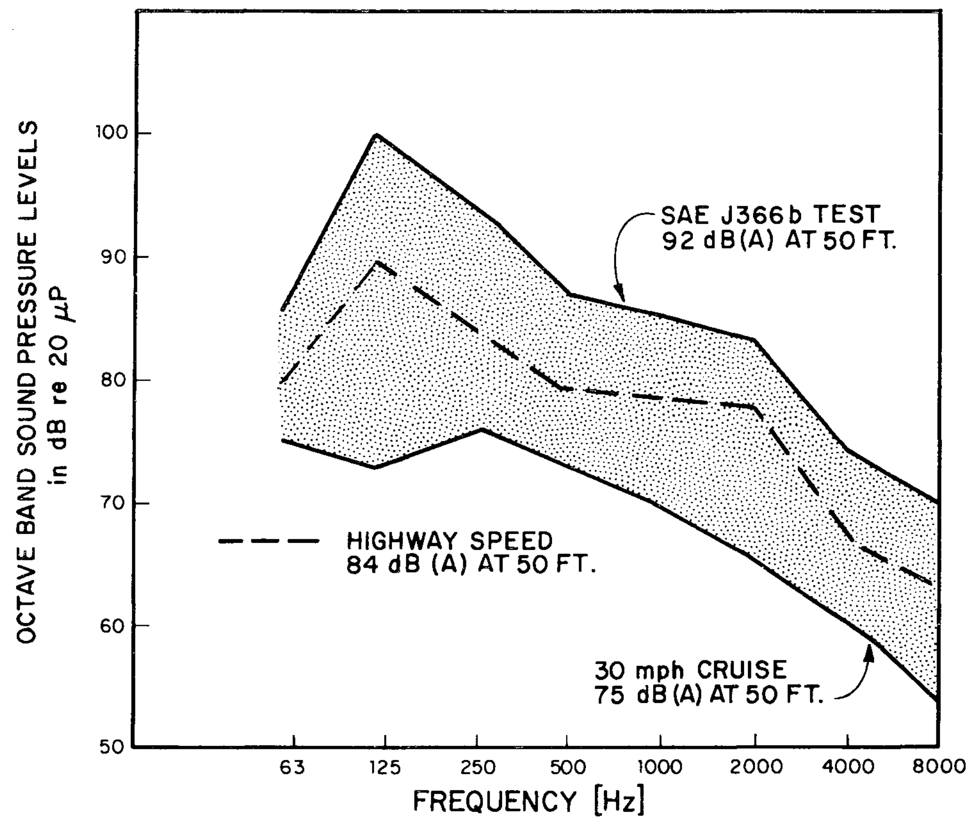

FIG. 2.4 TYPICAL EXAMPLE OF DIESEL TRUCK NOISE
REF. 2.6

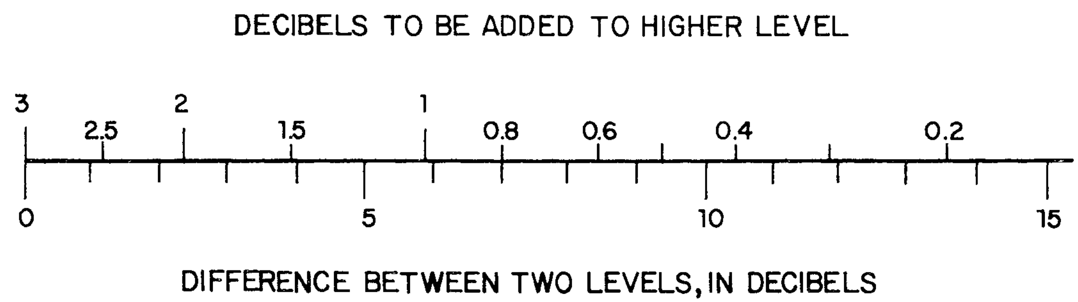

FIG. 2.5 LINE CHART FOR THE ADDITION OF SOUND PRESSURE LEVELS

To illustrate, consider the four levels; 72, 73, 76, 78. Combining 72 and 73 first, the difference is 1 dB so that according to Figure 2.5, the combined level is 2.6 dB more than the higher (73), to give 75.6 dB. Then combining 75.6 and 76, the difference is 0.4 dB so we add 2.8 to 76 to get 78.8 dB. The difference between 78 and 78.8 dB is 0.8, so we add 2.7 dB to 78.8 to get an overall level of 81.5 dB.

The chart of Figure 2.5 may be used to combine different frequency band levels, or any independent sound pressure levels. For example, the four levels in the preceding paragraph may be the independent contributions from four different noise sources -- the engine, fan, exhaust, and air intake. In this case, the measured level at the microphone from these four sources would be 81.5 dB.

2.6.2. Tape Recording

Tape recording is used when the storage of data for later retrieval in analog form is desirable. Since such recording reduces the dynamic range and fidelity of the original signal, and increases the time and costs required to obtain the data, tape recording should be employed only when unavoidable. The use of home or entertainment type magnetic tape recorders is not recommended for taping truck noise. A good discussion of the problems associated with recording noises which have large low frequency components (see Figure 2.4) will be found in reference [2.7].

Major considerations when taping noise data are maintaining calibration and keeping the noise signals within the dynamic range of the recorder. The first is achieved by "end-to-end" calibration, i.e., recording a signal for a known sound pressure at the microphone and keeping track of all attenuator settings. SAE Recommended Practice J184 "Qualifying a Sound Data Acquisition System" gives a method of determining the frequency response characteristics for the entire measurement system except the microphone.

1. The "fast" meter response should be used for varying sounds such as truck pass-by levels.
2. Use a wind screen supplied by the manufacturer of the sound level meter whenever making measurements in moderate wind. Measurements should not be made if wind speed exceeds 12 mph. The measured levels should be corrected as per the calibration supplied by the manufacturers.
3. Hold the instrument as far away from the body as possible. Use a tripod when available. The microphone should preferably be mounted on a gooseneck to reduce the effects of the instrument and operator on the measurement. An extension cable, along with a preamplifier, may be used if preferred. Follow the instruction manuals precisely.
4. When other intermittent high noise sources are present in the background, e.g., aircraft, the output of the SLM should be monitored by headphones to ensure that these noises are excluded from the data.
5. Always calibrate the instrument with a standard calibration source before starting measurements or after any interruption.

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3. MEASUREMENT OF NOISE LEVELS CONTRIBUTED BY VARIOUS COMPONENTS OF THE TRUCK

3.1. Introduction

For trucks and buses of current configurations, the important noise sources are the engine exhaust, engine, cooling fan, air intake system and tires. The relative contributions of these sources vary between large margins, as evidenced in Table 3-1. Tire noise becomes important at high speeds and is usually the dominant noise source at highway speeds. The SAE Standard J366b, for the measurement of exterior sound level of heavy trucks and buses is the standard test procedure used by the industry and several government organizations to determine new vehicle noise levels. This test procedure is designed to exclude tire noise as much as possible by measuring maximum noise level at speeds below 35 mph and with the truck unloaded. It is reprinted in Appendix II.

Table 3-1

Examples of Contributing Levels of the Sources of Noise from Diesel Trucks

Source	Contributed Noise Level dB(A) -- SAE J366b Procedure			
	Truck #1	Truck #2	Truck #3	Truck #4
Exhaust	86	79	82	77
Air Intake	74	67	75	67
Cooling Fan	75	91	82	79
Engine Mechanical	78	83	79	78
Transmission	75	70	67	68
Chassis	69	70	73	71
Total	87.5	92	86	83

Each contributing source listed in Table 3-1 can be broadly defined as follows:

1. Exhaust noise includes the noise produced by the exhaust gases at the tail pipe exit, noise from the muffler shell and piping and leakage of the exhaust system components (muffler, exhaust pipe and tail pipe).

2. Air intake noise includes the noise from the air inlet, the air cleaner shell and ducting and leakage of the air intake system components (air cleaner, silencer, piping). The blower and turbocharger noise are considered as portions of the engine mechanical noise, although the presence of such devices generally lowers the intake noise.

3. Cooling fan noise (as stated in Table 3-1) includes the various noise sources of the cooling system. Although the predominant noise source is the fan, the shrouds, radiators, shutters and grills affect the noise produced by the cooling system.

4. The mechanical radiation from the engine is associated with the combustion process and the mechanical components of the engine. These noises are the result of vibration of the engine structure, covers and accessories. The noise from fuel injection pumps and the valve train is generally included in this source.

5. Transmission noise is generally the result of casing vibration and gear whine. In general, the transmission is not a major noise source, although in some cases, it may become important after the other sources have been treated.

6. Chassis noise (as shown in Table 3-1) is the residual noise remaining when the truck is coasting by at 30 mph with the engine shut off and the transmission in neutral. This noise is due to tires, the creaks and groans of the chassis, body, etc. It is the lower limit of noise that a particular truck design is capable of, and is an important parameter in estimation of contributed noise levels of the other components of a truck.

Adding sound levels contributed by each source can be accomplished by successive use of Fig. 2.5 or the chart in Appendix III. In the case of Truck #1, the exhaust contributes 86 dB(A) out of the total level of 87.5 dB(A) i.e., the exhaust contributes more than half of the total noise energy at the receiver. Truck #2 noise is dominated by fan noise. In the case of trucks #3 and #4, no dominant single noise source exists, therefore reductions in exhaust, fan and engine mechanical noise should be considered.

Lowering the exhaust noise of Truck #1 from 86 dB(A) to 81 dB(A) will lower the total noise about 3dB(A). Further lowering of the exhaust noise to 76 dB(A) will reduce the overall noise by another 1 dB(A) only. As shown in Figure 3.1, the point of diminishing returns is reached very rapidly. Since the combined level of the remaining sources is 82 dB(A), quieting the exhaust alone will not lower the total noise below 82 dB(A). In the case of Truck #2, truck noise may be reduced by lowering the cooling fan noise level as shown in Figure 3.1. A large reduction in the cooling fan noise is required before the remaining noise sources start having any effect on the total noise level.

For any serious noise reduction program, therefore, the noise levels contributed by each of the major sources need to be determined at the very beginning, so that effort and cost are not expended in quieting sources that will have lesser effect on the overall vehicle noise level. Quantitative knowledge of contributed source levels also provides an information base from which to predict the maximum levels of individual noise sources which may be required to meet present and future noise regulations. These predicted individual noise levels can then be used as goals to be reached for the noise levels contributed by each of the major sources.

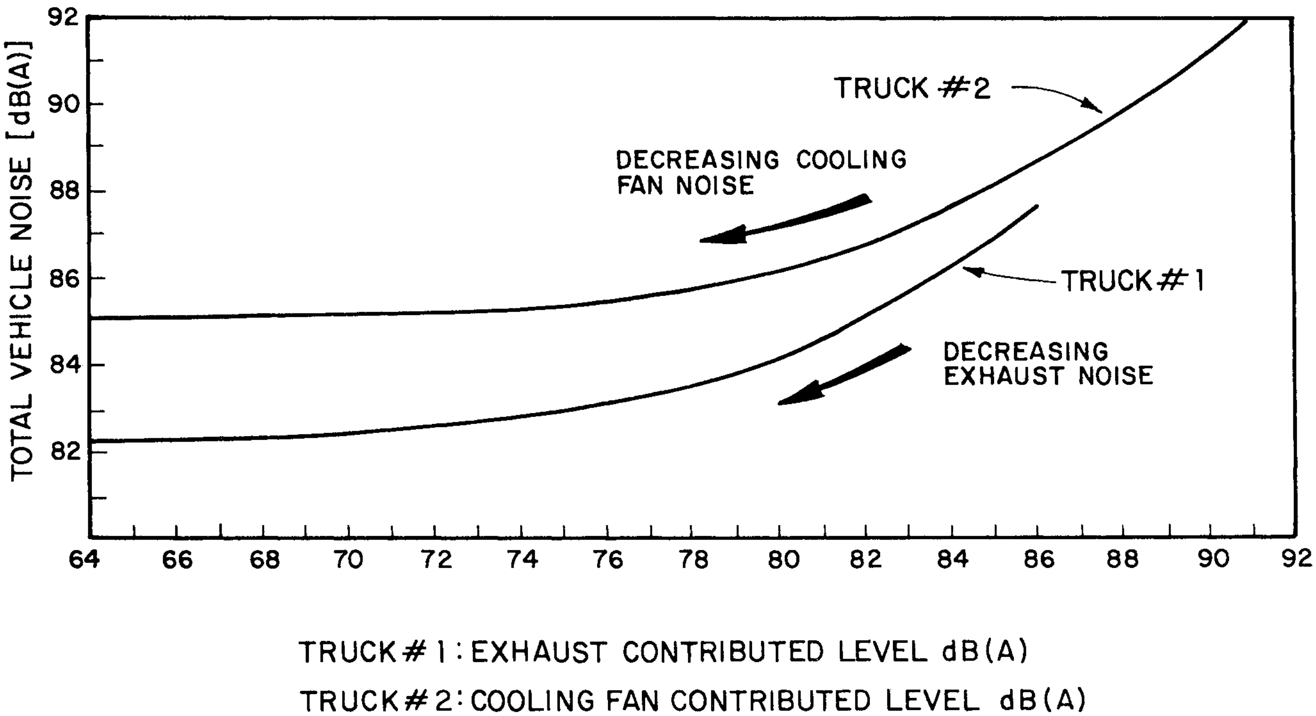


FIG 3.1 EFFECTIVENESS OF REDUCING NOISE FROM ONE VEHICLE COMPONENT ONLY

Acoustic materials for covering the vehicle components as required by the procedure which will be described in Section 3.5 should provide sound absorption as well as impede the transmission of sound energy. Materials providing sound absorption are known as absorptive materials. Materials that effectively impede sound transmission are known as barrier materials. Absorptive materials are in general unsatisfactory for applications where sound transmission has to be minimized and vice versa.

A layer of material in the path of a sound wave will either be set in motion by pressure of the incident sound wave, or if it has porous structure, will allow continued travel of the waves within the body of the material. If these processes result in the dissipation of energy in the material or in the transmission of sound waves into free space on the opposite side of the layer, the waves reflected back toward the source will have less energy than the incident waves, and we say that part of the incident energy has been absorbed by the material. The fraction of the energy absorbed depends on the nature of the material itself. its thickness, the frequency of the sound and the angle at which the sound wave strikes the surface of the material. Materials with fine open pores have good absorption properties. Since installation space limits the use of thick materials, the effectiveness of most absorptive materials in this application is limited to frequencies above 300 Hz.

Absorptive materials should be used in conjunction with a barrier material when transmission of sound is to be limited. Since our primary purpose is to limit the transmission of sound energy from the component to the microphone (or observer), the barrier should ideally reflect all the energy back towards the source. This can be accomplished by building an enclosure around the component. Unfortunately, this results in a build up of sound levels on the source side until the transmission of sound to the outside is equal to its value without the enclosure. Using an absorbent material on the inside increases the effectiveness of the barrier by preventing this build up. Dense, limp, non-porous materials like lead, asphalt composition and hard rubber are good barrier materials. Doubling the weight of the barrier will sandwich type construction with two impervious barrier layers separated by a compliant material makes a better sound barrier than a single layer construction.

Because of the increased sound levels inside an enclosure, even a small opening or crack in the barrier will severely limit its effectiveness. Packing the openings with glass fiber, foam, canvas boots, or other porous materials will not significantly improve the situation except at very high frequencies. Acoustically good seals can be made with heavy pliable putty-like materials. In the case when openings amount to 10% of the enclosure area, it can be shown that the noise reduction due to the enclosure will be limited to just 10 dB.

For covering the engine, transmission, etc., we recommend two layers of 1-inch open cell polyurethane foam with 1/32" lead sheet bonded to one side. Heavy cloth-back adhesive tape (also known as heating duct tape) or soldering the edges at short intervals can be employed to hold the covering in place. As far as possible, all openings should be covered with extra lead sheets held together with tape to form an air-tight seal.

For covering the intake filter, exhaust pipe, mufflers, stacks, etc., 3-1/2" thick building type glass fiber insulation may be used, with the foil on the outside. About 40 sq. ft. will be required. The material may be held together with wire or heating duct tape. Hot areas should be covered with mineral wool or several layers of asbestos paper to a thickness of 1 inch or so, before covering with the acoustic materials.

The effectiveness of acoustical materials will be reduced if oil or other liquids are allowed to saturate the pores. In addition, the fire hazard will be increased. The use of self-extinguishing materials is recommended. The selection of the foam will also depend on its ability to withstand the engine block temperatures without excessive deterioration. A list of known suppliers of the required acoustical materials appears in Appendix VII.

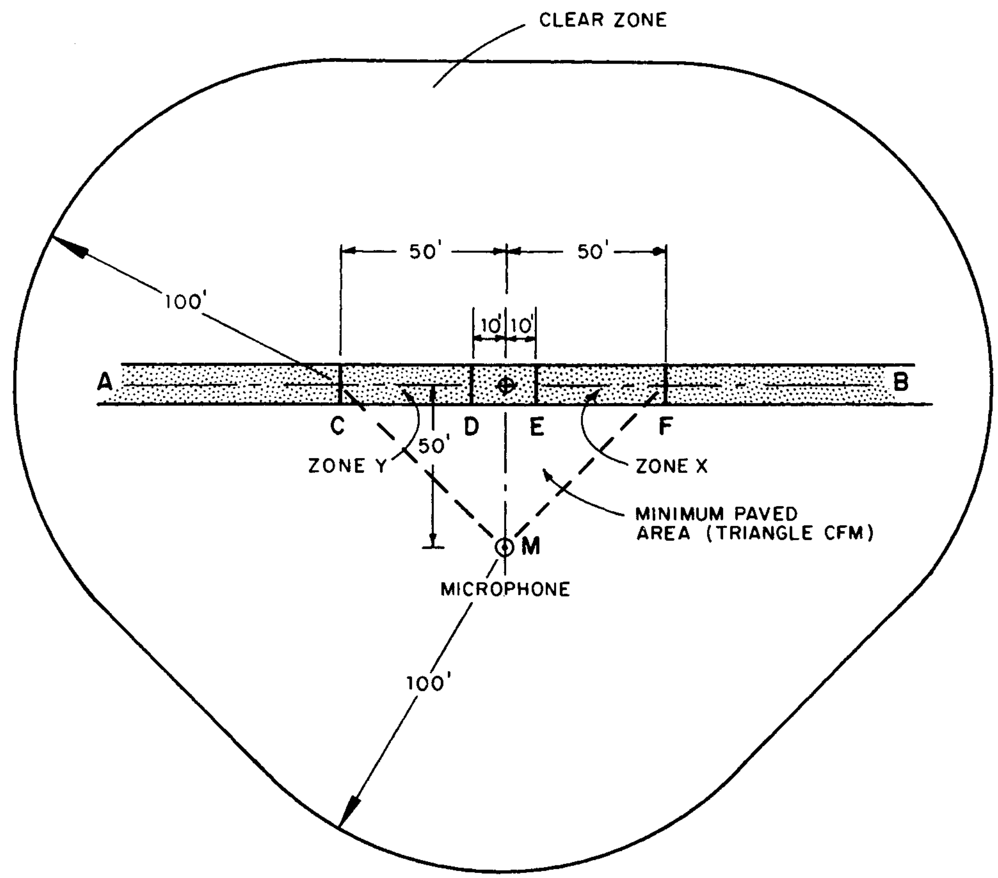
First of all, select the noise measurement procedure most appropriate for the application. Most noise regulations specify maximum levels at 50 ft., measured during acceleration according to SAE J366b (or the older SAE J366a) procedure. The proposed Interstate Motor Carrier noise standard specifies a stationary test procedure, as does the Bureau of Motion Carrier Safety (BMCS) regulation. For interior noise measurements the SAE Recommended Practice J 336 is well known. However, simpler procedures using the SAE J366b vehicle operation procedure (acceleration to 35 mph and maximum rpm) and noise measurement in the cab using the A-weighting only, have been employed (e.g. Ref. [7.3]) and can be used.

It is recommended that noise measurements be made using both the stationary and acceleration test procedure in each case. Whenever possible, cab interior noise measurement should also be made simultaneously with an exterior noise measurement . In the case of exterior noise measurement, readings should be recorded on both sides of the vehicle (using a bi-directional test site as described in Section 3.4). In all cases, the measurement should be repeated several times and the average of the two highest readings which are within 2 dB of each other should be used for analysis.

Once the procedures have been selected, the same procedures should be used on the same test site for all the tests described in Section 3.5.

3.4 Test Site

The SAE J366b (or J366a) procedure requires an asphalted or concrete roadway within a level open space free of reflecting surfaces such as parked vehicles, sign boards, buildings or hillsides located within 100 ft of either the vehicle path or the microphone. We suggest the use of a bi-directional test site as shown in Figure 3.2 to permit measurement of



VEHICLE TRAVELLING FROM A TO B SHOULD REACH
MAXIMUM RPM IN ZONE X

VEHICLE TRAVELLING FROM B TO A SHOULD REACH
MAXIMUM RPM IN ZONE Y

FIG 3.2 SUGGESTED BI-DIRECTIONAL TEST SITE FOR
EXTERIOR SOUND LEVEL MEASUREMENT

Possible effect of differences in terrain on the two sides of the roadway will also be eliminated. Pressure or light-interruption switches may be employed to provide definite signals when the truck crosses the end zone boundaries. If the entire area is seal coated asphalt or concrete surfaced, the measured levels will be slightly higher than typical roadside levels and as such will assure worst case conditions that offer assurance of compliance with regulations. Sites meeting only minimum requirements are relatively dependent upon seasonal variations in vegetation and hence the measurements will be somewhat inconsistent.

The stationary vehicle tests and cab interior tests should also be performed outdoors in a level open space free of large reflecting surfaces, although a marked test site will not be required.

3.5. Procedure for Determination of Contributed Levels

The procedure for determining the component contributions has been referred to as "window analysis". All major noise sources are temporarily quieted by covering with wrappings, removed or disconnected where practical or muffled. Each source is then uncovered or reconnected and the change in noise level measured to obtain a relative ranking of the noise sources. The procedure is summarized in Table 3-2 (Page 30). It has been field tested and used extensively by the Cummins Engine Co. [3.1] Columbus, Indiana, and by several vehicle and muffler manufacturers.

Preliminary Check

Performance test the vehicle to assure that the engine and fuel settings are set to manufacturer's maximum recommended fuel rate. This check is important for new vehicles that have not been in road service before testing. It is even more important for used vehicles because of possible hot-rod tuning (increased fuel pump/governor settings). Inspect the vehicle for proper exhaust and intake connections and fan position. Any features of the vehicle that differ from the standard production model should also be noted.

TABLE 3.2

NOISE SOURCE DIAGNOSIS PROCEDURE

Test (See Sec. 3.5)		Engine	Fan	Shutters	Intake System	Exhaust System	Transmission	Computa Obtaini buted
A	Untreated Baseline	Unwrapped	On	Open/Closed	Original	Original	Unwrapped	Magnitud indicate problem
B	Chassis Noise	Off, Unwrapped	Not Turning	Open/Closed	Original	Original	Unwrapped	
C	Treated Baseline	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	Minimum Level wi
D	Exhaust	Wrapped	Removed	Closed	Wrapped Auxiliary	Original	Wrapped	D-C= Exh buted No
C1	Treated Baseline	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	Should e
E	Intake	Wrapped	Removed	Closed	Original	Wrapped Auxiliary	Wrapped	E-C= Int buted No
C2	Treated Baseline	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	Should e
F1	Fan Baseline	Front Unwrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	
F2	Fan	Front Unwrapped	On	Open/Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	F_2-F_1 = buted N
G	Engine	Unwrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	G-C= En buted N
C3	Treated Baseline	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	Should
H	Transmis-sion	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Unwrapped	H-C= Tr Contrib Level
C4	Final Baseline	Wrapped	Removed	Closed	Wrapped Auxiliary	Wrapped Auxiliary	Wrapped	Should

*Note that restricted air flow caused by engine wrapping creates an unknown effect on the noise me this test.

below the lowest sound level to be measured. For example, if the chassis noise level is expected to be 70 dB(A), the background should be at 60 dB(A) or less. When ambient noise levels low enough to perform the baseline chassis noise tests are slightly exceeded, corrections can be applied to the data as shown in Fig. 2.2. Such data will be useful because the other test conditions produce higher levels of noise. However, the final configuration must be tested under the 10 dB lower ambient noise conditions.

A wind-screen should be used on the microphone at all times, but measurements may be made only when the wind velocity is below 12 mph. When using wind-screens, the manufacturer's calibration corrections, if any, should be observed.

The fuel tanks in the vehicle should be filled completely at the beginning of a test program. If the tanks must be refilled during the test, attempt to get fuel from the same batch as the prior fueling(s) and repeat the baseline tests to ensure that there has been no change attributable to fuel differences. A variation of up to 3 dB(A) has been observed between different batches of diesel fuel, corresponding to a variation in cetane number between 35 and 55.

The vehicle should be equipped with an accurate speedometer and engine tachometer, and should be in the "bob-tail" condition (without trailers) or otherwise unloaded if not a tractor. Care must be taken to prevent rattles, etc., from chassis and body components.

A. Baseline Noise Test

The vehicle should be in standard production condition. Measure exterior and interior noise levels using appropriate standard procedures with shutters open and closed.

B. CHassis Noise Test

The purpose of this test is to determine the lowest noise level that can be attained with all the engine-related noise sources effectively quieted. The measured sound level gives the contribution of the chassis, body, axles and tires to the overall noise level.

Accelerate the vehicle in standard production condition to approximately 35 mph. Determine a position on the test track where if the clutch is depressed and engine turned off, the vehicle will coast by the microphone at approximately 30 mph and the engine will have stopped soon enough to have no effect on the peak pass by sound level. When making measurements, shift into neutral at this location and shut off the engine. Record the dB(A) reading at 50 ft. according to SAE J366b, with the shutters open and closed. As noted earlier, it is desirable that the background noise level at the test site should be at least 10 dB(A) below the chassis noise level.

The test is not required if stationary test procedures only are to be employed.

C. Fully Treated Baseline Noise Test

(a) Fully Treating the Vehicle

All major noise sources should be completely covered. If after complete treatment, a new source becomes audible, it should be treated in a similar way. After complete treatment the measured noise level under acceleration testing should be no more than 3 to 6 dB(A) above the chassis noise level: every effort should be made to reach that goal. (In rare cases improper engine isolation may make it impossible to get within 10 dB(A) of the chassis noise level.)

Cooling Fan: Remove engine cooling fan if possible. If the installation prohibits removal of the fan, the fan belts should be removed and the fan restrained from moving by taping or blocking if necessary to prevent damage to the engine covering.

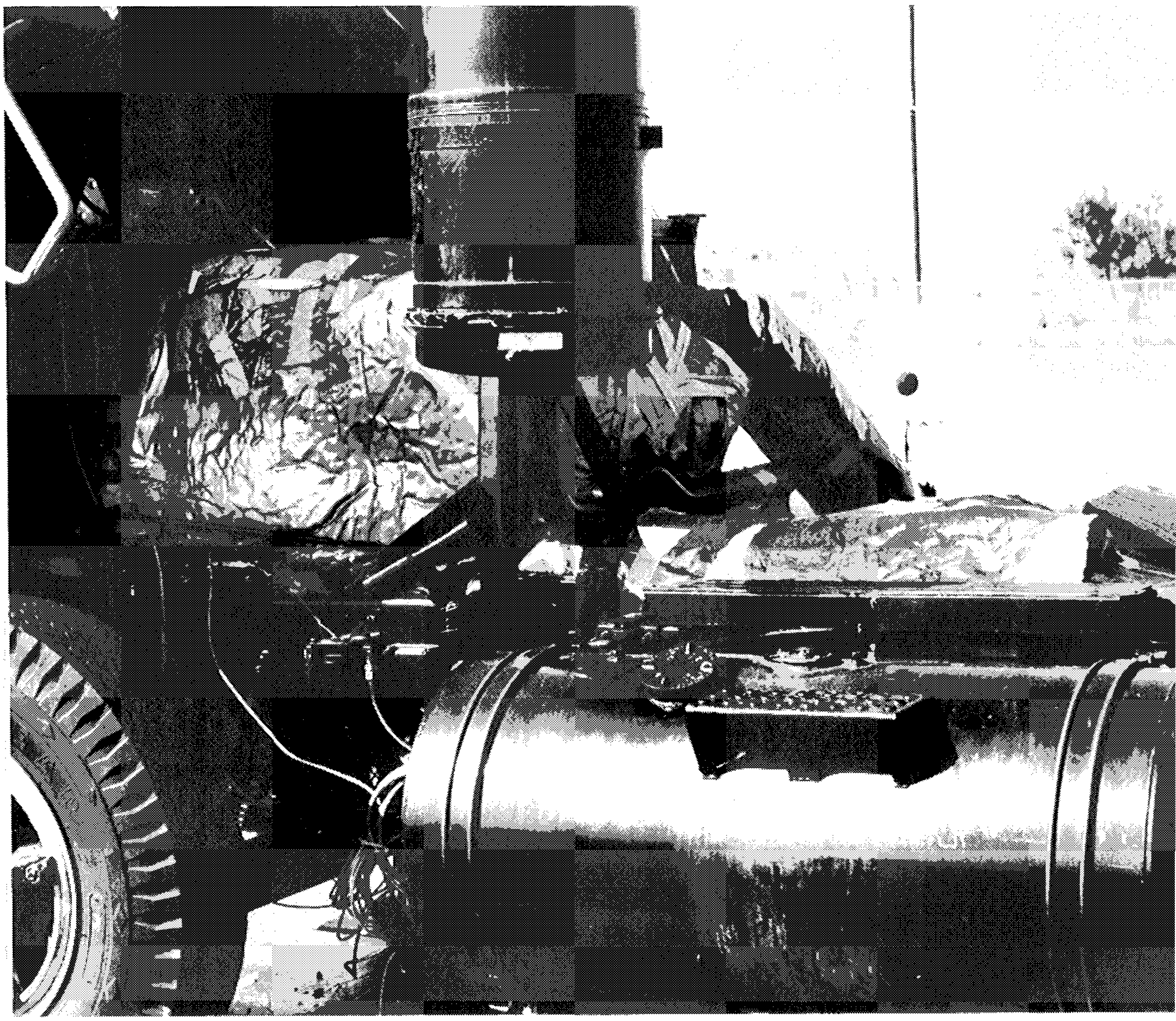
Transmission: Completely wrap the transmission in 1" thick open-cell polyurethane foam which has a lead sheet cemented on the outer side. It is permissible to use a separate uncemented lead sheet but care should be taken to seal all cracks and spaces. Pressure sensitive heating duct tape has been found to be suitable to hold the wrappings together. Alternatively, the edges can be soldered at short intervals. This wrap should terminate at the edge of the flywheel housing.

At least two layers of foam and lead should be applied alternately. Neat gap-free covering with lead is extremely important. 1/32 in. thick lead sheets have been found to be easy to work with and provide adequate attenuation. The materials have been discussed in Section 3.2 and suppliers listed in Appendix VII.

Engine: Carefully cover the exhaust manifold with heat-insulation such as 1" thickness of mineral wool or layers of 1/8" thick asbestos paper. Wrap the engine completely except for the front with two alternate layers of 1" foam and lead as in the case of the transmission. Special care should be taken to seal spaces around intake and exhaust pipes. The rear of the engine should be wrapped such that unwrapping the transmission does not necessitate unwrapping the engine. Now cover the front of the engine in such a way that the front covering can be removed without disturbing the rest of the covering. Note that the cooling fan and belts should be removed prior to this. Figure 3.3 shows a treated engine and transmission in a cab-over truck.

The treatment should not restrain the engine from its vibrations by constraining the mounts or by tightly packed lead sheets in confined spaces. In the cases that the wrapping has to go around the engine mounts or parts of the chassis, an extra layer of uncompressed foam should be used under the main covering.

Intake: The production air cleaner and intake piping should be removed from the engine. A new air cleaner selected from the charts in Appendix V C or other source should be wrapped in 3-1/2 inch thick foil-backed glass fiber insulation (the type used for home insulation), foil side out. The auxiliary system should be located so as not to interfere with the original installation. This will facilitate disconnecting and reconnecting to assess the noise contribution of the original air cleaner. Any rigid piping



COURTESY WHITE MOTOR CO.

FIG. 3.3 TREATED ENGINE AND TRANSMISSION

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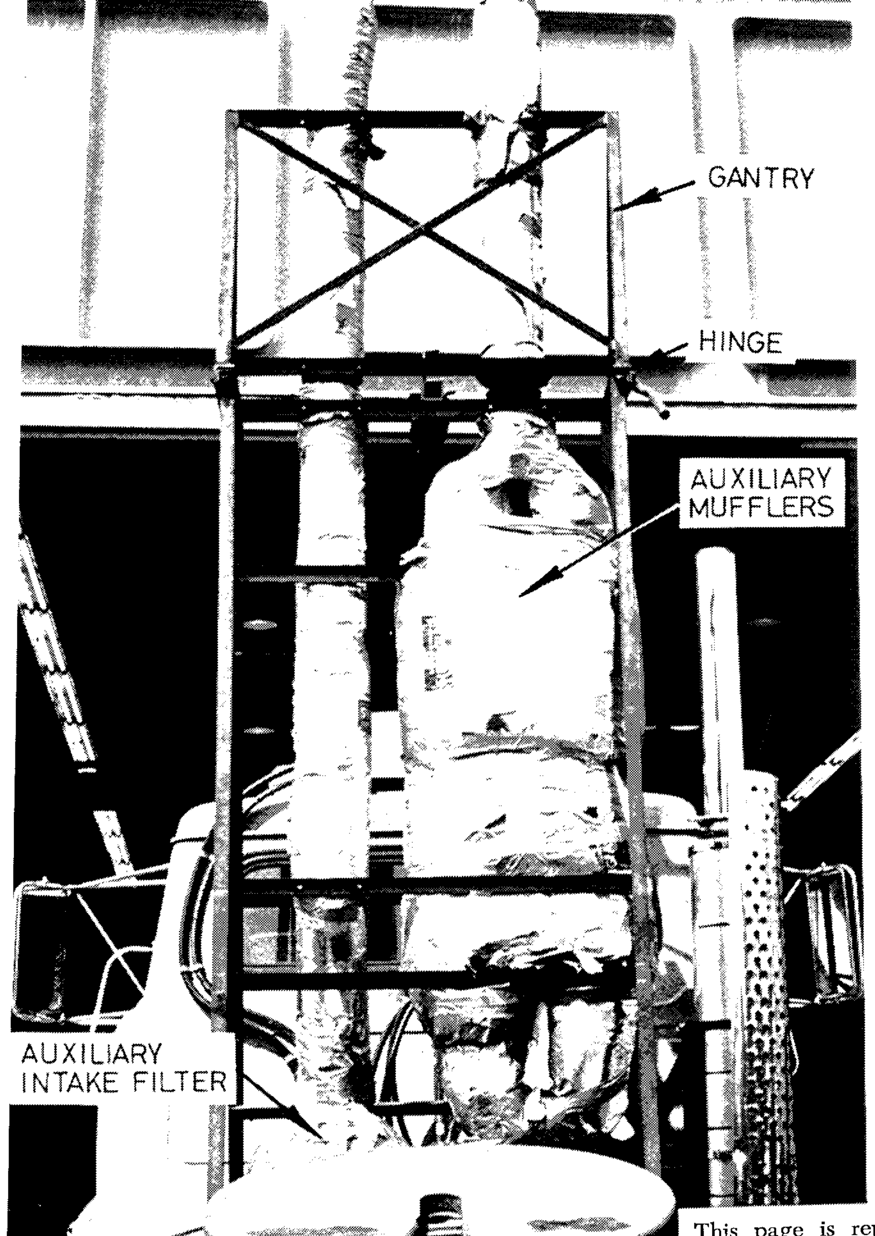
between the engine and this cleaner should also be wrapped with glass fiber insulation. Any flex-hose between the engine and this cleaner should be covered by a second, larger diameter flex-hose and wrapped if possible.

In order to minimize the intake flow noise, the auxiliary air cleaner should draw air through a long "snorkel". This rigid tube and all connections between it and the air cleaner should also be wrapped with glass fiber insulation and be securely mounted to the truck's chassis or a special truss or gantry mounted on the chassis. Fig. 3.4 shows the appearance of a vertical snorkel system with the auxiliary exhaust system that will be described in the next paragraph. It may be advantageous to hinge the vertical sections to permit safe legal operation of the vehicle on public streets. Although vertical snorkels are to be preferred, a long horizontal pipe system may also be used. These systems separate in time the engine noise from the residual inlet flow noise but add no additional separation between inlet and microphone at point of closest approach.

Exhaust: Disconnect the production muffler and route the exhaust through a special muffler/stack system. Connect two mufflers in parallel by means of "wyes" at both ends. The type of muffler to be used may be selected from the tables in Appendix V B. The muffler bodies should not touch one another. They should be wrapped individually and then together in foil-backed glass fiber insulation. A rigid stack should be attached to the muffler's outlet wye. Where feasible, a total stack height of about 20 ft. should be provided. The vertical system should be securely mounted to the truck's chassis or a gantry with hinges as described for the intake snorkel to permit safe legal operation on public streets. This type of system is shown in the photograph of Fig. 3.4. Alternatively, a long horizontal tail pipe may be employed to separate the exhaust flow noise in time from the noise of the remaining sources near the engine.

If the two wyed mufflers do not provide satisfactory exhaust silencing, additional muffling should be added ahead of the wyed mufflers. It is important that exhaust piping be sealed against leaks and be thoroughly wrapped in foil backed glass fiber insulation. The effectiveness of the muffling can



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COURTESY CUMMINS ENGINE CO.

FIG.3.4 PHOTOGRAPH OF AUXILIARY EXHAUST AND EXTENDED INLET SNORKEL requires subjective judgment. The contributions of the leaks and muffler and pipe shell vibration are sometimes overlooked although these can be major contributors when exhaust system noise levels are excessive. In case of most standard leak-free exhaust systems without special cover-ups, approximately half the sound energy is radiated directly from the muffler casing, exhaust pipe and tail pipe and half from the gas flow at the exit. The glass fiber and foil covering is required to reduce this casing and pipe noise.

Rigid mountings should not be used between the mufflers, stack or intake snorkel and the supporting structures. Securely tie the wrapped system to the structure with baling wire after the wrappings have been applied. A metal jacket may be used on top of the insulation to prevent it from being squeezed. Isolation mounts are recommended between such a jacket and the truck structure.

Precautions: The acoustic materials used for window analysis will absorb oil and increase the fire hazard. Large capacity fire extinguisher(s) should be carried within the fully treated truck. Since the fan will have to be disconnected for all but fan noise tests, the engine should not be operated or idled for prolonged periods. The radiator may be cooled between tests by spraying water on the radiator (care should be taken not to soak the wrappings) or by use of external fans.

(b) Fully Treated Baseline Noise Test

The vehicle should be in the fully treated condition as described above. Measure noise levels at 50 ft. on the driver side and on passenger side per acceleration test of SAE J366b with shutters open and closed. Measure cab interior noise levels per SAE J336 or the Bureau of Motor Carrier Safety Procedure again with shutters open and closed.

It is generally impractical to obtain a level less than 3 dB(A) above the chassis noise level measured in Test B. In special cases (e.g. excessive chassis resonance due to poor isolation from engine mounts), it may be impossible to get within 10 dB(A) of the chassis baseline level.

Reconnect the standard exhaust system, disturbing the auxiliary system and stack as little as possible. Perform the acceleration tests per SAE J366b and J336 and the stationary high idle tests of the Bureau of Motor Carrier Safety, with shutters closed.

The contributions of exhaust gas noise, muffler and pipe shell noise, or from one unit of a dual exhaust system, may be determined separately if desired, by appropriate partial covering or by routing the tailpipe through the auxiliary stack. For example, gas noise may be reduced substantially by routing the exhaust gases to the auxiliary stack so that the remaining noise will be predominantly shell noise. (One is cautioned, however, that during such re-routing the system back pressure should be maintained close to the original back pressure, otherwise direct comparison will be impossible between different tests.)

E. Intake Noise Test

The vehicle should be returned to the fully treated condition and the treated baseline test rerun according to Test C. If the new fully treated level is different from the initial fully treated level, the reason for this should be ascertained and improvements made in the covering etc. to obtain agreement. The original air cleaner and air cleaner piping (up to the engine wrapping) should be reconnected, leaving the auxiliary system in place, but disconnected. Perform the noise measurements with shutters closed.

It is possible to separate shell and gas noise as subcomponents of intake noise but the low intake noise levels that are generally found greatly complicate such separation.

F. Fan Noise Test

The vehicle should be returned to the fully treated condition and the treated baseline test rerun according to Test C. If the new fully treated level is different from the previously measured fully treated level, the reason for this should be ascertained and corrected to obtain agreement.

F1. Fan Baseline Noise Test

Unwrap the front of the engine only. Care should be taken not to expose any sound absorbing material (foam) to the fan environment, so that fan noise will not be absorbed in an unpredicable manner. Perform noise tests with shutters open and with shutters closed.

F2. Fan Noise Test

Now install the fan and fan belts. Perform the noise tests with shutters open and closed. At this time, tests should be performed with shutters in several different positions to determine the best aerodynamic condition for quiet operation.

G. Engine Mechanical Noise Test

Remove or disconnect fan and unwrap the rest of the engine up to the transmission casing. Perform noise measurements with shutters open only.

H. Transmission Noise Test

The vehicle should be returned to the fully treated condition and the treated baseline test rerun according to Test C. If the new fully treated level is different from the initial fully treated level, the reason for this should be ascertained and corrected to obtain agreement. Unwrap the transmission and perform the noise measurements with shutters closed.

I. Final Baseline Test

Return vehicle to fully treated condition and repeat noise measurment. Again, the new measurement should agree with the previously measured fully treated noise level.

This test is essential as a final check, especially if noise reduction hardware is to be tested next.

level that would be measured if that source alone were present during the measurement.

The measured noise level for each of the uncovered sources will be composed of the contributed level of that source plus the contributions from the remaining covered or quieted sources. For example, the measured noise level during the exhaust noise test (test D in Section 3.5) will include the contributed level of the original exhaust system as well as the noise leaking through the coverings of the engine, transmission, auxilliary intake piping and the contribution from the auxiliary intake. We can lump all the unwanted noises, that is, all the noises except the exhaust system noise, as a new background noise. If this background level were known, we could obtain the contributed level of the exhuast system from the noise level measured during test D by using the background noise correction chart of Fig. 2.2. The quieting procedure suggested in the previous section normally results in lowering the unwanted noise source levels sufficiently so that the chart for background noise correction can be used.

This new background noise level is reasonably well approximated by the fully treated baseline noise level (test C), and we shall use this level as the "background" for obtaining the contributed levels of each of the sources except the fan. In the case of fan noise, the background level is approximated by the fan baseline noise level (test F1)*. The steps required to obtain the contributed levels of each of the sources are given in the last column of Table 3-2. To illustrate this, let us look at an example.

---

*Fan noise can be identified unequivocally only if an independent low noise drive (e.g. hydraulic or electric motor) can be employed during a coast by test (engine off) to drive the fan in its normal environment with the engine unwrapped.

of the major noise sources of a heavy diesel truck using the procedure of Section 3.5, the following noise levels were measured on each side (SAE J366b acceleration test):

Test		Noise Level dB(A) Left Side	Noise Level dB(A) Right Side
A	Untreated Baseline Noise	86.0	86.0
B	Chassis Noise	73.0	73.0
C	Fully Treated Baseline Noise	76.5	76.5
D	Exhaust System Noise	82.0	83.0
E	Intake Noise	78.0	79.0
F 1	Fan Baseline Noise	79.0	79.0
F 2	Fan Noise	82.5	82.5
G	Engine Noise	81.0	81.0
H	Transmission Noise	78.0	78.0

To obtain the contributed levels, subtract the appropriate background noise levels with the help of **Fig.** 2.2.

	Left Side dB(A)	Right Side dB(A)
Exhaust Contributed Level	82.0 less 76.5 = 80.6	83.0 less 76.5 = 81.9
Intake Contributed Level	78.0 less 76.5 = 72.7	79.0 less 76.5 = 75.4
Engine Contributed Level	81.0 less 76.5 = 79.1	81.0 less 76.5 = 79.1
Transmission Contributed Level	78.0 less 76.5 = 72.7	78.0 less 76.5 = 72.7
Fan Contributed Level	82.5 less 79.0 = 80.0	82.5 less 79.0 = 80.0
Sum of Contributed Levels	85.2	85.9

As a check of the accuracy of the results, the contributed levels plus the measured chassis noise level should equal the untreated baseline noise level for the corresponding side of the truck. In our example, we have

	Left Side dB(A)	Right Side dB(A)
Sum of Contributed Levels	85.2	85.9
Chassis Noise (Test B)	73.0	73.0
Total	85.2 plus 73.0 = 85.5	85.9 plus 73.0 = 86.1
Untreated Baseline Noise (Test A)	86.0	86.0

Since the accuracy of measurement is not expected to be better than + or - 1 dB(A), agreement within 0.5 dB(A) is quite good. If there is a wider disagreement, a recheck of the calculations should be made. If any disagreement between the computed and measured untreated truck noise level is not due to a computation or measurement error, the causes for any uncertainties in measurements should be examined and the more complex computation procedure given in Appendix III should be followed.

One of the sources of inaccuracy in the determination of contributed levels arises when the difference between the measured component noise level and the fully treated baseline noise level is less than 3 dB(A). In the example shown above, the transmission noise level is only 1.5 dB above the treated baseline noise. The background noise correction chart of Fig. 2.2 cannot be used. If the rule for addition (or subtraction) of dB levels is employed, we find that a large uncertainty exists in the magnitude of the correction. This is illustrated in Fig. 3.5. The calculated contribution of the transmission should, therefore, be used only as a level indicative of the actual contributed level.

3.7. Time and Material Requirements

The following estimate of labor and materials was furnished by the Vehicle Noise Test Section of Cummins Engine Co. for exterior noise tests using the SAE Recommended Practice J366a with the procedures described in Sections 3.5 and 3.6.

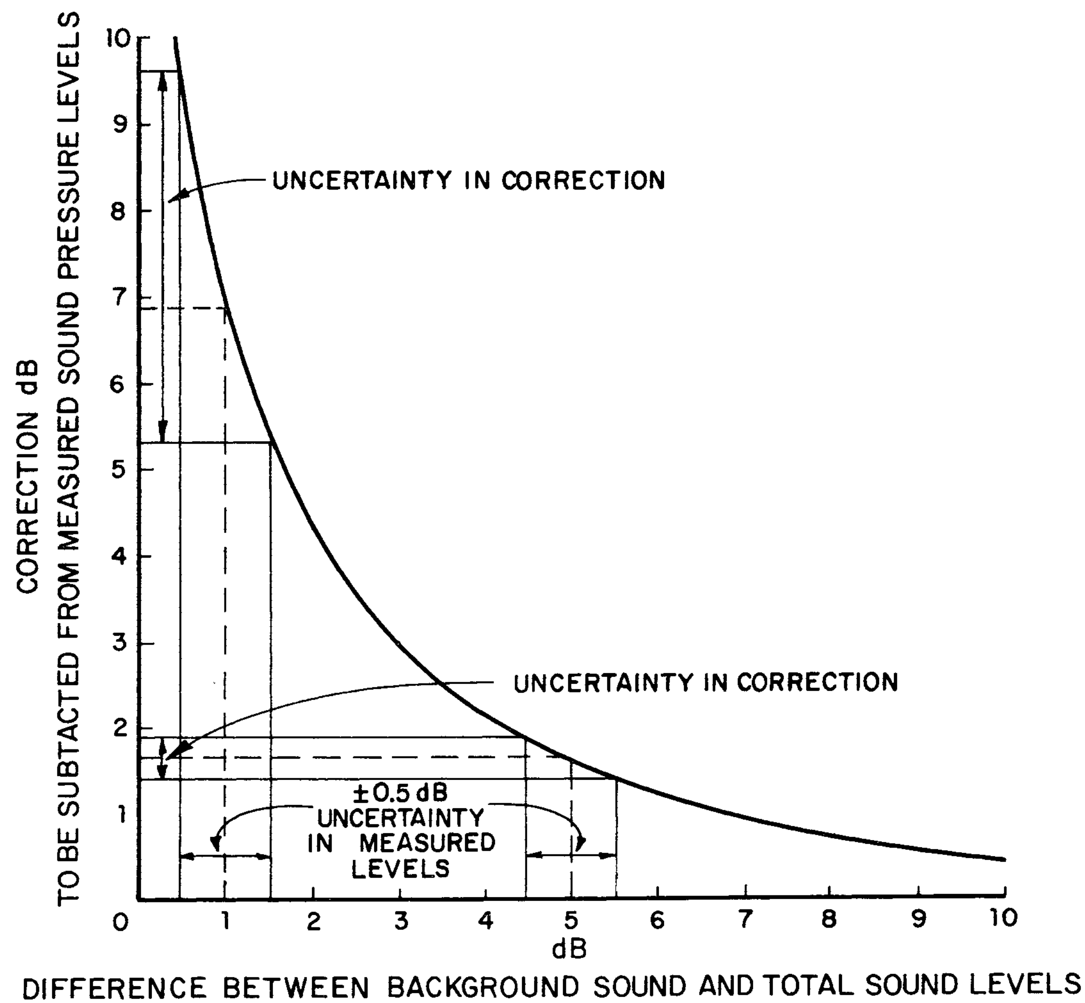

FIG 3.5 BACKGROUND NOISE CORRECTION SHOWING INCREASE IN THE UNCERTAINTY OF CORRECTION AS THE DIFFERENCE BETWEEN BACKGROUND AND TOTAL SOUND DIMINISHES

lead foil backing (approximately $...).

- 30 sq ft building type glass fiber insulation (3-1/2" thick) (approximately $5.00).
- 15 sq ft asbestos insulation paper (1/8" thick) (approximately $5.00).
- Duct tape, wire, extra muffler, intake filter and piping.

3.8 Shortened Procedures

The procedure outlined in Sections 3.5 and 3.6 does not require any skills in handling or tooling. The sequence has been selected to optimize labor expenditure and avoid mistakes in interpretation. It is tempting to eliminate some of the treated baseline test reruns. However, such "short-cutting" may jeopardize the validity of subsequent data and is not recommended for a manufacturer. For an operator, however, the resources available to control the noise of his vehicles are sufficiently limited that tests of the untreated baseline truck (test A), a baseline test with the fan removed (to ascertain if fan noise reduction will have immediate benefit) and a treated auxiliary exhaust test (to ascertain if exhaust treatment will have immediate benefit) should suffice for preliminary identification of noise sources.

If neither fan removal nor exhaust treatment show significant improvement, simultaneous fan removal and exhaust treatment should be tested to ascertain potential benefits. The absence of benefits following such tests indicates deep trouble beyond the expected capabilities of the operator. If the required maximum sound level is achieved or bettered with either the fan or exhaust treatment tests, a clear indication is obtained that the operator can indeed solve his problem.

3.8.1 Selecting an Optimum Exhaust System

Let us examine one specific instance when all the component contributed noise levels may not be required, and a shortened procedure may be adequate. The sequence of measurements would be as shown in Fig. 3.6.

If the level in (b) is close to the baseline untreated level (a), the original exhaust system is not a major contributor to overall noise. If level (b) is more than 3 dB(A) below the level (a), exhaust noise is a

EXHAUST NOISE EVALUATION SHORT PROCEDURE

a MEASURE UNTREATED BASELINE NOISE

b MEASURE NOISE OF TRUCK AFTER INSTALLING WRAPPED DOUBLE MUFFLER SYSTEM

c MEASURE NOISE OF TRUCK AFTER INSTALLING NEW MUFFLER

d ORIGINAL EXHAUST NOISE = level a − level b
NEW EXHAUST NOISE = level c − level b

FIG 3.6 ILLUSTRATION OF SHORTENED PROCEDURE FOR DETERMINING EXHAUST NOISE major contributor to overall noise and a search for a better [illegible]

system is warranted. A 3dB(A) difference implies that the exhaust noise equals the noise from all the other noise sources combined.

Obviously, the same strategy may be employed for selecting an optimum intake silencer, but the results may not be as conclusive unless the exhaust, fan and engine mechanical noise are exceptionally low or the exhaust system and the engine are also treated.

3.9 Combined Retrofit and Component Noise Evaluation

Selected quiet components can be tested during the same test series as the original component noise levels are being measured. The procedure would be as follows: (See Section 3-5 for details of the test procedures.)

a. Obtain untreated baseline noise level for original design truck.
b. Obtain chassis noise level.
c. Fully treat the vehicle.
d. Obtain fully treated baseline noise level.
e. Disconnect treated auxiliary exhaust system, reconnect original exhaust system and obtain noise level.
f. Disconnect original exhaust system and install new exhaust systems in sequence, obtaining noise levels in each case in the same way.
g. Reinstall auxiliary treated exhaust system and remeasure fully treated baseline noise level.
h. Repeat procedure for the other treated components as outlined in e, f and g above.
i. Obtain contributed noise levels of original and proposed components.
j. Select best combination of components to obtain desired truck sound level.

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3.1 "Noise Level Test - Source Identification," Bulletin No. 113, Test Methods Pilot Installation Center, Cummins Engine Co., Columbus, Indiana, June 1972.

4. EXHAUST NOISE

4.1. Introduction

Engine exhaust is one of the major sources of noise from vehicles powered by internal combustion engines. In the case of diesel powered vehicles, the constraints of space, weight, cost and engine performance often make adequate muffler design difficult. Figure 4.1 shows the contributed exhaust noise levels* for thirteen production model trucks along with the overall exterior noise levels of the trucks. Although all trucks had "standard" mufflers installed, the exhaust noise in some cases exceeds or equals the combined noise level of the remaining sources. Obviously, improvement in the original exhaust systems for such trucks will result in a substantial reduction of the overall truck noise levels. Besides the use of an inadequate muffler, excessive exhaust noise can also result from faulty or improper muffler installation that results in gas leakage or large vibration amplitudes. It is the purpose of this section to explain the current techniques of optimizing truck exhaust systems for noise as well as back pressure and to point out some of the installation practices that may result in unnecessarily loud exhaust noise levels from an otherwise adequate muffler.

The U. S. Department of Transportation has recently sponsored programs to measure the exhaust noise for a variety of muffler makes and configurations on several typical diesel engines. The results of these programs have been made available to the public [4.1] [4.2]. The programs have obtained much factual information that can be used to arrive at trends for the selection of mufflers for diesel truck applications. These trends are discussed in Section 4.4. The conclusions are derived from current muffler systems tested under the above-mentioned programs and from data supplied by a few of the muffler manufacturers. Improvement in muffler design may revise some of these trends and conclusions.

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*Measured by SAE J366b procedure, under acceleration.

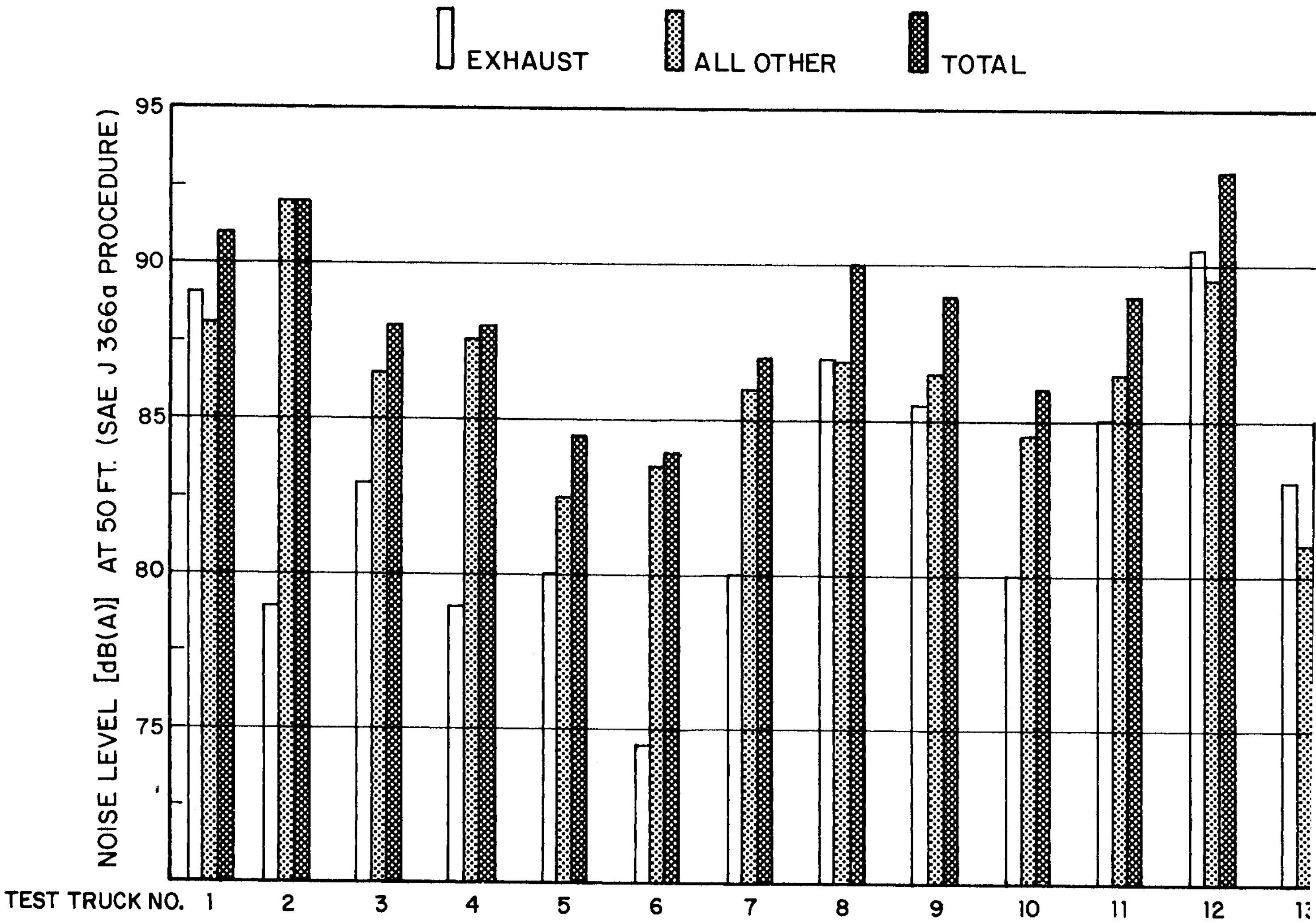


FIG. 4.1 CONTRIBUTED EXHAUST NOISE AND TOTAL NOISE LEVELS OF SOME PRODUCTION MODEL DIESEL TRUCKS

4.2. Mechanism of Exhaust Noise Generation

4.2.1. Naturally Aspirated Engines

Exhaust noise is produced by the sudden release of gas into the exhaust system by the opening of the exhaust valve. The amount of noise generated by this type of noise source is proportional to the rate of change of the volume flow velocity. The closing of the valve also produces noise but to a lesser degree, since the rate of change of volume flow velocity is smaller than when the valve opens.

A second source of noise results from the high velocity flow of gas over the exhaust valve. This flow noise may be the dominant source of high frequency exhaust noise.

The exhaust process generates a series of noise pulses with the largest component at the fundamental firing frequency f_F.

$$f_F = \frac{NZ}{120}\ \text{Hz} \qquad \text{[four-stroke engines]}$$

$$f_F = \frac{NZ}{60}\ \text{Hz} \qquad \text{[two-stroke engines]} \tag{4-1}$$

where

N = engine rpm

Z = number of engine cylinders connected to the independent exhaust system.

Hz = unit of frequency, Hertz (1 Hz = 1 cycle per second)

TABLE 4-1

Fundamental Firing Frequencies

Engine	No. of Strokes	f_F(Hz) @RPM
Cummins NHC-250	4	105 @ 2100
Cummins NTC-350	4	105 @ 2100
Mack ENDT-675	4	85 @ 1700
Detroit Diesel 6-71	2	210 @ 2100
Detroit Diesel 8V-71 (Single Exhaust)	2	280 @ 2100
Detroit Diesel 8V-71 (Dual Exhaust)	2	140 @ 2100

Typical firing frequencies f_F are given in Table 4-1. Moreover, a series of peaks occur at frequencies above the fundamental, spaced according to

$$f_m = \frac{mN}{60} \qquad m = 1, 2, 3, \ldots \qquad (4\text{-}2)$$

These peaks are clearly seen in Fig. 4.2 which shows the 1/10 octave-band spectrum for a typical unmuffled engine exhaust noise. To determine which of the peaks is important from the viewpoint of audibility, we can weight the spectrum according to the standard A-weighting curve of Fig. 2.1. The dashed curve in Fig. 4.2 shows that if we restrict our consideration to A-weighted sound levels, the noise at all audio frequencies becomes of equal concern.

Figure 4.3 shows octave band spectra and overall levels of unmuffled exhaust noise from five diesel engines. The greatest noise levels occur in the octave bands that contain the fundamental engine firing frequencies. Again, if these spectra were A-weighted, they would not show the high levels in bands below 1000 Hz.

4.2.2 Turbocharged Engines

Increased power outputs may be obtained from diesel engines by increasing the amount of air and fuel entering the engine per stroke. Modern truck engines employ air compressors to supply the added amount of air to the intake. In a turbocharger, the compressor is directly driven by a turbine powered by the exhaust and mounted on the same shaft. The exhaust flow is ideally suited to drive the turbine because of its high temperature. Since power is extracted from the exhaust flow, there is a small back pressure penalty. This can result in the lowering of the allowable muffler back pressures for turbocharged engines.

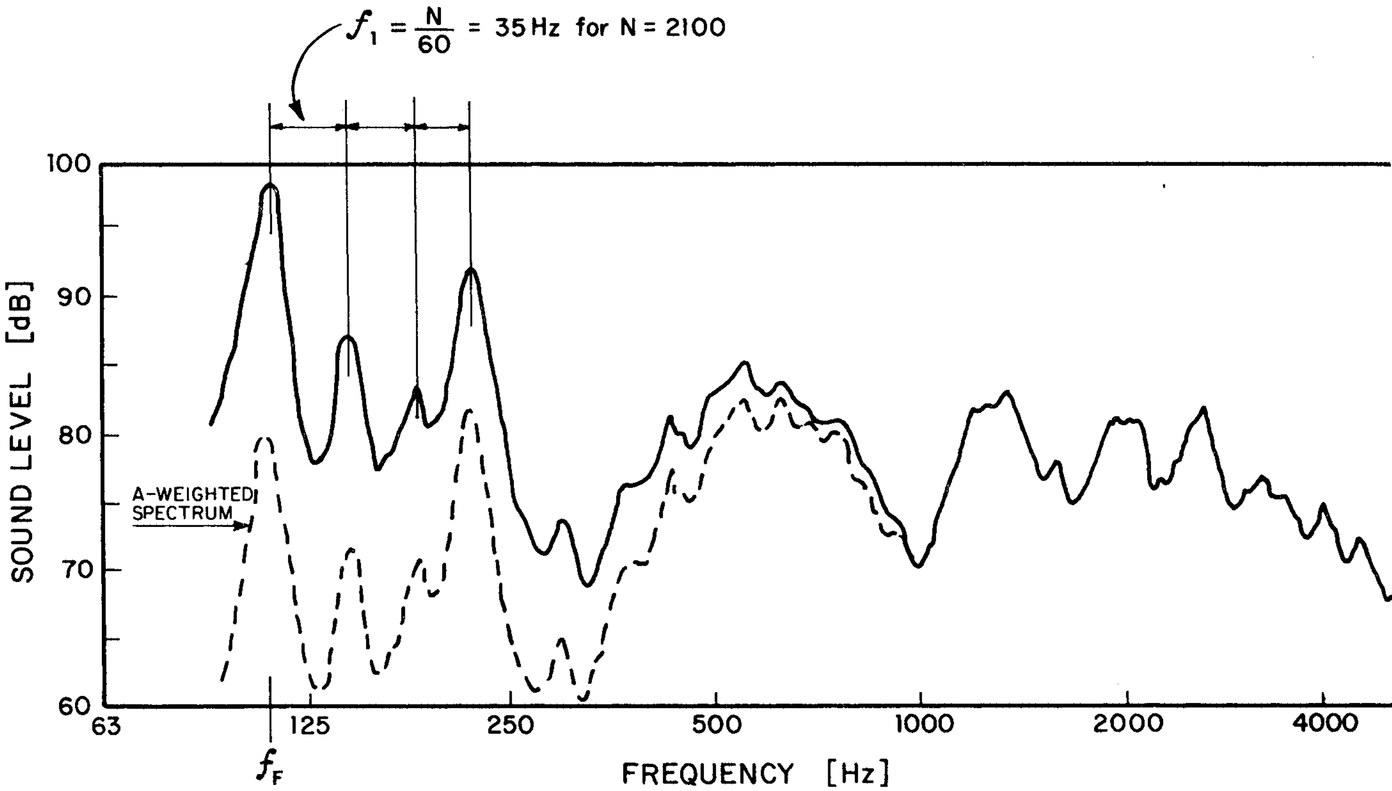


FIG. 4.2 1/10 OCTAVE SPECTRUM OF UNMUFFLED EXHAUST NOISE
CUMMINS NHC-250 ENGINE AT 2100 RPM

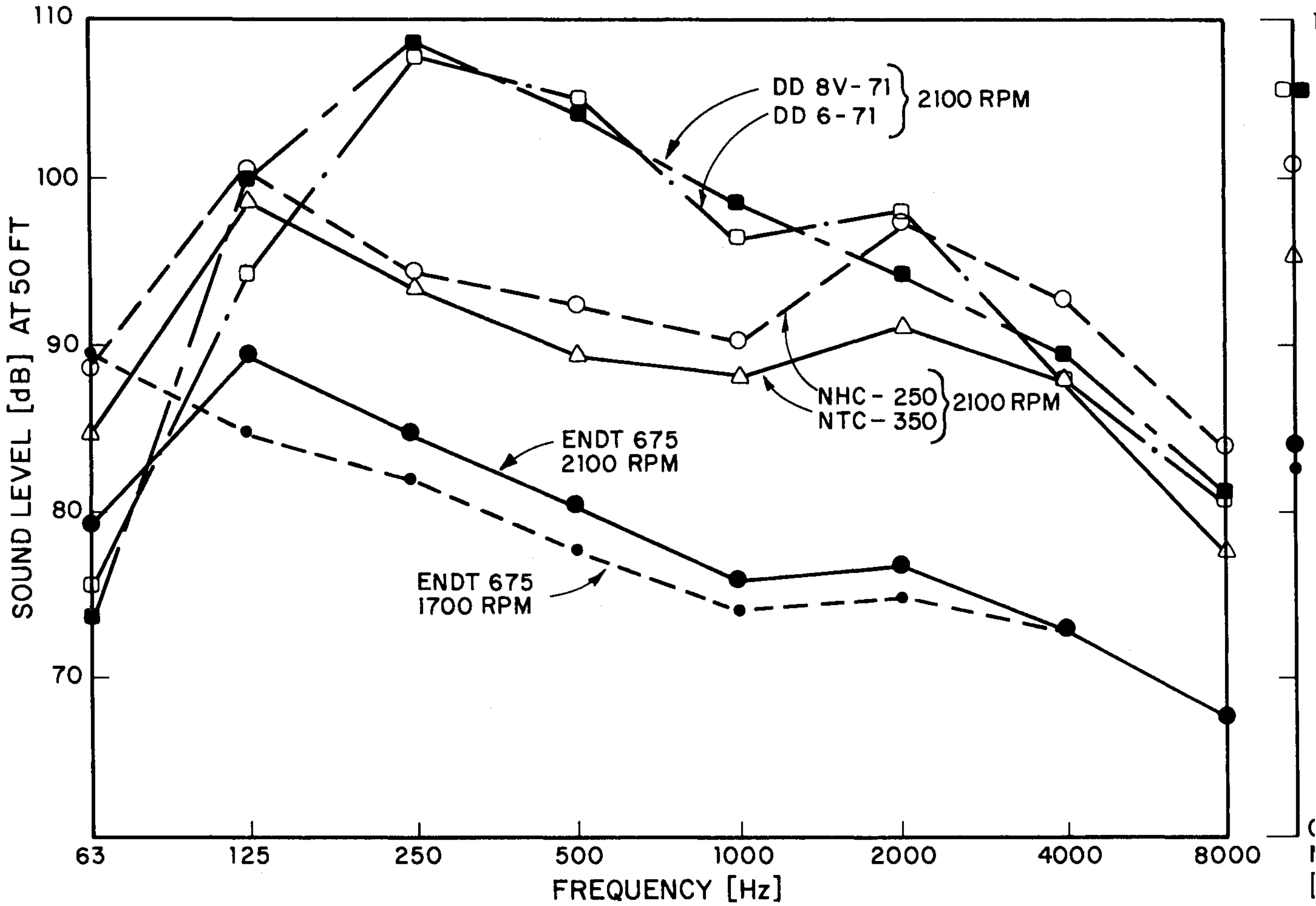


FIG. 4.3 OCTAVE BAND SPECTRA OF TYPICAL UNMUFFLED DIESEL ENGINE EXHAUS

existing in the turbine, unmuffled turbocharged engine exhaust noise is significantly lower than the corresponding noise of a naturally aspirated engine.

The turbines normally run at very high speeds (50,000 to 90,000 rpm) creating blade passage perturbations at frequencies in the range of 5,000 to 10,000 Hz. The increased mass flow rates alter the tuning of the exhaust system (see Sec. 4.3) and may necessitate larger muffler volumes. The greater mass flow also makes pressure pulsations at the end of the exhaust stroke important from the noise viewpoint, especially in engines with considerable valve overlap. Although the unmuffled noise level of the turbocharged engine is lower than that of a naturally aspirated engine, careless muffler choice and/or installation may result in unacceptable noise levels.

4.2.3. Effect of Engine Brakes on Exhaust Noise

The basic principle of engine brake operation is to temporarily convert a power producing diesel engine into a power absorbing air compressor, driven by the kinetic energy of the truck. This is accomplished by means of a mechanism that cuts off the fuel supply and opens the cylinder exhaust valves near the top of the normal compression stroke, releasing the compressed cylinder charge to the exhaust system. The net energy loss produces the braking effect, but the sudden discharge of highly compressed gases (up to 500 psia) into the exhaust system significantly increases the exhaust noise level. The average pressure at the exhaust system is increased, increasing the pressure pulsations at the fundamental firing frequency. Since the opening of the exhaust valve is relatively sudden as compared to the normal opening of the exhaust valve by means of a cam mechanism, a sharper pressure pulse is produced which results in sound radiation in the high frequency region of the spectrum. This produces the high-pitched noise characteristic of engine brake operation.

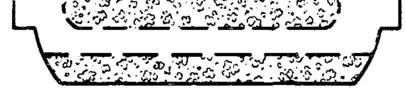
An unmuffled turbocharged engine in a truck which may be producing noise levels within the law without an engine brake, may exceed the permissible noise levels during engine brake operation. Manufacturers of engine brakes and mufflers are aware of this problem and offer special packages for exhaust noise control in engines equipped with engine brakes. Here, even more than in the case of ordinary exhaust systems, the effects of leaks in the system are important; the high pressure jet noise produced by the small leaks under engine braking conditions can be extremely loud, tonal in nature and hence highly annoying.

4.3. Fundamental Muffler Designs

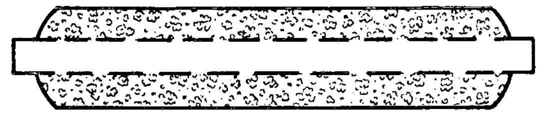
All mufflers operate on the basis of two physical mechanisms of noise reduction: sound absorption or dissipation and sound reflection. Different types of mufflers using these mechanisms are shown in Fig. 4.4.

Mufflers that operate primarily on the basis of sound absorption or dissipation are known as dissipative mufflers. They consist of porous materials such as glass fiber or steel wool, covered with perforated sheet metal. A straight-through design is typically employed to keep the pressure drop at a minimum. Dissipative mufflers can be very effective in reducing high frequency noise. Their low frequency performance is quite limited, however, unless the muffler is made very large. The dissipative muffler is also limited in its life because of the adverse effects of high temperatures and oil in the exhaust flow on the absorptive materials. Because of their limited frequency performance and limited life, dissipative mufflers are not commonly used on trucks.

Mufflers that operate primarily on the basis of sound reflection are known as reactive mufflers. Reflecting acoustical energy back toward the engine can be accomplished by introducing changes in cross-sectional areas and by means of short branches opening into large chambers. Reflecting the incident sound energy back toward the engine results in the formation of standing waves in the exhaust pipe between the exhaust manifold and the mufflers. The standing waves are continually reinforced during the operation of the engine until non-linearities take over and the sound energy

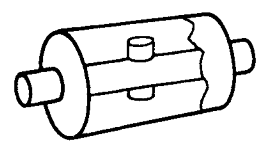


ANNULAR CORE

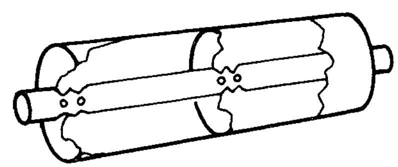


STRAIGHT - THROUGH

DISSIPATIVE MUFFLERS



SINGLE CHAMBER RESONATOR



DOUBLE CHAMBER RESONATOR

REACTIVE MUFFLERS

FIG. 4.4 TYPES OF MUFFLERS

be found in Refs. [4.3] and [4.4].

The acoustic performance of a muffler is usually judged by noise reduction measurements. Noise Reduction, also referred to as "insertion loss", is defined as the difference between the sound pressure levels, measured at the same point in space, before and after the muffler is inserted in the exhaust system. The assumption here is that the observed difference is due solely to the presence of the muffler. This is not always true, because by placing the muffler somewhere in the pipe, a long "tail pipe" has been changed into a short exhaust pipe plus a short tail pipe. In analyzing the acoustic performance of an exhaust system it is important to keep in mind that complex interdependent acoustic relationships exist between the different components of the systems. The exhaust pipe and tail pipe are important system elements, beside the muffler itself. The observed performance is actually a function of where the muffler is located; since exhaust pipe and tail pipe are acoustical elements in themselves.

The noise reduction of an exhaust system is easier to measure than to calculate. The noise reduction at discrete frequencies, when plotted against frequency, shows a series of spikes at frequencies corresponding to resonances in the tail pipe, exhaust pipe and the passages within the muffler. Figure 4.5 shows a typical plot of the noise reduction properties of a two-element reactive muffler system against frequency. By altering exhaust and tail pipe lengths and the lengths of connecting tubes and chambers within the muffler, a reactive muffler exhaust system can be tuned to reduce the noise level at a particular troublesome frequency. Note from Fig. 4.5 that the noise reduction may be negative at certain frequencies so that there may be increased noise levels at those frequencies. Reactive "resonators", which are in effect simple expansion chambers, can be added at appropriate points in an exhaust system to reduce the noise at a frequency that is ineffectively attenuated by the muffler, especially the low frequency components of exhaust noise.

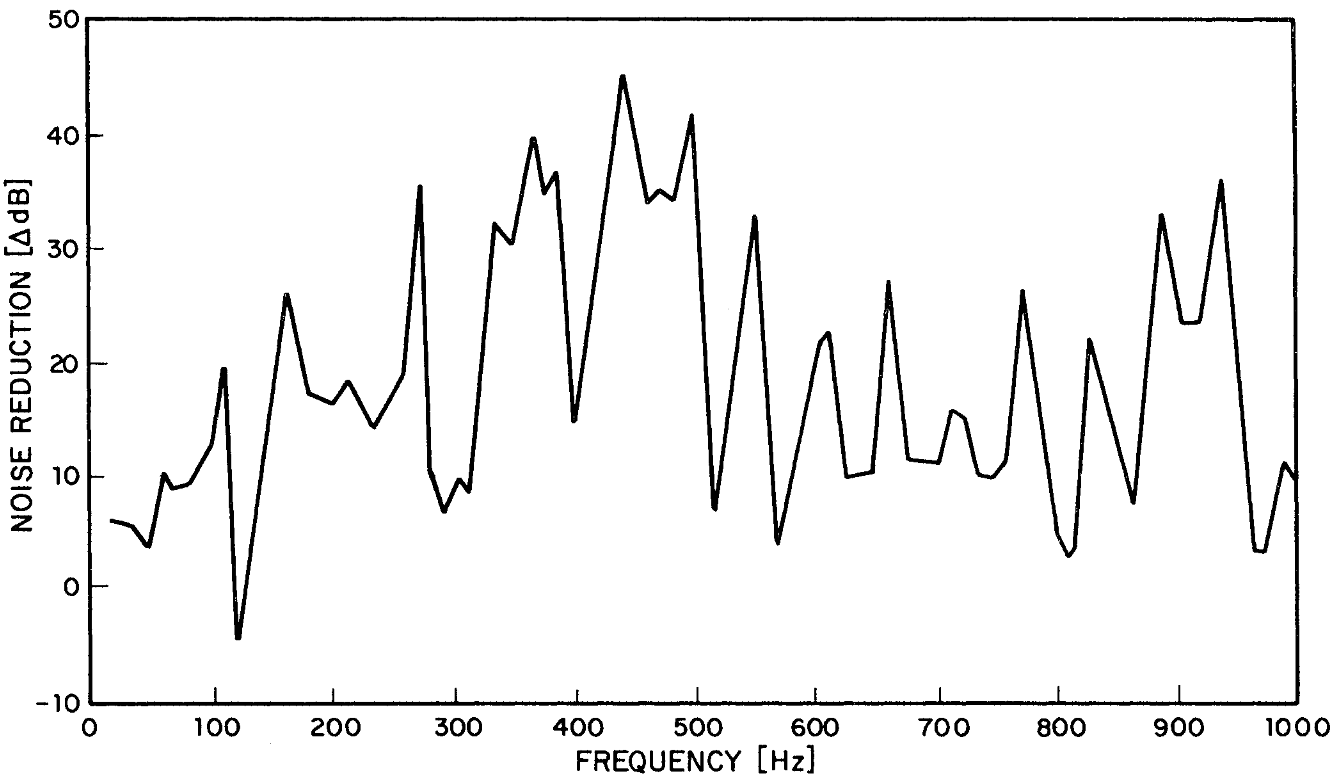


FIG. 4.5 VARIATION OF NOISE REDUCTION WITH FREQUENCY FOR A TWO-ELEMENT REACTIVE MUFFLER SYSTEM (ADAPTED FROM REF. [4.5])

Because of the many changes in cross sectional area that result in flow restrictions and separations, reactive mufflers in practice tend to have higher back pressures than dissipative mufflers. The noise reduction can be increased by increasing the number of cross-sectional changes or "elements" of the reactive mufflers, but the back pressure is also increased. However, the back pressures of most available mufflers are so low that the effect on engine horsepower is minimal. This is evident from Figure 4.6 which shows the engine brake horsepower for varying exhaust back pressures for a Detroit Diesel V71 engines. (The effect of intake restriction is more pronounced, as we shall see in Section 5).

The noise reduction performance of both reactive and absorptive mufflers is adversely affected when the average velocity of exhaust gases increases. Therefore, large diameter mufflers with low flow velocities offer potentially better performance over a wider range of engines. Because of the strong acoustical interdependence between all the elements that constitute an exhaust system, no generalizations can be made, however.

Reactive mufflers are commonly used on trucks, mainly because they offer a longer life than the dissipative mufflers, due to the better resistance of the sheet metal and pipe to hot exhaust gases in comparison with the absorptive material of dissipative mufflers; and because reactive mufflers can be properly tuned with the exhaust system and engine to significantly reduce the high energy low frequency components of exhaust noise. Maximum advantage of this last property is often not realized when original mufflers are replaced without consideration for the different lengths of exhaust and tail pipes that may be required for the new muffler. It is possible to combine the desirable properties of dissipative and reactive mufflers by using them in series. Some advanced exhaust systems are available for a few engines that utilize this idea.

Once the noise arising out of the unsteady flow of exhaust gases into the atmosphere has been sufficiently lowered, the secondary noise contributions due to the muffler and pipe shell vibrations become important.

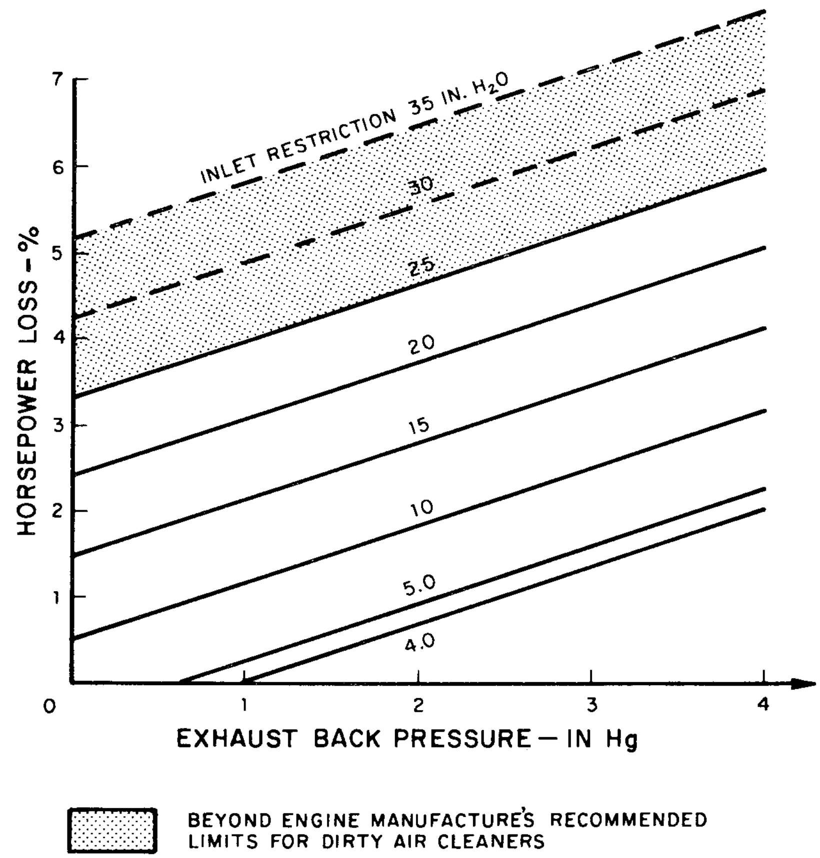


FIG. 4.6 EFFECT OF BACK PRESSURE ON ENGINE HORSEPOWER — DETROIT DIESEL V71 SERIES ENGINES

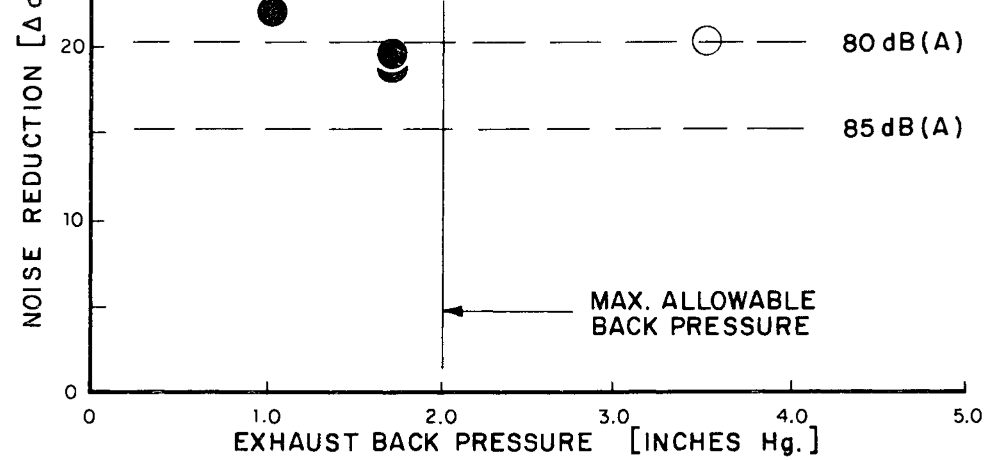
Mufflers offering improved noise reduction would therefore benefit from additional acoustical treatment of the exterior surfaces which would be in the form of an absorbent material covered by another layer of sheet metal. One large muffler manufacturer now offers a kit to reduce the shell noise that can be installed on some existing mufflers. Presently, no retrofit method appears to be available for reducing the noise radiated by vibrations of the exhaust manifold and exhaust pipe. The best course to take here is to use heavy gauge pipes, avoid long flexible sections of thin metal and to avoid rigid mounts between the pipe and the truck.

4.4. Acoustic Characteristics of Currently Available Mufflers

As presented herein, a muffler is considered "acceptable" if it achieves the necessary noise reduction and produces a back pressure less than or equal to the maximum allowable back pressure for the engine.

Figures 4.7 through 4.11 summarize the exhaust noise data for five engines [4.2]. A number of mufflers were tested on three naturally aspirated (Cummins NHC 250, Detroit Diesel 6-71 and Detroit Diesel 8V-71) and two turbocharged engines (Cummins NTC-350 and Mack ENDT-675). For each engine, the noise reductions [Δ dB(A)] obtained from the tested mufflers are shown against maximum back pressures produced by these mufflers, as well as their unit aftermarket prices. Mufflers that meet the engine manufacturer's back pressure limits are shown by solid symbols. All types of exhaust systems: horizontal, vertical, single and dual, are included in this data. The cost of a dual system is based on the cost of two mufflers.

There appears to be no general correlation between the back pressure produced by a muffler and its noise reduction, as seen from these figures. In the case of naturally aspirated engines (Figs. 4.7, 4.8 and 4.9), increased muffler price tends to correlate positively with greater noise reduction. The turbocharged Cummins NTC-350 does not show this correlation, but the turbocharged Mack ENDT-675 again shows improved noise reduction with increased muffler cost. Since the Detroit Diesel 8V-71



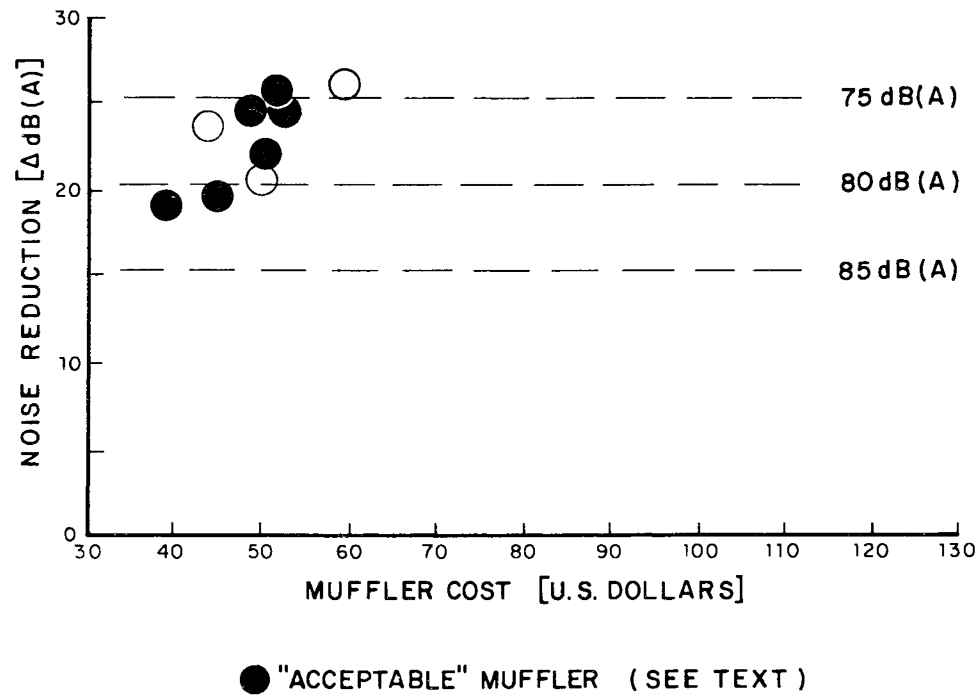


FIG. 4.7 NOISE REDUCTION FROM MUFFLERS
ENGINE: CUMMINS NHC-250

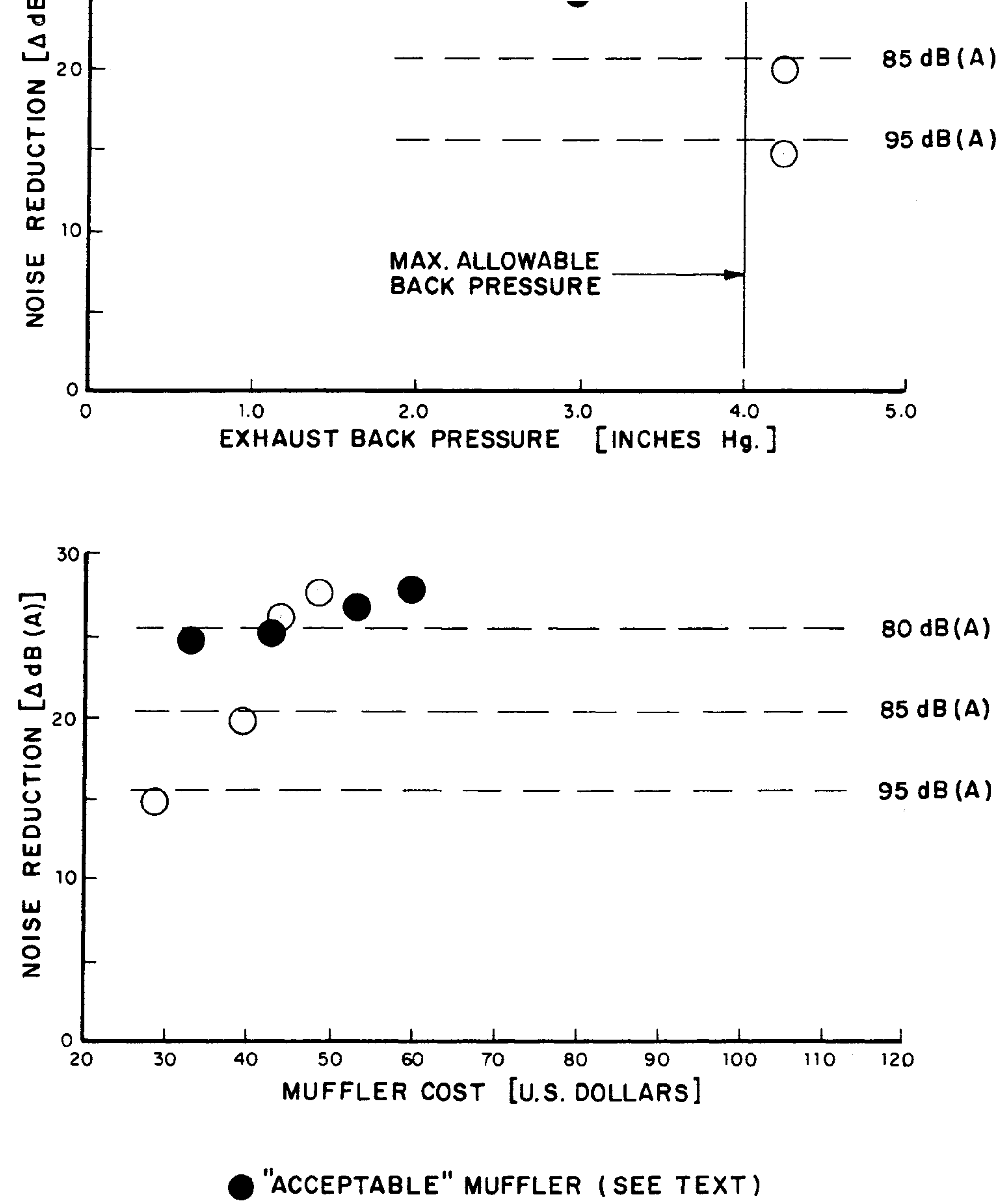


FIG. 4.8 NOISE REDUCTION FROM MUFFLERS
ENGINE: DETROIT DIESEL 6-71

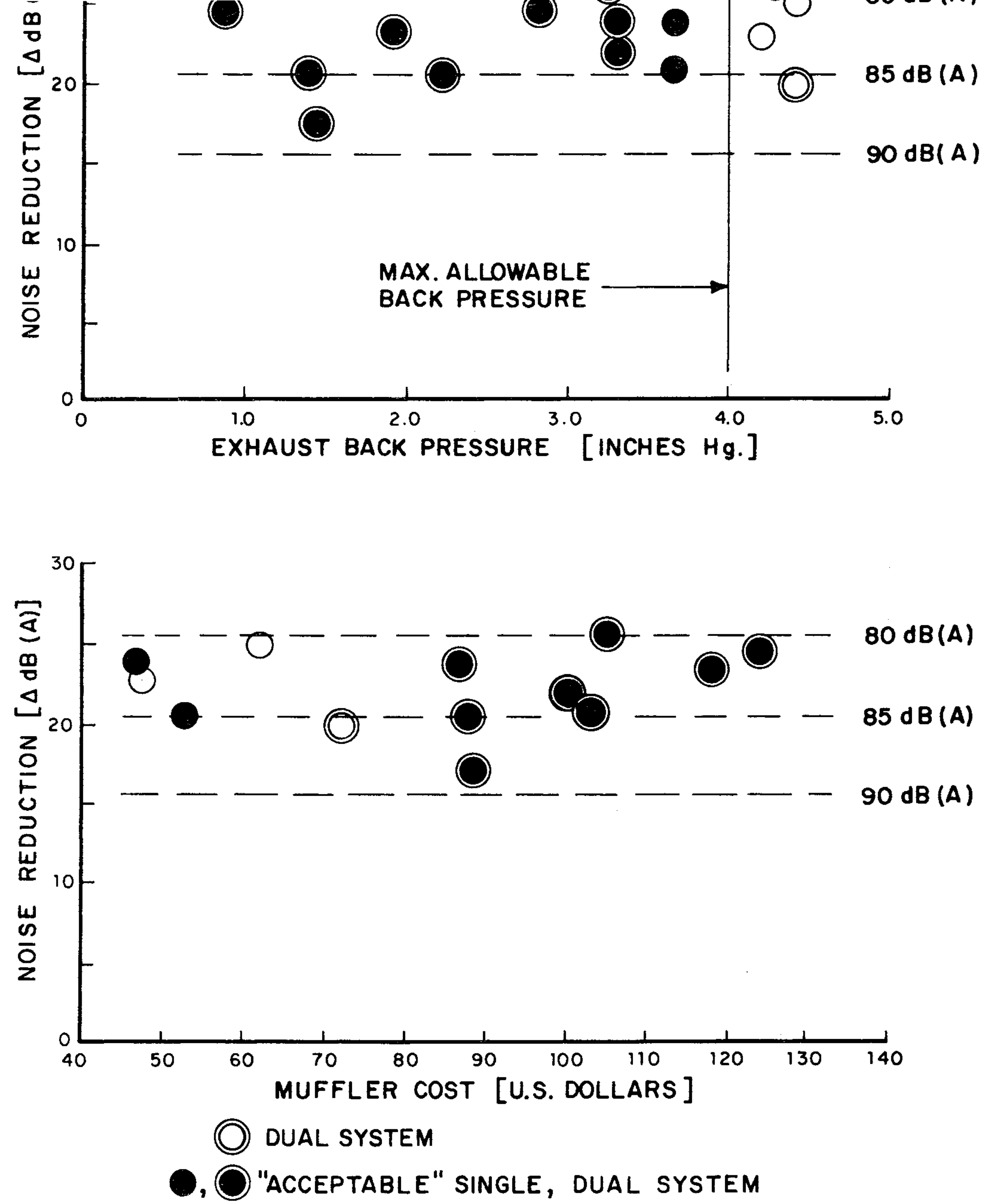


FIG. 4.9 NOISE REDUCTION FROM MUFFLERS
ENGINE: DETROIT DIESEL 8V-71

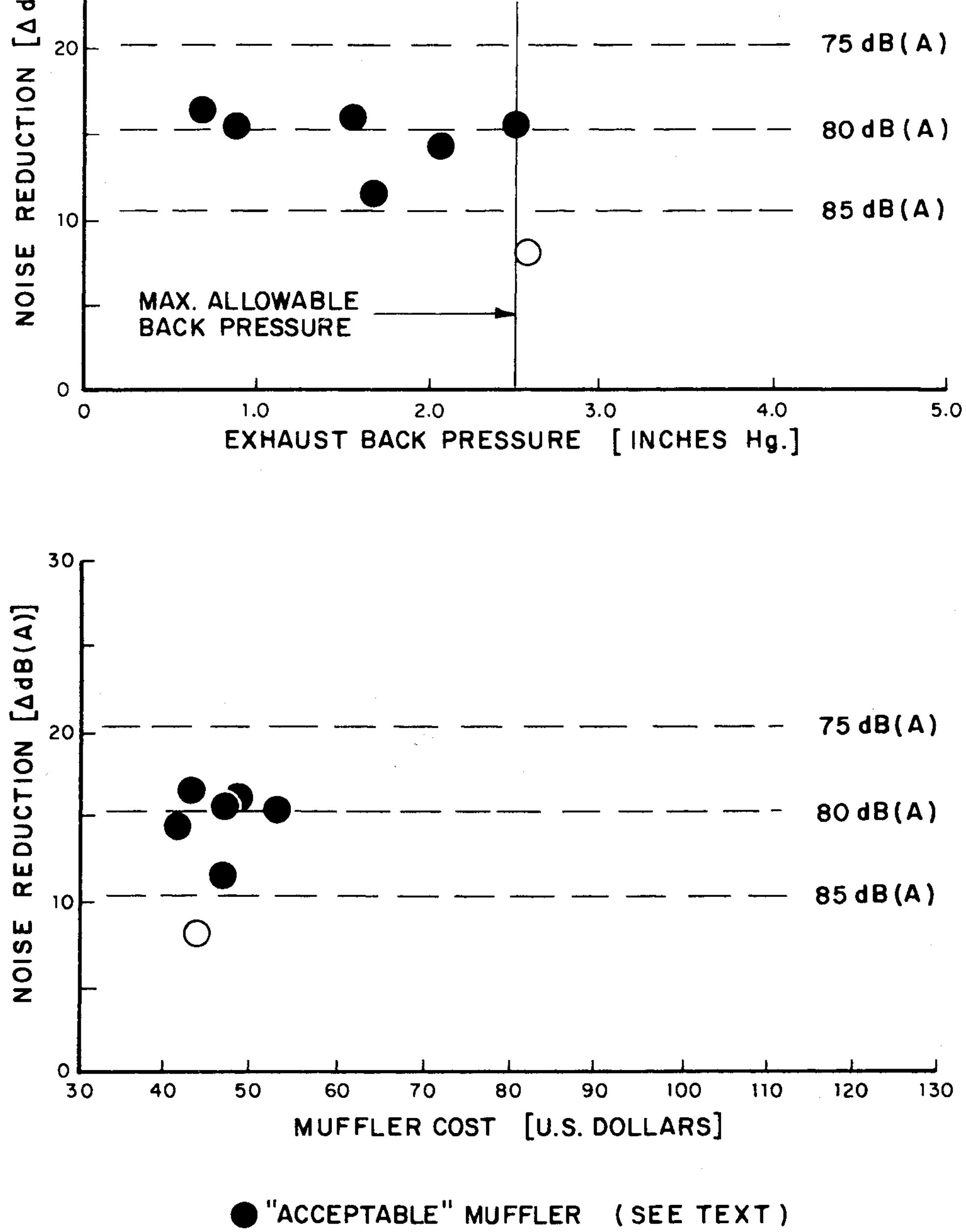


FIG. 4.10 NOISE REDUCTION FROM MUFFLERS
ENGINE: CUMMINS NTC-350 (TURBOCHARGED)

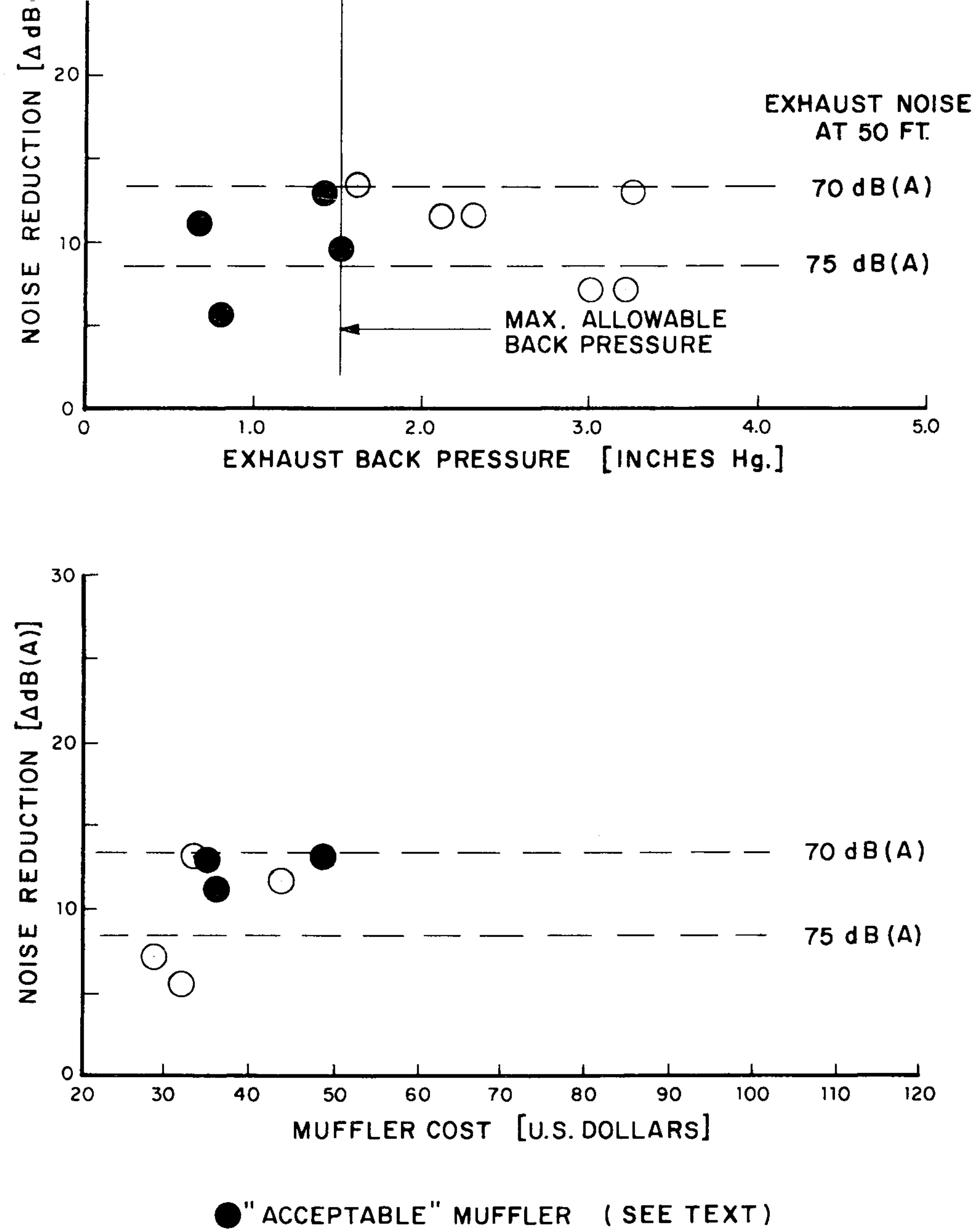


FIG. 4.11 NOISE REDUCTION FROM MUFFLERS
ENGINE: MACK ENDT-675

two distinct trends for this engine; one for single systems and one for dual systems (Fig. 4.9). It is seen that although some single muffler systems provide as much noise reduction as most dual systems, they are unacceptable because of higher back pressures. It is also seen that dual systems do not always result in very low back pressures, and in at least one case, a dual system has inferior noise reduction while exceeding the back pressure limit.

A muffler that produces greater back pressure is not always quieter than one with less restriction. However, it is true that a large percentage of the available mufflers that do provide good noise reduction are unacceptable from the back pressure viewpoint. The low level of allowed back pressure for the Mack ENDT-675 engine (Fig. 4.11) results in making several mufflers unacceptable that are superior from a noise viewpoint. It should be noted that many mufflers which are commercially available and which "fit" the engines tested were not made available for the particular test series reported in Ref. 4.2.

Noise reduction is a function of the design of that particular muffler and how well it is matched to the engine and exhaust pipe. In general, a short large-diameter muffler will silence very well over a narrow band of frequencies. A long slim muffler may give less attenuation, but will be effective over a wider range of frequencies. A claim often made is that turbocharged engines are "quieter" than naturally aspirated engines. The unmuffled exhaust noise level of the Cummins NHC-250 engine (naturally aspirated) is 4.5 dB(A) higher than the unmuffled exhaust noise level of the NTC-350 (turbocharged) engine. But the advantage appears to be lost when the exhausts are fitted with mufflers (see Figures 4.7 and 4.10). The increased air flow rates and sometimes lower allowable back pressure limits actually put much more severe demands on the mufflers for turbocharged engines. Exhaust noise from turbocharged engines has been discussed in Section 4.2.2.

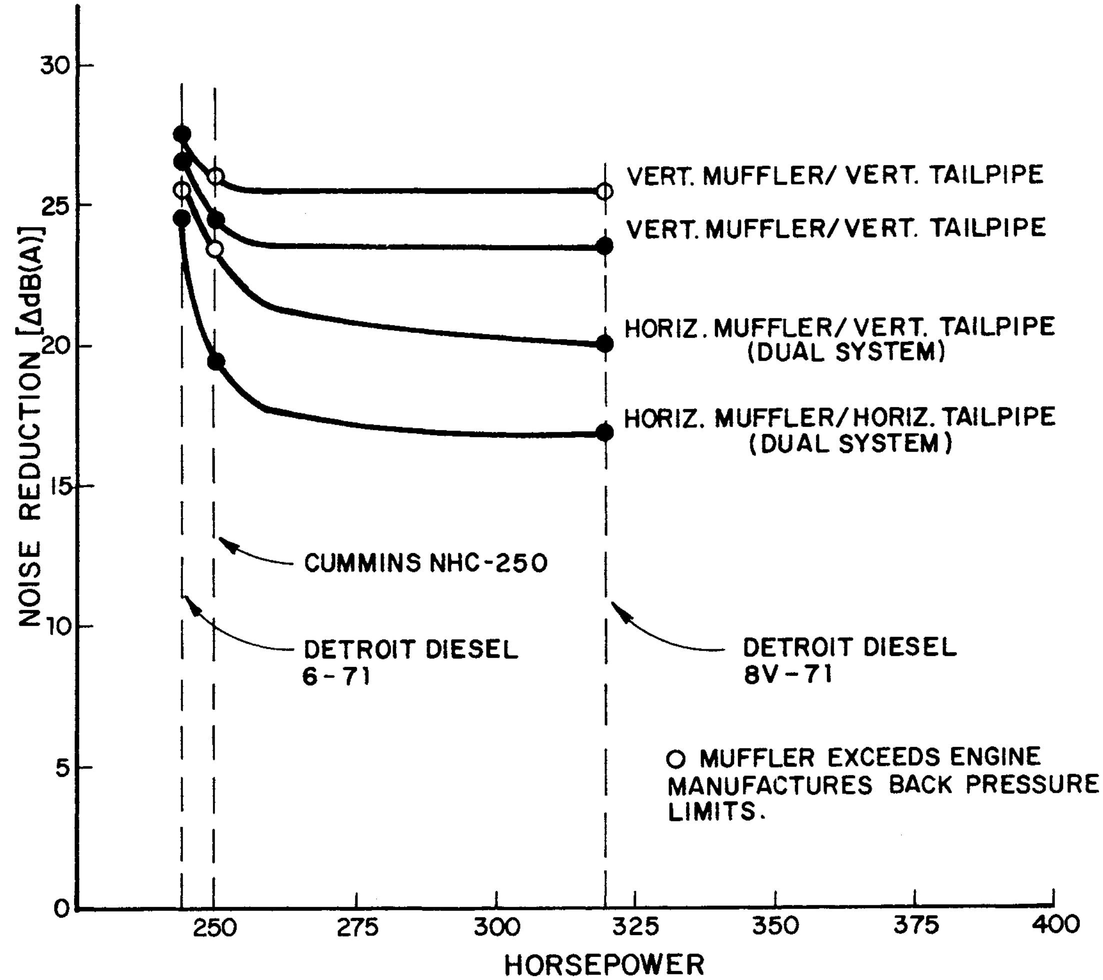


FIG. 4.12 CHANGE IN NOISE REDUCTION OF A MUFFLER WITH ENGINE HORSEPOWER

Noise reduction capability of a muffler also depends on the engine horsepower: it tends to decrease with increasing horsepower. This is shown in Figure 4.12 for four mufflers. The reason for decreased noise reduction is probably connected with the increased mass flow of the exhaust gases. Each muffler was used on three different engines. The pipe diameters were held constant. The figure clearly illustrates that the choice of a muffler becomes increasingly important as larger engines are employed. In general, the vertical muffler with vertical tailpipe gives the largest overall exhaust noise reduction for a particular engine [4.2]. Dual exhaust systems are sometimes preferred because of the reduced back pressures without extremely bulky muffler sizes. Because of size limitations and availability of parts and back pressure limits, duals can often outperform single verticals. Horizontal systems, with muffler and tailpipe under the truck body, are in general, louder than vertical systems because of the reverberation effect between the road surface and the truck underbody.

No correlation has been observed between muffler volume, length or diameter and noise reduction effectiveness. In fact, the range of sizes of mufflers available for diesel trucks is quite limited and the size appears to have been optimized by the muffler manufacturers for present day space and cost constraints.

4.5. Effect of Muffler Location on Exhaust Noise

It was emphasized in Section 4.3 that there is a strong acoustical interdependence of the three main elements: exhaust pipe, muffler and tail pipe, in controlling the overall exhaust noise.

Pressure waves in the exhaust pipe are reflected by the muffler and, therefore, interact with the succeeding exhaust pulses. Standing waves are set up along the pipe length at frequencies chiefly determined by the exhaust pipe length, gas temperature and engine fundamental firing frequency. If the muffler is located where the pressure fluctuations are largest in the pipe, it offers the highest noise reduction and the least back pressure. The length of the exhaust pipe that produces this optimum condition at the muffler may be found by the formula

$$L = \frac{24.5\, n\sqrt{T+460}}{f_F} \qquad (4\text{-}3)$$

where L is the required exhaust pipe length in ft; n is an integer, i.e. n = 1, 2, 3, ... etc., T is the average exhaust gas temperature in degrees Fahrenheit and f_F is the engine fundamental firing frequency (Equation 4-1).

Table 4-2 shows the effect of muffler location on a truck installation having a total exhaust system length of 23 ft, 2 in. (20 ft. of pipe and 3 ft. 2 in. of muffler length). The engine was a 200-bhp diesel, operated at full load at 2100 rpm [4.6].

TABLE 4-2

Variation of Sound Level and Back Pressure with Muffler Location on a Diesel Truck.

Muffler Location No.	Exhaust Pipe Length	Tail Pipe Length	Exhaust System Back Pressure	Noise Reduction ΔdB
1	4 ft.	16 ft.	.95" Hg.	13.0
2	7 ft.	13 ft.	.71" Hg.	16.5
3	10 ft.	10 ft.	.83" Hg.	7.5
4	13 ft.	7 ft.	1.01" Hg.	10.5
5	16 ft.	4 ft.	.73" Hg.	13.5
6	19 ft.	1 ft.	.72" Hg.	8.0
None	23 ft. 2 in.	0 ft.	.71" Hg.	Base

It can be seen that position No. 2 gives the greatest reduction of noise level with the smallest muffler back pressure. Least reduction of noise level occurs at position No. 3, while the highest back pressure is developed at position No. 4.

Since the optimal muffler location given by formula (4-3) depends on the fundamental firing frequency, that is to say, on the engine speed, calculations should be carried out at the design operating point. Final length adjustments to optimize the muffler location generally require experiment.

4.6. Conclusions

The most obvious conclusion from the available data is that suitable mufflers are available for all diesel engines that will lower exhaust noise levels below the noise levels of the engine and fan, while meeting the manufacturers' requirements for maximum back pressure. The back pressures of most available mufflers are so low that the effect on engine horsepower is not measurable with ordinary instruments.

The sound reduction properties of mufflers depend more on their internal design, materials, etc. than on their physical size. Since the physical sizes of available mufflers do not vary appreciably, the cost of a muffler can be an indication of its noise reduction potential, but is not a reliable guide.

The exhaust pipe should be of heavy material. Leaks in the exhaust pipe can be serious. Flexible joints and pipes should be avoided.

Turbocharging results in reduced exhaust noise levels but the selection of a muffler to take advantage of this noise reduction requires care. Engine brakes can be a troublesome source of noise and also require the selection of adequate mufflers and maintenance of leak-free exhaust systems.

Data should be available from manufacturers as to the acoustic performance of a given muffler on a given engine. However, changes in pipe routing, installation, etc. can have significant effects.

The data from Refs. 4.1 and 4.2 on a range of available mufflers for several diesel engines is condensed for reference in Appendix IV.

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5. INTAKE NOISE

5.1. Air Intake Requirements

Air intake systems of internal combustion engines are primarily designed to provide each cylinder with the maximum charge of dust-free air with as little pressure loss as possible. This dual requirement of efficient filtering and minimum pressure drop generally results in a compromise in design, with the pressure drop assuming primary importance. Fortunately, the filtering element has desirable acoustical properties for controlling the noise produced by the sudden induction of air into the engine. The increased area required to prevent excessive pressure drop at the filter also acts as an expansion chamber much like the reactive mufflers used in exhaust systems. The use of snorkels also has a beneficial effect on intake noise levels. In general, very little extra effort is required in the air intake filter selection and installation to achieve the necessary noise reductions.

5.2. Pressure Losses in the Intake System

The allowable pressure drops for diesel engine intake systems are smaller than the back pressures permitted in exhaust systems (1.0 to 1.5 in. of Hg compared to 3 to 4 in of Hg). This is because the upstream pressure for the inlet of naturally aspirated engines can at most be atmospheric (30.0 in Hg) and small pressure losses can exert an appreciable influence on the amount of air being admitted to the engine. The intake pressure losses are termed "intake restriction" and are generally expressed in **inches of water.** (1 inch Hg = 13.6 inches H_2O).

Figure 5.1 shows the results of varying the intake restriction and exhaust back pressure on Detroit Diesel V-71 series engines. An increase in intake restriction from 10 to 30 inches of H_2O results in a horsepower loss of nearly 4 percent accompanied by increased specific fuel consumption, exhaust temperature and smoke output. For a given intake restriction, a threshold increase in exhaust back pressure from 1 to 3 inches of Hg will result in a loss of horsepower of the order of 1 1/2 per cent only.

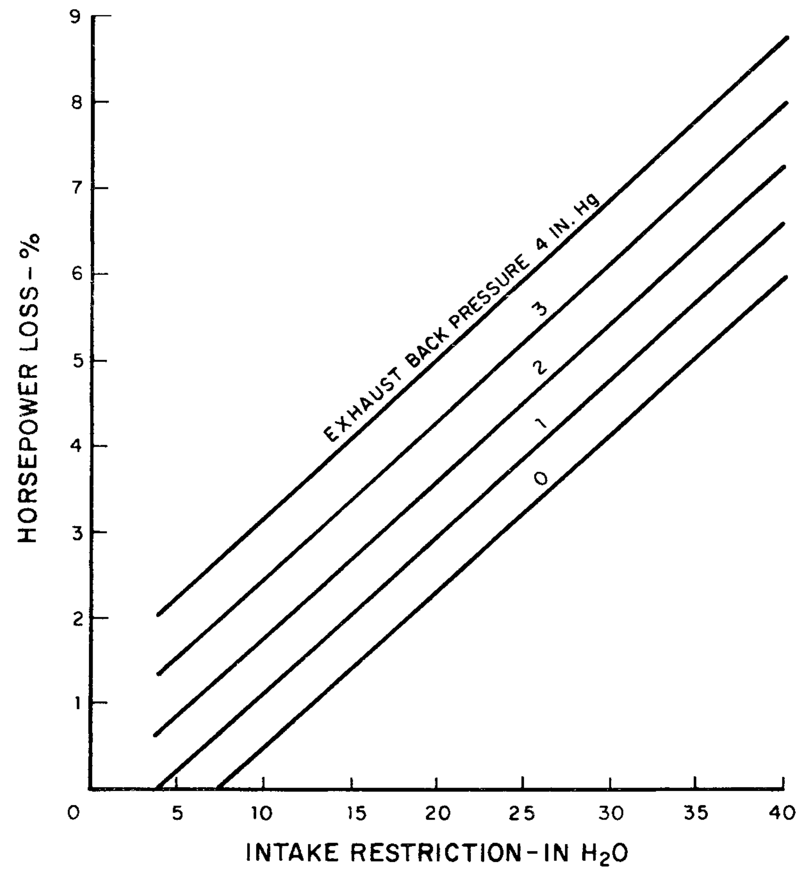

FIG. 5.1 EFFECT OF VARYING INTAKE RESTRICTION ON BRAKE HORSEPOWER OF DETROIT DIESEL V-71 SERIES ENGINES

Proper design of the intake system is thus extremely important from a performance viewpoint.

A good intake filter from the noise viewpoint is not necessarily one that has the most restriction. Restriction or pressure loss is a function of the flow rate, filter element active area, filter element pore size and the amount of dust collected in the pores.

5.3. Mechanism of Noise Generation

Intake noise is produced by both the opening and closing of the inlet valve. At its opening, the pressure in the cylinder is usually above atmospheric and a sharp positive pressure pulse sets the air in the inlet passage into oscillation at the natural frequency of the air column. This oscillation is rapidly damped by the changing volume caused by the piston motion downward. Closing of the inlet valve produces similar oscillations, which are relatively undamped. In the diesel engine, air inlet noise is generally observed in the low and middle frequencies (up to 1000 Hz). In the gasoline engine, this inlet noise may also be important in the higher octave bands owing to the flow noise produced in the carburetor.

Typical unsilenced intake noise levels for truck diesel engines at high idle vary between 70 dB(A) and 85 dB(A), measured at 50 ft from the engine inlet. Production air filters used on most trucks provide a noise reduction (Insertion Loss) of from 9 to 22 dB(A). In case of eleven trucks with Detroit Diesel Engines and production model intake filters [5.1], intake noise exceeded the noise levels from the remaining components in only one case. Six trucks had intake noise levels at least 14 dB(A) below the overall noise levels so that further reduction of intake noise would not be of any benefit. The remaining trucks would show overall noise reductions of 0.5 to 3 dB(A) for a 6 dB(A) reduction of intake noise. Of course, if the noise from remaining components were lowered, intake noise would assume greater importance for a greater proportion of trucks.

Intake filters act as silencers because of the sound absorption properties of the filter element and because of the area changes. Additional silencing may be provided by designing flow passages to restrict line-of-sight transmission.

Figure 5.2 shows the cutaway view of an oil bath type of intake filter. Heavy duty oil bath cleaners are good noise suppressors, have low restriction, but are not as efficient in removing smaller particles as dry type filters and have temperature limitations. Three typical dry type filter designs are shown in Figure 5.3.

Cleaners that have large, flat sections of sheet metal such as the one shown in Figure 5.3(a) can radiate significant amounts of noise from mechanical vibrations, especially for engine mounted intakes. If the use of such a cleaner cannot be avoided in an outside mount configuration, and the radiated sound is found to be excessive, damping treatment should be applied to the flat surfaces and heavy rubber connectors used in the piping for isolation. Air inlet hoods also fall under this precaution since they are almost always in a position that can radiate directly to the spectator. Large canister type dry filters (Fig. 5.3(b)) are designed to replace oil bath filters and generally give good acoustic performance with intermediate air cleaning efficiency.

Heavy duty two stage dry type cleaners can give optimum air filtration and still provide acceptable noise attenuation. Fig. 5.3(c) shows an advanced design of a two stage dry type air cleaner. Fins impart rotation to the entering air flow so that larger particles are separated out by centrifugal action before the air enters the filter element.

5.5. Obtaining Maximum Noise Reductions from Intake Filters

All types of filters are available with different duct configurations to allow mounting in all conceivable engine and body configurations. Use of

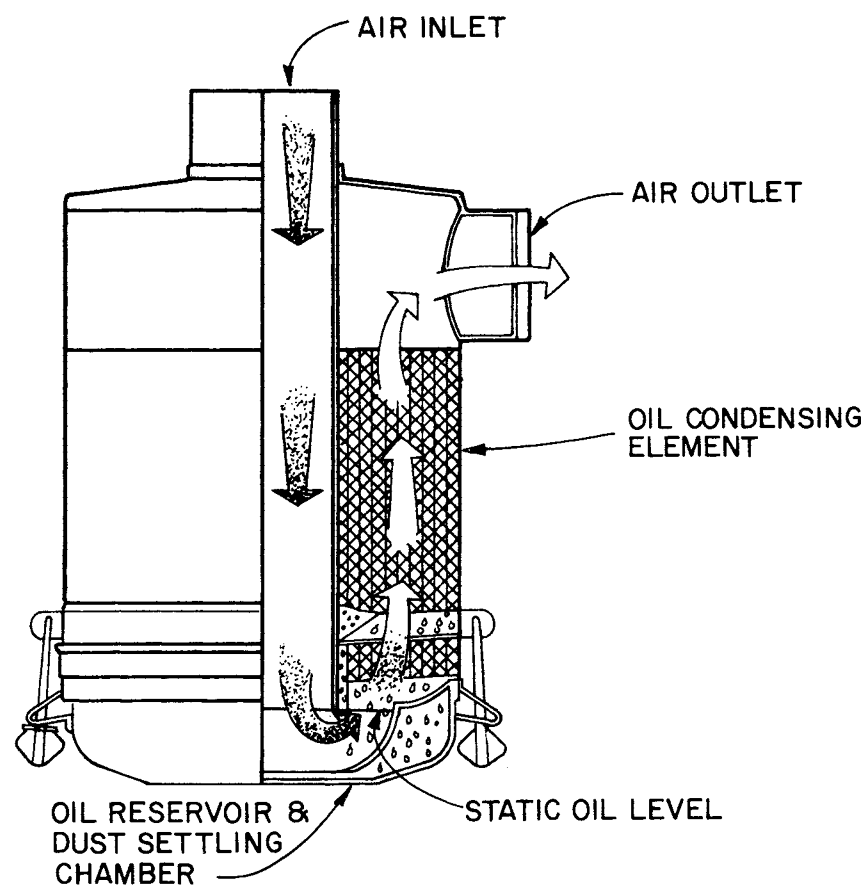

FIG. 5.2 CUTAWAY VIEW OF AN OIL BATH TYPE OF INTAKE FILTER

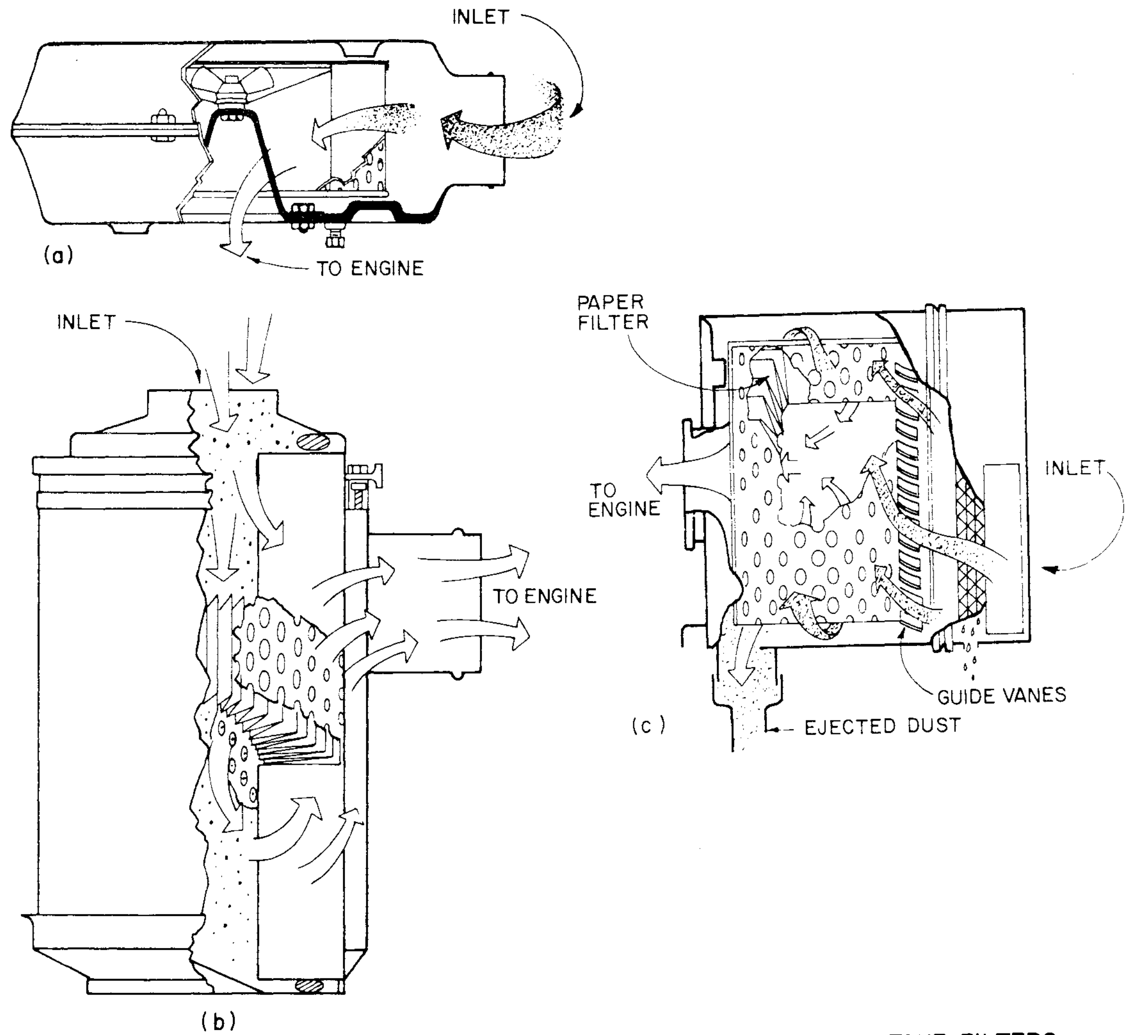

FIG. 5.3 CUTAWAY VIEWS OF THREE DRY TYPE INTAKE FILTERS

(a) SINGLE STAGE (b) CANISTER (c) TWO STAGE should be avoided as much as possible. Most rubber sections are not good acoustic barriers and radiate excessive amounts of noise because of their pulsating walls.

Attachment of the air cleaner or its piping to the cab can result in high noise and vibration levels in the cab. Isolation type mounts should be used to alleviate this problem and air inlet components, especially the inlet, should be located as far away from the operator as possible. The use of heavy wall round metal piping is recommended between the air cleaner and the engine air inlet. One short heavy rubber connector may be inserted in the piping to isolate vibrations.

Leaks can result in high noise levels.

Some characteristic intake filter placements for modern diesel trucks are shown in Fig. 5.4. The use of intake snorkels in cab-over configuration helps in reducing spectator noise. The intake contributed noise level at 50 ft. may be reduced by as such as 5 dB(A) with the use of a properly designed snorkel.

For maximum quieting, an additional intake silencer can be installed between the air cleaner and the engine inlet. These devices are not particularly expensive, are easy to install and will do a good job of absorbing the higher frequency noises. The silencer should be installed as close to the engine inlet as possible.

Data on intake filters for many truck diesel engines is presented in Appendix IV. This data was obtained from the results of other DOT sponsored programs reported in Refs. [4.1] and [4.2].

REFERENCES:

5.1 Law, R.M., "Diesel Engine and Highway Truck Noise Reduction ", SAE Paper No. 730240, Jan. 1973.

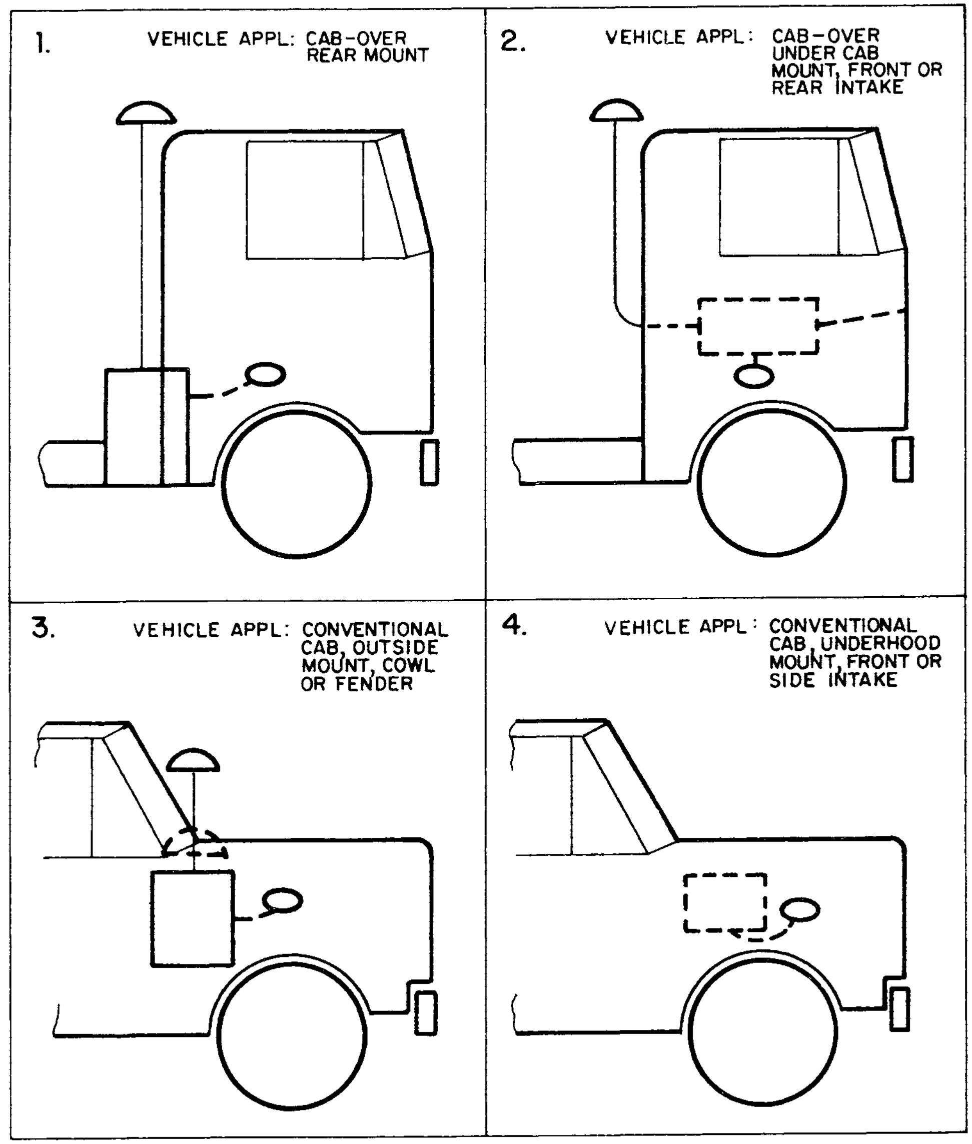

FIG 5.4 TYPICAL INTAKE FILTER PLACEMENT FOR MODERN DIESEL TRUCKS

6. CONTROL OF FAN AND COOLING SYSTEM NOISE

6.1. Introduction

The cooling system fan is a major source of noise for trucks. Sound levels of the fan noise at 50 ft* vary from near 70 dB(A) to almost 90 dB(A) depending predominantly on the fan-blade tip-speed and the position of radiator shutters. Other components of the cooling system generate noise but are of secondary importance. Noise from the water pump, belts and pulleys, water flow within the cooling system, and air flow through the radiator and through the engine compartment contribute very little to the overall truck noise.

6.2. Fan Noise

Figure 6.1 portrays the important parameters of a truck fan and radiator type cooling system. Design characteristics that significantly affect the cooling system noise are: fan tip speed, coverage of the fan by the shroud, and clearance between the radiator, fan and engine.

The 1/3 octave spectrum of the noise from a 26" diameter 6 bladed fan operating at 2500 rpm is shown in Fig. 6.2. The data was obtained by eliminating all other truck noise sources by turning off the engine and driving the fan with a secondary source. The fan noise is characterized by a pure tone at the blade passage frequency f_b, and its harmonics, where

$$f_b = \frac{N_f B}{60} \text{ Hz} \qquad (6\text{-}1)$$

N_f is the fan rpm and B is the number of blades**. In addition, there is a broad band noise component which is distributed over a wide range of frequencies. Although the overall sound pressure level with no frequency weighting (i.e., using the "lin" or C scale on the sound level meter) is

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* SAE J366a Test

** It is assumed for this equation that the blades are equally spaced.

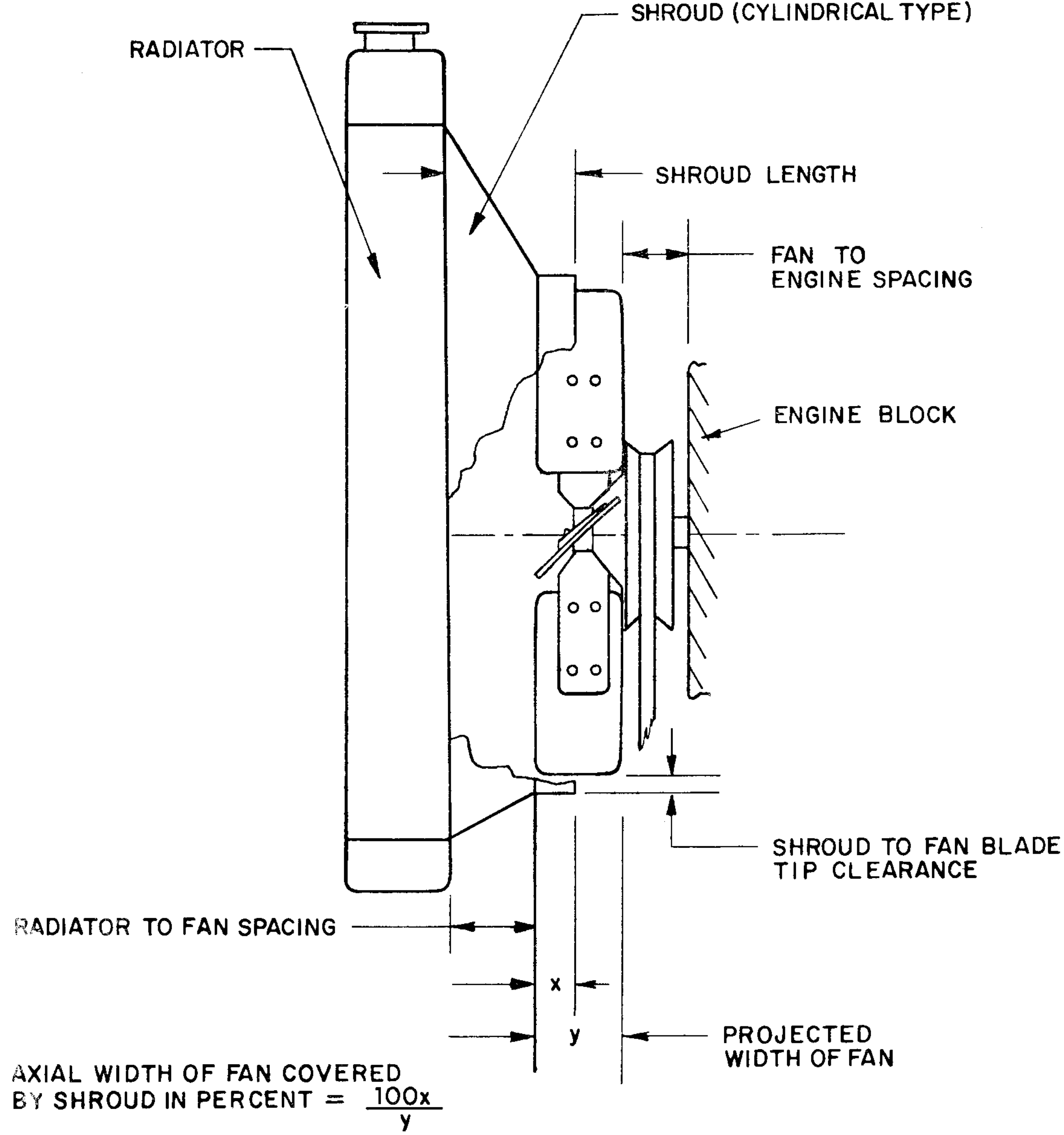
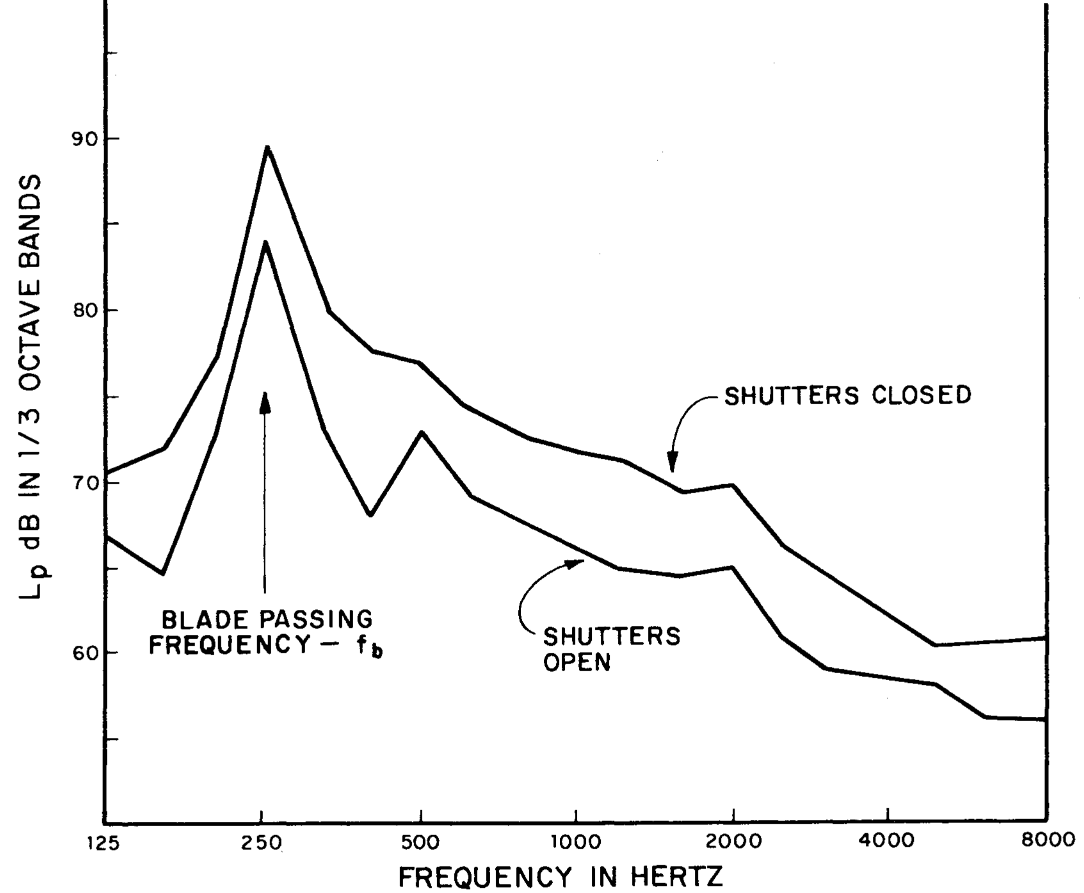


FIG 6.1 COOLING SYSTEM NOMENCLATURE



	SHUTTERS	
	CLOSED	OPEN
OVERALL LEVELS	84.2	79.0 dB(A)

INTERNATIONAL HARVESTER
DATA - SAE J 366a TEST

26" DIAMETER FAN
6 BLADES
2.4" PROJECTED WIDTH
2500 RPM FAN SPEED

FIG 6.2 SPECTRUM OF COOLING FAN NOISE

dominated by the pure tone at the blade passing frequency (250 Hz in this case), the pure tone components and the broadband component in general contribute equally to the A-weighted sound levels at 50 ft.

It is also seen from Fig. 6.2 that radiator shutters, when closed to block the flow of air through the radiator, increase the fan noise approximately 5 dB at all frequencies by causing the fan to operate in a stalled condition.

Radiator shutters also change the directional characteristics of the fan noise. When closed the shutters block sound transmission through the radiator causing the sound levels in front of the truck to decrease.

Data taken in studies of truck fan noise [6.1, 6.2, 6.3] show a strong dependence of the noise on the fan blade tip speed V_t. At tip speeds of interest to truck operators sound level is given by the relation

$$\text{Sound level dB(A)} = 50 \log_{10} V_t + \text{constant} \qquad (6\text{-}2)$$

where

$$V_t = \frac{\pi}{720} N_f D_f \text{ ft/sec} \qquad (6\text{-}3)$$

N_f = fan rpm

D_f = fan diameter in inches

The constant depends on other cooling system variables like shroud, radiator, engine compartment, etc. The dependence of sound level on fan rpm for a typical design is shown in Fig. 6.3.

Detailed studies of noise from tractor fans indicate that fan blade pitch and camber also affect the radiated noise [6.4]. However, the effects are very small and not of immediate importance.

The axial width of the fan covered by the shroud(see Fig. 6.1) influences the fan-noise as well as the air flow through the radiator at a given fan rpm. Both the noise and airflow increase as the fan coverage increases from zero to 50 or 60% of the blade width for all types of shrouds. However, the airflow increases by a much larger percentage (up to 25%) compared to the sound level increase which is generally less than 5%. This is a very important result, because with the optimum shroud coverage a reduction of fan rpm would be permissible without sacrificing the cooling capacity of the system, but resulting in significant noise reductions. This is discussed further in Section 6.3.1.

The clearance between the fan tip and the shroud also affects the flow rate and noise. Fan noise increases slightly with increasing clearance. The air flow rapidly increases with decreasing clearance for small clearances (less than 0.5 inch). Increases in clearance beyond about 0.5 inch do not materially diminish air flow.

Noise and air flow also increase with the number of blades on the fan, but in different proportions so that larger blade numbers deliver a given flow at a lower sound level than do fewer.

A number of different flexible fan designs have been proposed for use in trucks. These fans are designed so that the projected width of the fan decreases as the fan rpm increases. Thus, at high rpm the fan blades flatten out and reduce the horsepower required by the fan. However, the air flow delivered by the fan is also reduced so that in some applications these flexible fans cannot deliver the required air flow at any rpm. The flexible fans are only slightly quieter than conventional fans because projected width does not have a large effect on fan noise.

Conventional cooling fans used in diesel-engined trucks are not well designed aerodynamically. The blades are of uniform thickness and are considerably less aerodynamically efficient than contoured cross-sections. The spider or central portion of the fan is not aerodynamically designed so that air over this portion produces turbulence and noise. Rivets used to attach the blades to the spider also disturb the flow over the blades and increase drag. Despite these shortcomings, tests have shown that conventional fans can be used as components of acceptably quiet cooling systems.

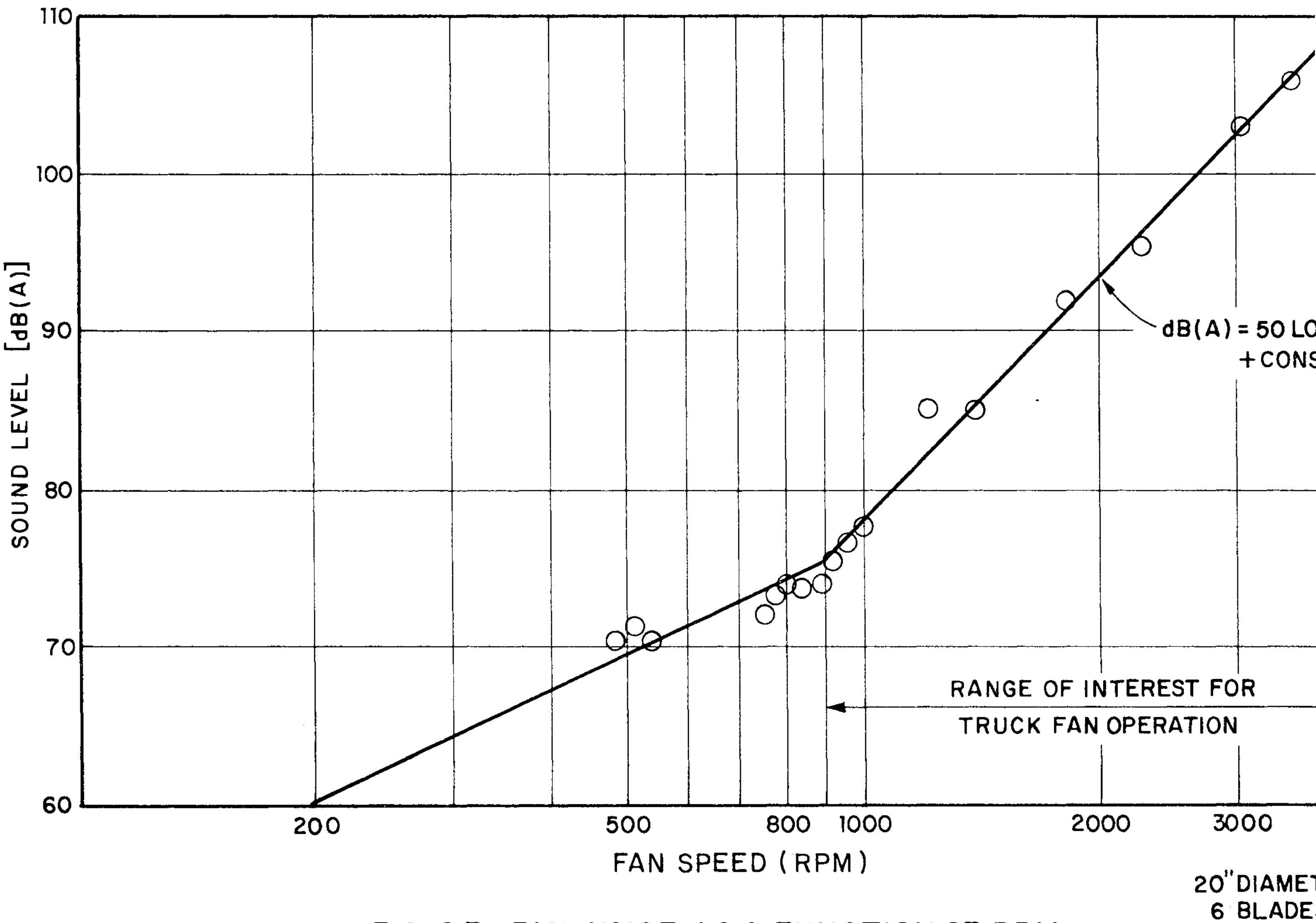


FIG. 6.3 FAN NOISE AS A FUNCTION OF RPM
(SOUND LEVELS AT 6 FT. FROM FAN SHAFT)

the engine compartment is not designed to allow a smooth flow of air out of the compartment. As a result a large amount of air recirculation and turbulence occurs so that the efficiency of the truck fan is approximately one-half that of an axial flow ventilating fan in a duct.

6.3 Methods of Noise Control

A number of truck and fan manufacturers have developed and evaluated new fan designs to determine whether or not they can be used to reduce cooling system noise. A few designs have led to small reductions in noise. Many other designs have either given no reduction or have been unable to move the required amount of air through the radiator. Thus, a program of fan noise control must go beyond fan redesign. Indeed, it is often possible to achieve the required noise control without changing the fan but by carrying out other improvements in the cooling system which allow the fan rpm to be reduced. This approach provides the additional benefit of reducing fan horsepower.

A second method of fan noise control is to use specially designed mufflers. Fan mufflers have not been used on trucks because of space limitations, the requirements for additional fan horsepower, cost and maintenance problems.

The potential for reducing fan noise hinges on the possibilities for maximizing the cooling rate at a given fan speed, thereby allowing the minimization of fan speeds and/or "fan on" time. There are several approaches to such an optimization, and the most promising of these are:

Improved fan shrouds
Increased coolant temperatures
Increased radiator to fan and fan to engine clearance
Radiator redesign

Variable speed fan drives circumvent the problems of improving the efficiency by reducing fan speeds when the engine does not require maximum cooling capacity. In many cases this results in significant noise reduction at highway speeds while reducing the required fan horsepower.

6.3.1. Fan Shrouds

The cooling fan shroud plays a major role in determining the effectiveness of the fan in moving air through the radiator. When no shroud is used the air recirculates around the blade tips so that the flow through the radiator is greatly reduced. In addition, the flow over the fan blades becomes more turbulent so that the fan noise increases. The shroud also results in a more uniform distribution of air flow over the radiator, thereby increasing the effectiveness of the radiator. Thus, addition of a shroud allows a significant reduction in fan rpm (and fan power) for the same heat transfer and a large reduction in fan noise. The exact amount of the reduction depends on the initial configuration and the shroud design. Therefore, generalized statements as to the noise reduction potential are difficult to make.

Three types of shrouds that are in use or have been proposed for use on trucks are shown schematically in Fig. 6.4. The simplest design is the orifice shroud. The cylindrical type shroud is also fairly simple in design and is commonly used on trucks. The venturi or contour shroud has been found to be effective when small tip clearances (1/4 inch or less) can be maintained. When the fan exhausts into open space the venturi shroud gives both a reduction in noise at a fixed fan rpm and an increase in air flow through the radiator.

When a shroud is used, it is important that all air gaps or leaks between the shroud and the radiator be sealed. Tests with a standard production shroud have shown that by carefully sealing all gaps the fan speed can be reduced while maintaining the same air flow through the radiator and the same heat transfer. The reduction in sound level due to the decreased fan speed was approximately 3 dB(A). An additional 2 dB(A) reduction in sound level was obtained by increasing the spacing between the fan and the radiator from 4" to 7". It is believed that the increased spacing improves the air flow distribution over the radiator.

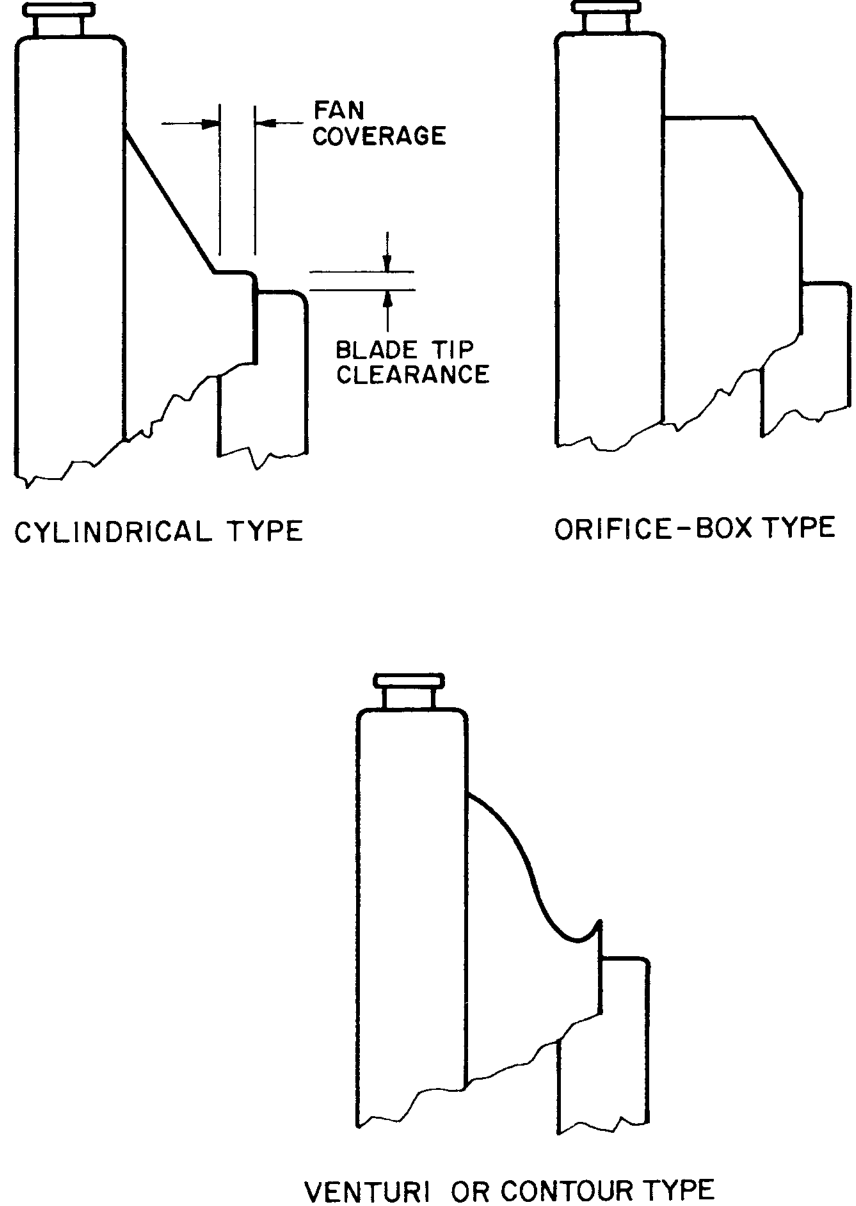


FIG 6.4 TYPICAL FAN SHROUDS

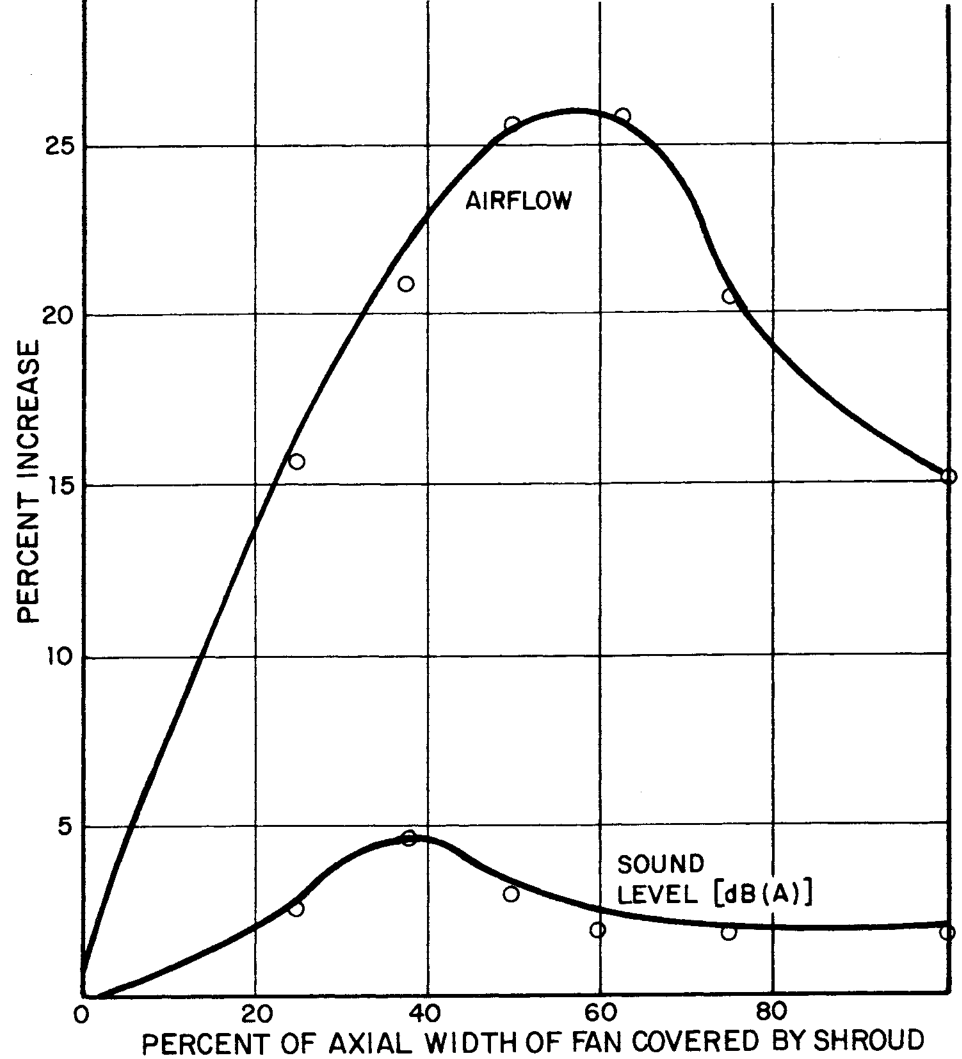
shown in Fig. 6.5. The air flow is increased as much as 25% with the optimum coverage equal to 50% to 60% of the blade projected width. This value of optimum coverage was found to be the same for all fan speeds and for four typical shroud designs. If the fan speed were adjusted to give constant air flow, we would expect the fan noise to vary ±2.5 dB(A) depending on fan coverage.

The effect of tip clearance on air flow and noise for the cylindrical shroud is shown in Fig. 6.6. For constant air flow the noise level would vary ±2 dB(A). Tests on the venturi shroud have indicated that the noise level with 1/4" clearance is about 8.5 dB(A) below the level for 1" clearance at the same heat transfer rate.

The minimum blade tip clearance in current production trucks is limited by the relative motion of the fan (connected to the engine) and the shroud (connected to the radiator). A clearance of 1" is acceptable on most existing trucks. Reduction of this clearance to 1/4" would require mounting the shroud from the engine and providing a flexible seal between the radiator and the shroud. The noise reduction achievable with small tip clearance, engine-mounted shrouds and slower turning fans can exceed 15 dB(A).

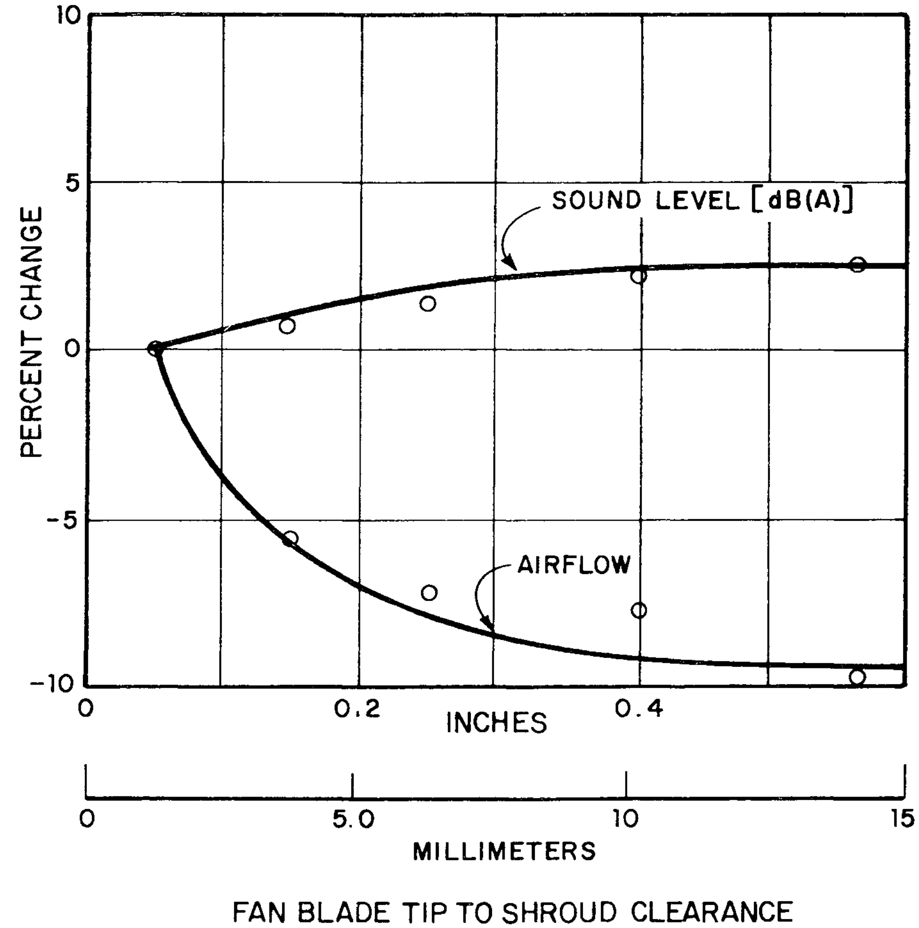
6.3.2 Variable Speed Fan Drives

In most trucks the fan is driven at a fixed ratio of engine speed. This ratio is selected so that the fan provides sufficient air flow through the radiator to meet engine cooling requirements under the most adverse conditions. Ambient air temperatures as high as 130° F normally set the designed cooling capacities. This design temperature is commonly referred to as Air-to-Boil or ATB, which is the ambient air temperature in degrees F at which the coolant entering the radiator will begin to boil under sustained low speed operation of the vehicle under full load. Because the fan speed is fixed by engine speed, the cooling system has sufficient over-capacity during most truck



NOTE: DATA PROVIDED BY INTERNATIONAL HARVESTER CO.

FIG. 6.5 EFFECT OF FAN COVERAGE ON AIR FLOW AND NOISE
(FAN SPEED HELD CONSTANT)



DATA : PROVIDED BY INTERNATIONAL HARVESTER CO.

FIG 6.6 EFFECT OF FAN TIP CLEARANCE ON AIR FLOW AND NOISE
(FAN SPEED HELD CONSTANT)

operations. Three specific problems arise because of ...
First, on cool days the excess cooling capacity reduces engine temperatures to such an extent that engine performance is significantly reduced. Second, engine warm-up times become long. And third, the required high fan speeds result in high fan horsepower consumption and high fan noise.

The first two problems are solved by using shutters located in front of the radiator. When closed, these shutters block the flow of air through the radiator and greatly reduce its cooling capacity. The fan blades operate in a stalled condition when the shutters are closed, which results in increased fan horsepower consumption and noise level. Measurements in SAE J366a pass-by tests indicate that closed shutters frequently increase the noise levels at 50 ft by as much as 5 dB(A). Fan horsepower increases approximately 30% with shutters closed.

The problems of excess cooling capacity can also be solved by using variable speed fan drives. A number of different designs have been used in automotive applications. However, each of these designs accomplish the function of reducing the fan speed so that the cooling system capacity better meets the engine cooling requirements. On-off and variable speed fan drives can be used in conjunction with shutters in such a way that the fan is not driven when the shutters are closed.

Heavy duty variable speed fan drives for truck applications have been available for some time*. At present one cooling system manufacturer offers two heavy duty fan drives. One unit is an on-off drive which senses the air temperature behind the radiator and switches the fan on when the air temperature exceeds a predetermined level. A second unit is a viscous clutch drive which adjusts the fan torque according to the air temperature behind the radiator. A second manufacturer plans to market a heavy duty viscous clutch drive which senses the temperature of the coolant and adjusts the torque applied to the fan according to the coolant temperature. The unit cost is expected to be between $100 and $200. The unit has to be fail-safe so that any failure results in full torque being applied to the fan.

* A list of manufacturers is included in Appendix VI.

Final selection of a drive is based on space requirements, costs, reliability and maintenance. Because of space limitations, variable speed drives are not currently available for all trucks.

The effectiveness of a variable speed fan drive in reducing fan noise and horsepower depends on utilization of the vehicle. Recent tests on interstate trucking operations indicate that the cooling fan can be free-wheeling 95% of the time. Operations in urban areas at low speeds or in hotter climates will result in more use of the cooling fan. However, it is probable that the variable speed fan drive would have a significant effect in reducing fan noise under many operating conditions. The savings in fan horsepower would eventually make up for the increased capital costs and maintenance costs of the drive unit. Table 6-1 gives some quantitative data on the noise reduction that can be achieved due to reductions in fan rpm as ambient temperature varies, for a coolant temperature of 200°F. The noise of a cooling system designed for a 120°F ATB (see p. 89 for definition of ATB) can be reduced by roughly 20 dB(A) on an 80°F day by reducing the fan rpm to 44% of the maximum design rpm, without compromising the cooling capacity.

TABLE 6-1

NOISE REDUCTION TO BE GAINED BY USING
A VARIABLE SPEED FAN DRIVE

Ambient Air Temperature	Percent Reduction in Fan rpm	Fan Noise Reduction
120°F	0%	0 dB(A)
110	21	6
100	36	11.5
90	47	16.5
80	56	21
70	62	25

Coolant temperature is 200°F

Data presented are for a fixed operating condition of the truck (fixed cooling requirement).

6.3.3. Increasing Coolant Temperatures

A substantial improvement in radiator heat transfer can be obtained by an increase in coolant temperature. This improvement means that the cooling requirements of the engine can be met with lower air flow through the radiator and consequently lower fan speeds and noise.

Increased coolant temperatures call for either pressurized cooling systems or use of coolants other than water. The added complexities of a pressurized system increases the cost of the cooling system. This cost must be balanced against the reduction in fan noise and horsepower that can be obtained.

The heat rejection, Q, for a given radiator design is approximately proportional to the square root of the air flow, q, and to the temperature difference between the incoming coolant and the incoming air, $T_{c,in} - T_{a,in}$

$$Q \propto \sqrt{q}\,(T_{c,in} - T_{a,in}) \qquad (6\text{-}4)$$

The air flow is proportional to the fan rpm, and the fan sound level is proportional to 50 $\mathrm{Log}_{10} V_t$, where V_t = fan tip speed. Combining these results and using decibel notation, we find that:

$$\text{Sound Level, dB(A)} \propto 100\ \mathrm{Log}_{10}\left[\frac{Q}{T_{c,in} - T_{a,in}}\right] \qquad (6\text{-}5)$$

Thus, a small increase in coolant temperature can bring about a large reduction in fan sound level. Table 6-2 presents illustrative results to show the noise reduction to be gained by increasing coolant temperature. Also shown are the pressures required to prevent boiling of the water entering the radiator. In this example it is assumed that the ATB requirement will be met if the coolant temperature is kept 12°F below its boiling point on a 100°F day.

TABLE 6-2

NOISE REDUCTION TO BE GAINED BY INCREASING THE COOLANT TEMPERATURE

Coolant Temperature	Coolant Pressure	Percent Reduction in Fan rpm	Fan Noise Reduction
200°F	0 psi	0%	0 dB(A)
210	3.2	17	5
220	6.8	31	9.5
230	11.0	41	13.5
240	15.8	49	17.5
250	22.8	56	21

Ambient air temperature is 100°F.
Data presented are for a fixed operating condition of the truck (fixed cooling requirement).

6.3.4. Increased Radiator to Fan and Fan to Engine Spacing

Increasing the space between the radiator and the fan and between the fan and the engine improves the distribution of air across the radiator and slightly reduces the fan noise. An increase in radiator to fan spacing from 4" to 7" can shown a 2 dB(A) reduction in fan noise for the same heat transfer. Similarly, an increase in fan to engine spacing from 4" to 8" gives approximately 2 dB(A) of noise reduction. This amount of noise reduction probably does not warrant the cost and difficulty of moving the engine in existing vehicles but is worthy of consideration in vehicle design especially when short block engines and/or conventional cabs are involved.

Although radiator to fan and fan to engine spacing are not of major importance in fan noise control, engine components should not be mounted very close to the fan. When objects are placed near the fan they create periodic blade forces which lead to high fan vibration and high radiated noise at the blade passing frequency.

6.3.5. Radiator Redesign

The radiator is the most expensive item in typical truck cooling systems. However, if a new radiator has to be installed anyway, the selection may be based on the following discussion.

Increasing the frontal area of the radiator is an effective means of noise control. For a given heat transfer requirement, increasing radiator area allows significant reductions in fan rpm and noise. For example, a 10% increase in radiator area can give a 4.5 dB(A) reduction in sound level.

Changes in radiator cross-section design can also lead to reductions in fan noise. Increases in the number of radiator fins per inch or increases in the radiator thickness increase the overall effectiveness of the radiator and thus requires less air flow for the same heat transfer. Similarly, use of corrugated or louvred fins increases the heat transfer and coefficient for a given air flow velocity and thereby allows the same overall heat transfer with lower flow velocities. Some data on different radiator designs are shown in Figs. 6.7 and 6.8.

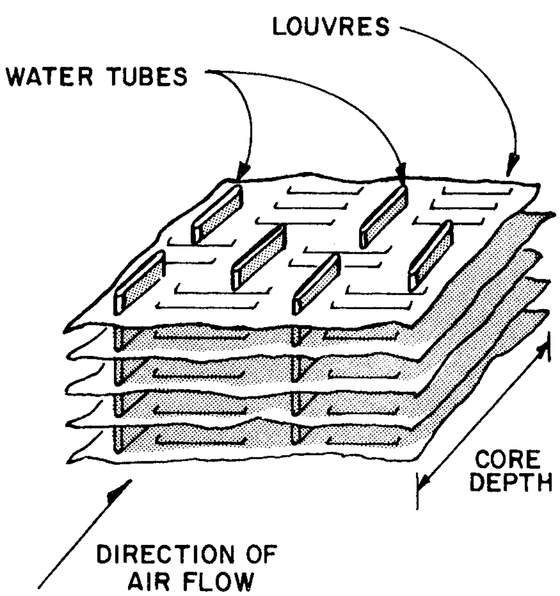


PLATE FIN TYPE OF RADIATOR CORE

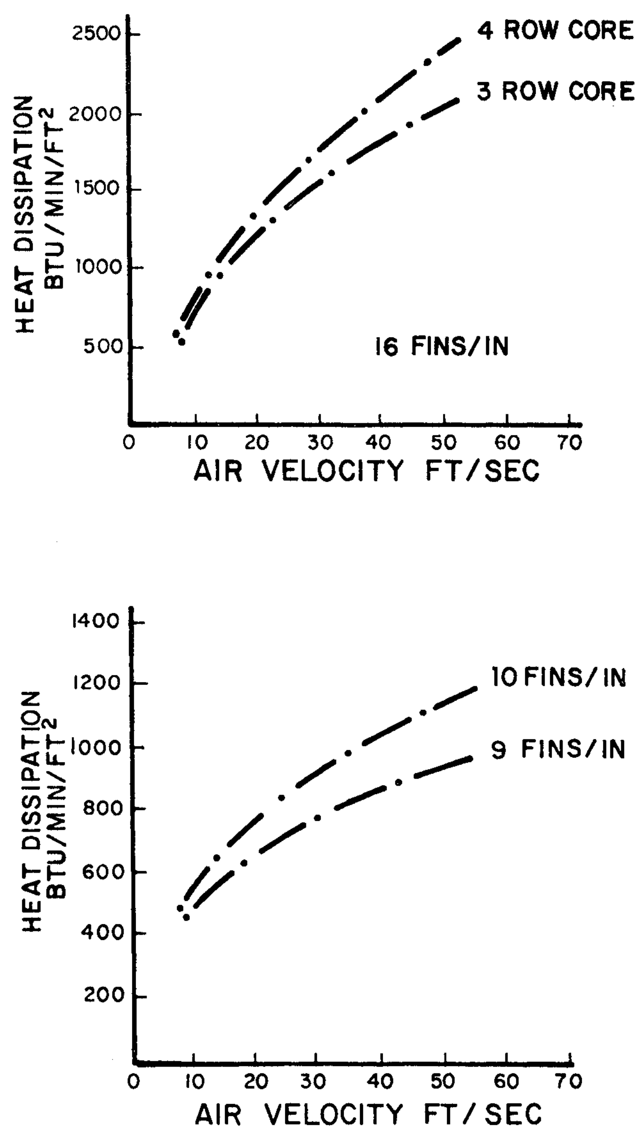
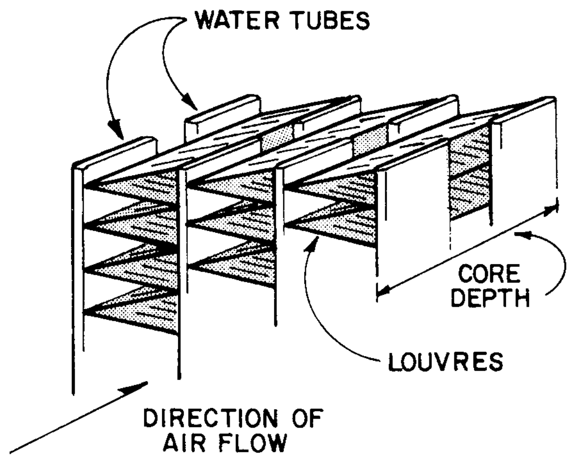
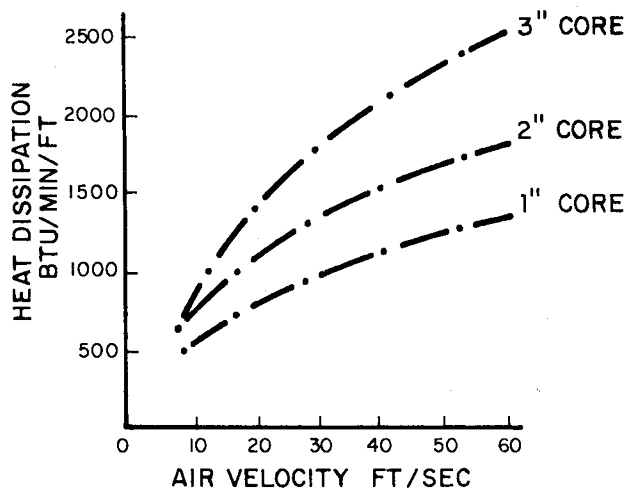


FIG. 6.7 MEASURED PERFORMANCE AS A FUCNTION OF DESIGN



CORRUGATED FIN TYPE OF RADIATOR CORE



180° F MEAN WATER TEMP
80° F INLET AIR TEMP
WATER FLOW 16.8 gpm / ft

FIG. 6.8 MEASURED PERFORMANCE AS A FUNCTION OF AIR FLOW VELOCITY AND CORE DEPTH

It should be remembered that changes in radiator design change the pressure drop through the radiator for a given air flow velocity. Thus, although the air flow velocity remains proportional to the fan rpm, the constant of proportionality varies with design.

6.4. Summary

- The fan is the dominant source of cooling system noise
- Fan noise control is best accomplished by reducing the fan rpm -- sound level varies as 50 Log $(rpm_f . D_f)$, where D_f is the fan diameter
- For a given cooling standard the greatest reductions in fan rpm and noise are gained by
  (1) use of variable speed drives, see Table 6-1
  (2) increasing coolant temperature, see Table 6-2
  (3) increasing radiator area
  (4) using improved shrouds
- Lesser reductions in fan rpm and noise are gained by
  (1) increasing fan to radiator and fan to engine spacing
  (2) radiator redesign
  (3) changing fan design
  (4) increasing coolant flow

Cooling system design considerations are given in greater detail in Appendix V.

REFERENCES:

6.1. Data provided by International Harvester Co., Fort Wayne, Indiana.

6.2. Data provided by Cummins Engine Co., Columbus, Indiana.

6.3. Law, R.M., "Diesel Engine and Highway Truck Noise Reduction," SAE Paper No. 730240, Jan., 1973.

6.4. Hummel, J.W. and Weber, J. A., Noise and Efficiency of Axial Flow Fans", SAE Paper No. 700702, September, 1970.

7. IN-CAB NOISE

7.1. Criteria for Maximum Noise Levels

Because of the potential safety hazard of permanent or temporary hearing loss, the basis for present government regulations has been the so-called hearing damage risk criteria [7.1]. These criteria are based on the probability of hearing loss resulting from exposure in specified proportions of the population. Provisions of motor carrier safety regulations which establish minimum driver hearing qualifications and which require drivers who must use a hearing aid to meet the qualification standard to wear the device while they are driving, are examples of the use of these criteria.

The Bureau of Motor Carrier Safety (BMCS) has recently promulgated a maximum interior sound level for commercial motor vehicles at 90 dB(A) [7.2]. This level is based upon an extrapolation of hearing-damage risk criteria to a 10 hour exposure time, which is the maximum allowable driving time, and relationship between the prescribed test and typical ten-hour driver noise exposure. The rule incorporates the stationary vehicle high-idle test procedure developed during detailed studies of both interior and exterior noise levels. The logic of this procedure and prescribed enforcement level are described in Appendix B of Ref. 7.3. Correlation studies between test data and noise exposure are reported in Ref. 7.4 and 7.5. The BMCS rule is applicable to all motor vehicles manufactured on or after October 1, 1974. On and after March 1, 1975, it will apply to motor vehicles irrespective of the date of manufacture.

7.2. In-Cab Levels

A summary of in-cab noise level measurements for 16 vehicles manufactured between 1967 and 1971 [7.5] is given in Table 7-1. For comparing the levels measured by the stationary high-idle test and the SAE J366a acceleration test that minimizes tire noise, the vehicle speed was limited to 35 mph as opposed to 50 mph as called for in the SAE 336 recommended practice for interior noise measurements. A-weighted sound levels were measured six inches from both sides of the operator, with windows open and closed. Simultaneous interior and exterior noise measurements were made in these tests since the J366a test procedure was observed. No significant difference for in-cab levels is apparent between cab-over and conventional configurations. On the average the window open tests showed only about 1 dB increase in sound levels as compared to the windows closed. Significantly, a number of these pre-1971 diesel trucks (trucks #2 and #16 have gasoline engines) exceed the maximum interior sound levels of 90 dB(A) with the windows closed as established by the Bureau of Motor Carrier Safely. Also, there is no apparent correlation between exterior and interior noise levels. Trucks with higher exterior noise levels do not necessarily have higher interior noise levels.

Table 7-2 lists interior and exterior noise levels of nine trucks with the following noise reduction treatments: removing the fan, covering the engine hood with leaded vinyl and installing a muffler with a high noise reduction (insertion loss) [7.6]. Even though the external noise levels at 50 ft are substantially reduced for the test trucks, the noise inside the cab remains high, indicating that noise transmitted from the engine through the body is the predominant source of in-cab noise.

TABLE 7-1

SUMMARY OF IN-CAB NOISE LEVEL MEASUREMENTS FOR 17 VEHICLES MANUFACTURED BETWEEN 1967 AND 1971

TRUCK #	BODY STYLE	EXTERIOR NOISE at 50 ft (J366a) dB(A)		INTERIOR NOISE at right ear dB(A)			
				BMCS Procedure (High Idle)		J366a Procedure (acceleration)	
		Right	Left	Open Windows	Closed Windows	Open Windows	Closed Windows
00	COE	88	88	85	83	91	88
01	COE	83.5	84	93	92.5	93.5	92
02	Conventional	80	76.5	--	--	84	83
03	Conventional	86.5	86	92	90	92.5	91
04	Conventional	88	85	93	91	94.5	94
05	COE Sleeper w/bedding	89	88	90	90	91	90
05	COE Sleeper without bedding	89	88	--	--	92.5	93
06	Conventional	85.5	85.5	93	92	93.5	94
07	COE Sleeper w/bedding	83	83	90	90.5	89.5	90
07	COE Sleeper without bedding	83	83	--	--	91	90
10	COE	88	89	89.5	88	90	87
11	COE	86.5	86	91.5	91.5	92	92

TABLE 7-1 (CONTINUED)

SUMMARY OF IN-CAB NOISE LEVEL MEASUREMENTS FOR 17 VEHICLES MANUFACTURED BETWEEN 1967 AND 1971

TRUCK #	BODY STYLE	EXTERIOR NOISE at 50 ft (J366a) dB(A)		INTERIOR NOISE at right ear dB(A)			
				BMCS Procedure (High Idle)		J366a Procedure (acceleration)	
		Right	Left	Open Windows	Closed Windows	Open Windows	Closed Windows
12	COE	88	86.5	89	88	90.5	90.5
13	Conventional	87.5	87	86.5	86.5	90.5	88
14	COE	86	87.5	87	87	87.5	87
15	COE Sleeper	87	87	90	90	91.5	92
16	Conventional	80.5	79	82	81	85.5	87.5
17	COE	88.5	89	89	88	92.5	89

TABLE 7-2

INTERIOR AND EXTERIOR NOISE LEVELS OF 9 TRUCKS WITH NOISE REDUCTION TREATMENTS

TRUCK #	EXTERIOR NOISE, dB(A) J366a		INTERIOR NOISE J366a at right ear; dB(A)			
			Std. Truck		Test Truck	
	Std. Truck	Test Truck	Windows Open	Windows Closed	Windows Open	Windows Closed
1	89	81	90	89	87	86
2	89	--	90	90	88.5	88
3	90.6	80.5	94	90	89.5	89.5
4	--	79	93	92	92.5	88
5	86	74.5	94.5	94	90.5	89
6	89.5	84	91	91.5	88	87
7	83	77	84.5	83.5	83.5	83
8	91	80	94	92	88	86.5
9	91	81	91	93	85.5	85

7.3. Control of In-Cab Noise

Interior noise levels of diesel engine trucks tend to be higher than trucks powered by gasoline engines, chiefly because of the higher noise levels and the problems associated with vibration isolation of the heavier diesel engine. Important factors controlling noise levels in cabs are the acoustical properties and workmanship of sound insulation around the operator and on the floor; the care in sealing openings for cables, shift levers, pedals, etc. in the floor and front bulkhead; engine mountings and vibration isolation of the cab from the chassis. Care is also required to prevent sound from entering through the ventilation system.

The purpose of acoustic treatment is twofold: to keep exterior noise from entering the cab by providing a barrier to the sound and to prevent the build-up of sound pressures inside the cab due to reverberation by minimizing the reflection of sound energy back into the cab. These different requirements call for two entirely different types of acoustic materials. Barrier type materials should be used on surfaces exposed to noise from the engine compartment like the floor and the lower portion of the cab. Sound absorption materials would be beneficial on all exposed solid surfaces in the cab, especially the roof, on side walls, doors, etc.

Heavy limp materials are necessary for barriers, the mass being the controlling factor. When limp materials cannot be used, damping may have to be added to control vibrations. For floors, heavy duty floor mats and thick rubber boots may be employed. In extreme cases, a layer of lead-loaded vinyl may be inserted under the mats. The lower portions of the cab side and front walls should be treated with a "septum type" material, which is a porous foam sheet with a thin non-porous barrier in between. This type of material would have an added advantage of providing some sound absorption inside the cab.

The best sound absorbing materials are porous open-celled plastics. The degree of absorption is a function of material thickness and frequency of the incident sound: large thicknesses being required for effectiveness at lower frequencies. To prolong the life of such materials in cabs, a perforated board or trim fabric may be fitted on its surface.

When sound absorption materials cannot be used, e.g. on windows and windshield, slanting the surfaces will keep noise from reflecting perpendicularly off one wall to another. This technique is useful also in improving the effectiveness of enclosures, where slanting may be used in addition to absorbing materials.

To prevent the transmission of engine vibrations to the cab, both the engine as well as the cab should be isolated from the chassis. Engine isolation may reduce the exterior noise as well and careful attention to this would prove beneficial. Isolation of the cab from the chassis would be straightforward for cab-over designs. Conventional cabs are more difficult to isolate but the use of rubber pads at mounting locations can be effective. It is generally advisable to consult specialists in the field of vibration isolation or the manufacturers of isolation mounts.

7.4. Side Benefits from Sound Insulation

Most sound absorbent materials are also good thermal insulators. Generous insulation at the roof will reduce the heating and air-conditioning horsepower requirement. Air-conditioning allows the windows and vents to be closed at all times, lowering the in-cab levels further. The ventilation or air-conditioning ducts have to be designed carefully to prevent exterior noise from entering the cab through them. Lining of the ducts with absorbent material and the use of baffles are two methods of reducing noise transmission through ducts.

The design of a sleeper cab can also have appreciable effects on in-cab noise levels. A well-padded sleeper compartment will not only add to the comfort of the sleeping person but reduce noise levels in the cab by 1 to 3 dB(A).

REFERENCES

7.1. "Walsh-Healy Act/Occupational Noise Exposure" Federal Register Vol. 34, No. 96, May 20, 1969. Revised Jan. 24, 1970.

7.2. "Motor Carrier Safety Regulation--Vehicle Interior Noise Levels", Federal Register, Vol. 38, No. 215, pp. 30880, Nov. 8, 1973.

7.3. Leasure, W. A., Jr., et al., "Interior/Exterior Noise Levels of Over-the-Road Trucks: Report of Tests," NBS Technical Note 737, U.S. Dept. of Commerce, Sept. 1972.

7.4. "Correlation of Truck Cab Interior Noise to Currently Legislated (Walsh-Healy) Limits" Reprot prepared for the Motor Vehicle Manufacturers Association, Inc. by Wyle Laboratories, El Segundo, Calif. June 16, 1972.

7.5. Close, W. H. and Clarke, R. M., "Truck Noise -- II. Interior and Exterior A-weighted Sound Levels of Typical Highway Trucks", Report No. OST/TST-72-2, U.S. Department of Transportation, July 1972. (Available through National Technical Information Service, Springfield, Va.).

7.6. Donnelly, T., Tokar, J. and Wagner W., "Truck Noise - VI B, A Baseline Study of the Parameters Affecting Diesel Engine Intake and Exhaust Silencer Design", Report No. DOT-TSC-OST-73-38. Prepared by Donaldson Co., Inc., for the U. S. Dept. of Transportation, Cambridge, Mass.,Jan., 1974.

APPENDIX I PROPOSED EPA NOISE EMISSION STANDARDS

From the Federal Register Vol. 38, No. 144, July 27, 1973

Subpart B - Interstate Motor Carrier Operations Standards

§202.10 Applicability

The provisions of Subpart B shall apply to any motor vehicle with a gross vehicle weight rating in excess of 10,000 pounds operated by a motor carrier in interstate commerce. These provisions apply to the total sound level emitted by a motor vehicle operated under the conditions specified.

§202.11 Standards for Highway Operations

No person shall operate a motor vehicle of a type subject to this regulation at any time or under any condition of highway grade, load, acceleration or deceleration in such a manner as to generate in excess of 86 dB(A) measured with fast meter response at 50 feet from the centerline of lane of travel on highways with speed limits of 35 mph or less; or 90 dB(A) measured with fast meter response at 50 feet from the centerline of lane of travel on highways with speed limits of more than 35 mph. This section shall not be construed as limiting or precluding the enforcement of any other provisions of Subparts B and C of this part.

§202.12 Standards for Level Street Operations 35 mph or Under

(a) Notwithstanding the provisions of §202.11, no person shall operate a motor vehicle upon any street with a speed limit of 35 mph or less and grade not exceeding plus or minus 1 percent in such a manner as to exceed 80 dB(A) measured with fast meter response at 50 feet from the centerline of lane of travel.

(b) This section shall not apply within 200 feet of any intersection controlled by an official traffic control device or within 200 feet of the beginning or end of any grade in excess of plus or minus 1 percent. This section shall not apply when the vehicle flow is not at a constant rate of speed and traffic is congested and requires noticeable acceleration or deceleration.

§202.13 Standard for Operation Under Stationary Test

(a) No person shall operate a motor vehicle which is powered by an engine with engine speed governor which generates more noise than 88 dB(A) measured with fast meter response at 50 feet from vehicle centerline when that engine is accelerated from idle with wide open throttle to govern speed with the vehicle stationary, transmission in neutral, and clutch engaged.

(b) This section applies to the total noise from the vehicle or combinations of vehicles excluding tire noise. It shall not be construed as limiting or precluding enforcement of any other provision of Subpart B of this part.

§202.14 Visual Exhaust System Inspection

No person shall operate a vehicle which has no expansion chamber, resonator or noise dissipative device in the exhaust system or is not equipped with an exhaust gas driven turbocharger, except that gas driven turbochargers alone will not be adequate on vehicles equipped with an engine brake unless such person can show that no such device is needed to enable said vehicle to meet noise standards under Subpart B of this part. Exhaust system components shall be in constant operation and properly maintained. No exhaust system shall be equipped with a cutout, by-pass, or similar device.

§202.15 Visual Tire Inspection

No motor vehicle shall be operated on tires at any time having tread pattern composed primarily of cavities in the tread (excluding sipes and local chunking or irregularities of wear) which are not vented by grooves to the tire shoulder or circumferentially to each other around the tire, unless such vehicle equipped with such tires can be shown not to exceed noise standards under Subpart B of this part.

§202.16 Enforcement Procedures

Under separate rulemaking procedures, the U.S. Department of Transportation will establish specific procedures for enforcement of these standards. Minimum requirements for instrumentation, test sites, and other conditions necessary to insure uniformity and minimum accuracy in testing shall be so prescribed. Procedures for measurement of vehicle sound levels under conditions of varying distance (other than 50 feet) may be prescribed in which case the measurement procedures and sound level limits shall be established to be equivalent to the sound level limits established in this Subpart measured at 50 feet.

sound level.

2. *Instrumentation*—The following instrumentation shall be used, where applicable, for the measurement required:

2.1 A sound level meter which meets the Type 1 requirements of ANSI S1.4-1971, Specification for Sound Level Meters.

2.1.1 As an alternative to making direct measurements using a sound level meter, a microphone or sound level meter may be used with a magnetic tape recorder and/or a graphic level recorder or indicating meter, providing the system meets the requirements of SAE J184.

2.2 A sound level calibrator (see paragraph 5.2.3).

2.3 An engine-speed tachometer (see paragraph 5.1.1).

3. Test Site

3.1 A suitable test site shall consist of a level open space free of large reflecting surfaces, such as parked vehicles, signboards, buildings, or hillsides, located within 100 ft (30 m) of either the vehicle path or the microphone. See Fig. 1.

3.2 The microphone shall be located 50 ft (15 m) from the centerline of the vehicle path and 4 ft (1.2 m) above the ground plane. The normal to the vehicle path from the microphone shall establish the microphone point on the vehicle path.

3.3 An acceleration point shall be established on the vehicle path 50 ft (15 m) before the microphone point.

3.4 An end point shall be established on the vehicle path 100 ft (30 m) from the acceleration point and 50 ft (15 m) from the microphone point.

3.5 The end zone is the last 40 ft (12 m) of vehicle path prior to the end point.

3.6 The measurement area shall be the triangular area formed by the acceleration point, the end point, and the microphone location.

3.7 The reference point on the vehicle, to indicate when the vehicle is at any of the points on the vehicle path, shall be the front of the vehicle except as follows:

3.7.1 If the horizontal distance from the front of the vehicle to the exhaust outlet is more than 200 in (5080 mm), tests shall be run using both the front and rear of the vehicle as reference points.

3.7.2 If the engine is located rearward to the center of the chassis, the rear of the vehicle shall be used as the reference point.

3.8 During measurement, the surface of the ground within the measurement area shall be free from powdery snow, long grass, loose soil, and ashes.

3.9 Because bystanders have an appreciable influence on meter response when they are in the vicinity of the vehicle or microphone, not more than one person, other than the observer reading the meter, shall be within 50 ft (15 m) of the vehicle path or instrument, and that person shall be directly behind the observer reading the meter, on a line through the microphone and the observer.

3.10 The ambient sound level (including wind effects) coming from sources other than the vehicle being measured shall be at least 10 dB(A) lower than the level of the tested vehicle.

3.11 The vehicle path shall be relatively smooth, dry concrete or asphalt, free of extraneous material such as gravel.

4. Procedure

4.1 Vehicle Operation—Full throttle acceleration and closed throttle deceleration tests are to be used. A beginning engine speed and proper gear ratio must be determined for use during measurements.

4.1.1 Select the highest rear axle and/or transmission gear ("highest gear" is used in the usual sense; it is synonymous to the lowest numerical ratio) and an initial vehicle speed such that at wide-open throttle the vehicle will accelerate from the acceleration point:

(a) Starting at no more than two-thirds (66%) of maximum rated or of governed engine speed.

(b) Reaching maximum rated or governed engine speed within the end zone.

(c) Without exceeding 35 mph (56 km/h) before reaching the end point.

4.1.1.1 Should maximum rated or governed rpm be attained before reaching the end zone, decrease the approach rpm in 100 rpm increments until maximum rated or governed rpm is attained within the

4.1.1.3 Should the lowest gear still result in reaching maximum rated or governed rpm beyond the permissible end zone, unload the vehicle and/or increase the approach rpm in 100 rpm increments until the maximum rated or governed rpm is reached within the end zone.

4.1.2 For the acceleration test, approach the acceleration point using the engine speed and gear ratio selected in paragraph 4.1.1 and at the acceleration point rapidly establish wide-open throttle. The vehicle reference shall be as indicated in paragraph 3.7. Acceleration shall continue until maximum rated or governed engine speed is reached.

4.1.3 Wheel slip which affects maximum sound level must be avoided.

4.1.4 For the deceleration test, approach the microphone point at maximum rated or governed engine speed in the gear selected for the acceleration test. At the microphone point, close the throttle and allow the vehicle to decelerate to one-half of maximum rated or of governed engine speed. The vehicle reference shall be as indicated in paragraph 3.7. If the vehicle is equipped with an exhaust brake, this deceleration test is to be repeated with the brake full on immediately following closing of the throttle.

4.2 Measurements

4.2.1 The meter shall be set for "fast" response and the A-weighted network.

4.2.2 The meter shall be observed during the period while the vehicle is accelerating or decelerating. The applicable reading shall be the highest sound level obtained for the run. The observer is cautioned to rerun the test if unrelated peaks should occur due to extraneous ambient noises. Readings shall be taken on both sides of the vehicle.

4.2.3 The sound level for each side of the vehicle shall be the average of the two highest readings which are within 2 dB of each other. Report the sound level for the side of the vehicle with the highest readings.

5. General Comments—Measurements shall be made only when wind velocity is below 12 mph (19 km/h).

5.1 It is strongly recommended that technically trained personnel select the equipment and that tests are conducted only by qualified persons trained in the current techniques of sound measurement.

5.2 Proper usage of all test instrumentation is essential to obtain valid measurements. Operating manuals or other literature furnished by the instrument manufacturer should be referred to for both recommended operation of the instrument and precautions to be observed. Specific items to be considered are:

5.2.1 The effects of ambient weather conditions on the performance

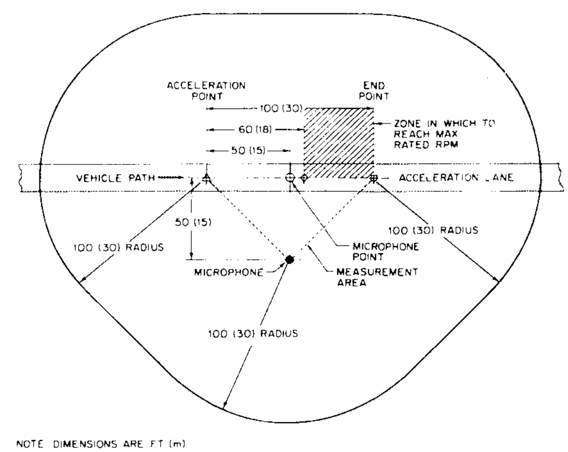


FIG. 1—MINIMUM UNIDIRECTIONAL TEST SITE (SEE PARAGRAPH 3.1)

5.2.2 Proper signal levels, terminating impedances, and cable lengths on multi-instrument measurement systems.

5.2.3 Proper acoustical calibration procedure, to include the influence of extension cables, etc. Field calibration shall be made immediately before and after each test sequence. Internal calibration means is acceptable for field use, provided that external calibration is accomplished immediately before or after field use.

5.3 Vehicles used for tests must not be operated in a manner such that the break-in procedure specified by the manufacturer is violated.

6. References—Suggested reference material is as follows:

ANSI S1.1-1960, Acoustical Terminology

ANSI S1.2-1967, Physical Measurement of Sound

the American National Standards Institute, Inc., 1430 Broadway, New York, New York 10018.

APPENDIX

The SAE recommends that a maximum A-weighted sound level of 88 dB when measured in accordance with the test procedure described above be used as a reference in the design and development of heavy trucks and buses.

An additional 2 dB allowance over the sound level limit is recommended to provide for variations in test site, temperature gradients, test equipment, and inherent differences in nominally identical vehicles.

SOUND LEVEL FOR TRUCK CAB INTERIOR — SAE J336

SAE Recommended Practice

Report of Vehicle Sound Level Committee approved June 1968.

1. Introduction—This SAE Recommended Practice suggests design criteria for maximum truck cab interior sound levels and describes the equipment and procedure for determining this sound level. This practice applies to new motor trucks and truck-tractors and does not include construction and industrial machinery as outlined in SAE J919.

2. Design Criteria

2.1 It is recommended that the octave band pressure levels as set forth in the following tables be used as references in the design and development of new vehicles.

Standard Octave Bands		Preferred Center Frequencies	
Band	Level	Center Frequency	Level
37.5- 75	103	63	101.5
75 - 150	97.5	125	96
150 - 300	92	250	90.5
300 - 600	86.5	500	85
600 -1200	81	1000	79.5
1200 -2400	75.5	2000	74
2400 -4800	70	4000	70
4800 -9600	70	8000	70
(USAS Z24.10-1953)		(USAS S1.11-1966)	

2.2 Trucks meet the design criteria if the sum of the measured band pressure levels does not exceed the sum of the criteria band pressure levels in paragraph 2.1, provided that no measured band level exceeds the corresponding criteria band level by more than 3 db.

3. Equipment

3.1 A sound level meter which meets the requirements of International Electrotechnical Commission Publication 179, Precision Sound Level Meter.[1]

3.2 Alternatively a microphone/magnetic tape recorder/indicating meter system whose overall response meets the requirements of the International Electrotechnical Commission Publication 179, Precision Sound Level Meters.

3.3 A set of octave band-pass filters meeting the requirements set forth in USAS S1.11-1966 or Z24.10-1953.

3.4 A sound level calibrator.

3.5 An engine speed tachometer.

4. Procedure

This specification is based on octave band analysis data (in decibels) measured under the following conditions:

4.1 Locate the microphone 6 in. to the right of, in the same horizontal plane as, and directly in line with, the driver's ear. Orient the microphone vertically upwards.

4.2 Vehicle windows and vents are to be in the fully closed position with all accessories "off."

4.3 The tests are to be conducted on smooth, dry concrete or asphalt road surfaces. No large sound reflecting surfaces should be within 50 ft of the test vehicle. Wind velocity should not exceed 15 mph.

4.4 Select a transmission and/or axle gear ratio so that approximately 50 mph is obtained at rated engine speed.

4.5 Obtain the maximum band pressure level reading in each octave band during accelerations at full throttle from a beginning engine speed of one-half rated engine speed up to the rated speed.

4.6 The average of the two closest readings from three runs shall be reported for each band.

5. General Comments

5.1 Data may be read directly or may be tape recorded for later analysis.

5.2 Sound level tests may be conducted with any type trailer or body on the vehicle, or may be conducted in the "bob-tail" condition.

5.3 On vehicles equipped with radiator shutters, the shutter position causing the maximum sound level should be determined and the tests conducted accordingly.

5.4 It is strongly recommended that technically trained personnel select equipment and that tests be conducted only by qualified persons trained in the current techniques of sound measurement.

[1] Available from U.S.A. Standards Institute, 10 East 40th Street, New York, New York 10016.

BUREAU OF MOTOR CARRIER SAFETY (BMCS) PROCEDURE

(1) Park the vehicle at a location so that no large reflecting surfaces, such as other vehicles, signboards, buildings, or hills, are within 50 feet of the driver's seating position.

(2) Close all vehicle doors, windows, and vents. Turn off all power-operated accessories.

(3) Place the driver in his normal seated position at the vehicle's controls. Evacuate all occupants except the driver and the person conducting the test.

(4) Use a sound level meter which meets the requirements of the American National Standards Institute Standard ANSI S1.4-1971 Specification for Sound Level Meters, for Type 2 Meters. Set the meter to the A weighting network, "fast" meter response.

(5) Locate the microphone, oriented vertically upward 6 inches to the right of, in the same plane as, and directly in line with, the driver's right ear.

(6) With the vehicle's transmission in neutral gear, accelerate its engine to its maximum governed or maximum rated engine speed. Stabilize the engine at that speed.

(7) Observe the A-weighted sound level reading on the meter for the stablized engine speed condition. Record that reading, if the reading has not been influenced by extraneous noise sources such as motor vehicles operating on adjacent roadways.

(8) Return the vehicle's engine speed to idle and repeat the procedures specified in paragraphs (6) and (7) of this section until two maximum sound levels within 2 dB of each other are recorded. Numerically average those two maximum sound level readings.

(9) The average obtained in accordance with paragraph (8) of this section is the vehicle's interior sound level at the driver's seating position for the purpose of determining whether the vehicle conforms to the rule

APPENDIX III. CALCULATION PROCEDURE FOR THE ESTIMATION OF CONTRIBUTED LEVELS

The procedure outlined below is somewhat more complex than the simple procedure described in Part I of this handbook (Sec. 3.6). The advantage of the present method is that it can give more accurate estimates of the contributed noise level of the truck component. However, the procedure requires more computations, and to some degree it also requires a "feel" for the effectiveness of the noise reduction treatment employed during the test procedure. Because of this, the simple procedure in Sec. 3.6 is to be preferred and this complex procedure used only as a second order correction on the simple procedure.

The "fully treated baseline noise" as measured by the procedure of Section 3.5C includes the reduced contributions from all the treated truck noise sources and the "chassis noise". To obtain the true "background" level existing during a measurement with <u>one</u> of the original components uncovered or reconnected, we should subtract from the measured fully treated baseline noise level, the noise level of that particular source when wrapped or the noise level of the auxiliary source. It is apparent that to do this, the noise level of the wrapped or auxiliary source in the absence of all the other sources should be known. This is where the question of "feel" comes in. Because these noise levels cannot be measured directly, estimated residual noise levels of the components are subtracted from the fully treated baseline noise level to obtain the background levels for each of the sources. These "source background" levels are then subtracted from the measured source levels to obtain the "source contributed" levels. To illustrate this, let us recalculate the contributed levels of the truck noise sources for the example given in Section 3.6.

Example:

During a series of measurements to obtain the contributed levels of the majcr noise sources of a heavy diesel truck using the procedure of Section 3.5, the following noise levels were measured on each side (SAE J366b acceleration test):

TEST		NOISE LEVEL, dB(A)	
		Left Side	Right Side
A	Untreated Baseline Noise	86.0	86.0
B	Chassis Noise	73.0	73.0
C	Fully Treated Baseline Noise	76.5	76.5
D	Exhaust System Noise	82.0	83.0
E	Intake Noise	78.0	79.0
F1	Fan Baseline Noise	79.0	79.0
F2	Fan Noise	82.5	82.5
G	Engine Noise	81.0	81.0
H	Transmission Noise	78.0	78.0

To obtain the source background levels, <u>assign</u> residual noise levels for the treated exhaust, intake and engine systems, based on experience and judgement. We choose the following levels:

Residual	Exhaust	noise	70.0 dB(A)
	Intake	noise	66.0
	Engine	noise	70.0
	Transmission	noise	60.0

As a check, add these assumed levels to the chassis noise level (test B). The sum should equal the fully treated baseline noise level (test C).

70 + 66 + 70 + 60 + 73 = 76.5 dB(A)

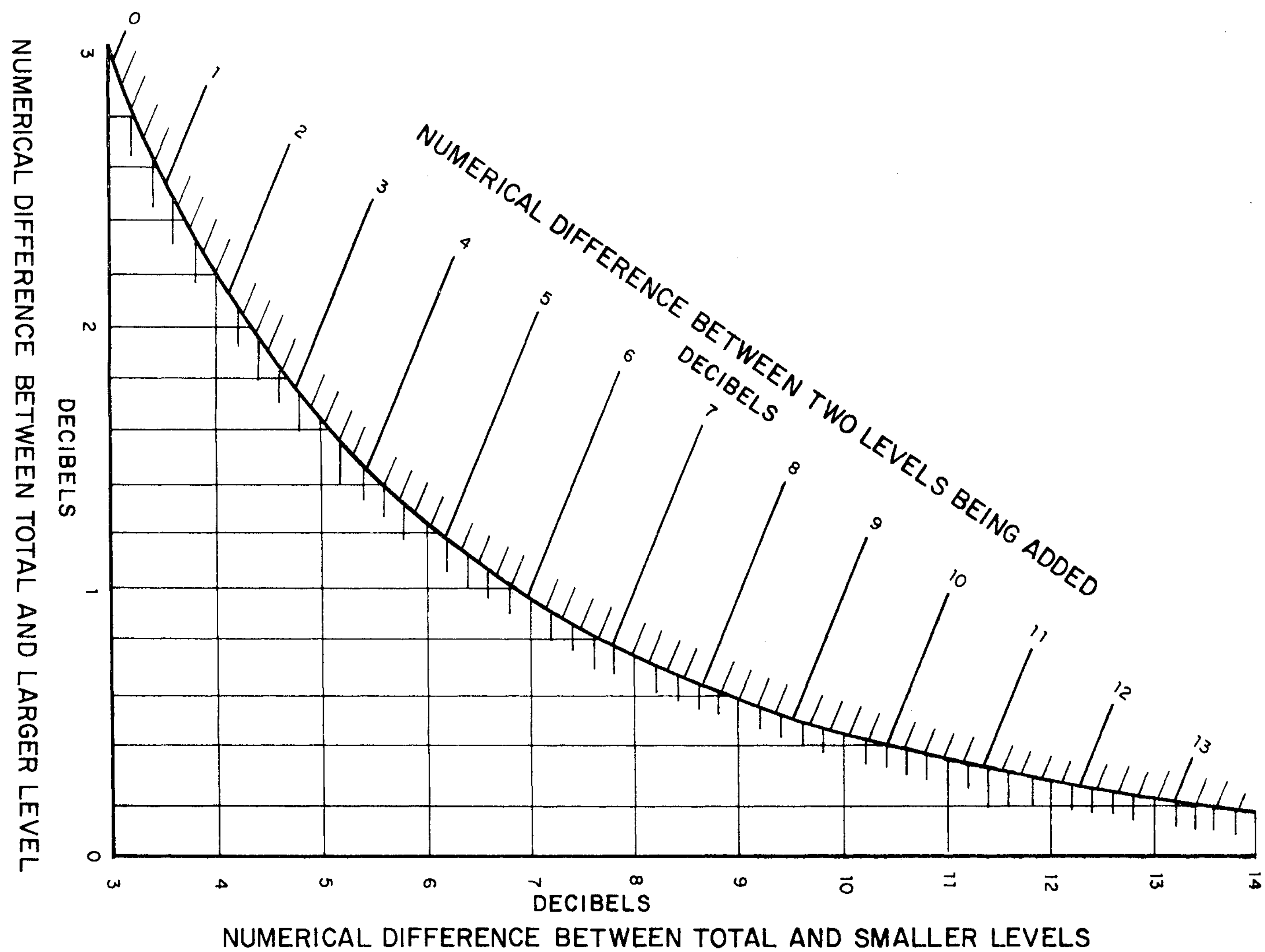
The source backgrounds are, therefore, given by:

Background for exhaust noise	= 76.5 less 70 = 75.4 dB(A)
Background for intake noise	= 76.5 less 66 = 75.0
Background for engine noise	= 76.5 less 70 = 75.4
Background for transmission noise	= 76.5 less 60 = 76.5

The contributed levels are then obtained as follows:

	Left Side dB(A)	Right Side dB(A)
Exhaust Contributed Level	82.0 less 75.4 = 80.9	93.0 less 75.4 = 82.2
Intake Contributed Level	78.0 less 75.0 = 75.0	79.0 less 75.0 = 76.8
Engine Contributed Level	81.0 less 75.4 = 78.2	81.0 less 75.4 = 78.2
Transmission Contributed Level	78.0 less 76.5 = 72.7	78.0 less 76.5 = 72.7
Fan Contributed Level	82.5 less 79.0 = 79.9	82.5 less 79.0 = 79.9
Sum of Contributed Levels	85.7	85.7

Finally, the chassis noise level plus the contributed levels equal: 73.0 + 85.7 = 85.9 dB(A). This checks satisfactorily with the Untreated Baseline noise level of 86.0 dB(A) on both sides.


NUMERICAL DIFFERENCE BETWEEN TWO LEVELS BEING ADDED
DECIBELS
0
1
2
3
4
5
6
7
8
9
10
11
12
13
NUMERICAL DIFFERENCE BETWEEN TOTAL AND LARGER LEVEL
DECIBELS
3
2
1
0
3
4
5
6
7
8
9
10
11
12
13
14
DECIBELS
NUMERICAL DIFFERENCE BETWEEN TOTAL AND SMALLER LEVELS

This appendix is divided into three parts as follows:

IVA: Engine Data and Index to Muffler and Intake Filter Data

IVB: Exhaust System Data*

IVC: Intake System Data

The data presented in this appendix was accumulated during Contracts DOT-TSC-532 and DOT-TSC-533 by Donaldson Co., Inc. and Stemco Manufacturing Company, for the U.S. Department of Transportation. It should be noted that only those mufflers and intake filters for which noise data is currently available are included in the charts that follow. The prices were obtained in late 1972. For details of test configurations, please refer to:

[4.1] Donnelly, T., Tokar J. and Wagner, W., "Truck Noise - VIB, A Baseline Study of the Parameters Affecting Diesel Engine Intake and Exhaust Silencer Design," Report No. DOT-TSC-OST-73-38. Prepared by Donaldson Co., Inc., for the U.S. Department of Transportation, Cambridge, Mass., Jan., 1974.

[4.2] Hunt, R.E., Kirkland, K.C., and Reyle, S.P., "Truck Noise - VIA, Diesel Engine Exhaust and Air Intake Noise," Report No. DOT-TSC-OST-12. Prepared by Stemco Manufacturing Co., for the U.S. Department of Transportation, Cambridge, Mass., July, 1973.

*In the tables, Exhaust system configuration is indicated as follows:

First letter: Single (S) or dual (D) system
Second letter: Muffler orientation, horizontal (H), vertical (V)
Third letter: Tailpipe orientation, horizontal (H), vertical (V)

IV - A

ENGINE DATA AND INDEX

Engine Manufacturer	Model No.	Total Displ. in^3	Number of Cylinders	Max. HP/ rated rpm	Max. Torque/ Rated rpm	Max. Exhaust Back Pres. (in. Hg)	Max. Intake Restrctn ($in. H_2O$)	Muffler Data on Page No
Caterpillar	1140	522	V8	150/3200	277/1800	2.50	25.0	4-4
	1145	522	V8	175/3200	326/1700	2.50	25.0	4-4
	1150	573	V8	200/3000	403/1600	2.50	25.0	4-5
	1160	636	V8	225/2800	474/1400	2.50	25.0	4-6
	1673	638	6	250/2200	690/1600	1.47	30.0	4-7
	1674	638	6	270/2200	805/1400	1.47	30.0	4-8
	1693 T	893	6	325/2100	1000/1450	1.47	30.0	4-9
	1693 TA	893	6	425/2100	1275/1400	1.47	30.0	4-1
Cummins	V8-210	504	V8	202/3300	387/1900	3.00	15.0	4-1
	V-555	555	V8	216/1800	425/1800	3.00	15.0	4-1
	V-903	903	V8	307/2600	707/2600	2.00	15.0	4-1
	VT-903	903	V8	320/2600	775/1800	2.50	15.0	4-1
	NH-220							4-1
	NH-220T							4-1
	NH-230	855	6	220/2100	644/1500	2.00	15.0	4-1
	NH-250	855	6	240/2100	658/1500	2.00	15.0	4-1
	NTC-270	855	6			2.50	15.0	4-2
	NHCT-270CT	855	6	270/2100	930/1300	2.50	15.0	4-2
	NTC-290	855	6	290/2100	837/1500	2.50	15.0	4-2
	NTC-335	855	6	335/2100	930/1500	2.50	15.0	4-2
	NTC-350	855	6	350/2100	1006/1500	2.50	15.0	4-2
	NTA-370	855	6	370/2100	1015/1500	2.50	15.0	4-2
	N-927	927	6	260/2100	720/1500	2.00	25.0	4-2

IV - A

ENGINE DATA AND INDEX

Engine Manufacturer	Model No.	Total Displ. in^3	Number of Cylinders	Max. HP/ Rated rpm	Max. Torque/ Rated rpm	Max. Exhaust Back Pres. (in. Hg)	Max. Intake Restrctn (in.H_2O)	Muffler Data on Page No.
Detroit Diesel	3-53N	159	3	94/2800	198/1800	4.0	16.0	4-28
	4-53N	212	4	130/2800	270/1800	4.0	16.0	4-29
	6V-53N	318	V6	197/2800	421/1500	4.0	16.0	4-30
	8V-53N	424	V8	240/2500	562/1500	4.0	14.0	4-31
	4-71N	284	4	140/2100	385/1200	4.0	15.9	4-32
	6-71N	426	6	210/2100	577/1200	4.0	15.9	4-33
	6-71T	426	6	262/2100	723/1600	2.5	12.0	4-34
	6V-71N	426	V6	210/2100	577/1200	4.0	15.9	4-35
	6V-71T	426	V6	262/2100	723/1600	2.5	12.0	4-36
	8V-71N	568	V8	280/2100	770/1200	4.0	15.9	4-37
	12V-71N	852	V12	420/2100	1154/1200	4.0	15.9	4-40
	8V-71T	568	V8	350/2100	965/1600	2.5	12.0	
Int.Harvester	DV462-B	461	V8	160/3000	307/2000	2.0	20.0	
	D301	301	6	112/3000	228/1600			
	DV550-B	548	V8	200/3000	389/2000	2.0	20.0	
Mack	ENDT475	475	6	190/2400	470/1500			
	ENDT673& ENDLT673	672	6	225/2100	653/1500			
	ENDT673C& ENDLT673C	672	6	250/2100	700/1600			
	END673E& ENDL673E	672	6	180/2100	540/1400			
	END 707& ENDL707	707	6	200/2100	557/1500			
	ENDT865	866	V8	325/2100	1100/1350			
	ENDT866	866	V8	375/2200	1040/1600			
	ENDT-675	672	6	235/1700	855/1200			4-41

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1140 & 1145

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHH	MOM09-0124							39.00	78	
	SHH	MOM09-0168							23.00	76	
	SHH	MTM10-0043	4			36	10	39	27.00	76	
	SHH	MTM10-0048							29.00	77	
	SHH	MOM12-0100	4			26	10 x 15	36	39.00	87	
	SHV	MBM10-0002							32.00	78	
	SHV	MBM10-0049							22.00	74	
	SHV	MOM12-0154	4			26	10 x 15	36	44.00	71.5	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	80	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	79	
	SVV	MSM09-0142							42.00	77	
	DHH	MZM08-5083							22.00	79	
	DHH	MOM09-0170							25.00	76	
	DHH	MTM08-5078							28.00	78	
	DHV	MBM08-5083							18.00	75	
	DVV	MTM08-5078							24.00	78	
Walker	SHV	21465								78	
	DHH	21476								79	

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1150

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHH	MOM09-0124							39.00	78	
	SHH	MOM09-0168							23.00	76	
	SHH	MTM10-0043	4			36	10	39	27.00	76	
	SHH	MTM10-0048							29.00	77	
	SHV	MBM10-0002							32.00	78	
	SHV	MBM10-0049							22.00	74	
	SHV	MOM12-0154	4			26	10 x 15	36	44.00	71.5	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	80	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	79	
	SVV	MSM09-0142							48.00	77	
	DHH	MZM08-5023							22.00	79	
	DHH	MOM09-0170							25.00	76	
	DHH	MTM08-5078							28.00	78	
	DHV	MBM08-5083							18.00	75	
	DVV	MTM08-5078							28.00	78	
Walker	SHH	21476								79	
	SHV	21465								78	

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1160

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHV	MBM10-0002							32.00	78	
	SHV	MBM10-0049							22.00	74	
	SHV	MOM12-0154							44.00	71.5	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	80	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	79	
	SVV	MSM09-0142							42.00	77	
	DHH	MZM08-5023							22.00	79	
	DHH	MOM09-0170							25.00	76	
	DHH	MTM08-5078							28.00	78	
	DHV	MBM08-5083							18.00	75	
	DVV	MTM08-5078							28.00	78	
Walker	SHV	21465								78	
	DHH	21476								79	

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1673T

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHH	MOM12-0108							47.00	77	
	SHV	MOM12-0131	5			26	10 x 15	34	49.00	74	
	SVV	MPM09-0161				44-1/2	9	41	31.00	73	
Walker	SVV	22829	5			44-1/2	9	41	--	73	

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1674TA

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	EXHAUST NOISE LEV @ 50 ft, dB(A) Supplied by Manufacturer	Measured (Ref. 4.1, 4.2)
Alex-Tagg	SVV	2418							62.00	84*	
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.00	79	
	SHV	MOM12-0131	5			26	10 x 15	36	49.00	75	
	SVV	MPM09-0161				44-1/2	9	41	31.00	74	
Walker	SVV	22829	5			44-1/2	9	41	--	74	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1693T

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref 4
Alex-Tagg	SVV	2418							62.00	84.5*	
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.00	81	
	SHV	MOM12-0131	5			26	10 x 15	34	49.00	76	
	SVV	MPM09-0161				44-1/2	9	41	31.00	75	
Walker	SVV	22829	5			44-1/2	9	41		75	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CATERPILLAR 1693TA

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHH	MOM12-2300							63.00	87	
	SHV	MOM12-0131	5			26	10 x 15	34	49.00	77	
	SVV	MPM09-0161				44-1/2	9	41	31.00	76	
Walker	SVV	22829	5			44-1/2	9	41	--	76	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS V8-210

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE L @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Alex-Tagg	SHH	2503							49.00	84*	
	SHH	2532							52.00	83*	
Donaldson	SHH	MOM09-0168							23.00	74	
	SHH	MTM10-0043	4			36	10	39	22.00	77	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	83	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	80	
	SVV	MSM09-0142							42.00	78	
	DHH	MOM09-0170							25.00	72	
	DHV	MBM08-5083							18.00	78	
	DVV	MTM08-5078							28.00	75	
Walker	SVV	22827								83	
	SVV	22828								80	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS V-555

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE L[...] @ 50 ft, dB(A[...] Supplied by Manufacturer	EXHAUST NOISE L[...] @ 50 ft, dB(A[...] Measured (Ref. 4.1,
Alex-Tagg	SHH	2503							49.00	84*	
	SHH	2532							52.00	83*	
Donaldson	SHH	MOM09-0168							23.00	74	
	SHH	MTM10-0043	4			36	10	39	22.00	74	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	83	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	80	
	SVV	MSM09-0142							42.00	78	
	DHH	MOM09-0170							25.00	72	
	DHV	MBM08-5083							18.00	78	
	DVV	MTM08-5078							28.00	75	
Walker	SVV	22827								83	
	SVV	22828								80	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS V-903

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Donaldson	SHV	MOM12-0131	5			26	10 x 15	34	48.68	85	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	82	
	SVV	MUM09-0022	4			44-1/2	9	33	32.00	85	
	DHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	86	
	DHH	MOM12-0100	4			26	10 x 15	36	38.63	86	
	DVV	MUM09-0022	4			44-1/2	9	33	32.00	85	
	DVV	MPM09-0063	4			44-1/2	9	34	33.00	84	
	DVV	MPM09-0115	3-1/2			44-1/2	9	32	38.81	82	
	DVV	MSM09-0135	4			44-1/2	9	35	41.46	78	
	DVV	MPM09-0141	4			44-1/2	9	43	47.89	80	
	DVV	MSM09-0142	4			44-1/2	9	39	42.48	82	
Riker	SHH	94507	5			31-1/2	9 x 14	48	52.37	86*	
	SHV	94006	4			35	9 x 14	49	51.51	85.7*	
	SHV	94506	5			35	9 x 14	52	53.77	83.4*	
	SVV	9XD505	5			44-3/4	9	45	69.32	86.5*	
	DHH	94007	4			31-1/2	9 x 14	45	49.90	84*	
	DVV	9XD354	3-1/2	156		44-3/4	9	39	57.07	83*	
	DVV	9XD404	4	164		44-3/4	9	40	59.10	86.5*	
	DVV	9XD505	5			44-3/4	9	45	69.32	87.3*	
Stemco	DVV	9338	4			44	9	39.5	41.76	71	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS V-903 (CONTINUED)

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A)	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Walker	SVV	22823	5			44-1/2	9	34		88	
	DVV	22809								85	
	DVV	22827								84	
	DVV	22828								80	
	DVV	21174								86	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS VT-903

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE I @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1.
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.00	83	
	SHV	MOM12-0131	5			26	10 x 15	34	49.00	83	
	SVV	MUM09-0074	5			44-1/2	9	34	39.00	82	
	SVV	MPM09-0161	5			44-1/2	9	41	31.00	81	
Walker	SVV	22823	5			44-1/2	9	34		82	
	SVV	22829	5			44-1/2	9	41		81	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NH 220

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE L @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Alex-Tagg	SHH	24	4			26	10 x 15	32	37.74	83.5	
	SHH	2470	4			26	10 x 15	32	37.74	83	
	SHV	2437	4			36	10 x 15	32	37.74	83	
	SVV	2435A	4			44-1/2	9	34	65.59	83	
Riker	SVV	9XD405	4	168		44-3/4	9	40.5	59.10	86	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NH-220 T

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured
Donaldson	SHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	82	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	81	
	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	82	
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	81	
	SHV	MOM12-1000	4			26	10 x 15	34	67.92	78	
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	78	
	SVV	MUM09-0063	4			44-1/2	9	34	33.51	77	
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	79	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	76	
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	51.12	78	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NH 230

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE L @ 50 ft, dB(A Supplied by Manufacturer	EXHAUST NOISE L @ 50 ft, dB(A Measured (Ref. 4.1,
Alex-Tagg	SHH	24	4			26	10 x 15	32	37.74	84	
	SHH	2470	4			26	10 x 15	32	37.74	84	
	SHV	2437	4			26	10 x 15	32	37.74	83	
	SVV	2435A	4			44-1/2	9	34	65.59	82.5	
Donaldson	SHH	MTM10-0043	4			36	10	39	44.34	80	
	SHV	MOM12-0154	4			26	10 x 15	36	43.78	81	
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	84	
	SVV	MPM09-0141	4	84	36	44-1/2	9	43	47.89	81	
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	51.12	85	
		MTM10-0038								81	
Walker	SVV	22828								82	
	SVV	22827								84	
	SVV	22829	5			44-1/2	9	41		85	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NH-250

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Alex-Tagg	SHH	24	4			26	10 x 15		37.34	84	
	SHH	2470	4			26	10 x 15	32	37.74	84	
	SHV	2437	4			26	10 x 15		37.74	83	
	SVV	2435A	4			44-1/2	9	34	65.59	84	
Donaldson	SHH	MOM12-0100	4			26	10 x 15	36	38.63	83	
	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	84	
	SHH	MTM10-0043	4	96	18	36	10-1/8	39	44.34	80	81
	SHV	MOM12-1000	4			26	10 x 15	34	67.92	80	
	SHV	MOM12-0154	4	96	10	26	10 x 15	36	43.78	79	77
	SHV	MOM12-0131	5			26	10 x 15	34	43.68	82	
	SVV	MTM10-0038							63.00	80	
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	79	
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	78	
	SVV	MPM09-0074	5			44-1/2	9	34	38.86	80	
	SVV	MOM09-0161	5	84	36	44-1/2	9	41	51.51	79	
	SVV	MPM09-0141	4	84	36	44-1/2	9	43	47.89	77	78
Riker	SHH	94007	4	156	16	31-1/2	9 x 14	45	49.90	86	80
	SHV	94006	4	216	96	35	9 x 14	49	51.51	85.5	75
	SVV	9XD405	4	168	40	44-1/2	9	40.5	59.10	86	74

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NH-250 (CONTINUED)

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE I @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1.
Stemco	SHH	9400	4			26	10 x 15	40	39.36	81	81.
	SHH	9864	4	112		33	10	45	50.58	80.5	
	SHH or SVV	9344	4	144		44	9	39-1/2	45.10	78.5	
	SHV	9855	4	197		28	10	29	41.06	78	
	SHV	9854	4	208		40-1/2	10	40	48.56	76	76
	SVV	9349	4	144	24	44	9	46	52.23		80
		9349		144	48						78
		9349		144	72						78
	SVV	9300	4	144		44	9	39	35.87	80	
Walker	SVV	22827								84	
	SVV	22828	4			44-1/2	9	43		82	77
	SVV	22829	5			44-1/2	9	41		85	79

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NTC 270

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured
Donaldson	SHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	83	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	83	
	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	84	
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	82	
	SHV	MOM12-0131	5			26	10 x 15	34	48.58	83	
	SHV	MOM12-1000	4			26	10 x 15	34	67.92	81	
	SVV	MOM09-0022	4			44-1/2	9	33	32.01	79	
	SVV	MUM09-0063	4			44-1/2	9	34	33.51	78	
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	81	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	77	
	SVV	MPM09-0161	5			44-1/2	9	41	51.12	79	
Riker	SVV	9XD405	4			44-3/4	9	40.5	59.10	86	
	SVV	9XD505	5	240		44-3/4	9	45	69.32	72.1	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NHCT-270-CT

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE I @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1.
Alex-Tagg	SVV	2435 A5	5			44-1/2	9	34	70.02	38	
Donaldson	SHH	MTM10-0043	4			36	10	39	27.00	82	
	SHV	MOM12-0154	4			26	10 x 15	36	44.00	81	
	SVV	MPM09-0063	4			44-1/2	9	34	34.00	84	
	SVV	MPM09-0141							48.00	82	
	SVV	MPM09-0161	4			44-1/2	9	43	31.00	85	
	SVV	MPM10-0038							63.00	80	
Walker	SVV	22827								84	
	SVV	22828								82	
	SVV	22829	5			44-1/2	9	41		85	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NTC-290

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB Measured
Donaldson	SHH	MOM12-0100	4			26	10 x 15	36	39.00	83	
	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.00	85	
	SHV	MOM12-0131	5			26	10 x 15	34	49.00	83	
	SHV	MOM12-1000							68.00	82	
	SVV	MUM09-0022	4			44-1/2	9	33	32.00	80	
	SVV	MUM09-0074	5			44-1/2	9	34	39.00	81	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	78	
	SVV	MPM09-0161				44-1/2	9	41	31.00	80	
	SVV	MPM09-0197							33.00	77	
Riker	SHV	94006							52.00	83.7*	
	SVV	9XD-404							59.00	82.7*	
Walker	SHH	21174								83	
	SVV	22809								80	
	SVV	22823	5			44-1/2	9	34		81	
	SVV	22828								78	
	SVV	22829	5			44-1/2	9	41		80	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NTC-335

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE I @ 50 ft, dB(A Supplied by Manufacturer	EXHAUST NOISE I @ 50 ft, dB(A Measured (Ref. 4.1.
Alex-Tagg	SHH	2397	5			45	9	33	62.02	85	
	SVV	2418	5			44-1/2	9	34	62.02	84	
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	86	79
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	84	81
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	81	78
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	83	81
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	31.00	81	79
Riker	SHV	94506	5			35	9 x 14	52	53.77	84	7.
	SVV	9XD404	4	168		44-3/4	9	40	59.10	82.4	81
	SVV	9XD405	4	204		44-3/4	9	40.5	59.10	82	7
	SVV	9XD505	5	240		44-3/4	9	45	69.32	72.3	7
Stemco	SVV	9327	5			44	9	38.5	41.28	79	8
Walker	SVV	22829	5			44-1/2	9	41		81	
	SVV	22809								81	
	SVV	22823	5			44-1/2	9	34		83	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NTC-350

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured [illegible]
Donaldson	SHH	MOM12-0108	5	96	18	26	10 x 15	36	47.11	80	8
	SHH	5080B254								77	
	SHV	MOM12-0131	5	96	10	26	10 x 15	34	48.68	82,84	8
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	51.12	80,82	8
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	84	
		MPM09-0197							33.00	79	
Riker	SHH	94007	4	168	16	31-1/2	9 x 14	45	49.90	86.5	
	SHV	94006	4	216	96	35	9 x 14	49	51.51	85.6	
	SVV	9XD404	4	168	40	44-3/4	9	40	59.10	82.4	
Stemco	SHH	9401	5	112	20	26	10 x 15	39.5	41.31		8
	SVV	9327	5	150	48	44	9	38.5	42.93	79	7
Teck	SHH	D179	5	112	20	26	10	29	44.01		8
	SHV	D146	4			41-1/4	10	30	42.27		
	SVV	505D9T	5	150	48	44-3/4	9	27	46.85		8
Walker	SVV	22823	5			44-1/2	9	34			8
	SVV	22829	5			44-1/2	9	41			8

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS NTA 370

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE LE @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Alex-Tagg	SVV	2418	5			44-1/2	9	34	62.02	84	
Donaldson	SVV	MUM09-0074	5			44-1/2	9	34	38.86	84	
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	31.00	82	
Walker	SVV	22823	5			44-1/2	9	34		84	
	SVV	22829	5			44-1/2	9	41		82	

IV-B EXHAUST SYSTEM DATA

Engine: CUMMINS N-927

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Donaldson	SHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	84	
	SHH	MTM10-0043	4			36	10-1/8	39	44.34		8
	SHH	MTM10-0048	4			27-3/4	10-1/8	34		82	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	85	
	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	89	
	SHV	MOM12-0154	4			26	10 x 15	36	43.78	81	
	SHV	MOM12-0176	5			26	10 x 15	37		88	
	SHV	MOM12-1000	4			26	10 x 15	34	67.92	87	
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	86	
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	84	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	82	
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	51.12	85	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 3-53N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Donaldson	SHH	MOM09-0170							25.00	77	
	SHV	MBM08-5083							18.00	81	
	SVV	MTM08-5078							28.00	81	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 4-53N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Donaldson	SHH	MOM09-158							23.00	78	
	SHV	MOM09-158							23.00	75	
	SVV	MSM09-146							26.00	76	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 6V-53N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE L @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	SHH	MTM10-0043	4			36	10-1/8	39	44.34	82	
	SHH	MTM10-0048	4			27-3/4	10-1/8	34		85	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	87	
	SHV	MOM12-0154	4			26	10 x 15	36	43.78	79	
	SVV	MPM19-0063	4			44-1/2	9	34	33.51	85	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	83	
	SVV	MSM09-0142	4			44-1/2	9	39	42.48	81	
	DHV	MBM08-5083								82	
	DVV	MTM08-5078								82	
Riker	SHH	94007	4			31-1/2	9 x 14	45	49.90	88	
	DVV	9XD304	3	156		44-3/4	9	38	55.01	82.8	
Stemco	SVV	9336	4			44	9	39.5	40.40	78	
	DVV or DHH	9345	3.5			36	9	35.3	34.68	78.5	
Walker	SHH	21174								87	
	SVV	22828								83	
	SVV	22827								87	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 8V-53N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured
Donaldson	SHH	MTM10-0043	4			36	10	39	27.00	83	
	SHV	MBM10-0002							32.00	86	
	SVV	MPM09-0141	4			44-1/2	9	43	48.00	84	
	DHH	MOM09-0140							30.00	80	
	SHV	MOM09-0140							30.00	77	
	DVV	MSM09-0146							26.00	78	
Walker	SHH	21465								86	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 4-71N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Donaldson	SHH	MZM08-5008							27.00	85	
	SHV	MBM10-0002							32.00	82	
	SVV	MSM09-0135							41.00	80	
	SVV	MPM09-0115							39.00	84	
Walker	SHH	22808								85	
	SHV	21465								82	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 6-71N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Alex-Tagg	SHH	2458	4			48	9	37	58.94	86	
	SVV	2417C	4			21	10 x 15	28	53.90	84.5	
Donaldson	SHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	86	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	88	
	SHH	MTM10-0043	4			36	10	39	44.34		8
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	89	
	SHV	MOM12-0154	4			26	10 x 15	36	43.78	83	7
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	88	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	85	
	SVV	MSM09-0142	4			44-1/2	9	39	42.48	82	8
Riker	SHH	94307	3.5	156	16	31-1/2	9 x 14	43	48.03	87.2	8
	SH	94007	4			31-1/2	9 x 14	45	49.90	87.2	
	SHV	94306	3.5	216	96	35	9 x 14	43	48.03	86	7
	SVV	9XD405	4	192	40	44-3/4	9	40.5	59.10	87	7
Stemco	SVV	9336	4			44	9	39.5	40.40	78	
	SVV	9349	4			44	9		50.22	77	7
Teck	SHH	D38	4							90.5	
	SHV	D146	4							80.5	
	SVV	405 D9T	4							85	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 6-71T

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(— Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(— Measured
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	81	
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	80	
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	77	
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	81	
	SVV	MUM09-0161	5	84	36	44-1/2	9	41	51.12	78	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 6V-71

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured
Alex-Tagg	SHH	2458	4			48	9	37	58.94	85	
	SVV	2517C	4			21	10 x 15	28	53.90	84.5	
Donaldson	SHH	MTM10-0006	4			27-3/4	10-1/8	32	32.96	86	
	SHH	MOM12-0100	4			26	10 x 15	36	38.63	88	
	SHV	MOM12-0154	4			26	10 x 15	36	43.78	84	
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	81	
	SVV	MUM09-0022	4			44-1/2	9	33	32.01	78	
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	88	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	86	
	SVV	MSM09-0142	4			44-1/2	9	39	42.48	83	
	SVV	MTM10-0038	5			43-1/2	10-1/8	44		84	
	DVV	MTM08-5078	3			29	8-1/2	25		81	
	DVV	MPM09-0115	3.5			44-1/2	9	32	38.81	87	
	DVV	MSM09-0135	4			44-1/2	9	35	41.46	82	
Riker	SHH	94307	3.5			31-1/2	9 x 14	43	48.03	86 *	
	SHH	94007	4					45	49.90	85.6*	
	SHV	94306	3.5			35	9 x 14	43	48.03	84 *	
	DVV	9XD304	3	156		44-3/4	9	38	55.01	85.2*	
	DVV	9XD354	3.5	156		44-3/4	9	39	57.07	86.3*	
Walker	SHH	21174								88	
	SVV	22828								86	

*Vehicle Test Data

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 6V-71T

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured (Ref. 4.1
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	47.11	82	
	SVV	MUM09-0074	5			44-1/2	9	34	38.86	82	
	SVV	MPM09-0161	5	84	36	44-1/2	9	41	51.12	79	
	DHH	MTM08-5078	3			29	8-1/2	25		83	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 8V-71

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Alex-Tagg	SHH	2478	5			42	9	32	70.02	84.5	
	SVV	2533	3.5			56	9	39	73.09	83	
	SVV	2435 A5	5			44-1/2	9	34	70.02	86	
	DVV	2531	4			44-1/2	9	33	66.72	84	
	DVV	2435 A	4			44-1/2	9	34	65.59	84	
Donaldson	SHH	MOM12-0108	5			25-3/4	10 x 15	36	34.00		9
	SHV	MOM12-0131	5			26	10 x 15	34	48.68	86	9
	SHV	MOM12-0176	4			26	10 x 15	37	53.22		8
	SVV	MPM09-0063	4			44-1/2	9	34	33.51	87	8
	SVV	MPM09-0141	4			44-1/2	9	43	47.89	84	8
	SVV	MTM10-0038								83	8
	SVV	WSM09-0212								77	
	SVV	MKM10-0064								81	
	SVV	MPM09-0141	4			44-1/2	9	43	47.89		8
	DHH	MZM08-5008									8
	DHH	MTM10-0043	4			36	10	39	44.34		8
	DHV	MBM10-0002							32.00	83	
	DHV	MOM12-0154	4			26	10 x 15	35	43.78		8
	DVV	WSM09-0211								76	
	DVV	MSM09-0146	3.5			44-1/2	9	39	43.42	77	8
	DVV	MPM09-0115	3.5			44-1/2	9	32	38.81	85	8
	DVV	MSM09-0135	4			44-1/2	9	35	41.46	81	8

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 8V-71 (CONTINUED)

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOISE @ 50 ft, dB(Supplied by Manufacturer	EXHAUST NOISE @ 50 ft, dB(Measured (Ref 4.1
Riker	SHH	94007	4	156	16	31-1/2	9 x 14	45	49.90	85	
	SHV	94006	4	216	96	35	9 x 14	49	51.51	83	82
	SHV	94306	3.5			35	9 x 14	43	48.03	83.2	79
	SHV	94506	5			35	9 x 14	52	53.77	83.7	79
	SVV	9XD405	4	192	40	44-3/4	9	40.5	59.10	86.6	80
	SVV	9XD505	5			44-3/4	9	45	69.32	85.7	79
	SVV	81002	4			46	8-1/4 x 11-1/2	52		83	88
	SVV	10005	4			44-3 /4	10	49		82	85
	DHH	94007	4	156	16	31-1/2	9 x 14	45	49.90	85.4	82
	DHH	94307	3.5			31-1/2	9 x 14	43	48.03	86.2	78
	DHH	94507	5			31-1/2	9 x 14	48	52.37	87	
	DHV	94306	3.5			35	9 x 14	43	48.03	83.9	8
	DHV	94006	4	228	96	35	9 x 14	49	51.51	83.9	8
	DHV	94506	5			35	9 x 14	52	53.77	86	8
	DVV	9XD404	4	158		44-3/4	9	40	49.10	85.7	8
Stemco	SHV	9416	4								8
	SVV	9866								78	8
	SVV	9344	4	243		44	9			80	8
	SVV	9344 & Wye 9867	4								7
	DHH	9345	4								8
	DHV	9416	4								8

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 8V-71 (CONTINUED)

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER Length (in.)	MUFFLER Diameter (in.)	MUFFLER Weight (lb.)	MUFFLER Retail Cost ($)	EXHAUST NOIS[E] @ 50 ft, dB Supplied by Manufacturer	EXHAUST NOIS[E] @ 50 ft, dB Measured
Stemco	DVV	9350	4							78	
	DVV	9338	4	199						80	
Walker	SHV	21465								83	
	SVV	22827								87	
	SVV	22828								84	
	DHH	22808								85	

IV-B EXHAUST SYSTEM DATA

Engine: DETROIT DIESEL 12V-71N

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE L @ 50 ft, dB(A	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured (Ref. 4.1,
Donaldson	DHH	MTM10-0006							33.00	86	
	DHV	MOM12-0154	4			26	10 x 15	36	44.00	84	
	DVV	MPM09-0063	4			44-1/2	9	34	34.00	89	
	DVV	MPM09-0141	4			44-1/2	9	43	48.00	86	
	DVV	MSM09-0142							42.00	83	
	DVV	MTM10-0038							63.00	85	
Riker	DHH	94007							50.00	87.7	
	DVV	9XD-404	4			44-3/4	9	40	59.00	87	
Walker	DVV	22827								89	
	DVV	22828								86	

IV-B EXHAUST SYSTEM DATA

Engine: MACK ENDT 675

Muffler Manufacturer	Exh. System Configuration	Muffler Model No.	Exhaust Pipe Diam. (in.)	Exhaust Syst. Length (in.)	Tail Pipe Length (in.)	MUFFLER				EXHAUST NOISE @ 50 ft, dB(	
						Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	Supplied by Manufacturer	Measured
Alex-Tagg	SHH	1393							46.00	85	
	SHV	1393							46.00	84	
Donaldson	SHH	MTM12-0043	4						44.34		7
		MOM12-1000							68.00	76	
Mack	SHV	2ME361-P3	4								7
	SHV	2ME361-P5	5						34.73	85	7
	SVV	2ME335A	4								7
	SVV	2ME336B	4								7
Stemco	SHH	9400	4								7
	SHV	9926	4						48.78		7
	SVV	9300	4						36.00	71	7
Teck	SHH	D38	4						28.62		7
	SHV	419DT	4						33.94		7
Walker	SVV	22409								75	

IV-C INTAKE SYSTEM DATA

Engine: CUMMINS - MODEL NO. V-555

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	FWA14-0033		1	Dry				131.00
	EBA13-0018		1,3	Dry	25.0	13.0	35.0	113.00

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: CUMMINS - MODEL NO. V-903

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	FHG14-0121	6.0	1	Dry	21.5	14.0	41.0	183.00
	EBA13-0018		1,3	Dry	25.0	13.0	35.0	113.00
	EBB22-0022		2	Dry				163.00

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: CUMMINS - MODEL NO. NH250

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications					Intake Noise Level
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	
Vortox	AB120 A4	5-1/2	1,3	Dry	19-3/8	12	23.0	74.21	70
	G135 AC2	5-1/2	2,3	Oil	19-9/16	13-9/16	38.0	77.78	71
			1	Oil					67
Farr	T-519		2	Dry	10	24 x 27	62		64
	T-519		1	Dry	10	24 x 27	62		59

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: CUMMINS-MODEL NO. NTC-335

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	EBA15-0005		1,3	Dry				45.00
	FHG16-0116	6.0	1	Dry	23.0	16.0	50.0	134.00
	EBB16-0007		4	Dry				122.00
Vortox	AB160 B4	6.0	1,3	Dry	24-5/8	16-1/16	40.0	123.00
	G160 BE2		1,3	Oil				102.00
	AE160 B4		2	Dry				123.00

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: CUMMINS - MODEL NO. NTC-350

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications					Intake Noise Level
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	
Donaldson	FHG16-0116	6.0	1	Dry	23.0	16.0	50.0	134.00	61
Fram	242338								63

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: DETROIT DIESEL - MODEL NO. 6-71N

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	FHG16-0116	6.0		Dry	23.0	16.0	50.0	133.66
Farr	L-49284	5.0		Dry	11.0	24 x 27	68.0	

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: DETROIT DIESEL - MODEL NO. 6V-53N

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	EBA13-0018		1,3	Dry				112.00
	FHG14-0121	6.0	1	Dry	21.5	14.0	41.0	183.00
	EBB22-0004		2	Dry				157.00

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: DETROIT DIESEL - MODEL NO. 8V-71N

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications				
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)
Donaldson	EBA15-0005		1,3	Dry				145.00
	FHG16-0116	6.0	1	Dry	23.0	16.0	50.0	134.00
	EBB22-0003		2	Dry				164.00
	EBB16-0007		4	Dry				122.00
	EBA15-0002	6.0		Dry	25.0	15.0	40.0	144.65
Vortox	AB160 A4	6.0	1,3	Dry	24-5/8	16-1/16	40.0	123.00
	G160 AC2		1,3	Oil				103.00
	AE160 A4		2	Dry				123.00
Farr	L-467845			Dry	10-1/2	24 x 27	62.0	

(*) Numbers refer to Fig. 5.4, page 79

IV-C INTAKE SYSTEM DATA

Engine: MACK - MODEL NO. ENDT-675

Air Cleaner Manufacturer	Air Cleaner Model No.	Intake Pipe Dia. (in.)	Intake System Config. (*)	Air Cleaner Specifications					Intake Noise Level
				Type	Length (in.)	Diameter (in.)	Weight (lb.)	Retail Cost ($)	
Vortox	AB120 A	5.5		Dry	19.0	12.0	23.0	74.21	73
Mack	2MD455 AP2	5.0			19.0	14.0	20.0		76

(*) Numbers refer to Fig. 5.4, page 79

APPENDIX V. COOLING SYSTEM DESIGN CONSIDERATIONS

A5.1 Introduction

A truck cooling system is required to keep the temperatures of several critical components at specified levels. The most obvious component that needs cooling is the engine. Other components that contribute to the heat load of the cooling system are the transmission, torque converter, lube oil, hydraulic oil cooler and the air conditioner. The effects of engine superchargers and retarders should also not be overlooked.

Hydraulic oil temperatures are generally limited to 180°F. Jacket water is limited to 200°F for diesel engines and 210°F for gas engines by most engine manufactures' design and warranty criteria. Therefore, if the jacket water is to be used for cooling the hydraulic oil, the water temperature has to be limited to below 180°F. This problem can be easily overcome, however, by using a separate hydraulic oil-to-air cooler upstream or at the side of the jacket water cooler. Transmission and torque converter oils are generally permitted to rise to 250°F. Because of the efficiency of water as a coolant, cost and convenience would suggest the use of an oil-to-jacket water heat exchanger for transmission cooling. The entire heat load is then transferred to the air through the jacket water radiator.

The cooling fan is usually selected to move sufficient air through the radiator to achieve the cooling capacity required to comply with the engine manufacturer's installation specifications and warranty provisions. The air flow rate per engine horsepower will depend to a large extent on the radiator area and its heat transfer characteristics.

A5.2 Basic Heat Transfer Equation

The fundamental heat transfer equation for a radiator is:

$$Q = h\, A_w \,(LMTD) \qquad \text{(A5-1)}$$

where

Q = heat load, (btu/hr)

h = heat transfer coefficient of cooling surface, (btu/hr) (ft^2)(°F)

A_w = Surface area wetted by air flow, (ft^2)

LMTD = Logarithmic mean temperature difference (°F)

Copper, aluminum and steel are the most commonly used cooling surfaces. Since the overall heat transfer coefficients of these materials cannot be varied to any significant degree, we are limited to controlling our cooling capacity through variations of area and/or temperature difference. For heat exchangers with air and coolant flowing in parallel directions, LMTD is defined as

$$LMTD = \frac{(T_{a,out} - T_{a,in}) + (T_{c,in} - T_{c,out})}{\log_e \left(\frac{T_{c,in} - T_{a,in}}{T_{c,out} - T_{a,out}} \right)} \qquad \text{(A5-2)}$$

where

$T_{a,out}$ = temperature of air leaving the radiator

$T_{a,in}$ = temperature of air entering the radiator

$T_{c,out}$ = temperature of coolant leaving the radiator

$T_{c,in}$ = temperature of coolant entering the radiator

Vehicle radiators are cross-flow heat exchangers, and as such, correction factors are needed in expressing the heat transfer in terms of the LMTD. However, in typically designed cooling systems the coolant temperature drop through the radiator is sufficiently small that these correction terms can be neglected.

At the high flow rates commonly employed for truck radiators the LMTD is independent of air flow and is approximately equal to

$$LMTD \simeq T_{c,in} - T_{a,in} \qquad \text{(A5-3)}$$

The dependence of the heat transfer coefficient on air flow varies with radiator design. As a general rule one can assume that for truck radiators

$$h \propto \sqrt{V_{air}} \qquad \text{(A5-4)}$$

where V_{air} is the velocity of the air flowing through the radiator. Thus, the heat transfer is approximately given by the equation:

$$Q \simeq \alpha \, A_r \sqrt{V_{air}} \; (T_{c,in} - T_{a,in}) \qquad \text{(A5-5)}$$

where α is a constant which depends on radiator design and A_r is the frontal area of the radiator.

Heat transfer characteristics of a typical truck radiator are shown in Fig. A5.1. The data was obtained with a radiator having the following core parameters:

Tubes: 4 rows, 1/2 inch dia at 5/8" spacing across face

Fins: .003 inch rolled copper, spacings as shown in figure A 5.1.

The heat transfer rate is measured at LMTD of 100°F which is the average difference between water in and air in temperatures. In this application LMTD is also referred to as the "potential" across

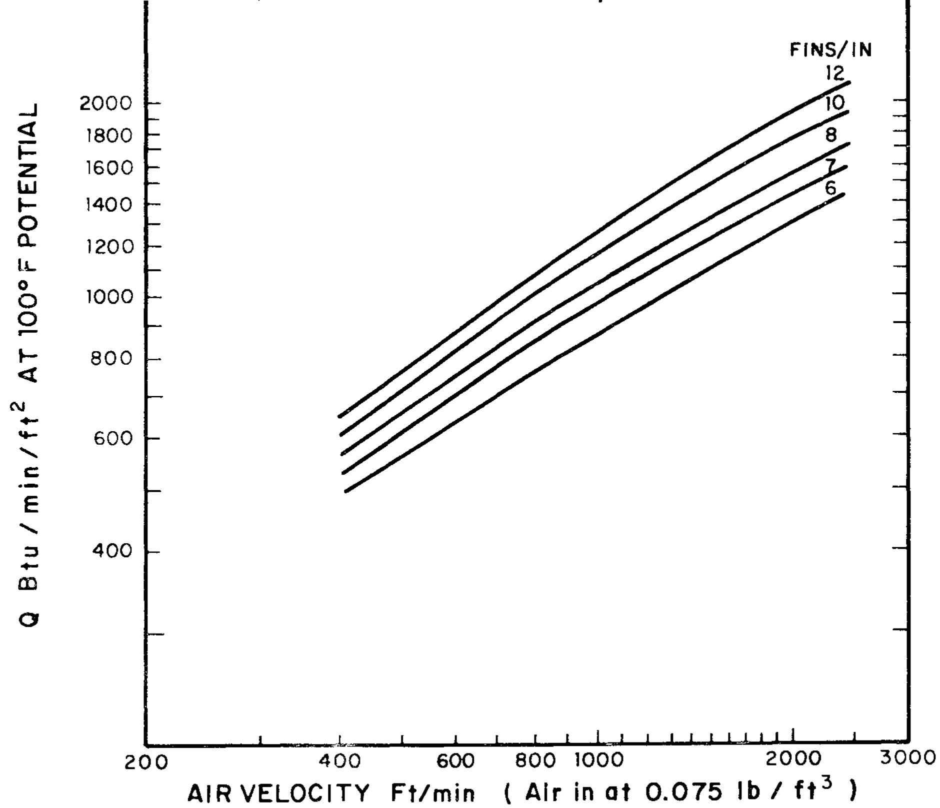


FIG. A5.1 HEAT TRANSFER PERFORMANCE OF A TRUCK RADIATOR
CORE : 4 ROWS OF 1/2" TUBES AT 5/8" SPACING
FINS : .003 IN. COPPER, SPACED AS SHOWN

the radiator.

A5.3 Air Flow Characteristics

The relationship between pressure drop and air flow rate through the radiator and through the fan are quite different in nature. This is because the radiator produces a "loss" of flow energy while the fan introduces energy into the flow. It is very important to match the fan to the radiator for optimum operation as a system.

The cooling system fan is selected to give the required air flow at a pressure set by the pressure drop through the radiator and through the engine compartment. The performance of a given fan at a set rpm is specified by a fan curve in which the static and total pressure across the fan is plotted as a function of flow as shown in Fig. A5.2. Also plotted is the percent efficiency of the fan which is given by the formula:

$$\eta = \frac{q \quad \Delta P}{550\,(HP)} \times 100 \qquad \text{(A5-6)}$$

where η = the percent efficiency, q = volume flow rate of air in cubic feet per second, HP = horsepower absorbed by the fan, and ΔP = pressure difference across the fan in lb/ft^2. A static efficiency is based on the static pressure across the fan while a total efficiency is based on total pressure.

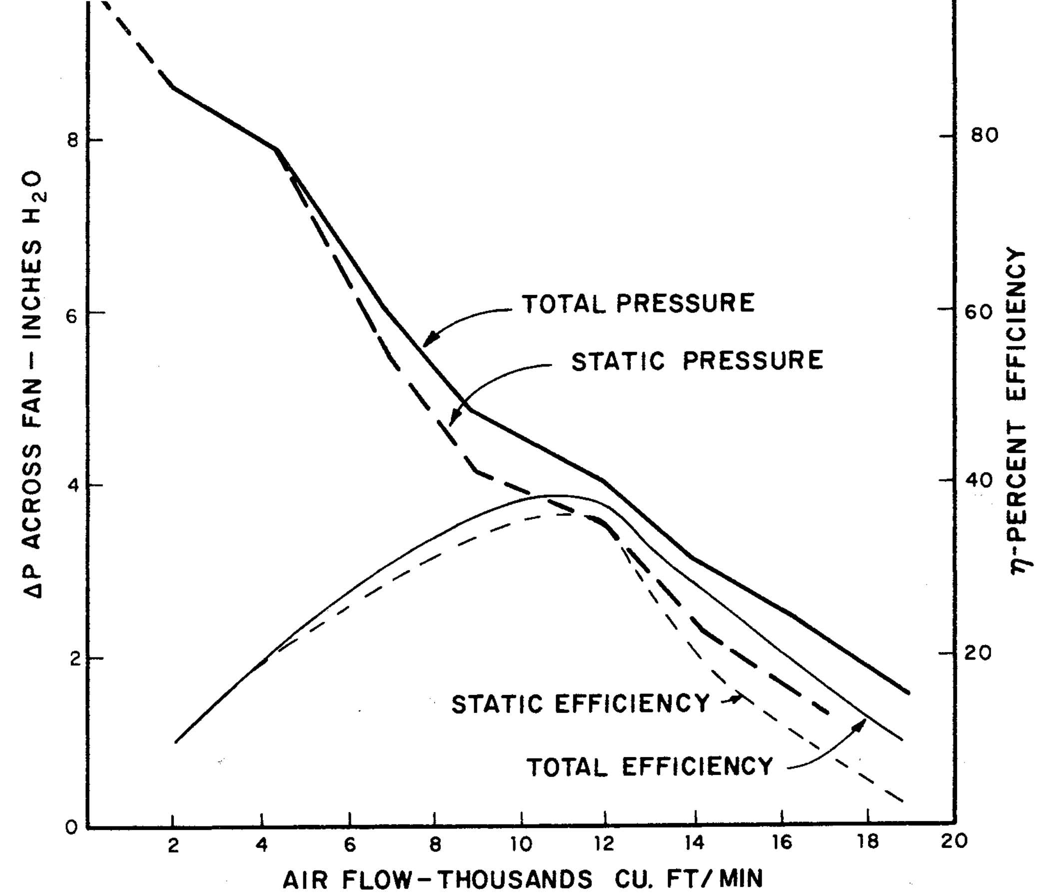


FIG. A5.2.A TYPICAL FAN CURVE

In truck applications the dynamic or velocity pressure delivered by the fan is mostly lost in the engine compartment. Therefore, the static fan curve is used for design.

Data obtained with a conventional truck cooling system show that the air flow increases approximately in proportion to fan rpm as shown in Fig. A5.3. Fig. A5.4 shows the fan horsepower and airflow as a function of rpm and blade pitch. It should be recognized that these performance curves were obtained with a particular fan and radiator. Different designs will show somewhat different performance. However, the curves shown can be relied on to establish general trends.

The static pressure drop through the radiator as a function of flow speed is shown in Fig. A5.5. As a general rule, the pressure drop is proportional to the air speed squared. The intersection of this curve with the corresponding fan curve determines the operating point of the fan as shown in Fig. A5.6. Higher flow is achieved either by decreasing the pressure drop through the system or by changing the fan rpm and its design so as to move the fan curve up and to the right.

In Section A5.2, we showed that for flow rates normally encountered in truck cooling systems, the heat transfer rate, Q, is directly proportional to the square root of the air flow velocity.

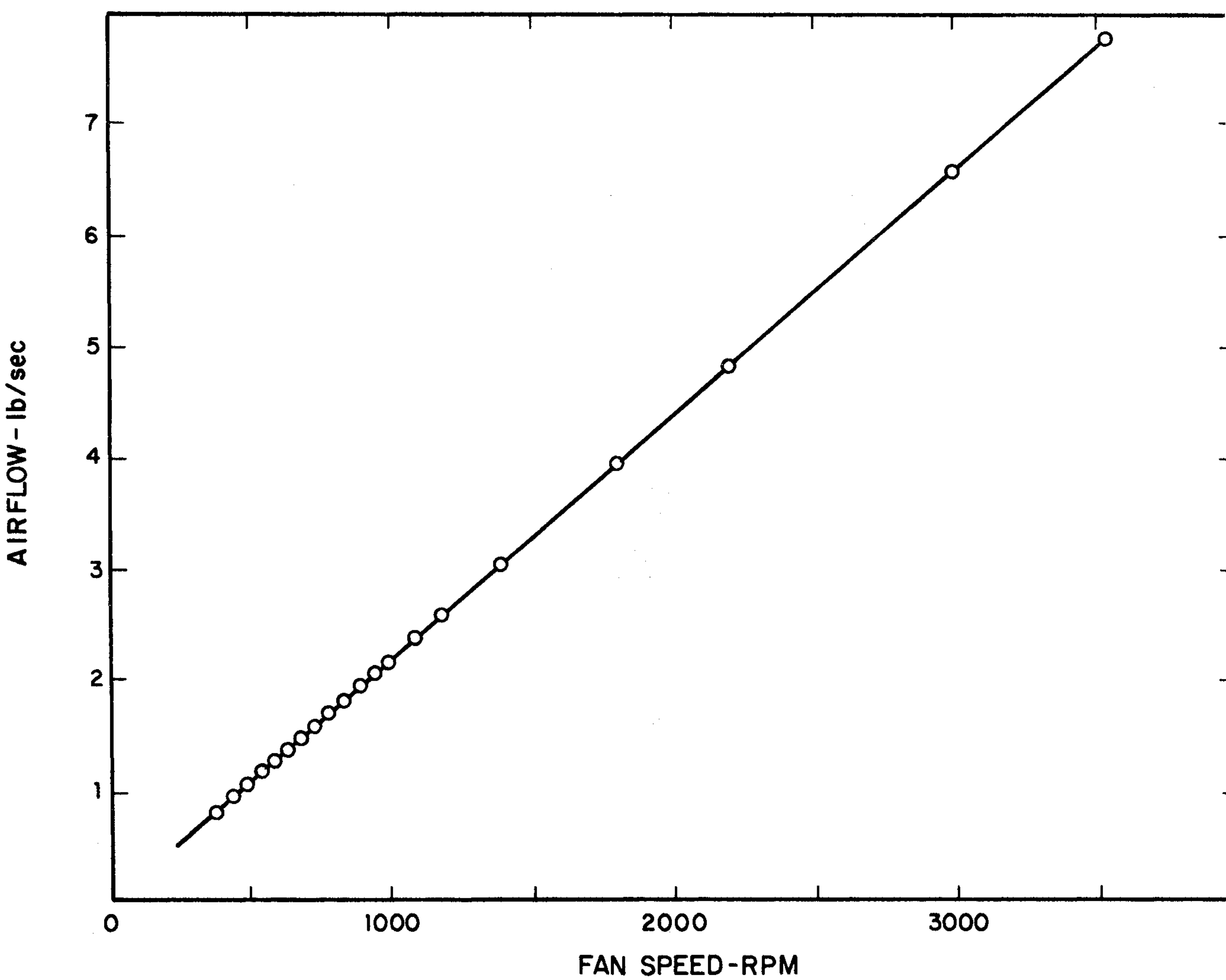
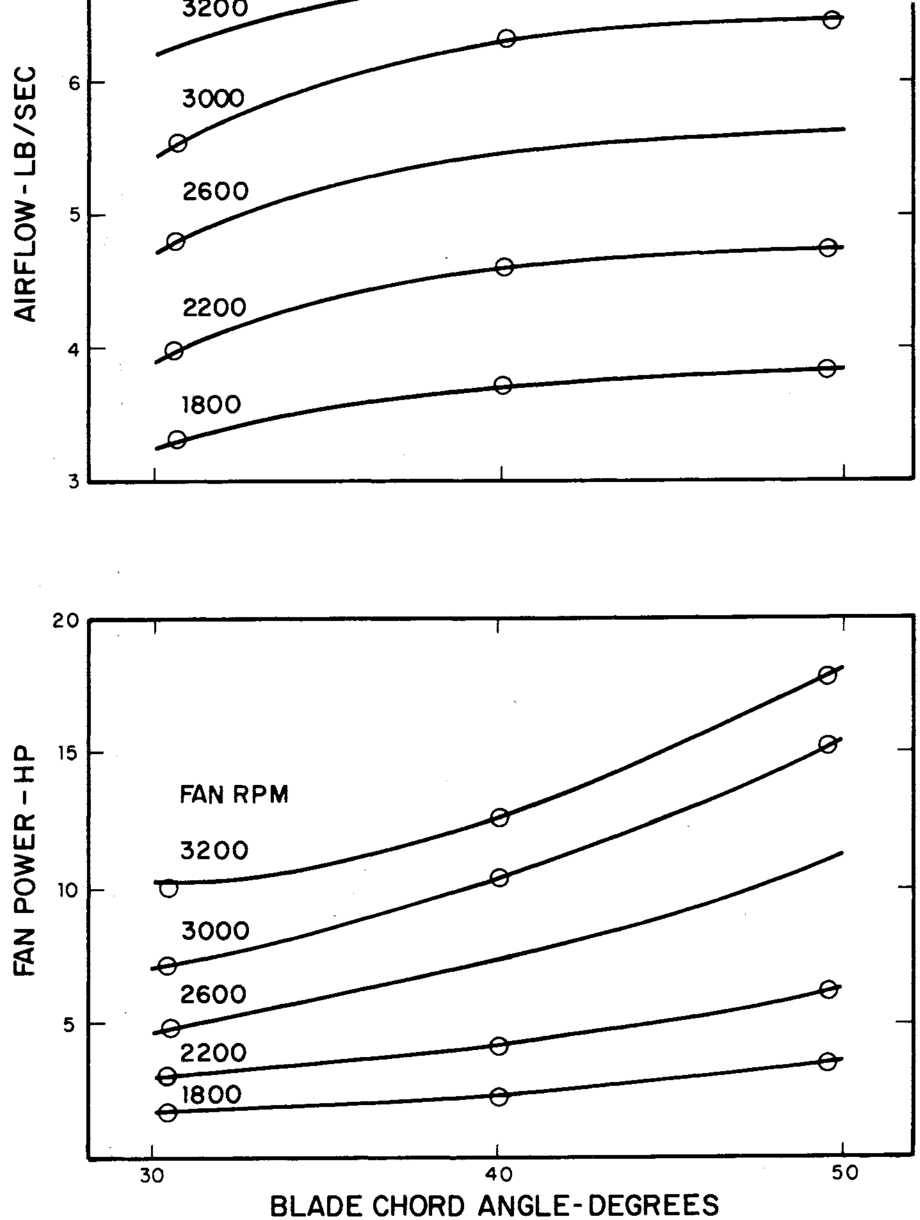


FIG. A5.3 AIR FLOW VS FAN RPM
FOR A GIVEN FAN AND SHROUD COMBINATION



NOTE : 20" DIAMETER FAN, 6 BLADES, CYLINDRICAL TYPE SHROUD
NOTE : DATA PROVIDED BY INTERNATIONAL HARVESTER CO.

FIG. A5.4 AIR FLOW AND POWER VS FAN RPM AND BLADE PITCH

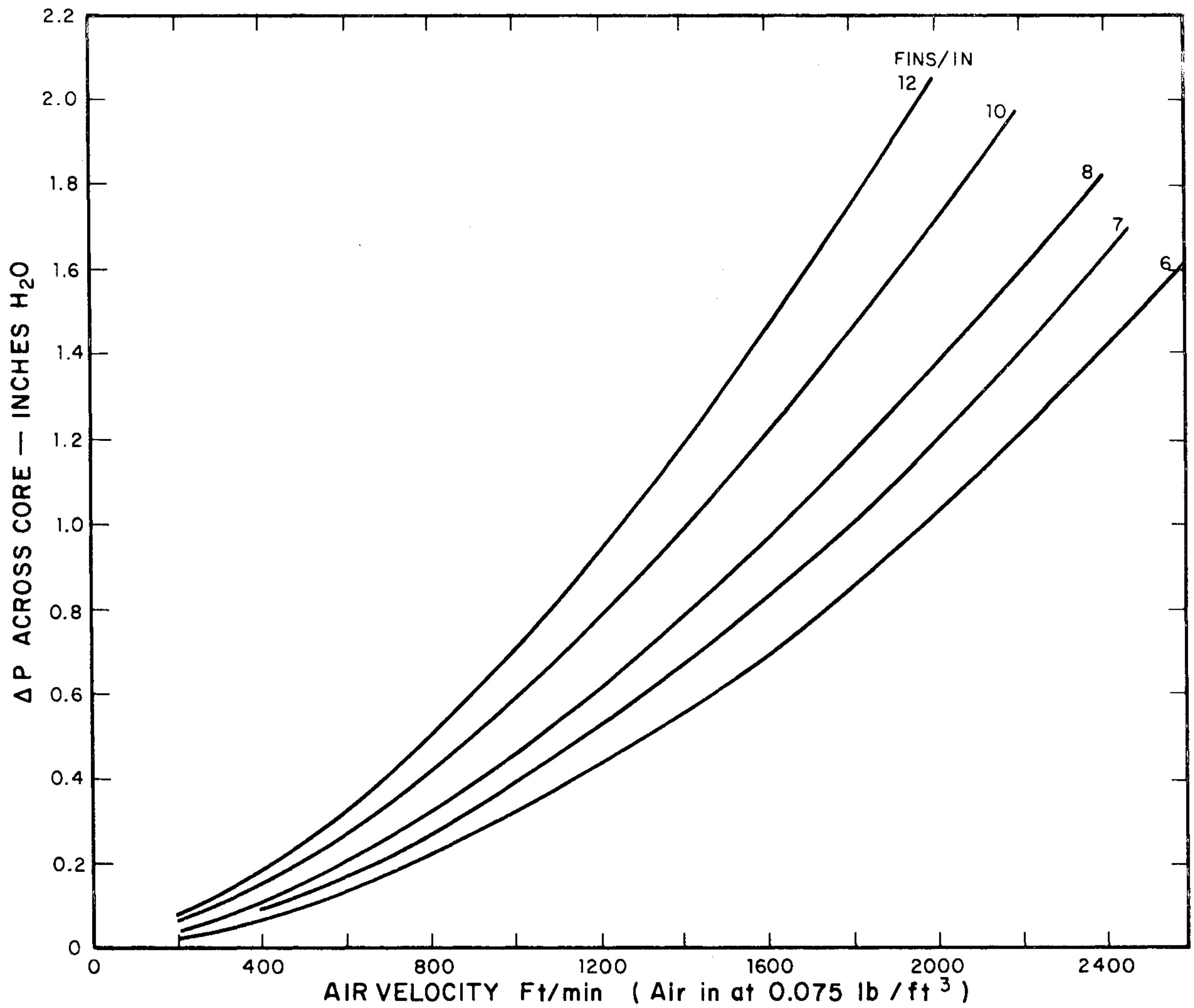


FIG. A5.5 STATIC PRESSURE DROP THROUGH A RADIATOR CORE : 4 ROWS OF 1/2" TUBES AT 5/8" SPACING FINS:.003 IN COPPER, SPACED AS SHOWN

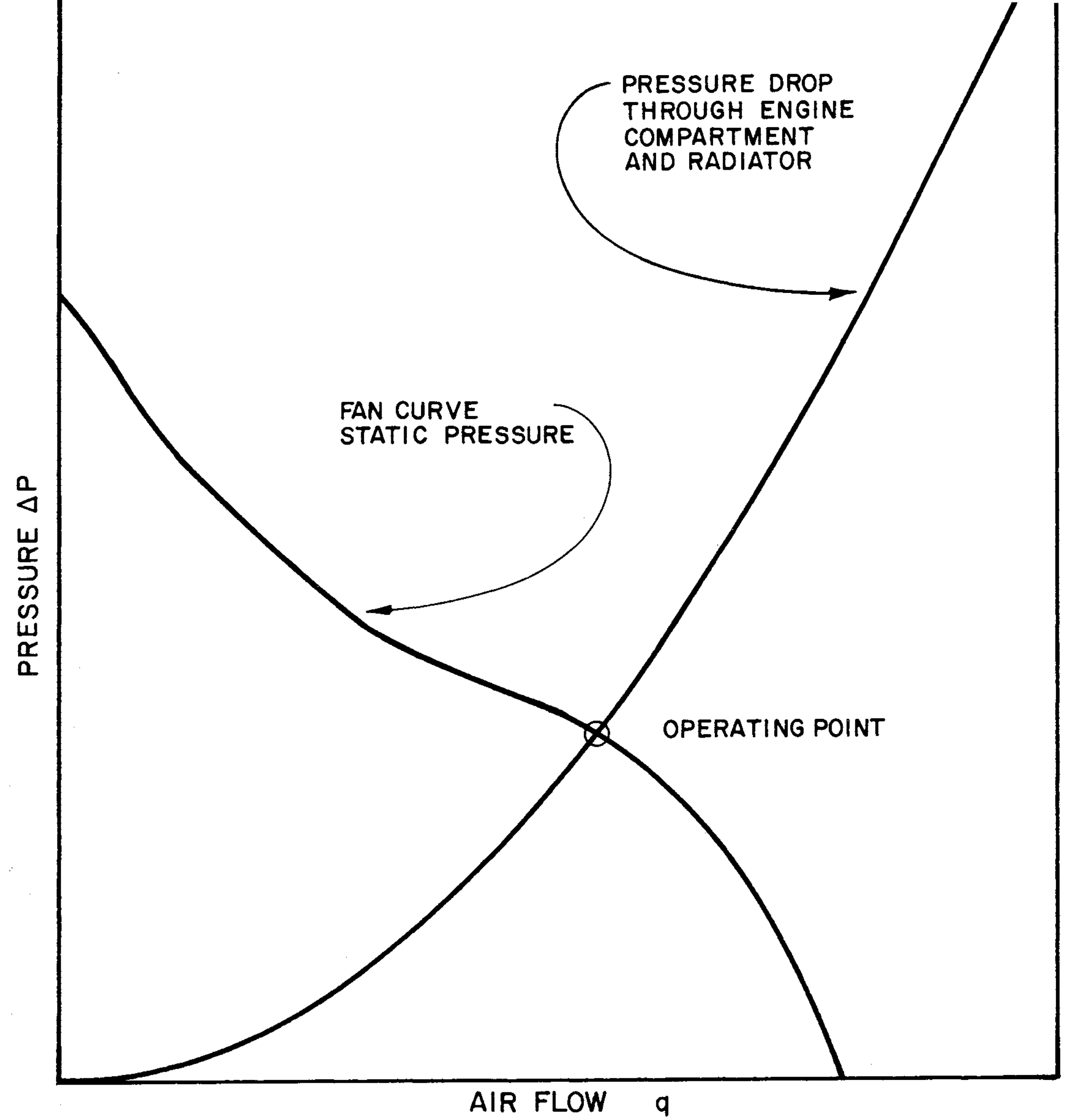


FIG. A5.6 DETERMINATION OF AIR FLOW

Since the air flow varies in proportion to the fan rpm N_f, we get the relation:

$$Q \propto \sqrt{N_f} \; . \qquad \text{(A5-7)}$$

Heat transfer is directly governed by fan rpm if all other parameters are kept constant.

A5.4 Noise Level

When we talk about cooling system noise, we are primarily concerned with the noise of the fan. Other components of the cooling system: water pump, belts and pullies, air flow through the radiator and water flow within the system, contribute very little to the overall truck noise unless steps have been taken to significantly reduce the levels of the truck's primary noise sources.

Fan noise consists of both pure tones and broad band noise, a typical spectrum is shown in Fig. 6.2 in Sec. 6 of this handbook. The pure tones are at the blade passage frequency, f_b, and its harmonics when the blades are equally spaced:

$$f = \frac{N_f B}{60} \text{ Hz} \qquad \text{(A5-8)}$$

where

N_f = fan rpm

B = number of blades

The pure tone components of fan noise are commonly referred to as rotational noise. The annoyance caused by rotational noise can be reduced somewhat by using randomly spaced blades. This has the effect of producing pure tones of greatly reduced intensity at side-band frequencies around f_b, and a predominant pure tone at a frequency $f_{b1} = \frac{N_f}{60}$. For most truck applications, this frequency is very low and hence nearly inaudible to the human ear. However, cases have been reported when the presence of this pure tone at, say, 40 Hz, produces a vague sense of discomfort to the driver of the vehicle.

The broadband component of fan noise is commonly referred to as vortex noise. This noise results from the random forces on the fan blades due to the incident air flow.

Studies show that both rotational and vortex noise levels are strongly dependent on the blade tip speed, v_t . Ventilation system fan laws predict that the noise level in dB(A) increases as 50 $\text{Log}_{10} v_t$. Aircraft fan and helicopter rotor noise studies, on the other hand, show a dependence close to 60 $\text{Log}_{10} v_t$. Truck

The ambient air temperature at which the coolant will reach boiling point during sustained low speed operation at full load is known as Air-To-Boil temperature (ATB). ATB is generally specified by the engine manufacturers and may be as high as 130°F for diesel truck engines.

A5.5.1. Fan Design

Use of more fan blades and greater projected width allows the fan to deliver the same airflow at a reduced fan rpm. As discussed in Sec. A5.3 the air flow is determined from the fan curves and the pressure drop characteristics of the radiator and engine compartment. The effect of increasing the number of fan blades is to increase the pressure developed by the fan at all flows. The pressure developed by the fan is approximately proportional to the number of blades. The approximation is very good when the number of blades is small. We expect it to be valid for up to 10 blades on a truck fan

$$\frac{\Delta P'}{\Delta P} = \frac{B'}{B} \qquad \text{(A5-10)}$$

where B = original number of blades;

B'= new number of blades.

fans operate in a significantly more turbulent flow than ventilation and aircraft fans or helicopter rotors. However, studies of truck fan rotational noise have shown a dependence closely approximating $50 \text{ Log}_{10} v_t$, while fan vortex noise level obeys a relationship given by $10 \text{ Log}_{10} BA_b$ where A_b is the area per blade. Hence, for truck fans, we can use the following equations to predict noise levels

$$\text{Fan Noise Level, dB(A)} = 50 \text{ Log } v_t + 10 \text{ Log}_{10} BA_b + x \tag{A5-9}$$

or

$$\text{Fan Noise Level, dB(A)} = 50 \text{ Log } N_f + 50 \text{ Log } D + 10 \text{ Log } B A_b + y$$

where D is the fan diameter and x, y are constants to be determined from experiments.

A5.5 Design Considerations

The cooling fan on conventional diesel powered vehicles is selected to move sufficient air through the radiator to meet the engine manufacturers' installation specifications to meet their warranty provisions. The required air flow is set by sustained low speed operation of the truck at peak torque or full load on very hot days. The ram effect at high vehicle speeds reduces the required performance from the fan but is of no help at low speeds.

The effect of reducing the fan rpm is to move the fan curve in Fig. A5.6 down and to the left. A new fan curve is constructed by moving each point to the left such that

$$\frac{q'}{q} = \frac{N_f'}{N_f} \qquad \text{(A5-11)}$$

and down such that

$$\frac{\Delta P'}{\Delta P} = \left[\frac{N_f'}{N_f} \right]^2 \qquad \text{(A5-12)}$$

The effect of increasing the projected width or blade pitch of the fan is to move the fan curve to the right such that

$$\beta' - \tan^{-1} \frac{4q'}{\pi^2 D^3 N_f} = \beta - \tan^{-1} \frac{4q}{\pi^2 D^3 N_f} \qquad \text{(A5-13)}$$

where

β = original blade pitch

β' = new pitch

D = fan diameter (kept constant)

N_f = fan RPM (kept constant)

q = air flow with blade pitch β, and

q' = air flow with blade pitch β'.

The net effect of these changes can be to arrive back at the same fan curve with reduced rpm. The procedure is to set the increase in pressure due to increasing the number of blades, $[\Delta P]_B$, equal to the decrease in pressure due to reducing fan rpm $[\Delta P]_{N_f}$ and to set the decrease in flow due to the decrease in rpm, $[\Delta q]_{N_f}$, equal to the increase in flow due to the increase in blade pitch. With these conditions

$$\left[\frac{N_f'}{N_f}\right]^2 = \frac{B}{B'} \qquad \text{(A5-14)}$$

and

$$\beta' - \beta = \left(1 - \frac{N_f'}{N_f}\right) \tan^{-1} \frac{4q}{\pi^2 D^3 N_f'} \qquad \text{(A5-15)}$$

In Sec. A5.4 the fan noise level is shown to be

$$\text{Fan Noise Level, dB(A)} = 50 \text{ Log}_{10} N_f + 50 \text{ Log}_{10} D + 10 \text{ Log}_{10} BA_b + Y \qquad \text{(A5-9)}$$

From this relation we see that the change in noise level due to a change in number of blades and the appropriate increase in fan projected width is given by

$$[\Delta dB(A)]_{new\ fan} = 50 \ Log_{10} \frac{N_f'}{N_f} + 10 \ Log \frac{B'}{B}$$

$$= 25 \ Log_{10} \left(\frac{B}{B'}\right) - 10 \ Log \frac{B}{B'}$$

$$= 15 \ Log_{10} \frac{B}{B'} \qquad \text{(A5-16)}$$

Increasing the number of blades from 6 to 8 without changing pitch or fan diameter generally produces a 2 dB decrease in noise. Increasing the number of blades from 6 to 10 generally produces a 4 dB decrease in noise. International Harvestor company has evaludated 6 and 8 bladed fans both with a 2.4" projected width. The 8 bladed fan was found to deliver the same air flow with 100 RPM less, resulting in a 2 dB decrease in noise level.

A small reduction in fan noise can also be gained by increasing the blade pitch or projected width of the fan without increasing the number of blades. Fig. A5.4 shows that increasing the pitch from 30° to 50° allows the fan speed to be reduced approximately 400 rpm without reducing airflow. This would result in noise reduction in the range of 2 to 4 dB(A). Thus, in some cases it may be possible to reduce the noise by up to 8 dB(A) by increasing the number of blades and the blade pitch. For cases in which the fan curve is reasonably flat at the desired operating point, little benefit is gained by increasing the projected width of the fan.

A number of different flexible fan designs have been proposed for use in trucks. These fans are designed so that the projected width of the fan decreases as the fan rpm increases. Thus, at high rpm the fan blades flatten out and reduce the horsepower required by the fan. The air flow delivered by the fan is also reduced so that in truck applications these flexible fans should be selected to deliver the required airflow in the high-speed flattened condition. The flexible fans are only slightly quieter than conventional fans because projected width does not have a large effect on fan noise.

Many truck fan manufacturers have current development activities in which they are looking for more efficient and quieter fans. A common approach in their search is to use fans with an improved aerodynamic design. Fiberglass fans with airfoil-shaped cross-sections and bullet hubs are being proposed for truck use. Also proposed are fans with many blades (in one case, 16), fans with blade twist and taper, and fans with non-uniform blade spacing.

The potential improvement in fan performance gained by improved aerodynamic design is often nullified by the environment in which the fan must operate. Typically, the clearance between the fan and nearby hoses, pumps, engine accessories, and the engine block is minimal. Also, the engine compartment is not designed to allow a smooth flow of air out of the compartment. As a result, a large amount of air recirculation and turbulence occurs so that the efficiency of the truck fan is approximately one-half that of an axial flow ventilating fan in a duct.

A5.5.2 Radiator Design

From Section A5.2, the heat transfer relation for a truck type cooling system is given by the equation:

$$Q \simeq \alpha A_r \sqrt{V_{air}} \quad (T_{c,in} - T_{a,in}) \qquad \text{(A5-5)}$$

For radiators of similar design (same tube and fin spacing, core depth etc.) and equal temperature potential $T_{c,in} - T_{a,in}$:

$$Q \propto A_r \sqrt{V_{air}} \qquad \text{(A5-17)}$$

If geometrically similar fans are used, air flow velocity V_{air} will be proportional to the fan tip speed V_t. Hence, we can write the fan noise level in terms of the heat transfer and size of the radiator as follows:

$$\text{Fan noise level, dB} = 50 \text{ Log } v_t + 10 \text{ Log } B\ A_b + x \qquad \text{(A5-9)}$$

$$= 100 \text{ Log } Q - 100 \text{ Log } A_r + 10 \text{ Log } A_r + \text{constant}$$

$$= 100 \text{ Log } Q - 90 \text{ Log } A_r + \text{constant} \qquad \text{(A5-18)}$$

directly with the radiator area A_r. Thus, for a given heat transfer requirement Q, the effect of increasing radiator area is to allow significant reductions in fan rpm and noise.

<u>EXAMPLE:</u> Let us calculate the effect of a 10% increase in radiator frontal area.

We will use subscript 1 for original sizes and noise levels and subscript 2 for new sizes and noise levels.

$$A_{r_2} = 1.1 \ A_{r_1} .$$

Fan diameters will be related as square roots of the areas

$$D_2 = \sqrt{1.1} \ D_1 = 1.05 \ D_1 .$$

Fan tip speed, for constant Q, will be proportional to $1/A_r^2$. Therefore,

$$V_{t_2} = \left(\frac{1}{1.1}\right)^2 V_{t_1} = 0.827 \ V_{t_1} .$$

Fan rpm's will be related as

$$N_{f_2} = \frac{0.827}{1.05} N_{f_1} = 0.781 \ N_{f_1} .$$

$$\text{Change in noise level} = 100 \text{ Log}_{10} \ 0 - 90 \text{ Log}_{10} \ (1.1)$$
$$= - 3.75 \text{ dB(A)}$$

Hence, the 10% increase in radiator size will result in making a reduction of fan rpm by about 22% and thereby reduce the fan noise by about 3.5dB(A). Greater increases in radiator area will result in proportionate noise reduction.

Changes in radiator cross section design also affect the heat transfer and may be used advantageously to reduce fan noise. Recall from Sec. A5.2 that for a fixed LMTD the heat transfer from the radiator Q is controlled by the heat transfer coefficient h and the surface area of the radiator "wetted" by the airflow, A_w:

$$Q = h\,A_w\,(LMTD) \qquad (A5\text{-}1)$$

Increases in the number of radiator fins per inch (fig. A5-1) or increases in radiator core thickness increase the overall effectiveness by increasing the wetted area A_w. Copper radiator fins are rolled to .003 in. and brass tubes as low as .004 to increase fin densities. Further reductions in thickness would seriously reduce the durability of these components. Increasing the wetted area without increasing the radiator frontal area does have a cost penalty and could also result in increased pressure loss across the radiator. The use of corrugated or louvred fins increases the heat transfer coefficient, h, for a given air flow velocity and thereby allows the same overall heat transfer with lower flow velocities. Some results for automobile radiators are shown in Figs. 6.7 and 6.8 in Sec. 6 of this handbook.

Heating, ventilation and air conditioning systems engineers use a "noise factor" to predict the effect of design changes on fan noise. The noise factor is defined as

$$\text{Noise factor, dB} = q(\Delta P)^2$$

where

$q =$ volume flow of air

$\Delta P =$ static pressure across fan.

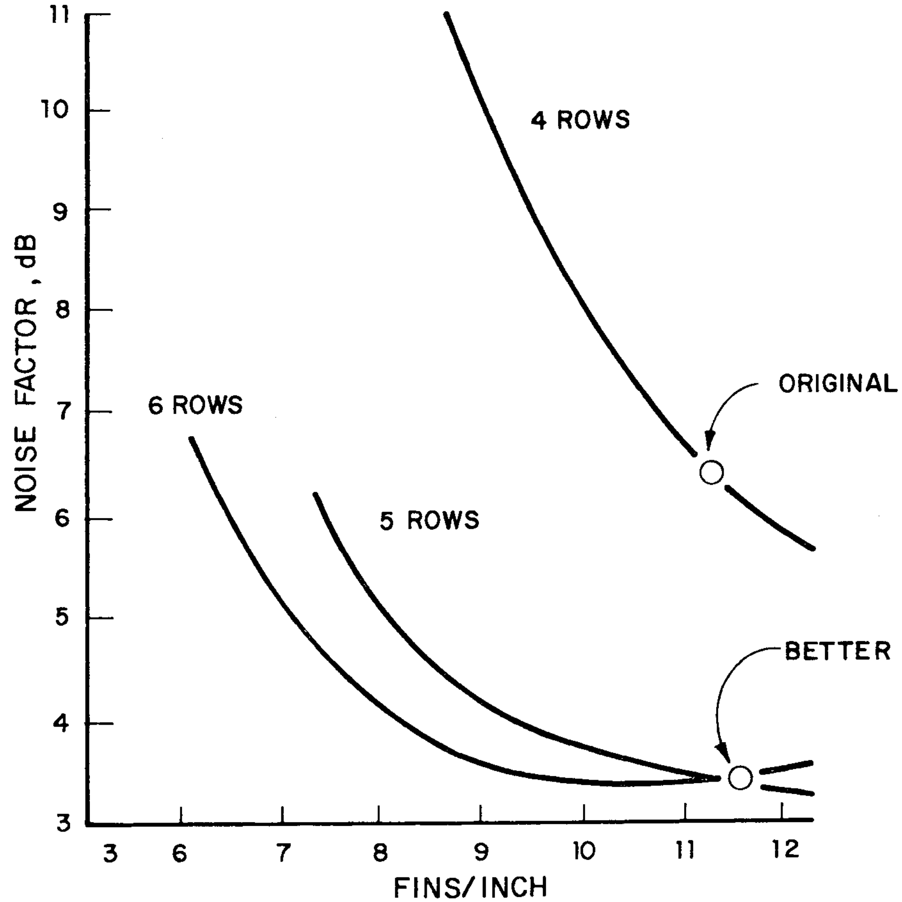
Fan noise is assumed to vary as $10 \log_{10} \Delta P^2$ and it is assumed that the fan is designed to operate at peak efficiency. The noise factor for three radiator core designs is shown in Fig. A5.7. The design with 5-tube rows and 11 fins per inch has a noise factor that is 4 dB less than that of the 4-row design because of the reduced volume flow of air that is required to achieve the same heat rejection.

A.5.5.3 Engine Compartment Design

The engine compartment must provide adequate area for the air flow to exit. However, a tradeoff exists. Large exit area minimizes the back pressure for the fan and results in high air flow. A small exit area contains the noise within the engine compartment and reduces the roadside level. Data indicates that the exit area for the air to flow out of the engine compartment should be in the range of 400 to 600 sq. in. for mass flows up to 7.5 lb/sec.

A.5.6 Summary

Performance and noise are interrelated and an improvement in performance allows noise and fan horsepower reduction from reduced fan rpm. The procedures given here are intended to permit preliminary



NOTE: DATA PROVIDED BY INTERNATIONAL HARVESTER CO.

FIG. A5.7 NOISE FACTOR FOR VARIOUS RADIATOR ARRANGEMENTS computation to determine the feasibility of optimizing existing cooling packages. Fan and radiator manufacturers should always be consulted before making radical changes in a truck's cooling system.

Other methods for reducing cooling system noise using available retrofit components are given in Sec. 6 of this handbook.

APPENDIX VI - SUPPLIERS OF ACOUSTIC MATERIAL AND RETROFIT COMPONENTS

A. Buyer's Guide to Materials for Noise/Vibration/Shock Control (from Sound and Vibration, July 1973)

B. Additional Suppliers of Acoustic Materials (Those not listed in A.)

C. Muffler and Air Cleaner Manufacturers

D. Fans, Fan Clutches, Shrouds

ing each of the four major categories for the various classifications by material type.

- ☐ Sound absorption materials are porous or fibrous materials which absorb sound energy—they do not block the transmission of sound.
- ☐ Sound barrier materials are dense, impervious materials which block the transmission of sound energy—they generally reflect rather than absorb sound.
- ☐ Vibration and shock isolation materials are resilient materials that absorb vibrational energy or restrain the vibratory motion of mechanical structures.

For easy reference to the company or product wanted, find the appropriate category in the table below. Then locate the manufacturer by referring to the company listings and product sub-classifications. Companies which appear in bold-face type are advertisers in this issue.

I. Sound Absorption Materials
1. Formboard
2. Glass Fiber & Foam
3. Metal Felt & Porous Metals
4. Mineral Wool
5. Perforated Ceramic Tile
6. Perforated Sheet Metal
7. Plastic Foam Sheet
8. Slotted Masonry Units
9. Spray-on Absorptive Coatings
10. Wood Fiber & Fiberboard
11. Woven & Nonwoven Felt & Cloth

II. Sound Barrier Materials
1. Asphalted Felt & Fiberboard
2. Constrained Sheet Metal Composites
3. Gypsum Board
4. Insulation Board
5. Lead Sheet
6. Loaded Plastic Sheet
7. Loaded Plastic/Plastic Foam Composites
8. Loaded Rubber Sheet
9. Mastic/Cellulose Composites
10. Particle Board
11. Sealants & Sealing Tapes
12. Semi-Liquid Compounds
13. Sheet Metal/Mineral Wool Composites
14. Sheet Metal/Plastic Foam Composites

III. Vibration/Shock Isolation Materials
1. Elastomeric Pads & Foams
2. Elastomers
3. Fibrous Blankets & Boards

IV. Vibration Damping Materials
1. Adhesives
2. Elastomers
3. Mastic Sheet & Tile
4. Plastic Sheet & Tile
5. Semi-Liquid Compounds, Mastic Based
6. Semi-Liquid Compounds, Plastic Based
7. Tapes

I. Sound Absorption Materials

1. **Formboard**
2. Glass Fiber & Foam
3. **Metal Felt & Porous Metals**
4. Mineral Wool
5. Perforated Ceramic Tile
6. Perforated Sheet Metal
7. Plastic Foam Sheet
8. Slotted Masonry Units
9. Spray-on Absorptive Coatings
10. Wood Fiber & Fiberboard
11. **Woven & Nonwoven Felt & Cloth**

Accessible Products Co., 1350 E. 8th St., Tempe, AZ 85281 (7)
ACS Industries, Inc., 71 Villanova St., Woonsocket, RI 02895 (4)
Advanced Acoustical Research Corp., 2259 Sawmill River Rd., Elmsford, NY 10523 (2,4,6,7,9)
Air-O-Plastik Corp., Asia Place, Carlstadt, NJ 07072 (2,7)
Airtex Industries, Inc., Flexible Products Div., 3558 2nd St. N., Minneapolis, MN 55412 (7)
Alpro Acoustics Div., P.O. Box 30460, New Orleans, LA 70190 (6)
American Acoustical Products, 9 Cochituate St., Natick, MA 01760 (2,4,7)
Architectural Products Div., Erdle Perforating Co., Inc., P.O. Box 1568, Rochester, NY 14603 (6)
Arrow Sintered Products Co., 7650 Industrial Dr., Forest Park, IL 60130 (3)
Barry Div./Barry Wright Corp., 700 Pleasant St., Watertown, MA 02172 (3)
H. L. Blachford, Inc., 1855 Stephenson Hwy., Troy, MI 48084 (2,7)
Brunswick Corp., Technical Products Div., One Brunswick Plaza, Skokie, IL 60076 (3,6)
The Celotex Corp., P.O. Box 22622, Tampa, FL 33602 (7,10)
Chemical Coatings & Engineering Co., Inc., 221 Brook St., Media, PA 19063 (9)
Consolidated Kinetics Corp., 249 Fornof Lane, Columbus, OH 43207 (2,7)
Conwed Corp., 2200 Highcrest Rd., St. Paul, MN 55113 (10)
Diamond Perforated Metals Co., A Div. of Whittaker Corp., 17915 S. Figueroa St., Gardena, CA 90248 (6)
Donn Products, Inc., 1000 Crocker Rd., Westlake, OH 44145 (6)
Duracote Corp., 350 N. Diamond St., Ravenna, OH 44266 (7)
Eckoustic Div., Eckel Industries, 155 Fawcett St., Cambridge, MA 02138 (2)
Electro-Ionic Systems, Inc., 1085 Memorex Dr., Santa Clara, CA 95050 (7)
Environmental Services and Products, Inc., P.O. Box 1281, OH 45401 (7)
Fansteel/Reflective Laminates, 851 Lawrence Dr., Newbury Park, CA 91320 (2,4,6)
Ferguson Perforating & Wire Co., Inc., 133 Ernest St., Providence, RI 02905 (6)
Ferro Corp., Composites Div., 34 Smith St., Norwalk, CT 06852 (7)
GAF Corp., Industrial Products Div., Glenville Station, Greenwich, CT 06830 (11)
Gaska-Tape, Inc., 1801 Minnie St., Elkhart, IN 46514 (7)
General Noisecontrol Corp., 101 E. Main St., Little Falls, NJ 07424 (2,4)
Globe Industries, Inc., Acousti-Pad Div., 2638 E. 126th St., Chicago, IL 60633 (11)
The Harrington & King Perforating Co., Inc., 5655 Fillmore St., Chicago, IL 60644 (6)
Hecht Rubber Co., 484 Riverside Ave., Jacksonville, FL 32202 (7)
Industrial Acoustics Co., Inc., 380 Southern Blvd., Bronx, NY 10454 (2,4,6)
Inescon, Inc., 5235 Darrow Rd., Hudson, OH 44236 (2,7)
Insul-Coustic/Birma Corp., Jernee Mill Rd., Sayreville, NJ 08872 (2,11)
Johns-Manville, Greenwood Plaza, Denver, CO 80217 (2)
Korfund Dynamics Corp., Cantiague Rd., Westbury, NY 11590 (2,7)
MBI Products Co., 1176 E. 38th St., Cleveland, OH 44114 (2,11)
National Cellulose Corp., 12315 Robin Blvd., Houston, TX 77045 (9)
National Gypsum Co., Gold Bond Building Products Div., 325 Delaware Ave., Buffalo, NY 14202 (1,4)
National Perforating **Corp.**, Parker St., Clinton, MA 01510 (6)
Nichols Dynamics, Inc., 740 Main St., Waltham, MA 02154 (2,7,9)
Noise Measurement & Control Div., National Research Corp., 322 E. Lancaster Ave., Wayne, PA 19087 (2,6,7,10)
Norton Co., Sealant Operations, 12 Bennett Dr., Granville, NY 12832 (7)
Owens-Corning Fiberglas Corp., 1 Fiberglas Tower, Toledo, OH 43659 (1,2)
Pittsburgh Corning Corp., Three Gateway Center, Pittsburgh, PA 15222 (2)
The Proudfoot Co., Inc., P.O. Box 9, Greenwich, CT 06830 (8)
Ray Proof Corp., 50 Keeler Ave., Norwalk, CT 06856 (6)
Scason, Inc., 112 Main St., Norwalk, CT 06430 (7)
Scason Canada, Ltd., 2220 Midland Ave., 31 Administration Park, Scarborough, Ontario M1P 3E6, Canada (7)
Scott Paper Co., Foam Div., 1500 E. Second St., Chester, PA 19013 (7)
Sound/Eaze Products-John Schneller and Assoc., P.O. Box 386, Kent, OH 44240 (7)
Singer Partitions, Inc., 444 N. Lake Shore Dr., Chicago, IL 60611 (7)
The Soundcoat Co., Inc., 175 Pearl St., Brooklyn, NY 11201 (7)
Sound Solutions Corp., 601 Washington St., Lynn, MA 01901 (2,6,7,9)
Specialty Composites Corp., Delaware Industrial Park, Newark, DE 19711 (7)
Standard Felt Co., 115 S. Palm Ave., Alhambra, CA 91802 (11)
Stark Ceramics, Inc., P.O. Box 8880, Canton, OH 44711 (5)
Steel-Fab, Inc., 430 Crawford St., Fitchburg, MA 01420 (2)
Tull Environmental Systems, Div. J. M. Tull Industries, Inc., 285 Marietta St., NW, Atlanta, GA 30302 (2,6,7)

3. Gypsum Board
4. **Insulation Board**
5. Lead Sheet
6. **Loaded Plastic Sheet**
7. **Loaded Plastic/Plastic Foam Composites**
8. **Loaded Rubber Sheet**
9. **Mastic/Cellulose Composites**
10. Particle Board
11. Sealants & Sealing Tapes
12. Semi-Liquid Compounds
13. **Sheet Metal/Mineral Wool Composites**
14. **Sheet Metal/Plastic Foam Composites**

Accessible Products Co., 1350 E. 8th St., Tempe, AZ 85281 (5-7,11)
ACS Industries, Inc., 71 Villanova St., Woonsocket, RI 02895 (13)
Advanced Acoustical Research Corp., 2259 Sawmill River Rd., Elmsford, NY 10523 (2,5,6,13)
The Aeroacoustic Corp., P.O. Box 65, Amityville, NY 11701 (13,14)
Aeronca, Inc./Environmental Control Group, P.O. Box 688, Pineville, NC 28134 (13,14)
Air-O-Plastik Corp., Asia Place, Carlstadt, NJ 07072 (7,11,14)
Airtex Industries, Inc., Flexible Products Div., 3558 2nd St. N., Minneapolis, MN 55412 (5,7,8)
Alpro Acoustics Div., P.O. Box 30460, New Orleans, LA 70190 (13)
American Acoustical Products, 9 Cochituate St., Natick, MA 01760 (5-7,13,14)
American Smelting & Refining Co./Federated Metals Div., 150 St. Charles St., Newark, NJ 07105 (5)
Architectural Products Div., Erdle Perforating Co., Inc., P.O. Box 1568, Rochester, NY 14603 (13,14)
Barley-Earhart Co., 233 Divine Hwy., Portland, MI 48875 (1,9)
Bar-Ray Products, Inc., 209 25th St., Brooklyn, NY 11232 (5-8)
Barry Div./Barry Wright Corp., **700 Pleasant St.,** Watertown, MA 02172 (2,8)
H. L. Blachford, Inc., 1855 Stephenson Hwy., Troy, MI 48084 (5-7)
Canada Metal Co., Noise Control Div., 721 Eastern Ave., Toronto, Ontario M4M 1E6, Canada (5)
The Celotex Corp., P.O. Box 22622, Tampa, FL 33602 (1,3,4)
Certain-Teed Products Corp., CSG Group, P.O. Box 860, Valley Forge, PA 19482 (4)
Chemical Coatings & Engineering Co., Inc., 221 Brook St., Media, PA 190363 (6-8,11)
Chemprene, Inc., 579 South Ave., Beacon, NY 12508 (6-8)
Consolidated Kinetics Corp., 249 Fornof Lane, Columbus, OH 43207 (6,7,11)
Conwed Corp., 2200 Highcrest Rd., St. Paul, MN 55113 (4)
Cowl Div. of James B. Carter, 88 Fennel St., Winnipeg, Manitoba, Canada (12)
Dow Corning Corp., Midland, MI 48640 (11)
Duracote Corp., 350 N. Diamond St., Ravenna, OH 44266 (6,7)
Eckoustic Div., Eckel Industries, 155 Fawcett St., Cambridge, MA 02138 (7)
Electro-Ionic Systems, Inc., 1085 Memorex Dr., Santa Clara, CA 95050 (14)
Environmental Services and Products, Inc., P.O. Box 1281, Dayton, OH 45401 (6-8)
Fansteel/Reflective Laminates, 851 Lawrence Dr., Newbury Park, CA 91320 (13)
Ferro Corp., Composites Div., 34 Smith St., Norawlk, CT 06852 (6,7)
GAF Corp., Industrial Products Div., Glenville Station, Greenwich, CT 06830 (1,9)
Gaska-Tape, Inc., 1801 Minnie St., Elkhart, IN 46514 (11)
General Electric Co., Silicone Products Dept., Waterford, NY 12188 (11)
General Noisecontrol Corp., 101 E. Main St., Little Falls, NJ 07424 (5,13)
Globe Industries, Inc., Acousti-Pad Div., 2638 E. 126th St., Chicago, IL 60633 (1,9,12)

MBI Products Co., 1176 E. 38th St., Cleveland, OH 44114 (4)
National Gypsum Co., Gold Bond Building Products Div., 325 Delaware Ave., Buffalo, NY 14202 (3,4,10)
National Research Corp., Concord Rd., Billerica, MA 01821 (6)
Nichols Dynamics, Inc., 740 Main St., Waltham, MA 02154 (5-8)
Noise Control Products, Inc., 969 Lakeville Rd., New Hyde Park, NY 11040 (13)
Noise Measurement & Control Div., National Research Corp., 322 E. Lancaster Ave., Wayne, PA 19087 (1,2,4-7,13)
Norton Co., Sealant Operations, 12 Bennett Dr., Granville, NY 12832 (11)
Owens-Corning Fiberglas Corp., 1 Fiberglas Tower, Toledo, OH 43659 (4)
Quaker, State Oil Refining Corp., P.O. Box 989, Oil City, PA 16323 (9,12)
Ray Proof Corp., 50 Keeler Ave., Norwalk, CT 06856 (5,13)
Scason, Inc., 112 Main St., Norwalk, CT 06430 (2,6)
Scason Canada, Ltd., 2220 Midland Ave., 31 Administration Park, Scarborough, Ontario M1P 3E6, Canada (2,6)
Scott Paper Co., Foam Div., 1500 E. Second St., Chester, PA 19013 (7)
Sound/Eaze Products-John Schneller and Assoc., P.O. Box 386, Kent, OH 44240 (6,7)
Singer Partitions, Inc., 444 N. Lake Shore Dr., Chicago, IL 60611 (6,7,9)
The Soundcoat Co., Inc., 175 Pearl St., Brooklyn, NY 11201 (2,5-7,14)
Sound Solutions Corp., 601 Washington St., Lynn, MA 01901 (5-8,11)
Specialty Composites Corp., Delaware Industrial Park, Newark, DE 19711 (6,7)
Tull Environmental Systems, Div. J. M. Tull Industries, Inc., 285 Marietta St., NW, Atlanta, GA 30302 (2,5-7,14)
United States Gypsum Co., 101 S. Wacker Dr., Chicago, IL 60606 (3,4,11)
Veneered Metals, Inc., Woodbridge Ave. at Main St., Edison, NJ 08817 (2)
Webster Products Co., 1261 W. Wright St., Santa Ana, CA 92705 (7,9)
Wilshire Foam Products, Inc., 2665 Columbia St., Torrance, CA 90503 (14)

III. Vibration/Shock Isolation Materials

1. **Elastomeric Pads & Foams**
2. **Elastomers**
3. **Fibrous Blankets & Boards**

Aeroflex Laboratories, Inc., Isolator Products Div., 35 South Service Rd., Plainview, NY 11803 (1)
Air-Loc Products, Fisher St., Franklin, MA 02038 (1)
Amber/Booth Co., 7914 Westglen, Houston, TX 77042 (1-3)
American Acoustical Products, 9 Cochituate St., Natick, MA 01760 (1)
Barley-Earhart Co., 233 Divine Hwy., Portland, MI 48875 (1)
Louis P. Batson Co., P.O. Box 3978, Greenville, SC 29608 (1)
Barry Div./Barry Wright Corp., 700 Pleasant St., Watertown, MA 02172 (1,2)
Beltran Associates, Inc., 1133 E. 35th St., Brooklyn, NY 11210 (1)
Chemical Coatings & Engineering Co., Inc., 221 Brook St., Media, PA 19063 (2)
Consolidated Kinetics Corp., 249 Fornof Lane, Columbus, OH 43207 (1,3)
Conwed Corp., 2200 Highcrest Rd., St. Paul, MN 55113 (3)
Dow Corning Corp., Midland, MI 48640 (2)
Environmental Services and Products, Inc., P.O. Box 1281, Dayton, OH 45401 (1)
Fabreeka Products Co., 1190 Adams St., Boston, MA 02124 (1)
GAF Corp., Industrial Products Div., Glenville Station, Greenwich, CT 06830 (3)

Lord Kinematics, 1635 W. 12th St., Erie, PA 16512 (1)
Machinery Mountings, Inc., 41 Sarah Dr., Farmingdale, NY 11735 (1-3)
Martec Acoustical Products Co., Div. of Martec Associates, Inc., 1645 Oakton St., Des Plaines, IL 60018 (1)
Mason Industries, Inc., 92-10 182 Place, Hollis, NY 11423 (2)
MBI Products Co., 1176 E. 38th St., Cleveland, OH 44114 (3)
National Research Corp., Concord Rd., Billerica, MA 01821 (1)
Nichols Dynamics, Inc., 740 Main St., Waltham, MA 02154 (1)
Norton Co., Sealant Operations, 12 Bennett Dr., Granville, NY 12832 (1)
Scott Paper Co., Foam Div., 1500 E. Second St., Chester, PA 19013 (1)
Sound Solutions Corp., 601 Washington St., Lynn, MA 01901(1,2)
Specialty Composites Corp., Delaware Industrial Park, Newark, DE 19711 (1,2)
Standard Felt Co., 115 S. Palm Ave., Alhambra, CA 91802 (3)
Tull Environmental Systems, Div. J. M. Tull Industries, Inc., 285 Marietta St., NW, Atlanta, GA 30302 (1)
Unisorb Machinery Installation Systems, 22 West St., Millbury, MA 01527 (1)
VibraSonics, P.O. Box 14098, Fort Worth, TX 76117 (1,2)
Vibration Mountings & Controls Inc., Post Office Box 776SV, Butler, NJ 07405 (1,2)

IV. Vibration Damping Materials

1. Adhesives
2. Elastomers
3. **Mastic Sheet & Tile**
4. **Plastic Sheet & Tile**
5. **Semi-Liquid Compounds, Mastic Based**
6. **Semi-Liquid Compounds, Plastic Based**
7. **Tapes**

Air-O-Plastik Corp., Asia Place, Carlstadt, NJ 07072 (7)
Airtex Industries, Inc., Flexible Products Div., 3558 2nd St. N., Minneapolis, MN 55412 (4,7)
American Acoustical Products, 9 Cochituate St., Natick, MA 01760 (2,4)
Barley-Earhart Co., 233 Divine Hwy., Portland, MI 48875 (7)
Barry Div./Barry Wright Corp., 700 Pleasant St., Watertown, MA 02172 (1,2,4)
H. L. Blachford, Inc., 1855 Stephenson Hwy., Troy, MI 48084 (4,6)
Chemical Coatings & Engineering Co., Inc., 221 Brook St., Media, PA 19063 (1-6)
Consolidated Kinetics Corp., 249 Fornof Lane, Columbus, OH 43207 (1,4,6,7)
Dow Corning Corp., Midland, MI 48640 (1,6)
Duracote Corp., 350 N. Diamond St., Ravenna, OH 44266 (4)
Electro-Ionic Systems, Inc., 1085 Memorex Dr., Santa Clara, CA 95050 (6)
Environmental Services and Products, Inc., P.O. Box 1281, Dayton, OH 45401 (4)
Fabreeka Products Co., 1190 Adams St., Boston, MA 02124 (2)
Ferro Corp., Composites Div., 34 Smith St., Norwalk, CT 06852 (6)
GAF Corp., Industrial Products Div., Glenville Station, Greenwich, CT 06830 (3)
Gaska-Tape, Inc., 1801 Minnie St., Elkhart, IN 46514 (7)
General Electric Co., Silicone Products Dept., Waterford, NY 12188 (1,2)
General Rubber Corp., 9 Empire Blvd., South Hackensack, NJ 07606 (2)
Gilmore Industries, Inc., 3355 Richmond Rd., Cleveland, OH 44122 (2)
Globe Industries, Inc., Acousti-Pad Div., 2638 E. 126th St., Chicago, IL 60633 (3,5)
Hecht Rubber Co., 484 Riverside Ave., Jacksonville, FL 32202 (1,2,7)
IMS Co., **24050 Commerce Park Rd., Cleveland,** OH 44122 (4)

Insul-Coustic/Birma Corp., Jernee Mill Rd., Sayreville, NJ 08872 (1,5)

Korfund Dynamics Corp., Cantiague Rd., Westbury, NY 11590 (6)

Lambda Corp., Box 181, Whippany, NJ 07981 (6)

Lord Kinematics, 1635 W. 12th St., Erie, PA 16512 (2)

Mason Industries, Inc., 92-10 182 Place, Hollis, NY 11423 (2)

3M Co., Industrial Specialties Div., 3M Center, St. Paul, MN 55101 (7)

National Research Corp., Concord Rd., Billerica, MA 01821 (2,4)

Nichols Dynamics, Inc., 740 Main St., Waltham, MA 02154 (1,5-7)

Noise Measurement & Control Div., National Research Corp., 322 E. Lancaster Ave., Wayne, PA 19087 (3,4)

Norton Co., Sealant Operations, 12 Bennett Dr., Granville, NY 12832 (7)

Philadelphia Resins Corp., P.O. Box 454, Montgomery, PA 18936 (1,4-6)

Quaker State Oil Refining Corp., P.O. Box 989, Oil City, PA 16323 (5)

Scason, Inc., 112 Main St., Norwalk, CT 06430 (4,6)

Scason Canada, Ltd., 2220 Midland Ave., 31 Administration Park, Scarborough, Ontario M1P 3E6, Canada (4,6)

Singer Partitions, Inc., 444 N. Lake Shore Dr., Chicago, IL 60611 (1,3)

The Soundcoat Co., Inc., 175 Pearl St., Brooklyn, NY 11201 (2,4,6)

Sound Solutions Corp., 601 Washington St., Lynn, MA 01901 (3,5)

Specialty Composites Corp., Delaware Industrial Park, Newark, DE 19711 (4)

Tull Environmental Systems, Div. J. M. Tull Industries, Inc., 285 Marietta St., NW, Atlanta, GA 30301 (1,2,4)

United States Gypsum Co., 101 S. Wacker Dr., Chicago, IL 60606 (1)

VibraSonics, P.O. Box 14098, Fort Worth, TX 76117 (2)

Vibration Mountings & Controls Inc., Post Office Box 776SV, Butler, NJ 07405 (1,2,5,6)

Webster Products Co., 1261 W. Wright St., Santa Ana, CA 92705 (3)

Wilshire Foam Products, Inc., 2665 Columbia St., Torrance, CA 90503 (7)

B. ADDITIONAL ACOUSTIC MATERIALS SUPPLIERS

(Those not listed in A.)

Company	Products
SONO THERM, INC. 90 Arthur Street Buffalo, New York 14207	Low Con Fiberflax Blankets Aluminum backed for high temperature use
3 M COMPANY Adhesives, Loadings & Sealers Div. 3 M Center St. Paul, Minnesota 55101	Contact cement, adhesives

AMF BEAIRD, INC.
Maxim Products Group
P.O. Box 1115
Shreveport, Louisiana 71102

ALEXANDER-TAGG IND., INC.
395 Jacksonville Road
Warminster, Pennsylvania

BURGESS-MANNING DIVISION
8101 Carpenter Freeway
Dallas, Tesas 75257

COWL INDUSTRIES, LTD.
88 Fennel Street
Winnipeg, Manitoba, CANADA

DONALDSON COMPANY, INC.
1400 West 94th Street
Minneapolis, Minnesota 55431

NELSON MUFFLER COMPANY
Box 51
Stoughton, Wisconsin

RIKER MANUFACTURING, INC.
4901 Stickney Avenue
Toledo, Ohio 43612

STEMCO MANUFACTURING CO., INC.
300-312 Industrial Blvd.
Longview, Texas 75601

UNIVERSAL SILENCER CORP.
P.O. Box 268
Libertyville, Illinois 60048

WALKER MANUFACTURING COMPANY
1201 Michigan Blvd.
Racine, Wisconsin 53402

AIR CLEANER MANUFACTURERS

AIR-MAZE DIVISION
25008 Miles Road
Cleveland, Ohio 44128

CANADIAN FILTERS, LTD.
277 Williams Street, South
Chatham, Ontario, CANADA

DONALDSON COMPANY, INC.
1400 West 94th Street
Minneapolis, Minnesota 55431

FARR COMPANY
2301 Rosecraus Blvd.
El Segundo, California 90245

FRAM CORPORATION
105 Pawtucket Avenue
Providence, Rhode Island 02916

PUROLATOR CORPORATION
970 New Brunswick
Rahway, New Jersey 07065

UNITED FILTRATION
9707 Cottage Grove Avenue
Chicago, Illinois 60628

VORTOX COMPANY
121 So. Indian Hill Blvd.
Claremont, California 91711

FANS

AIR TURBINE PROPELLER CO.
Box 218
Zelienople, Pennsylvania 16063

BROOKSIDE CORPORATION
McCordsville, Indiana 46055

FLEX-A-LITE CORPORATION
5916 Lake Grove Ave. S. W.
Tacoma, Washington 98499

HARTZELL PROPELLER CO., INC.
1964 Shroyer Avenue
Piqua, Ohio 45356

SCHWITZER DIVISION
WALLACE-MURRAY CORPORATION
1125 Brookside Avenue
Indianapolis, Indiana

FAN CLUTCHES:

HORTON INDUSTRIES, INC.
1170 15th Avenue S. E.
Minneapolis, Minnesota 55414

BENDIX CORPORATION
MOTOR COMPONENTS DIVISION
Elmira, New York 14903

FAN CLUTCHES (Continued)

BENDIX CORPORATION
HEAVY VEHICLE SYSTEMS GROUP
901 Cleveland Street
Elyria, Ohio 44035

EATON CORPORATION
FLUID POWER DIVISION
1101 W. Hanover
Marshall, Michigan 49068

ROCKFORD CLUTCH
1200 Windsor Road
Rockford, Illinois 61101

SCHWITZER DIVISION
WALLACE-MURRAY CORPORATION
1125 Brookside Avenue
Indianapolis, Indiana

WEB CONTROLS CORPORATION
320 Briarcliffe Road
West Englewood, New Jersey

SHROUDS

MEMPHIS METAL MANUFACTURING CO., INC.
795 Tanglewood Street
Memphis, Tennessee

SIPLER PLASTICS, INC.
Doyleston, Pennsylvania 18901

APPENDIX VII - REPORT OF INVENTIONS

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