

Bolt Beranek and Newman Inc.



Noise Control of an Underground Load-Haul-Dump Machine

Prepared for:

**United States Department of the Interior
Bureau of Mines**

By

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50 Moulton Street
Cambridge, MA 02138**

National Mine Health & Safety Academy
Learning Resource Center

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Final Report

on

Contract No. USBM ~~H026013~~ H0262013
Noise Control of Underground Load-Haul-Dump Machines

OFR
78-125

July 1977

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(this page replaces old Title Page)

REPORT DOCUMENTATION PAGE

1. Report No.	2.	3. Recipient's Accession No.	
4. Title and Subtitle NOISE CONTROL OF AN UNDERGROUND LOAD-HAUL-DUMP MACHINE		5. Report Date July 1977	
		6.	
7. Author(s) Geoffrey G. Huggins, Richard Madden, Bruce S. Murray		8. Performing Organization Report No. BBN Report No. 3595	
9. Performing Organization Name and Address Bolt Beranek and Newman Inc. 50 Moulton Street Cambridge, MA 02138		10. Project/Task/Work Unit No.	
		11. Contract or Grant No. USBM No. H0262013	
12. Sponsoring Organization Name and Address U.S. Department of the Interior Bureau of Mines 4800 Forbes Avenue Pittsburgh, PA 15213		13. Type of Report Final Report	
		14.	
15. Supplementary Notes			
16. Abstract Diagnostic tests were performed on a production ST-5A Load-Haul-Dump machine manufactured by Wagner Mining Machinery Company to evaluate and rank order noise sources. The diagnostic evaluation indicated that the major noise sources were the transmission, the engine, and the cooling fan. This phase was followed by implementation of recommended noise control treatments, which included vibration isolation of the transmission, a transmission enclosure, an engine enclosure, and modifications of the inlet to the cooling fan. The final step in the program was an underground operational period which began on 31 August 1976 at the National Gypsum Mine in Shoals, Indiana. The noise control treatments successfully reduced the LHD operator noise exposure by 7 dB(A), thus increasing its allowable operation time by a factor of 2.7. Actual measurements made underground using noise dosimeters have indicated that, during a typical shift, the noise control treatments have reduced the fractional exposure to an allowable level. The report discusses the mechanical aspects of the LHD, its noise sources, the treatments applied, the final levels achieved, and field experience with the noise-controlled machine.			
17. Originator's Key Words Noise control, mining machines, Load-Haul-Dump machines		18. Availability Statement	
19. U. S. Security Classif. of the Report UNCLASSIFIED	20. U. S. Security Classif. of This Page UNCLASSIFIED	21. No. of Pages	22. Price

FOREWORD

This report was prepared by Bolt Beranek and Newman Inc., Cambridge, MA under USBM Contract No. H0262013. The contract was administered under the technical direction of PMSRC, with Dr. H. K. Sacks acting as the Technical Project Officer. Mr. David Vila was the contract administrator for the Bureau of Mines.

Work performed between April 1976 and July 1977 is summarized in this report, which was written in July 1977.

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1. INTRODUCTION

There has been considerable interest in the noise exposure of underground miners since the enactment of the Federal Metal and Non-Metallic Health and Safety Act.* This law prompted the U.S. Bureau of Mines to issue a research contract (Contract No. HO346046) to identify those pieces of mining machinery which are principal contributors to the noise exposure of underground metals and nonmetallic miners. In the course of this contract, the diesel-powered load-haul-dump (LHD) machine was identified as a major contributor not only because of high noise levels of individual machines, but also because of the large number of these machines in use.

The final report on this study [1], which was submitted in March 1975, estimated that 4000 diesel-powered mining machines were in use in underground operations. This 4000 total includes approximately 690 LHDs. Operator sound pressure levels measured at 100 dB(A) were typical on LHDs. Since the estimated duty cycle for this equipment is 0.8, the equivalent noise exposure is 100 dB(A) for 6 hours or a fractional noise exposure of 3, according to the Health and Safety Act.* In other words, the operator is exposed to three times his daily allowable exposure each day. This figure, coupled with the 690 machines in use, gives an *equivalent* of 2070 persons exposed to the limit of noise each working day. This problem led the U.S. Bureau of Mines to award a contract (HO262013) to demonstrate noise control on a LHD. This report describes the results of that demonstration program.

The machine selected for the program was the Wagner Mining Equipment Company's Model ST-5A, Scooptram[®], the most widely

*Code of Federal Regulations, Title 30, Chapter I, Part 57, Public Law 89-577.

used LHD. The ST-5A is powered by an air-cooled 8-cylinder Deutz diesel engine and has a 5 cu-yd bucket capacity.

During the course of the program, numerous diagnostic measurements were performed to rank-order the noise sources. The diagnostic evaluation indicated that the major noise sources were the transmission, the engine, and the cooling fan. This evaluation phase was followed by implementation of recommended noise control treatments, which included vibration isolation of the transmission, a transmission enclosure, an engine enclosure, and modifications of the inlet to the cooling fan. The final step in the program was an underground operational period which began on 31 August 1976 at the National Gypsum Mine in Shoals, Indiana. The noise control treatments successfully reduced the LHD operator noise exposure by 7 dB(A), thus increasing its allowable operation time by a factor of 2.7. Actual measurements made underground using noise dosimeters have indicated that, during a typical shift, the noise control treatments have reduced the fractional exposure to an allowable level.

Mechanical aspects of the LHD, its noise sources, the treatments applied, the final levels achieved, and field experience with the noise-controlled machine, are discussed in the following pages.

2. PHYSICAL DESCRIPTION OF LOAD-HAUL-DUMP MACHINE

The machine selected for the noise control demonstration was the Wagner ST-5A Scooptram[®] pictured in Fig. 1. The ST-5A is one of the most popular and widely used load-haul-dump machines for underground mining. During the course of the program, Wagner, as subcontractor to BBN, provided a machine for noise source diagnosis and a second for fabricating and installing the noise control treatments. The machine is a low-profile loader (67 in. high) with a 5-cu-yd bucket mounted on the front. Just to the rear of the bucket is the center-articulated steering which allows short-radius turns in the mine. The driver is seated at the side of the aft section of the vehicle, facing inward, since the vehicle has the same tram capabilities in either forward or reverse. Also located in the aft section is the air-cooled diesel engine, the torque converter, and the transmission.

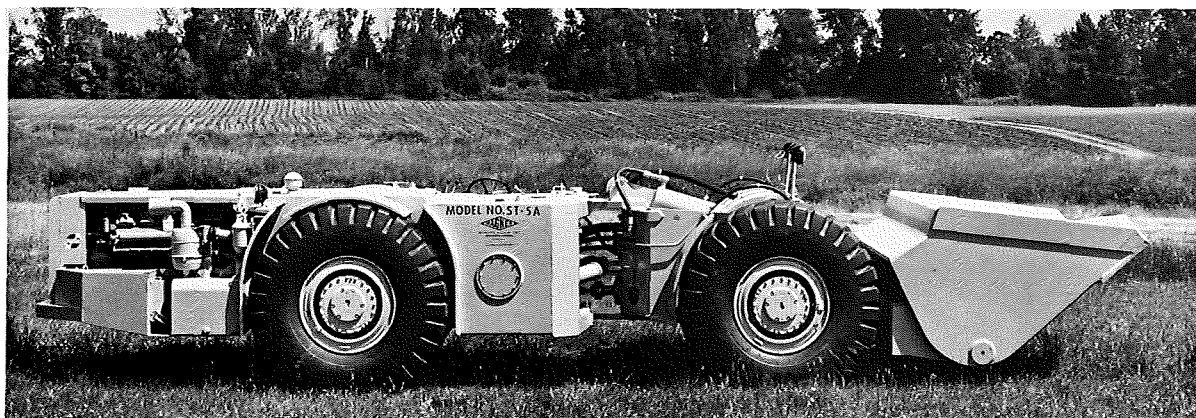


FIG. 1. LOAD-HAUL-DUMP MACHINE.

A schematic of the power train of the ST-5A is shown in Fig. 2. The diesel engine is a Deutz 8-cylinder model F8L-714, which is rated by U.S. Bureau of Mines Schedule 24 at 180 hp at 2300 rpm. Connected directly to the engine is the single-stage Clark Model C-8402 torque converter. A shaft from the torque

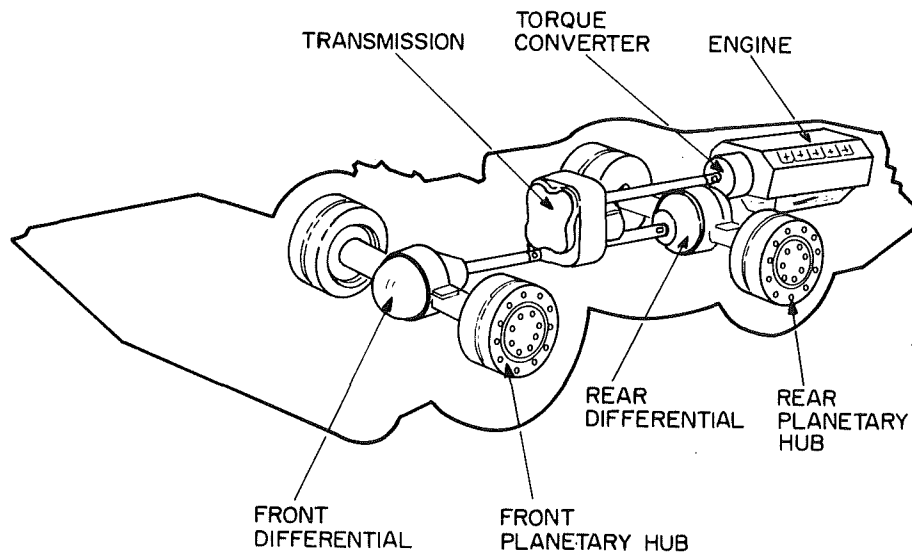


FIG. 2. DRIVE TRAIN FOR LOAD-HAUL-DUMP MACHINE.

converter drives the 4-speed forward and reverse Clark Model 3421 transmission which, in turn, drives the front and the rear differentials. The individual differentials then drive the front and rear planetary hubs on the wheels.

The location of the components relative to the operator is shown in Fig. 3, which also illustrates the location of other components relevant to the discussion of noise sources and noise control treatments. These components are the hydraulic pumps, the combustion air intake for the engine, the cooling fan for the engine, and the engine exhaust.

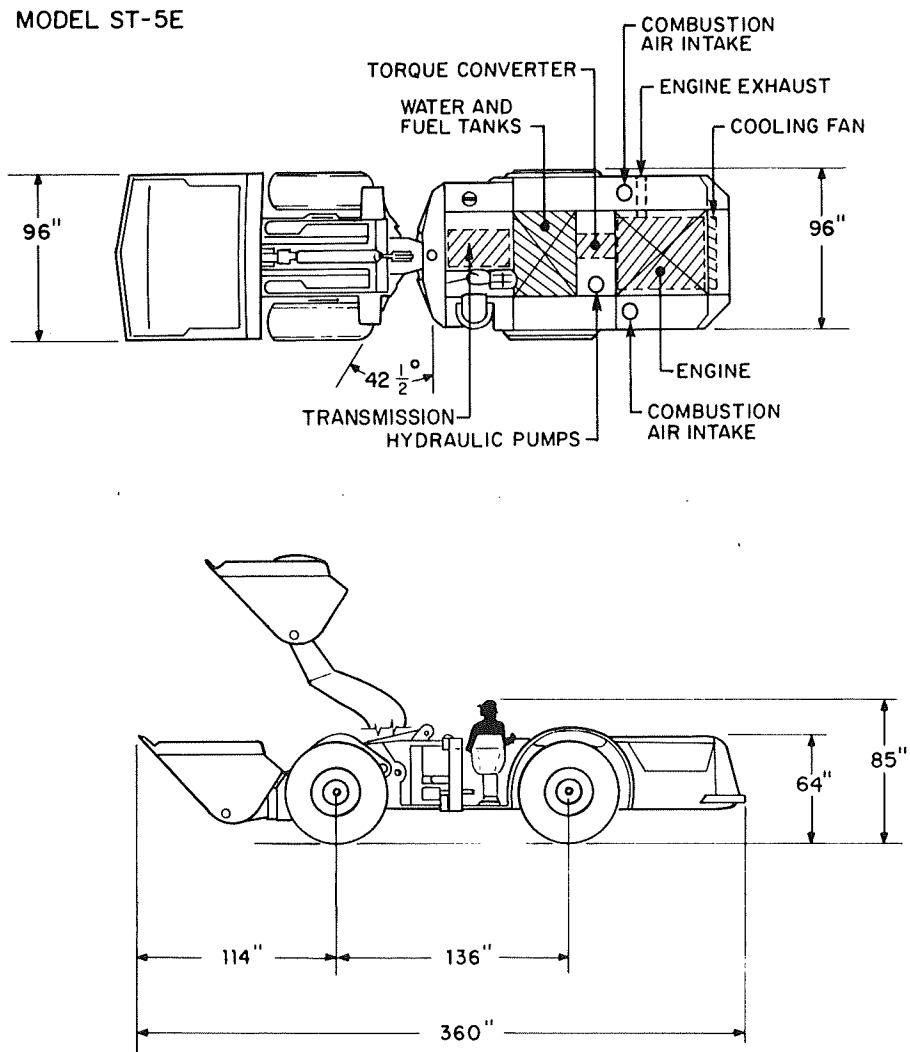


FIG. 3. LOCATION OF COMPONENTS IN LHD.

3. ACOUSTIC DESCRIPTION OF LOAD-HAUL-DUMP MACHINE

The physical description of the LHD in Sec. 2 gives some insight into the noise sources associated with the machine. The major sources are the engine, its intake and exhaust, the cooling fan for the engine, and the drive train. Hydraulic pumps were not considered, since the operator's exposure to this noise is of short duration at irregular intervals and, consequently, does not make a significant contribution to his fractional exposure. Each of these sources contributes sound to the operator by a direct airborne path. Some of the sources also contribute by structure-borne paths, whereby vibrational energy is transmitted to surfaces near the operator which, in turn, vibrate and radiate sound at the operator.

Establishment of the various source levels and paths was difficult, because some components could not be operated separately. In addition, many sources share common structureborne paths. A two-part approach was used to isolate the contributions of various sources. First, a cover-and-expose technique was employed wherein the major sources were covered with lead sheets and the exhaust was piped away to give a baseline "blanket" quiet condition. Second, the individual components were exposed to give their contributions. In addition, operating conditions of high idle, torque converter stall, and operation in fourth gear were used to accentuate the various sources. Vibration of various components was measured to determine the structureborne paths, and noise data were obtained to determine operator exposure. The data were analyzed in terms of 1/3-octave bands to quantify the source levels.

Noise control was also a two-step program. First, the noise source diagnosis was performed on an ST-5A directly off

the production line. Treatments were designed and implemented on a second ST-5A, which was then placed in service underground. This procedure involves some risk because it did not provide a true "before" and "after" noise picture on the same machine. However, it was the only logical course because of the constraint of minimum disruption of production. The fact that we were able to quiet the second machine to a level where it is in compliance illustrates that this risk is minimal. In addition, our achievement in quieting a different machine than the one used for source diagnosis gives us confidence that the noise control treatments designed for this machine should be equally applicable to others.

3.1 Initial Noise Level

The sound spectrum of an unquieted ST-5A is presented in Fig. 4. This measurement was taken at the operator position for both high idle and torque converter stall while the machine was parked outdoors and on grass. Also shown is another operator position measurement, made on a Wagner ST-5B underground in the high idle operating condition. There is a difference of approximately 4 to 5 dB between the aboveground and underground measurements. Interestingly, there is a great similarity between measurements at high frequencies, presumably because of the large room underground where the measurements were made. The spectra are broad, with only one tone present in the 100-Hz 1/3-octave band. This tone, as will be shown later, is contributed by the exhaust.

The noise source diagnosis was conducted indoors in the shop area at Wagner. The room in which the diagnostic tests were conducted augmented the sound pressure levels measured at the operator position, increasing levels by 4 to 6 dB(A). A comparison of the previously presented levels measured outdoors

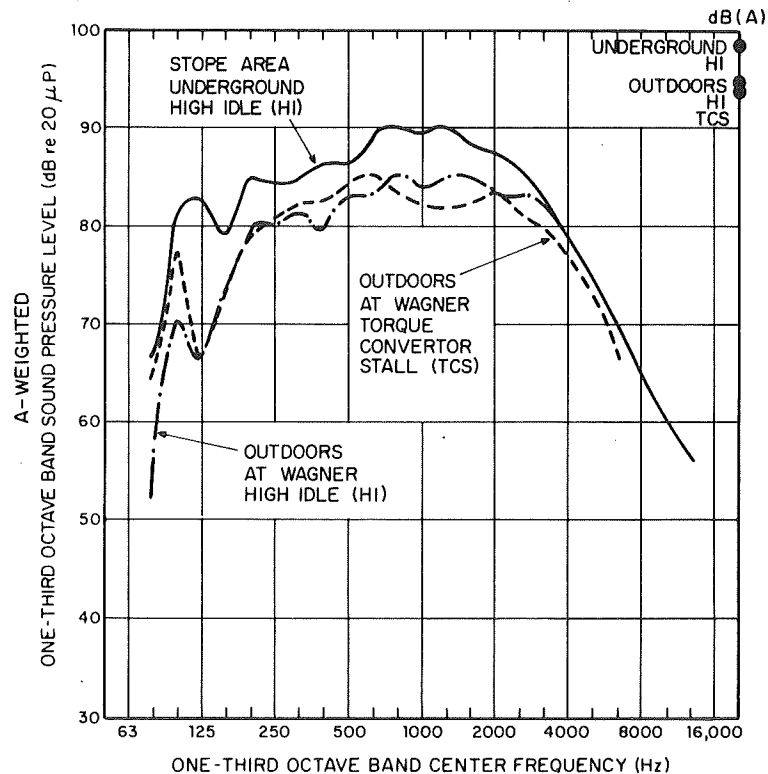


FIG. 4. COMPARISON OF ABOVEGROUND AND UNDERGROUND SOUND PRESSURE LEVELS MEASURED ON WAGNER LHD.

with those measured indoors is given in Fig. 5 for the high idle case and in Fig. 6 for torque converter stall. The machine was tested indoors on jacks as shown in Fig. 7. All diagnostic tests were performed while the LHD was in this position, with the wheels raised so that the transmission could be operated without moving the vehicle.

Table 1 lists the sources of noise, the paths that contribute to the operator noise exposure, and the test condition that would enhance the contribution of that source. For this machine, the speed at high idle is typically 100 rpm higher than that torque converter stall (i.e., 2400 rpm vs 2300 rpm).

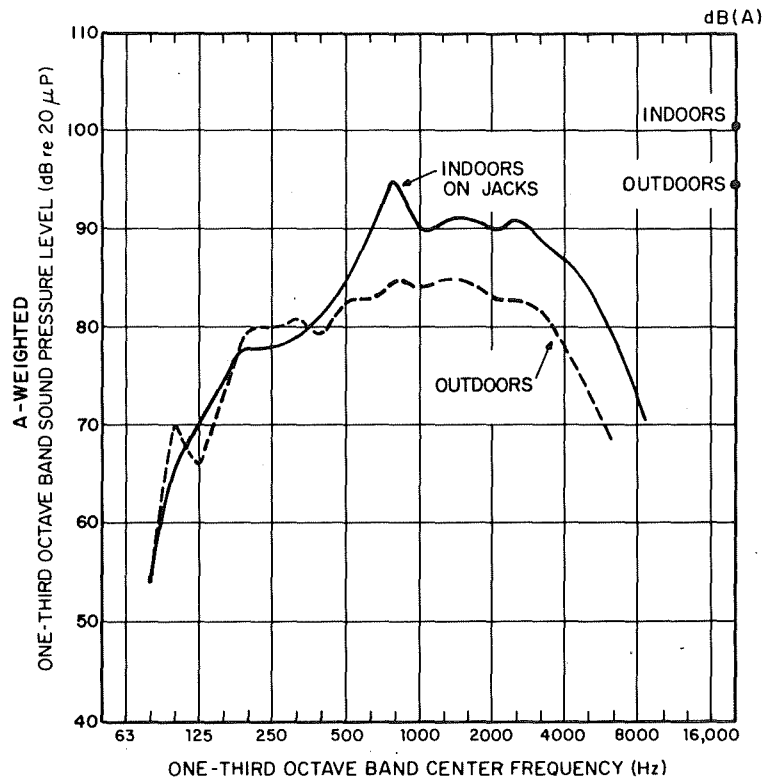


FIG. 5. COMPARISON OF INDOOR AND OUTDOOR MEASUREMENTS ON UNTREATED ST-5A IN HIGH IDLE CONDITION AT OPERATOR POSITION.

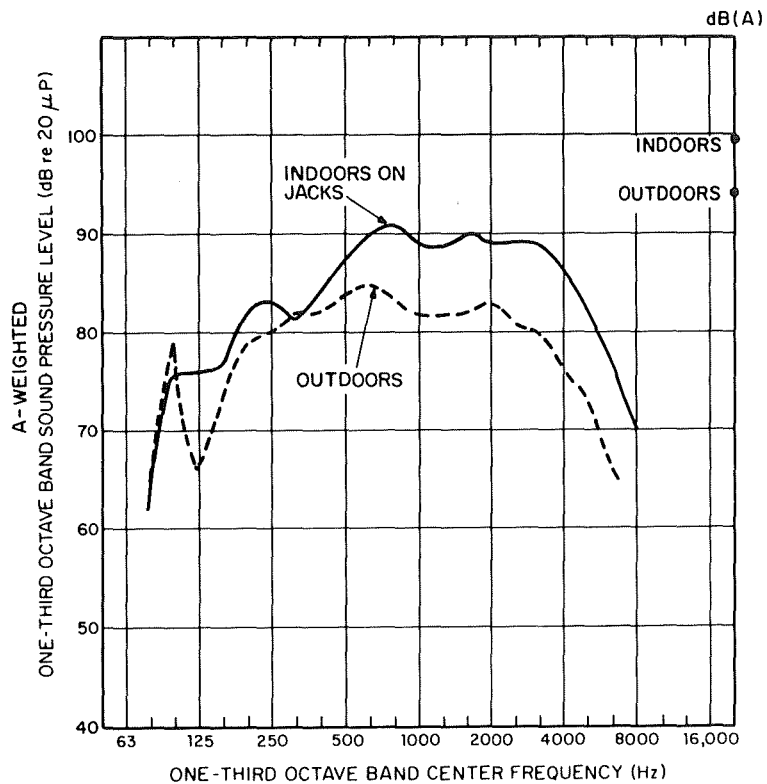


FIG. 6. COMPARISON OF INDOOR AND OUTDOOR MEASUREMENTS ON UNTREATED ST-5A IN TORQUE CONVERTER STALL CONDITION AT OPERATOR POSITION.

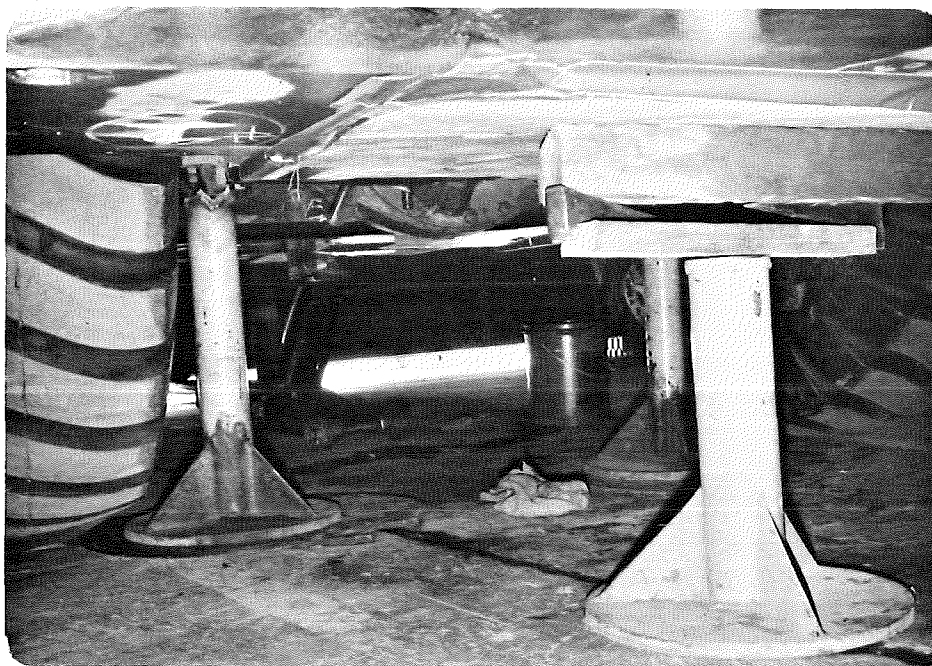


FIG. 7. JACKS USED TO RAISE WHEELS FOR DIAGNOSTIC TESTS.

TABLE 1. TEST CONDITIONS FOR SOURCE DIAGNOSIS.

Source	Path	Test Condition	Comments
Engine	Airborne Structureborne	High Idle	Proportional to rpm
Engine Air Intake	Airborne	High Idle	Proportional to rpm
Engine Cooling Air Fan	Airborne	High Idle	Proportional to 6 th power of rpm
Engine Exhaust	Airborne	Torque Converter Stall	Proportional to engine load
Transmission	Airborne Structureborne	Rated Speed in 4 th gear	Must be in gear

3.2 "Blanket" Noise Control Treatment

Individual noise source levels were evaluated through the use of a "blanket" noise control treatment, wherein each of the major contributors to the overall level was covered with lead sheet and the exhaust was piped away. The drive shaft was disconnected for all tests except those evaluating transmission noise. This procedure established the "noise floor" for the diagnostic measurements by providing a minimum noise level which was reasonably achievable in the shop area. The components of the blanket treatment are shown in Figs. 7, 8 and 9. Figure 7 shows the plywood sheets used to seal the area under the engine. Figure 8a shows the 1.5-lb/ft² lead sheet which was used to cover the engine.* Also shown is the flexible hose which was used to

*Absorptive material was placed under the lead to prevent reverberant sound buildup in the engine compartment.

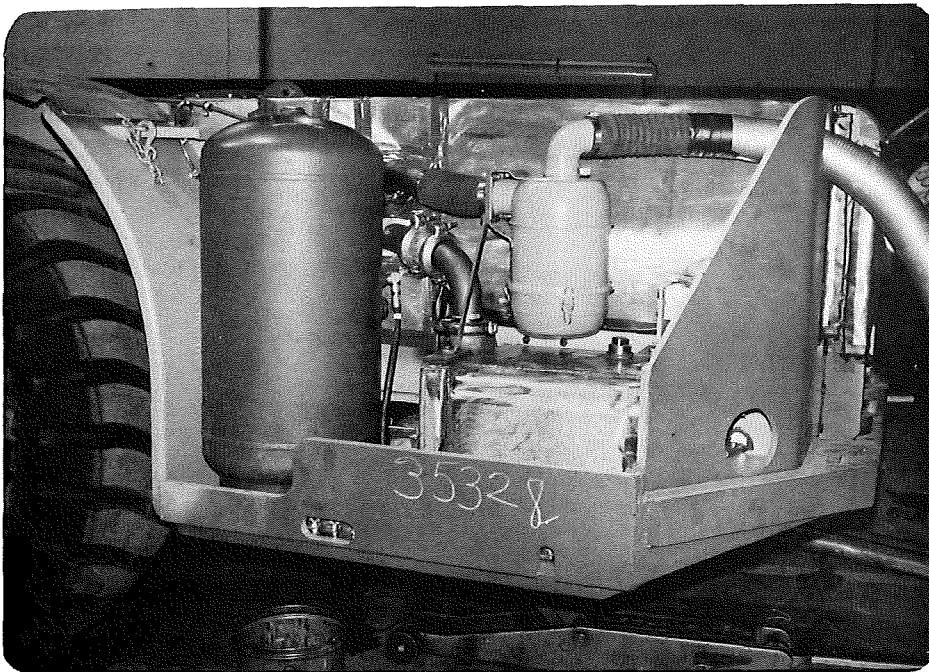


FIG. 8a. BLANKET NOISE CONTROL TREATMENT OF ENGINE AND DUCTING OF COMBUSTION AIR INTAKE.

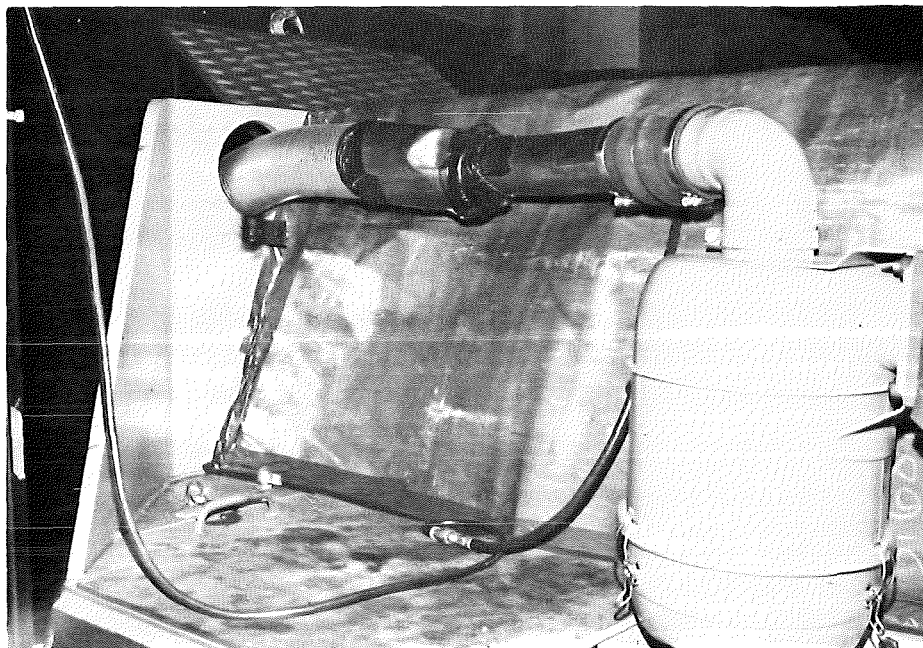


FIG. 8b. SIMILAR TO 8a BUT OTHER SIDE.

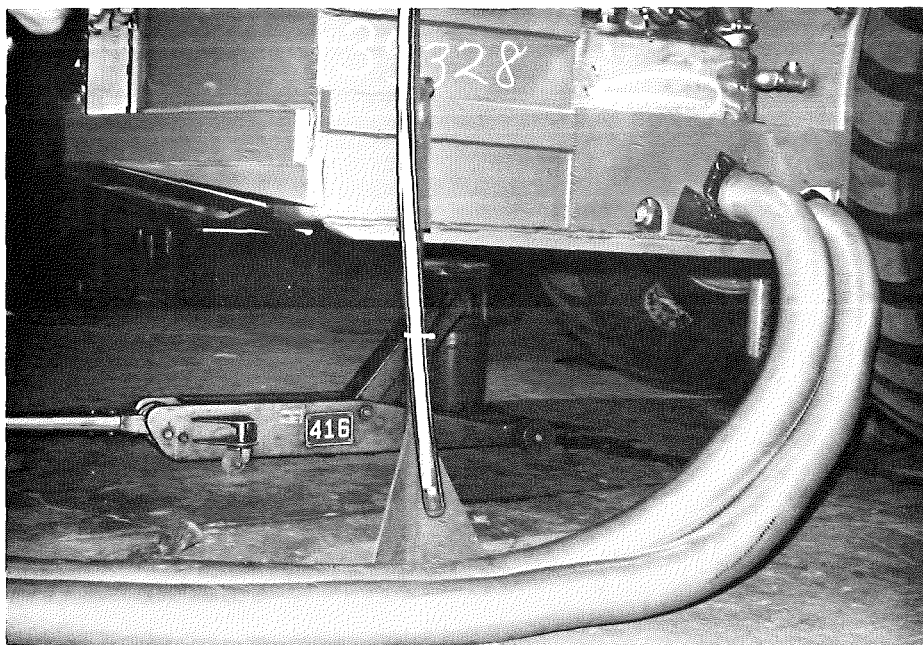


FIG. 8c. DUCTING OF ENGINE EXHAUST.

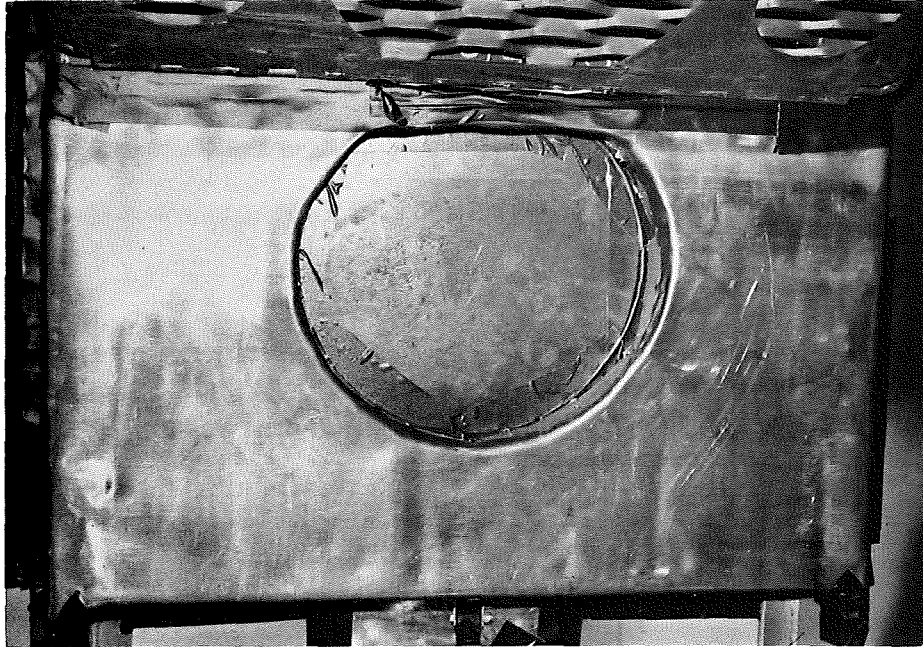


FIG. 9a. BLANKET
NOISE TREATMENT
NEAR COOLING FAN.

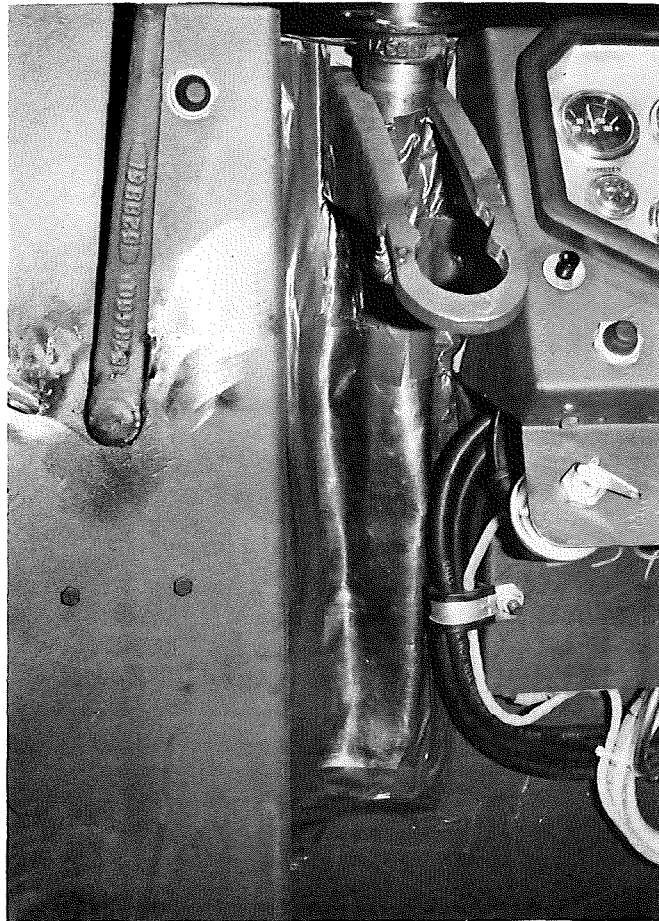


FIG. 9b. SEALING THE
OPERATOR COMPARTMENT.

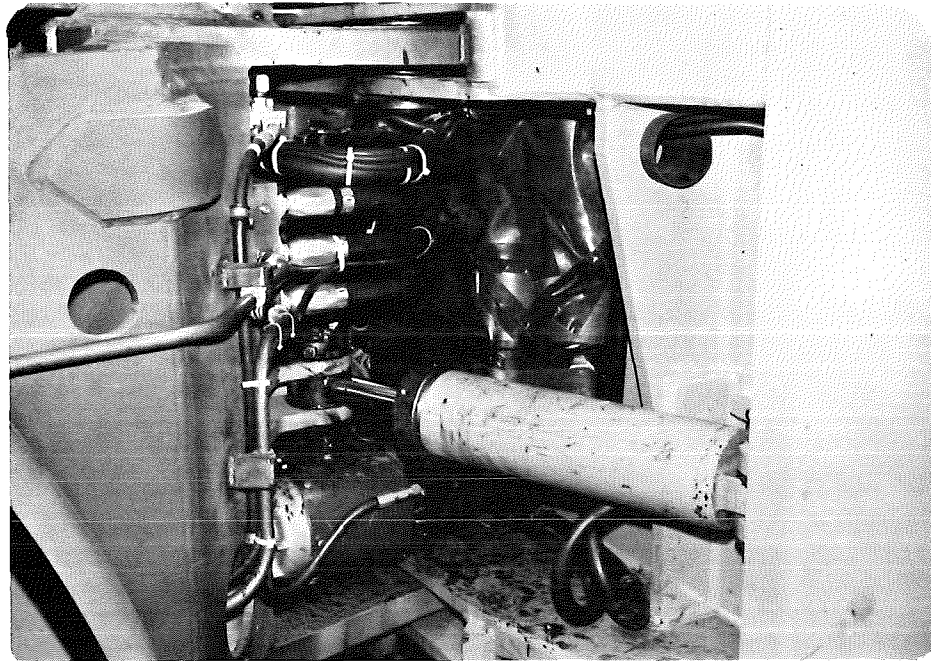


FIG. 9c. SEALING OF HOLES AROUND TRANSMISSION NEAR CENTRAL PIVOT.

duct away the combustion air intake, and Fig. 8b shows the same view from the other side of the vehicle. Figure 8c shows the ducting of the exhaust from the water scrubbers. Figure 9a shows the 1.5-lb/ft² lead sheet in the cooling fan area. Figures 9b and 9c illustrate the sealing of holes near the transmission; Fig. 9b is the sealing around the steering column, and Fig. 9c is the sealing in the area where the vehicle is articulated. Additional components of the blanket treatment, not shown, included absorptive material applied to the transmission cover, and sealing in the floor area of the operator's position.

The effectiveness of the blanket noise control treatment is illustrated in Fig. 10a and 10b for the high idle and torque converter stall operating modes, respectively. The figure presents the A-weighted spectrum of the LHD, measured indoors prior

FIG. 10a. COMPARISON OF LEVELS BEFORE AND AFTER BLANKET NOISE TREATMENT AT HIGH IDLE.

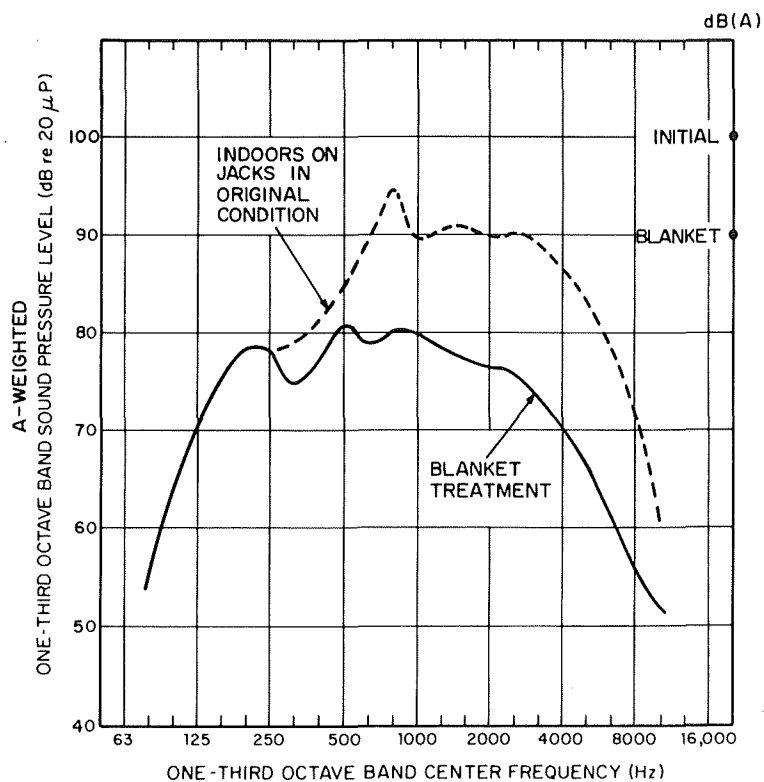
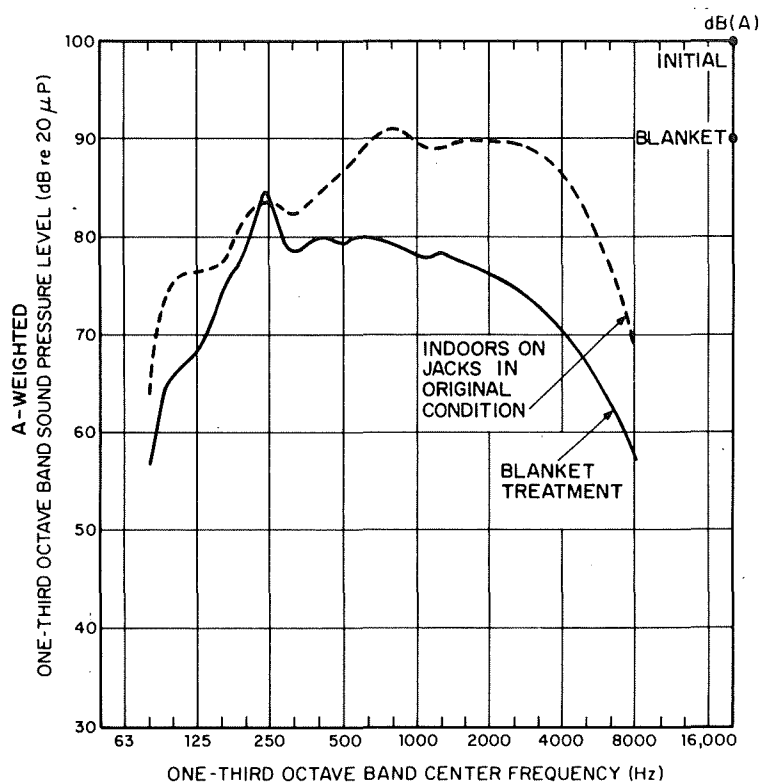


FIG. 10b. COMPARISON OF LEVELS BEFORE AND AFTER BLANKET NOISE TREATMENT AT TORQUE CONVERTER STALL.



to the blanket treatment, compared to the spectrum measured after the blanket treatment, with the drive shaft disconnected. The blanket treatment resulted in approximately 10 dB(A) of noise reduction, in both operating modes. The major attenuation occurring at frequencies above 500 Hz.

3.3 Individual Source Diagnosis

The magnitude of the individual sources was obtained by removing various components of the blanket treatment and by operating the machine in a condition which would accentuate the source under consideration. The procedure used for each source, and the resultant levels, are discussed in the following paragraphs. The overall source levels, including both airborne and structureborne path noise, are treated first and are followed by an estimate of the structureborne component of the engine and transmission noise.

3.3.1 Overall source levels

Engine Exhaust

The contribution of the engine exhaust was determined by removing the exhaust ducting only, leaving the remainder of the blanket noise control treatment. The machine was operated in the torque converter stall mode. The A-weighted 1/3-octave band levels measured under this condition, as compared to the blanket treatment, are shown in Fig. 11. The high idle test condition is also shown, since it will be used in evaluating engine noise. The speed of the engine in torque converter stall is approximately 2300 rpm, and since the engine has 8 cylinders, the firing rate is

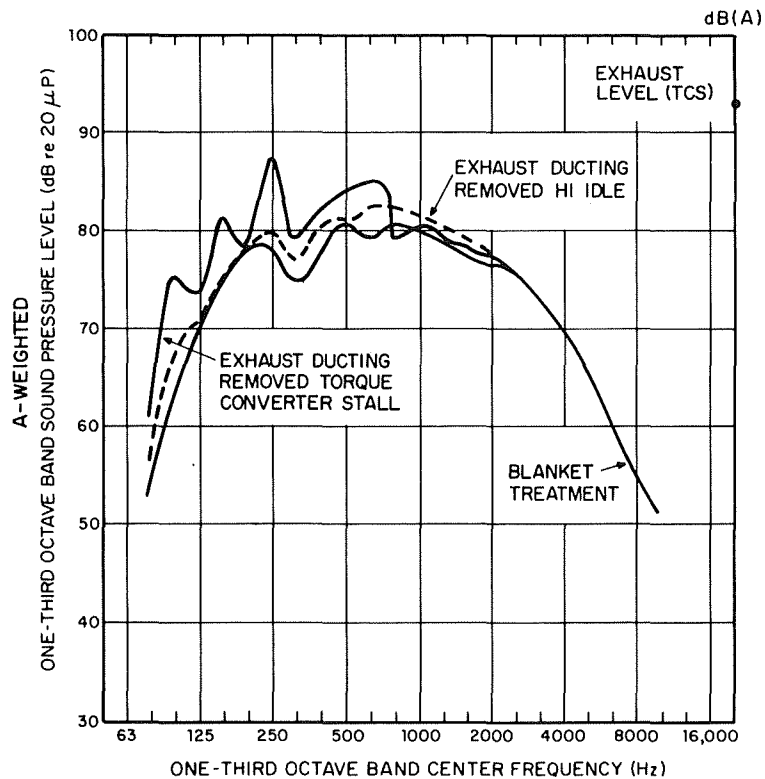


FIG. 11. EXHAUST SOURCE LEVELS AT OPERATOR POSITION.

$$f_r = (\text{rpm}/60) (\text{No. cyl}) (\text{cyl ignitions/rev})$$

$$= (2300/60) (8) (1/2) \approx 153 \text{ Hz}.$$

This tone is evident in the 1/3-octave band centered at 160 Hz. Also evident are tonal contributions in the 100- and 250-Hz bands. The source of this tone at 100 Hz appears to be the firing frequency being modulated by the rpm of the engine, i.e., $153 - (2300/60) = 114 \text{ Hz}$. The tone at 250 Hz is then the second harmonic of the firing rate, again modulated by the engine speed. The overall level for the exhaust is 93 dB(A) at the operator position.

Engine Air Intake

The contribution of the engine air intake was determined by removing the inlet ducting only, leaving the remainder of the blanket treatment. The machine was then operated in the high idle mode to maximize the contribution of this source.

The A-weighted 1/3-octave band levels for this source are shown in Fig. 12, compared with the noise floor achieved under the blanket quieting treatment. Note that only minor increases in the noise level resulted from removal of the air intake ducting. The only recognizable contributions which appear are in the

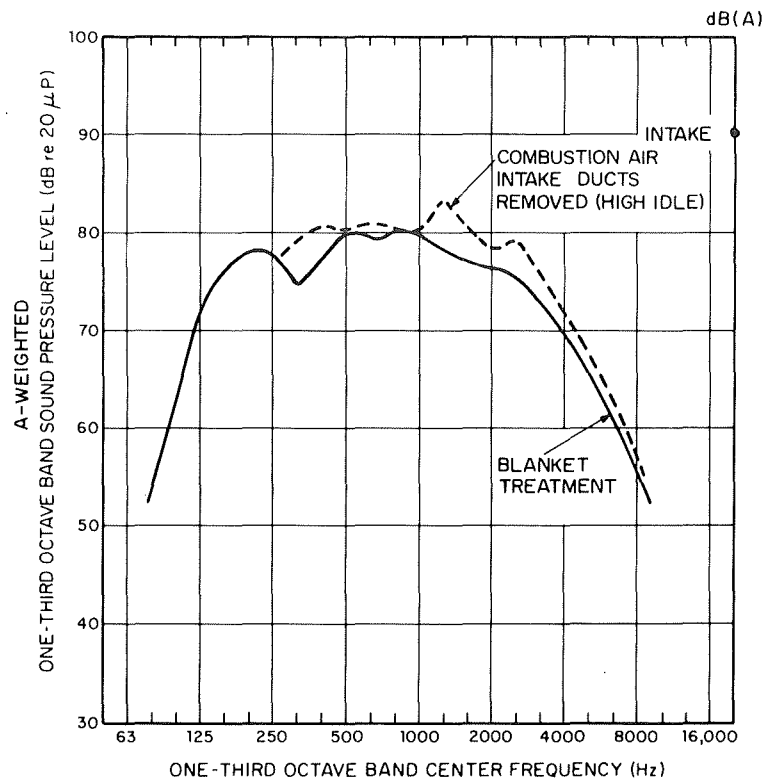


FIG. 12. COMBUSTION AIR INTAKE LEVELS AT OPERATOR POSITION.

1250-Hz 1/3-octave band and a harmonic in the 2500-Hz 1/3-octave band. The overall level for this source is approximately 90 dB(A) at the operator station. Since there is so little difference between the blanket treatment and that with the intake ducts removed, the spectrum should be regarded as an upper bound.

Cooling Fan

Cooling air from the Deutz engine is ducted over the top of the engine and then exhausted following passage by the transmission oil heat exchangers. Cooling fan airborne noise can arrive at the operator by two paths: One starts at the intake to the fan and travels on the outside of the machine to the operator; the other follows the cooling air over the engine through the exhaust ducts and to the operator. In the latter case, it is not possible to isolate the cooling fan noise from some engine noise. With this constraint, the contribution of the cooling fan was determined at the high idle test condition, with the cover over the cooling fan opening removed and the plywood sheets under the engine removed from the blanket quieting treatment. All other treatments remained intact. The 1/3-octave band levels associated with this source are given in Fig. 13. The overall source level was 94 dB(A) when the doors to the shop area were closed, and 92.5 dB(A) when they were open. In addition to the broadband noise of the fan, there is a tone in the 800-Hz 1/3-octave band. The tone corresponds to the blade rate of the 8 bladed fan [i.e., $[\text{fan rpm}/60] \times \text{no. of blades}$: $(5300/60) \times 8 = 706 \text{ Hz}$].

Engine

The contribution of the engine source was obtained by subtracting the 1/3-octave band levels of the fan, exhaust, and intake at the high idle test condition from the original spectrum

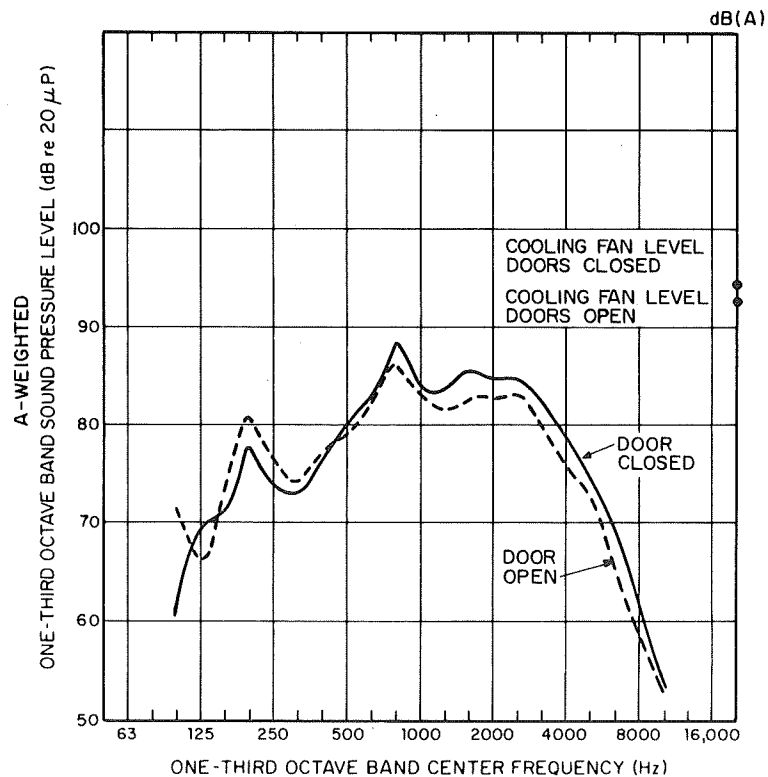


FIG. 13. COOLING FAN SOURCE LEVELS AT OPERATOR POSITION.

presented in Fig. 5. The resultant engine airborne 1/3-octave band levels are presented in Fig. 14. The overall level associated with the engine is 96 dB(A).

Transmission

The final source is the transmission. The levels associated with this source were determined by connecting the drive shaft and operating the transmission in fourth gear at maximum rated speed. The 1/3-octave band levels for the unquieted condition and the blanket treatment condition are presented in Fig. 15. The upper curve presents data for the unquieted condition,

FIG. 14. ENGINE AIRBORNE
SOURCE LEVEL AT
OPERATOR POSITION
AT HIGH IDLE.

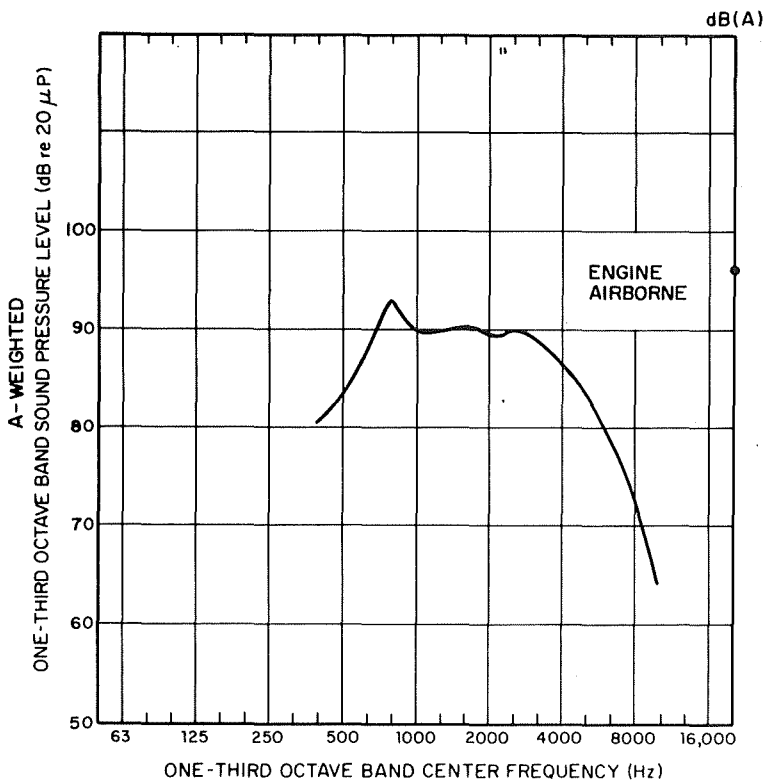
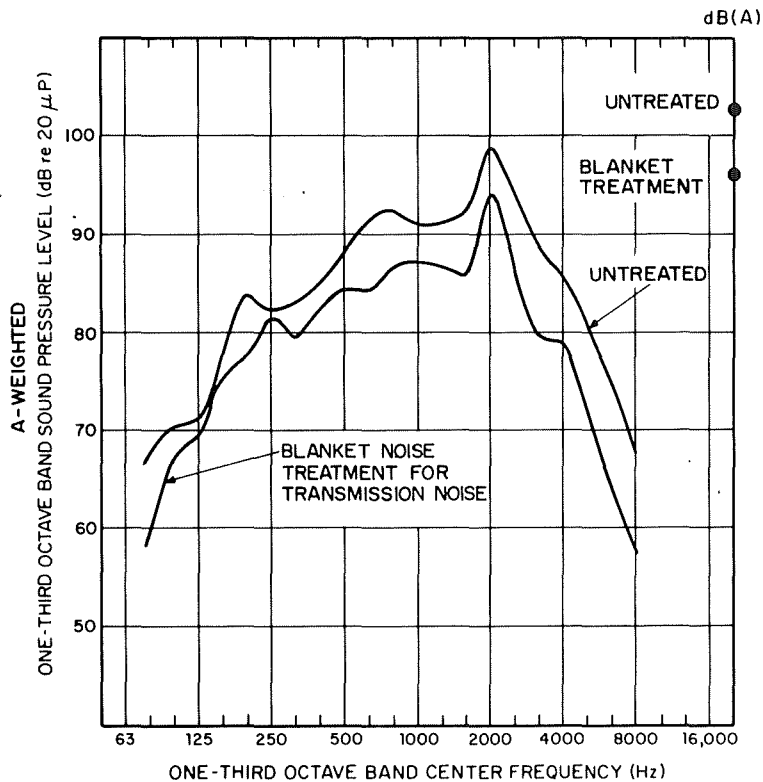


FIG. 15. TRANSMISSION NOISE
SOURCE EVALUATION
OBTAINED BY OPER-
ATING IN FOURTH
GEAR AT RATED
SPEED.



whereas the lower curve represents data with the blanket treatment.

The 1/3-octave band levels for the transmission exposed are presented in Fig. 16. The two curves are the results of the test in (1) raw data form and (2) after correcting for the noise floor as established by the blanket treatment. The transmission contains a strong tone in the 2000-Hz 1/3-octave band. Comparison of the spectra between high idle, torque converter stall, and operating in fourth gear with wheels spinning indicates that the principal contribution of the transmission is in this 2000-Hz 1/3-octave band. The overall level associated with transmission noise was determined to be 101 dB(A).

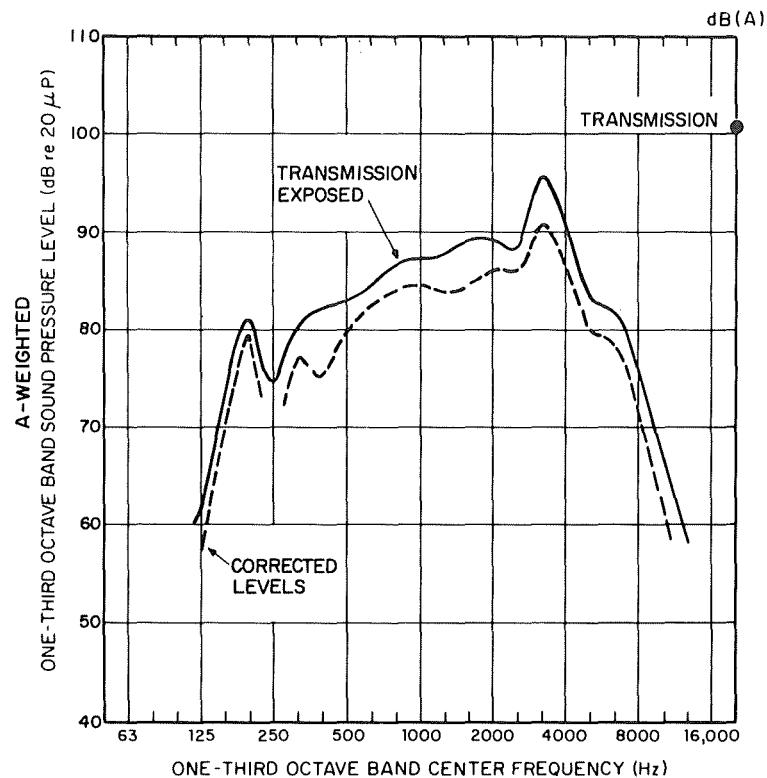


FIG. 16. TRANSMISSION SOURCE LEVELS AT OPERATOR POSTION.

3.3.2 Structureborne noise

Two components were assumed to be the major contributors to the structureborne noise. These are the engine and the transmission. For both components, the procedure for estimating the structureborne noise was the same. A transfer function was developed between sound and vibration determined by exciting the structure at a mounting point with an electric chipping hammer, while measuring the vibration on the main structure below the mount and the sound pressure level at the operator position. This transfer function was then used with accelerations measured during operation (high idle mode for the engine, and fourth gear at rated speed for the transmission) to predict the sound pressure level at the operator position. The sound pressure level was compared to the data obtained under the same operating condition with the blanket treatment in place. This comparison gave an estimate of the frequency bands where structureborne sound is important.

Engine

The engine is mounted to the frame rails at one point on each of its four corners. The mounting system employs four Huntington M500-70CR vibration isolators. The chipping hammer was applied to the left front engine mount (nearest to the fan) and the left rear engine mount (nearest to the operator). Figures 17 and 18 present the acceleration measured on the frame rails adjacent to the mounting points, the sound pressure level at the operator station, and the resulting transfer function for the front and rear mounts, respectively.

Vibration measurements were made at the same locations during operation at high idle. Figure 19 presents these vibration levels and the predicted sound pressure levels at the operator position, using the transfer functions from Figs. 17 and 18. Figure 20 compares the sum of these two sound pressure levels,

FIG. 17. TRANSFER FUNCTION
FOR LEFT FRONT
ENGINE MOUNT.

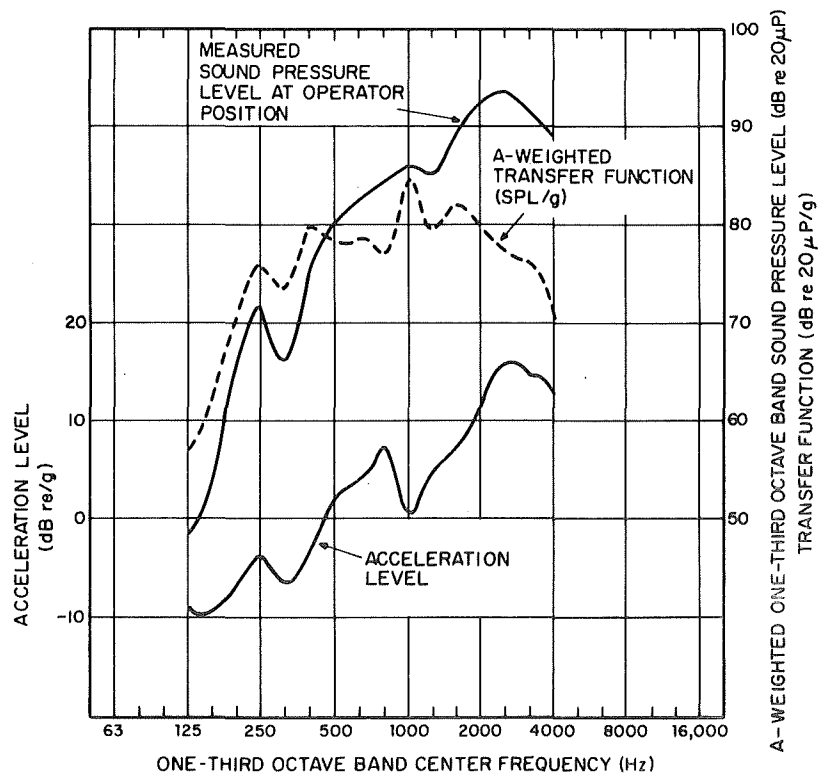


FIG. 18. TRANSFER FUNCTION
FOR LEFT REAR
ENGINE MOUNT.

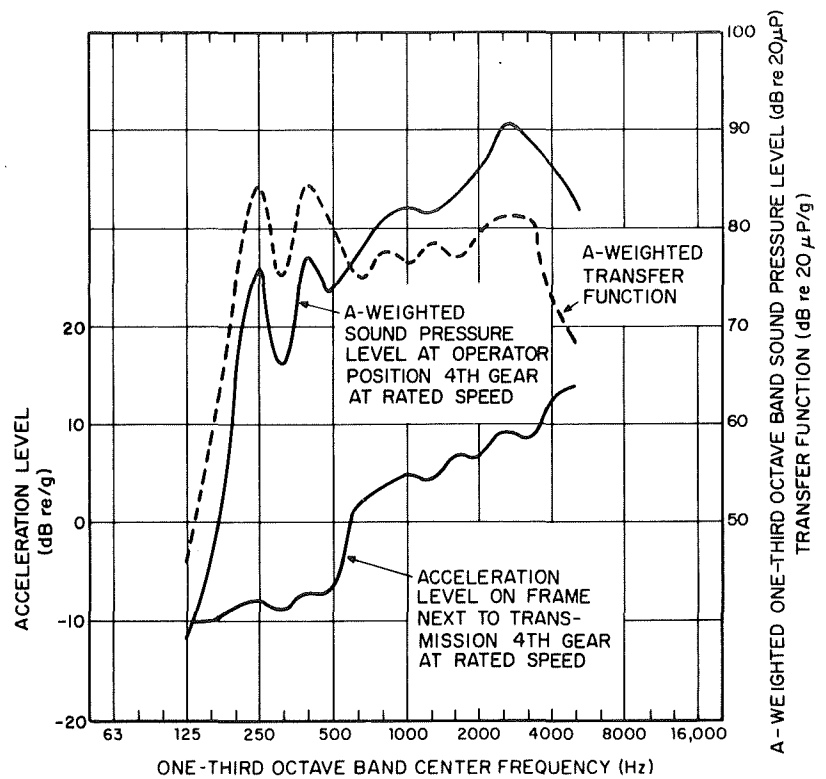


FIG. 19. COMPONENTS OF PREDICTED STRUCTURE-BORNE SOUND BASED ON TRANSFER FUNCTION FOR ENGINE VIBRATION.

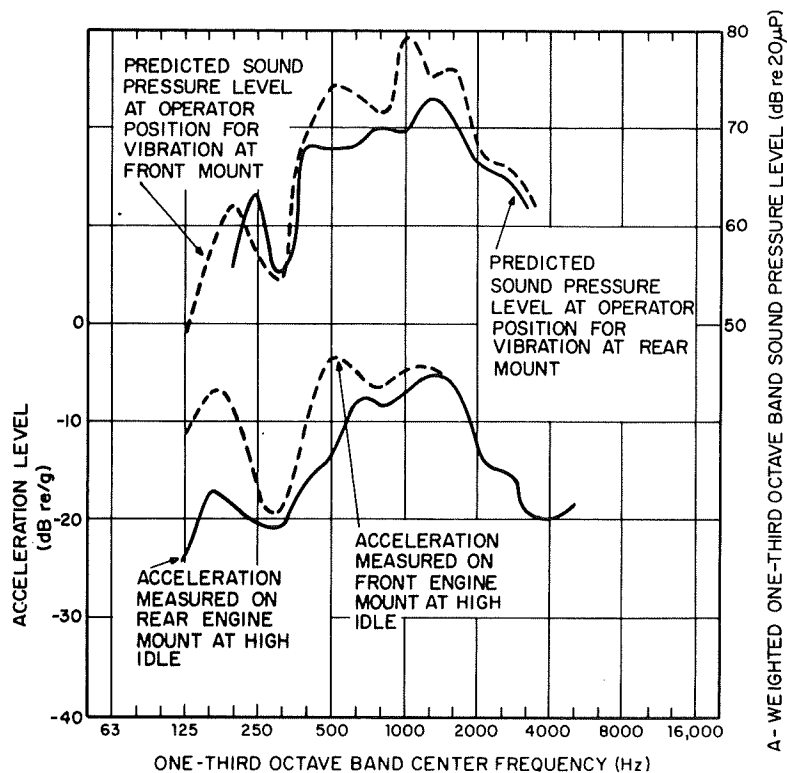
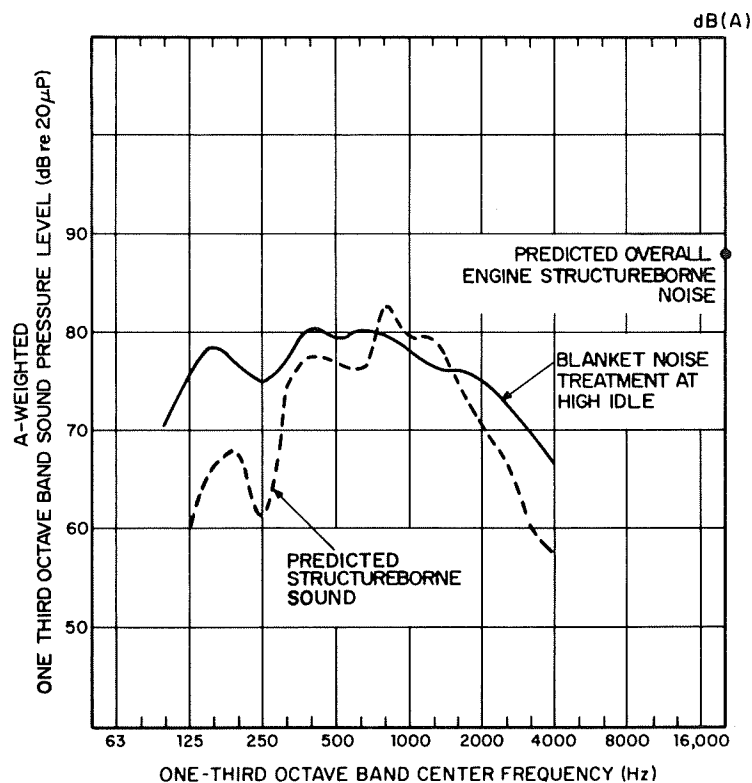


FIG. 20. PREDICTED STRUCTUREBORNE SOUND BASED ON TRANSFER FUNCTION FOR ENGINE VIBRATION.



increased by 3 dB because there are mounting points on both sides of the engine, with the 1/3-octave band levels obtained with the blanket noise treatment. We note that the predicted and measured results are in reasonable agreement in the range from approximately 400 Hz to 3150 Hz. Structureborne engine noise is probably the principal contributor in this frequency range. The overall level associated with the structureborne sound is estimated by calculation to be 88 dB(A).

Transmission

The transmission was mounted rigidly to a bulkhead at four points. The chipping hammer was applied to the transmission, and the acceleration was measured at an adjacent point on the bulkhead. The transfer function between this acceleration and 1/3-octave band sound pressure level measured at the operator position is given in Fig. 21, along with the raw data used to calculate it. Figure 22 presents vibration levels measured at the same location during operation at rated speed in fourth gear, the predicted structureborne noise based on the transfer function of Fig. 21, and the 1/3-octave band sound pressure levels for the blanket noise treatment condition. The structureborne sound predicted is in reasonable agreement with the blanket noise treatment measurements in the range from 630 Hz to 5000 Hz; therefore, the sound in this range probably arrives by a structureborne path. The overall level for the structureborne sound is estimated by calculation to be 96 dB(A).

3.4 Summary of Source Diagnosis Data

A summary of the sources with their levels and frequency ranges of importance is presented in Table 2.

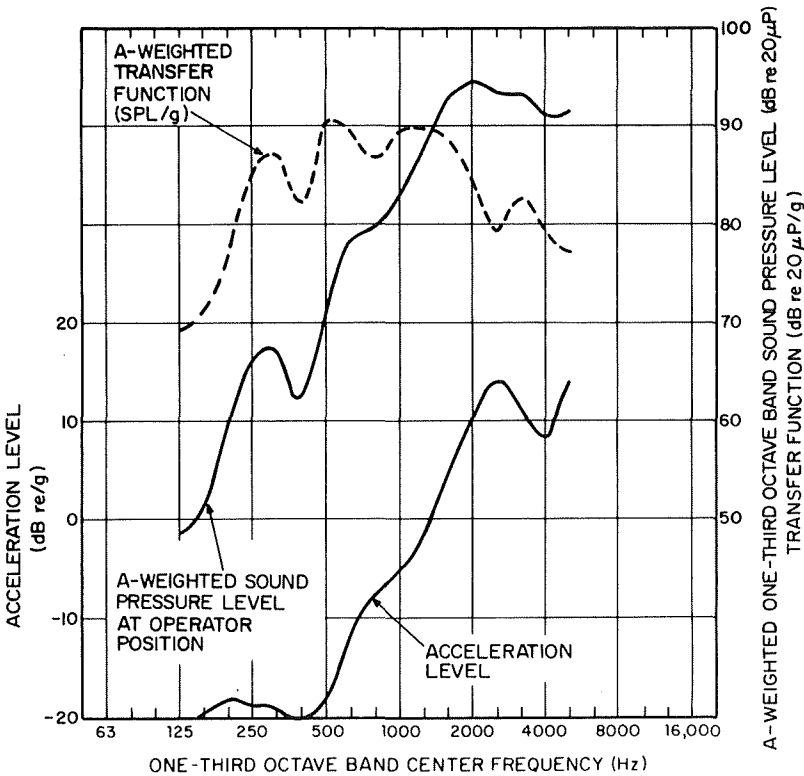


FIG. 21. TRANSFER FUNCTION FOR TRANSMISSION.

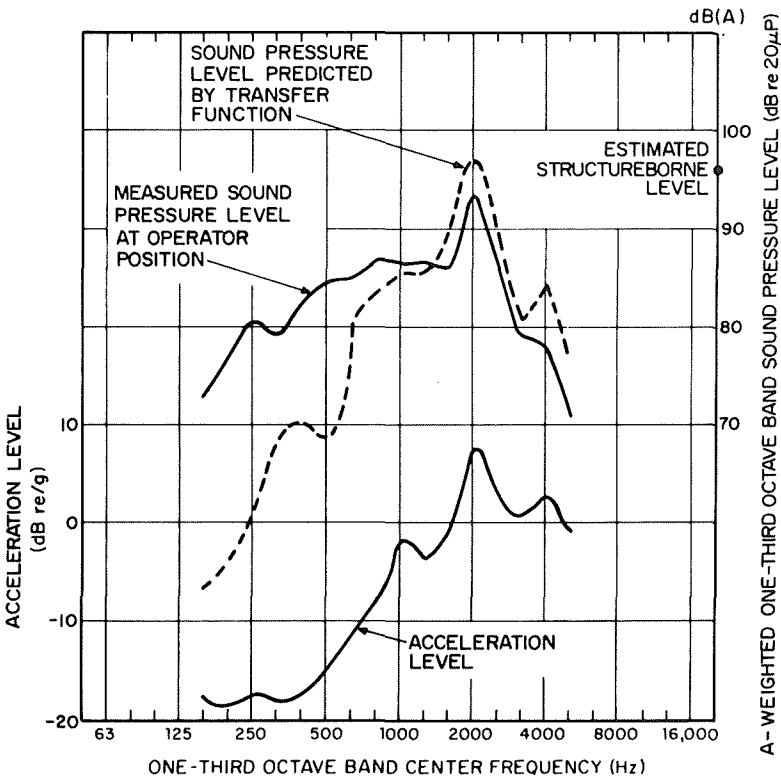


FIG. 22. STRUCTUREBORNE COMPONENT OF TRANSMISSION NOISE.

TABLE 2. SUMMARY OF NOISE SOURCES.

Source	Sound Pressure Level at Operator Position dB(A)	1/3-Octave Bands Above 85 dB(A)	1/3-Octave Bands Above 80 dB(A)
Transmission	101	400/4000	200/6300
Airborne	99	400/800, 2500/3000	200/400
Structureborne	96	800/2500	630/5000
Engine	97	630/6300	400/8000
Airborne	96	630/4000	400/630, 5000
Structureborne	88		400/1600
Cooling Fan	94 ¹	800	315/2500
Engine Exhaust	93	250	160, 250, 400/630
Engine Air Intake	90	1250	400/800, 1250

¹Obtained with cooling fan adjacent to closed door, thus giving a direct reflection to the operator. Level reduced to 92.5 when door was opened.

4. NOISE CONTROL DESIGN

In Sec. 3, we discussed the individual noise sources in detail. In this section, we present an estimate of the degree of quieting required for each source, and discuss treatments that will achieve this goal. The individual source levels presented in Table 2 were obtained in a shop area with the doors closed; to make them useful, we must develop a scheme for translating these values into equivalent values when the machine is finally placed in service underground. Our estimate will be made based on comparisons of data taken in the shop, data taken outdoors, and acoustic data taken in the mine.

4.1 Estimate of Required Noise Reduction

Figures 5 and 6 supply an estimate of the difference between sound levels measured in the shop during the diagnostic testing program and sound levels that would be measured outdoors under the same conditions - approximately 5 dB.

An estimate of the increase in sound level from outdoors to underground may be obtained using the procedure of Ref. 1. This procedure requires data on the average absorption coefficient of the space underground, typical dimensions of the space, and estimates of the source/receiver distances.

Measurements performed in the National Gypsum Mine indicate that the space is acoustically hard with an average Sabine absorption coefficient $\alpha \approx 0.2$. Typical dimensions of the room are 11 ft high by 25 ft wide. Typical source/operator distances are 3 ft for the transmission and 11 ft for the fan. The ratio of source/operator distance to room width is therefore in the range of 0.12 to 0.44, and the ratio of source/operator distance to room height is 0.27 to 1. Using Fig. 3.10 from Ref. 1, gives the correction factors shown in Table 3:

TABLE 3. CORRECTION FACTORS ABOVEGROUND TO UNDERGROUND MEASUREMENTS.

<u>Source/Operator Distance</u> <u>Room Dimension</u>	Tunnel dB(A)	Room dB(A)
0.1	1	
0.27		1
0.44	4	
1		4

A conservative estimate is to use the maximum of 4 dB increase underground. This increase compares very well with the results presented in Fig. 4. Sources will therefore be decreased by 5 dB to account for the difference between indoors and outdoors, and then incremented by 4 dB to account for the difference between underground and aboveground measurements. The net result is a decrease of 1 dB.

The adjusted source levels and the required reduction to bring the total to 90 dB(A)*are given in Table 4. The noise reduction required to bring all of the sources to a level less than 90 dB(A) is substantial and, in fact, some reductions may not be achievable within a reasonable cost. However, we note that these levels were obtained from test conditions which were designed to maximize the source contribution. In addition, they do not occur simultaneously. Consequently, the noise reductions presented are extremely conservative and, in fact, may not be needed to quiet the machine to an operational level which satisfies the allowable noise exposure as specified by the Health and Safety Act for a typical operating scenario for the machine.

For these reasons, we removed two of the sources from further consideration. First, the air intake was deleted because its

*Assuming 3 sources as discussed in the next paragraph.

TABLE 4. ESTIMATED NOISE REDUCTION TO REACH 90 dB(A) IN UNDERGROUND OPERATION.

Source	Estimated Maximum Level Underground dB(A)	Level Required for Sum to be 90 dB(A) dB(A)	Required Noise Reduction dB(A)
Transmission	100	85	15
Airborne	98		
Structureborne	95		
Engine	96	85	11
Airborne	95		
Structureborne	87		
Fan	93	85	8

level was so close to the noise floor that no reliable measurement could be made. Second, the exhaust was eliminated because (1) its principal contribution was in the low-frequency range and (2) it would require the design of a new muffler to satisfy the space constraints with minimum backpressure. Off-the-shelf mufflers did not meet these constraints.

4.2 Individual Source Treatments

Treatments for the transmission, engine, and fan sources are discussed in the following paragraphs. A summary table follows the discussion.

4.2.1 Airborne sound

Transmission Enclosure

The treatment of the transmission airborne source involved sealing holes in the operator's compartment and installing covers around the transmission compartment. The principal hole which

required sealing was around the steering column (see Fig. 9b). Design of a seal for this area must allow for the fact that the steering column can assume the three positions indicated in Fig. 9b. This seal consisted of a shroud with an elongated hole through which the column passes, thus giving the required adjustment. It is bolted through a gasket to the existing structure. Figure 23 shows the transmission compartment cover in the raised position, allowing access to the transmission. The interior surface was lined with 1-in.-thick, 2-lb/ft³ polyurethane foam with a 0.0005-in.-thick aluminized Mylar facing on it. This was covered with a 22-gauge (0.031 in.) perforated steel sheet with 51% open area. This sandwich was held together with studs welded

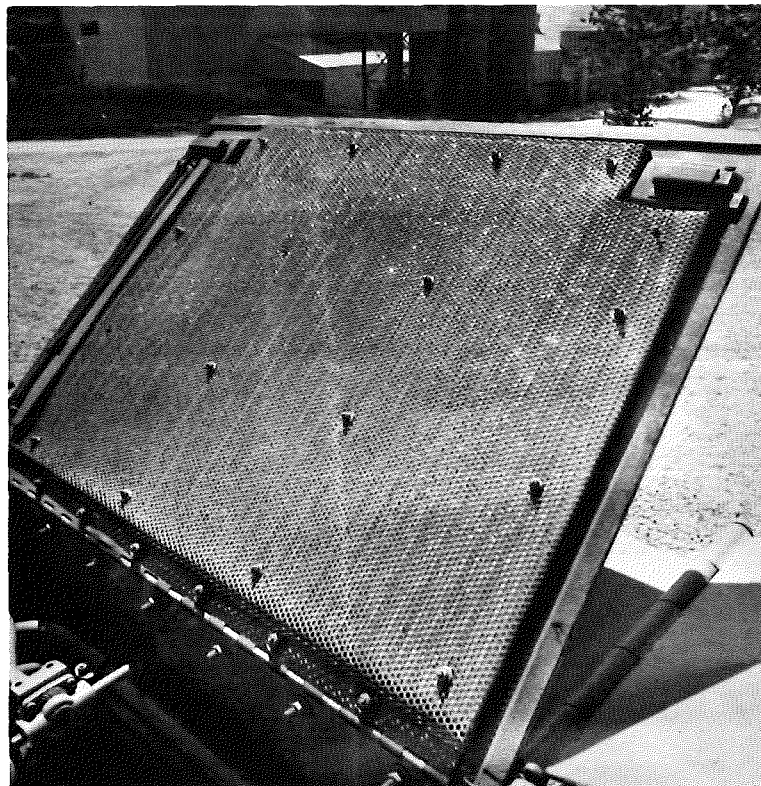


FIG. 23. TRANSMISSION COMPARTMENT COVER.

to the cover plate and capped with washers and nuts. The transmission hood was vibration-isolated from the main structure of the LHD at the hinge by using a rubber washer arrangement, shown in Fig. 24. In addition, a 1/4-in.-thick by 1-in.-wide rubber

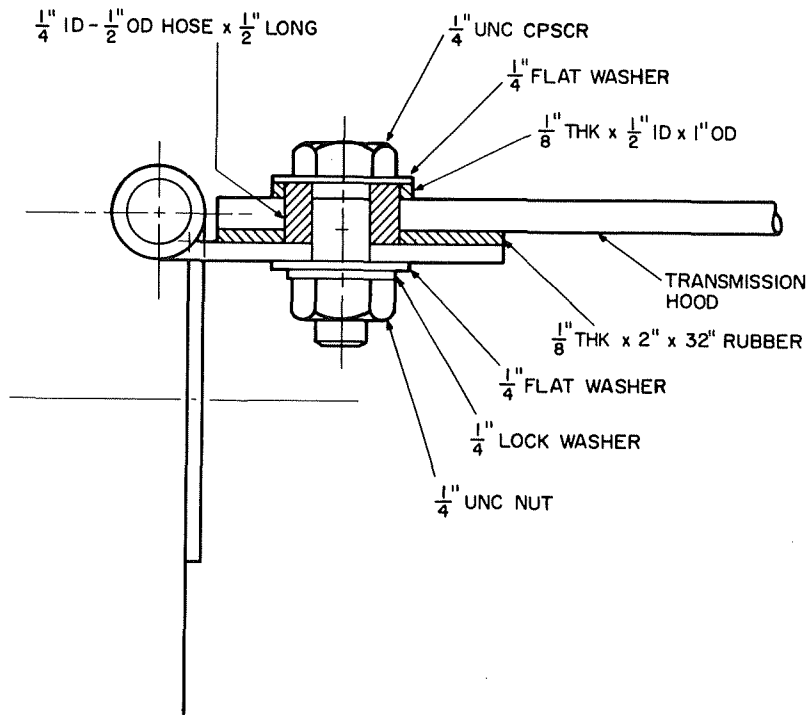


FIG. 24. TRANSMISSION HOOD VIBRATION ISOLATION.

strip was glued to the other three sides of the cover to attenuate structureborne vibrations which excite the hood and thus radiate sound very close to the operator. The bottom of the transmission compartment was sealed with a belly pan made of 10-gauge steel (0.139 in.). A drain hole was provided in the pan for changing the transmission fluid.

Figure 25 shows a cover plate which seals holes from the transmission compartment in the center where the machine is



FIG. 25. TRANSMISSION FRONT COVER.

pivoted. Noise coming through this area is reflected off the forward structure of the machine to the operator. This cover, which was made of 10-gauge steel, had 1/2 in. of polyurethane foam with 1/8-in. leaded rubber bonded to it to provide acoustical absorption and mass addition. The acoustical treatment weighed 1.7 lb/ft². A final path for transmission noise included radiation from the belly pan, ground reflection, and passage through the grate at the operator's feet to the operator. This path was eliminated by welding a plate underneath the grate.

Water and Fuel Tank Cover

In the final stages of the noise control effort, we noted that the existing fuel and water tank cover was radiating a

substantial amount of sound to the operator because of its high vibration level. Consequently, an additional cover was placed over the existing cover. This additional cover was vibration-isolated from the original with a scheme similar to that of Fig. 24. The cover, which can be seen in Fig. 26, was made of 10-gauge steel and had a 1.7 lb/ft^2 mass addition and absorption treatment applied to it, similar to the transmission front cover.

Water and Fuel Tank Compartment Treatments

The interior of the fuel and water tank compartment was treated with acoustical absorption material, as shown in Fig. 27. Approximately 14 sq ft of the 1-in. Mylar-faced polyurethane foam was stud-welded to the front, back, and side surfaces. Since

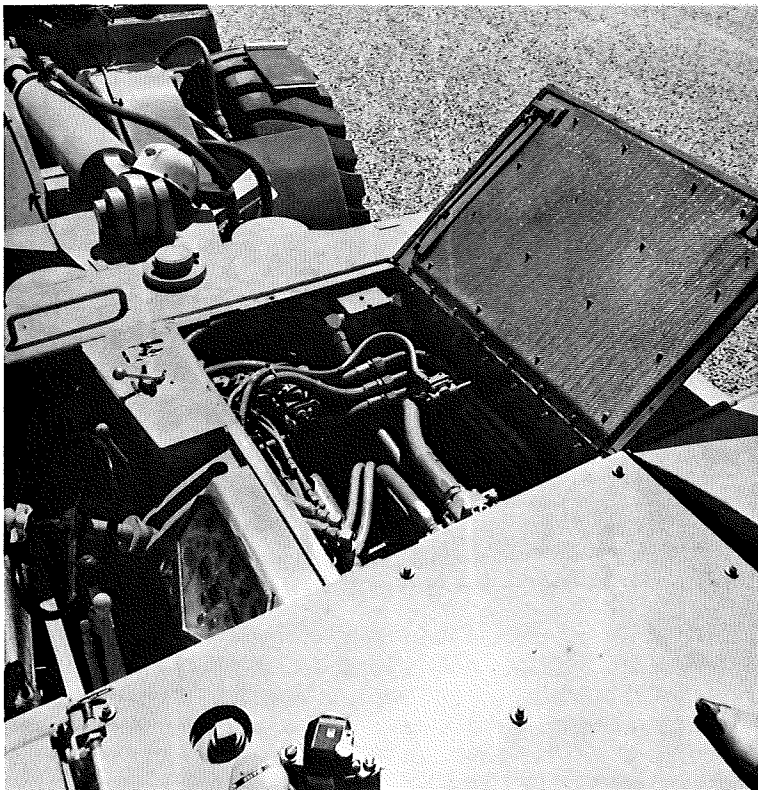


FIG. 26. WATER AND FUEL TANK COVER.

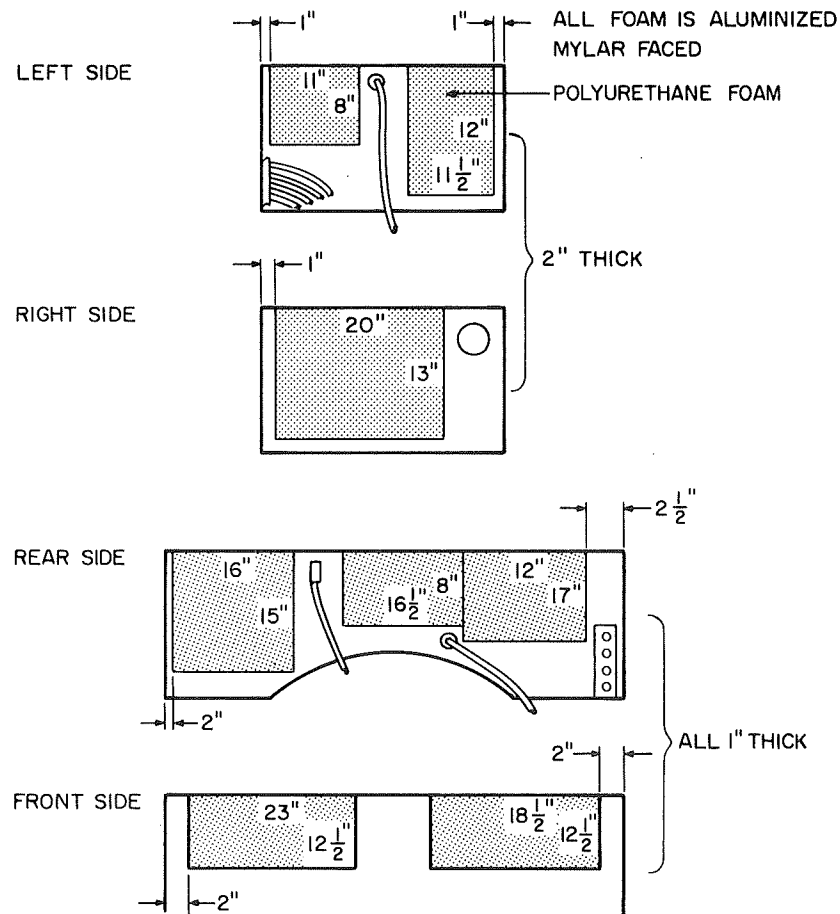


FIG. 27. ACOUSTICAL TREATMENT FOR INTERIOR OF WATER AND FUEL TANK COMPARTMENT.

there is limited access to this area, the acoustical material was not protected by perforated plate.

Torque Converter Cover

Another significant source is radiation of sound from the area of the torque converter. Consequently, a cover, shown in Fig. 28, was designed and fabricated from 10-gauge steel. The 1-in.-thick Mylar-faced polyurethane foam was attached to it with the stud welding technique. Large buttons (Fig. 29) were placed to cover the studs and to hold the acoustical material in place.

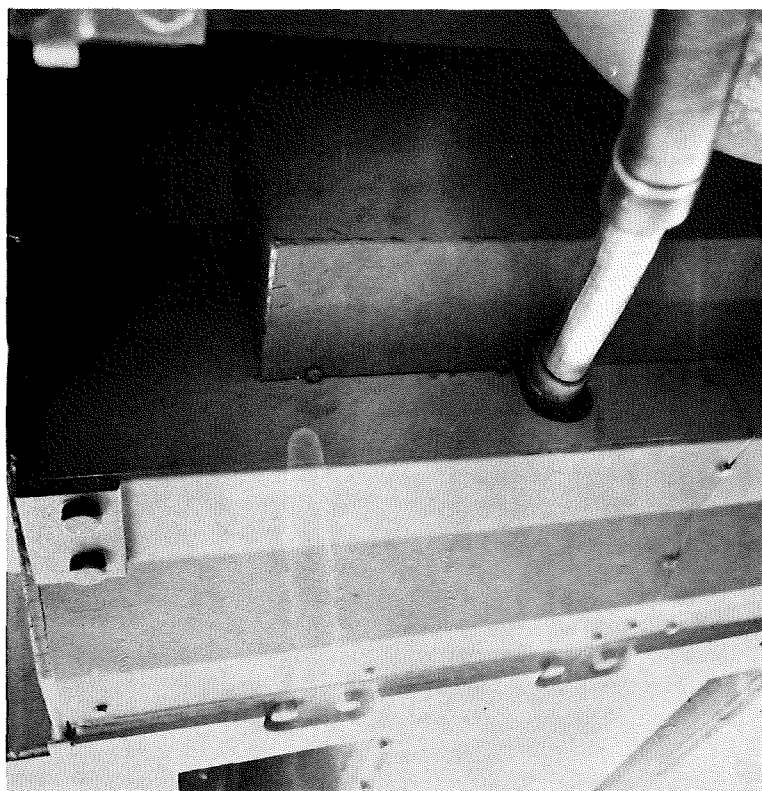


FIG. 28. TORQUE CONVERTER COVER.

Torque Converter Compartment Treatments

The interior of the torque converter compartment was treated with acoustical absorption material as shown in Fig. 30. Approximately 22 sq ft of the Mylar-faced polyurethane foam was stud-welded to the front, back, side, and top surfaces. The perforated protective plate was not used, again because of limited access.

Engine Enclosure

A major noise treatment for the machine involved the development of an enclosure for the diesel engine. The complete enclosure, shown in Fig. 31, consisted of a hood, side panels, and a

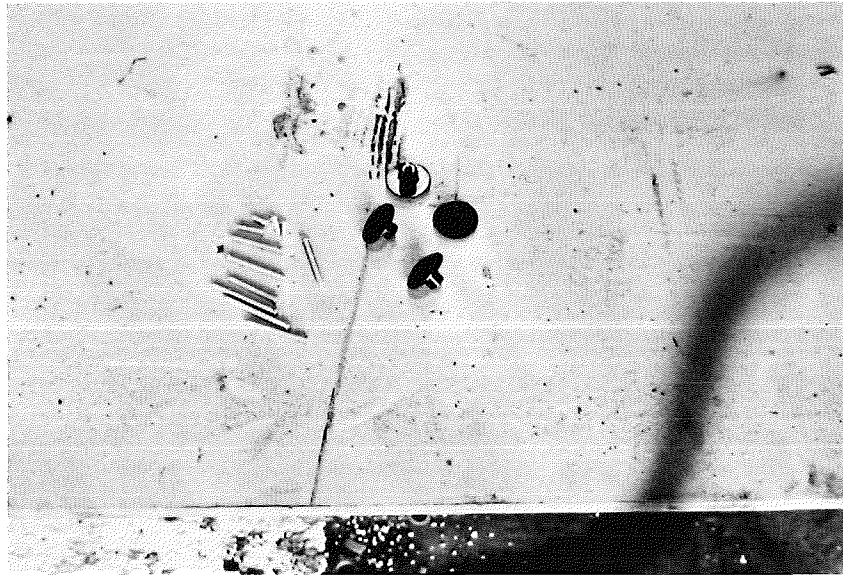


FIG. 29. STUDS AND BUTTONS FOR STUD-WELDING ATTACHMENT TECHNIQUE.

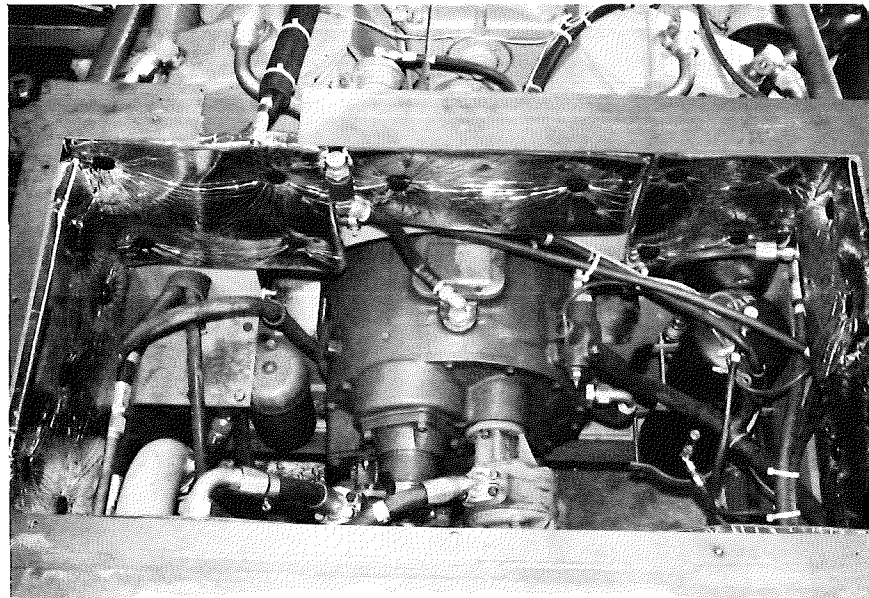
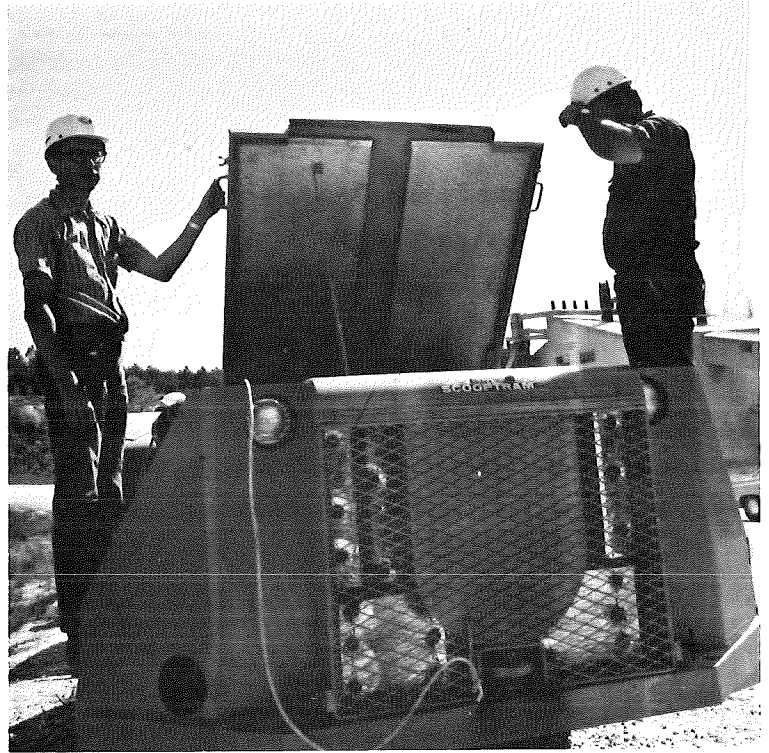


FIG. 30. ABSORPTIVE MATERIALS APPLIED TO THE INTERIOR OF THE TORQUE CONVERTER COMPARTMENT.



FIG. 31. ENGINE ENCLOSURE.

belly pan with ducts for cooling air exhaust. The components are shown in Fig. 32. The major problem with enclosing the air-cooled engine was to ensure that sufficient cooling was provided for the engine and that the temperature did not build up in the compartment to a point where other components are damaged. This problem was solved with a two-step procedure: First, an enclosure and lined exhaust duct were designed; second, they were simulated in a test wherein temperature and pressure differential tests were conducted on the machine during operation. These tests were conducted in conjunction with the engine manufacturer, who, in addition, performed his own set of cooling tests. The results of the cooling tests, which indicate that there is

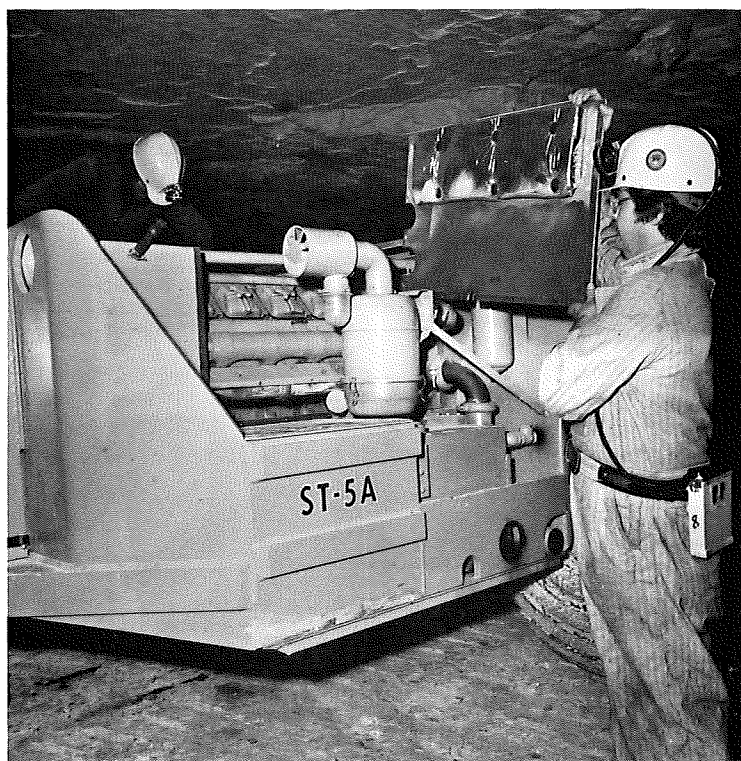


(a) Hood

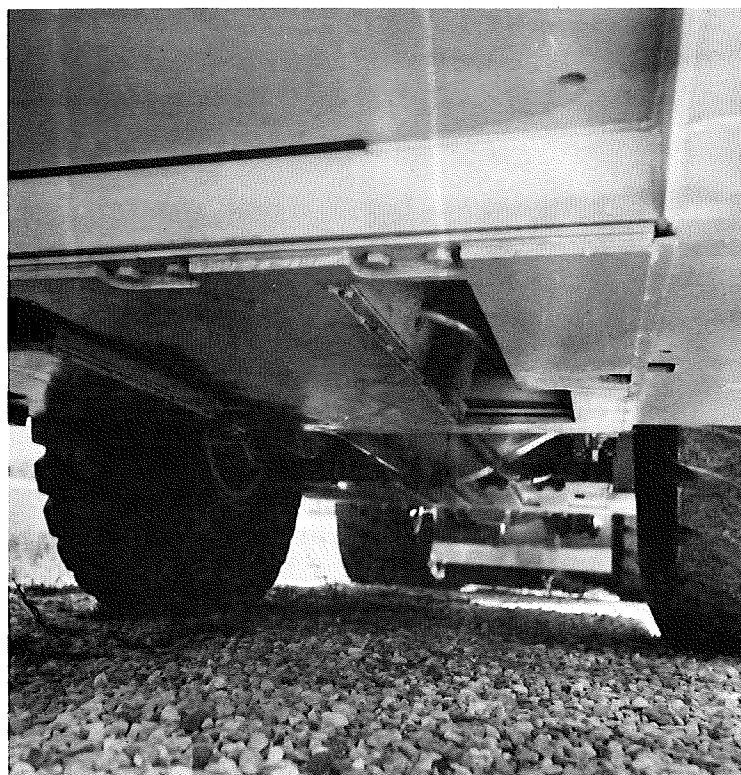


(b) Left Side Enclosure

FIG. 32. COMPONENTS OF ENGINE ENCLOSURE.



(c) Right Side Enclosure



(d) Belly Pan and Cooling
Air Exhaust Ducts

FIG. 32. COMPONENTS OF ENGINE ENCLOSURE.

sufficient cooling capacity, are presented in Appendix A. The engine manufacturer's cooling tests also indicated that the enclosure allowed sufficient cooling air to pass over the engine.

The hood of the engine enclosure, shown in Fig. 32a, is very similar to the transmission cover in that it is made of 10-gauge steel, is hinged to the main structure, and has rubber gasketing material applied to the edges to vibration-isolate it from the main structure. It also has the 1-in.-thick polyurethane foam with 0.0005-in.-aluminized Mylar facing protected by perforated metal.

The left and right side enclosures shown in Figs. 32b and 32c are again 10-gauge steel, but they have 2-in.-thick polyurethane foam with Mylar facing applied and held on with studs and buttons similar to those used in the fuel and water components. Note, in Figs. 31, 32b, and 32c, that there is a removable section of the side enclosures which is shorter than the full length of the hood and that there is a fixed section at each end of the enclosure. As we will discuss later, the fixed sections somewhat restrict access to the engine. The removable section is attached to the fixed sections by a number of hexhead bolts.

The belly pan and its cooling air exhaust ducts are shown in Fig. 32d. The pan is made of 5/8-in.-thick steel with two cutouts for the cooling air to exhaust. The cut-outs provide a total exhaust area of 4.7 ft^2 , which exceeds the manufacturers' minimum requirement of 3.2 ft^2 .* The configuration of the lined ducts inside the engine enclosure is sketched in Fig. 33. The ducts are fabricated as separate pieces which are bolted to the belly pan. The ducts are lined with the 1-in.-thick 2-lb/ft³

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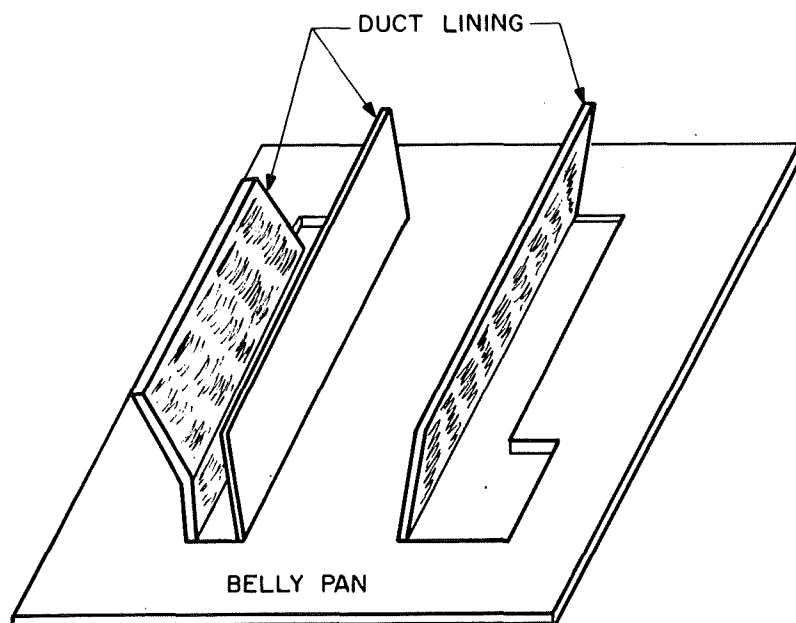
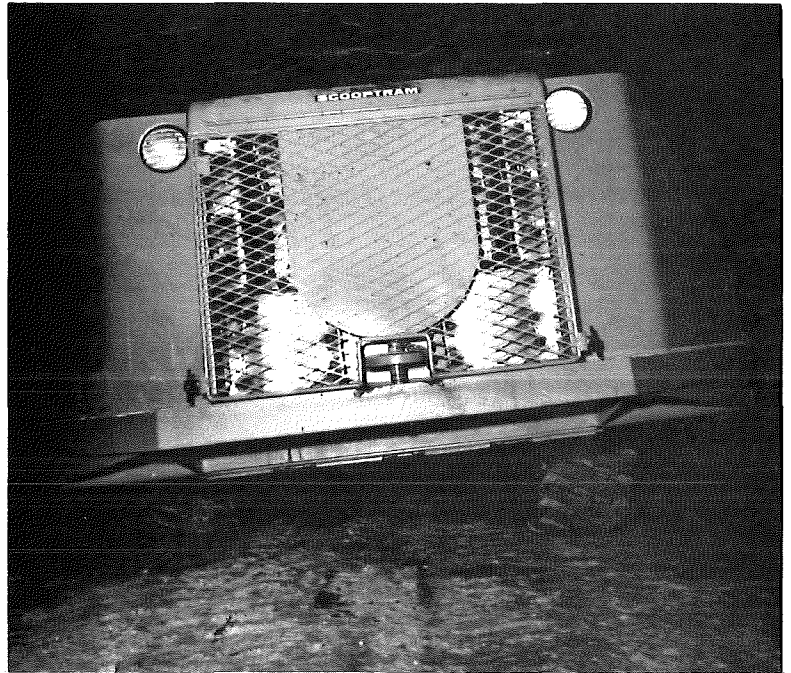


FIG. 33. BELLY PAN WITH COOLING FAN EXHAUST DUCT LINING.

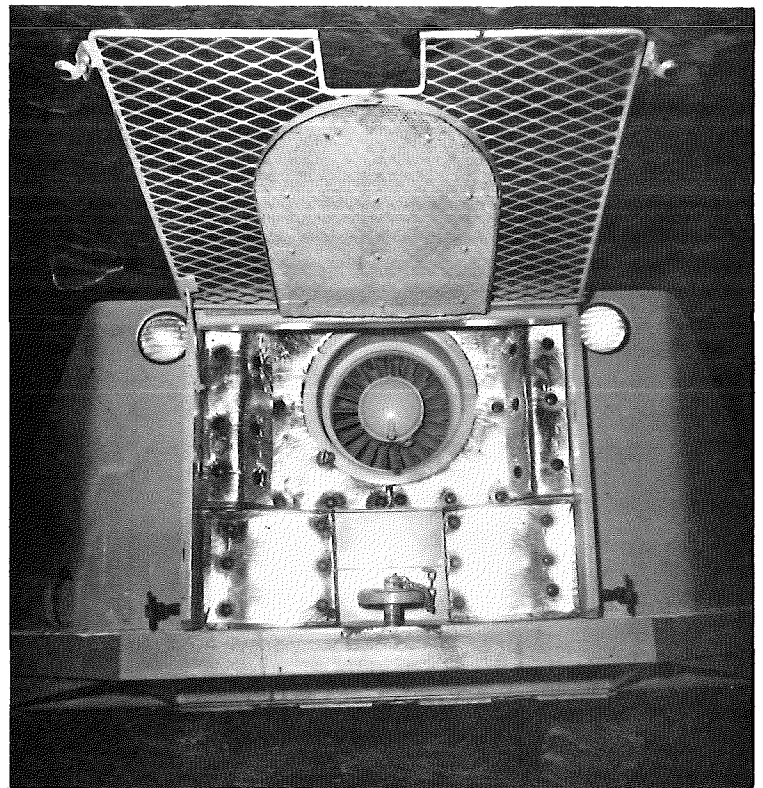
polyurethane foam with 0.005-in. aluminized Mylar protected by the 22-gauge perforated steel sheet. Note that one of the exhaust cutouts does not have a lined duct on one side. This was eliminated because of space restrictions.

Cooling Fan

The treatments applied to the area of the cooling fan are shown in Fig. 34. Figure 34a shows the baffle and grille lowered in its normal position for operating; Fig. 34b shows the baffle and grille raised to illustrate the acoustical treatments. As can be seen in Fig. 34b, the entire area around the fan was sealed. The absorptive material was 1-in. polyurethane foam with 0.005-in. aluminized Mylar covering, stud-welded to the structure. The perforated cover sheet was not used since this area has limited access. The fan grille was equipped with a baffle, as



(a) Baffle in Lowered Position



(b) Baffle in raised Position

FIG. 34. TREATMENTS APPLIED TO COOLING FAN AREA.

shown in Fig. 34. The baffle was covered with the 1-in. polyurethane foam and covered with the 22-gauge perforated steel plate.

4.4.2 Structureborne sound

Measurement locations for evaluating the vibration isolation and resultant reduction in structureborne sound for both the engine and transmission are shown in Fig. 35. The figure also presents a designation for the locations to be referred to in subsequent figures.

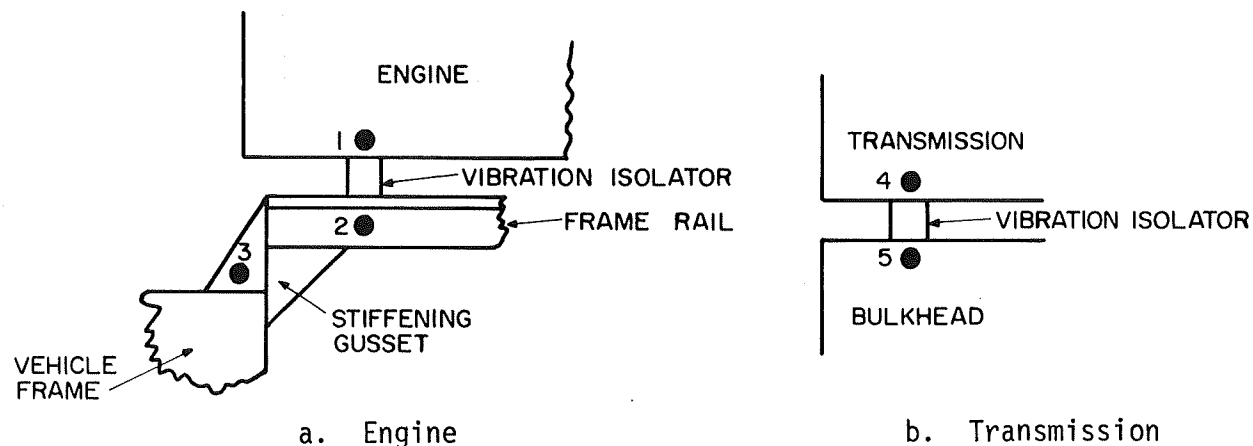


FIG. 35. MEASUREMENT LOCATIONS FOR EVALUATION OF VIBRATION ISOLATION.

Engine

As we pointed out in the source diagnosis section, there is reason to suspect that structureborne noise may be unacceptably high in the frequency range between 400 and 1600 Hz. To evaluate this premise, we applied the chipping hammer excitation to the engine mounts and measured the acceleration on both the engine and frame rail side of the mount. This gave the transmissability in octave bands (i.e., acceleration level on engine side of the mount minus acceleration level on the frame rail side) presented

in Fig. 36 for all four corners of the machine. Note that the front engine mounts are ineffective in reducing vibration below 500 Hz. In fact, they actually increase the level of vibration. Additional measurements on the vehicle frame, shown in Fig. 37, further illustrate the problem. This situation is typical of the case where the mounting structure is not sufficiently stiff. The noise control treatment was to stiffen the frame rail by welding 6-in. $\times$ 6-in. $\times$ 1/2-in. thick triangular gussets to the ends of the frame rails, as shown in Fig. 38. A comparison of the

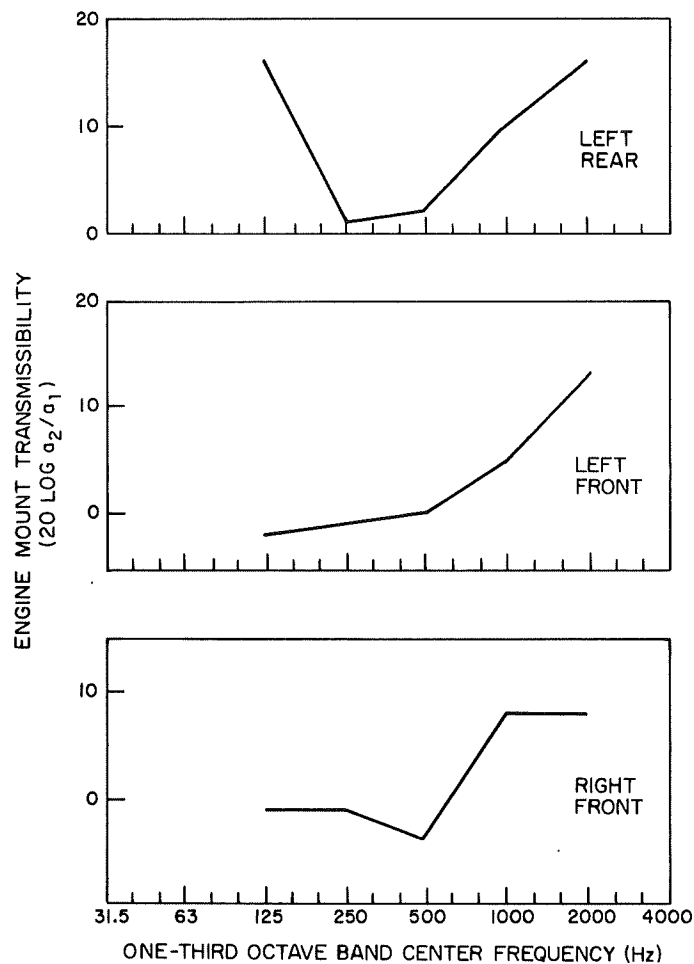


FIG. 36. TRANSMISSABILITY OF ENGINE MOUNTS.

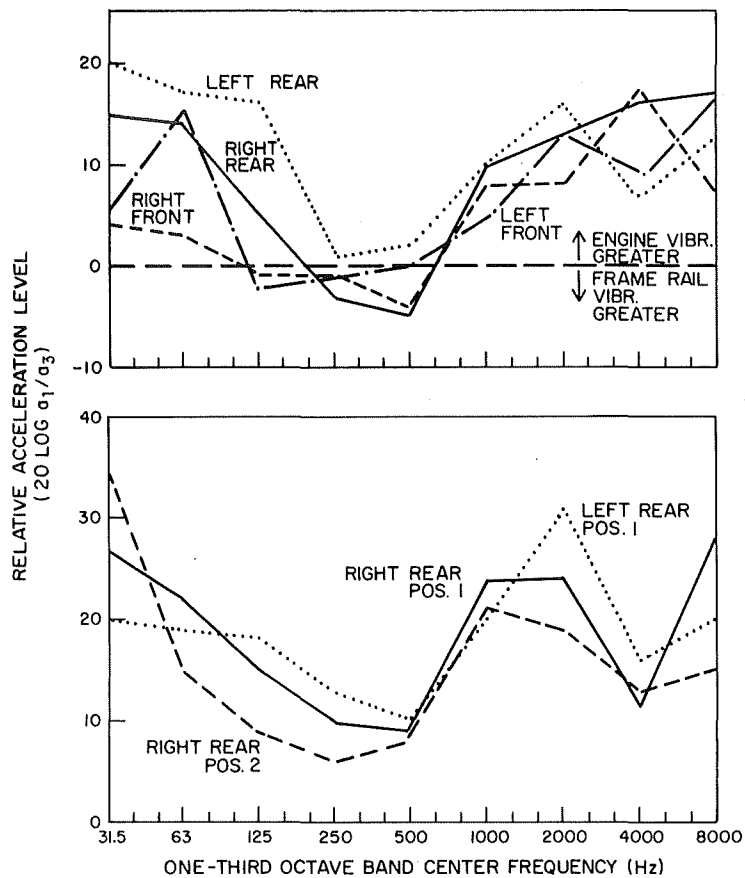


FIG. 37. ENGINE MOUNT TRANSFER FUNCTIONS.

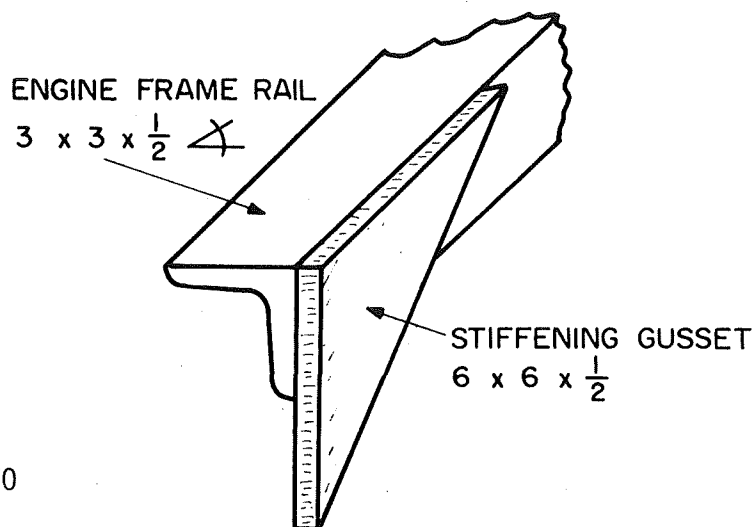


FIG. 38. STIFFENING APPLIED TO ENGINE FRAME RAIL.

transmissability across the mounts, following the increase in stiffness, with the unstiffened case is presented in Fig. 39. The stiffening resulted in an increased attenuation of at least 5 dB at all frequencies. This increase should translate into a decrease of at least 5 dB(A) in the structureborne sound, and, therefore, should reduce the structureborne sound to a level of 83 dB(A), which should be acceptable.

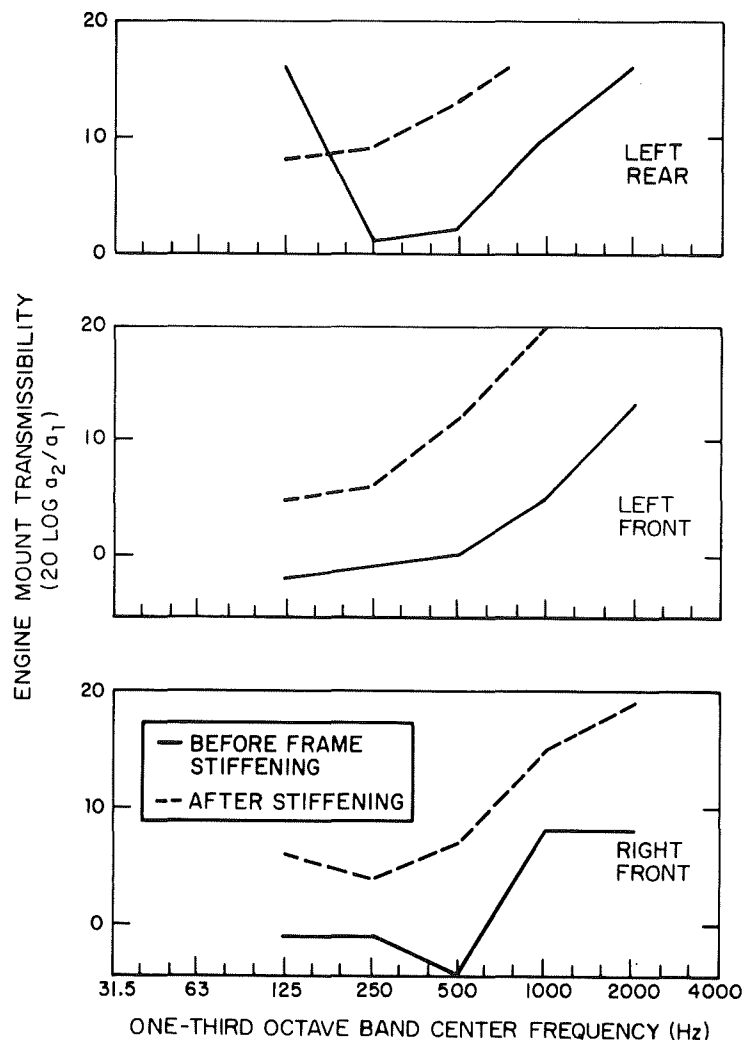


FIG. 39. COMPARISON OF ENGINE MOUNT TRANSMISSABILITY BEFORE AND AFTER STIFFENING.

Transmission

Structureborne noise from the transmission was estimated to be important in the range from 630 Hz to 5000 Hz. Since the transmission was hard-mounted to the main structure of the machine, the obvious noise control treatment was to use vibration isolation mounts between the transmission and the structure. Shock loading and maximum torque are the governing design conditions for the mounts. Shock loading occurs during violent maneuvers on shifting. Maximum torque occurs in low gear at maximum engine torque.

The transmission weighs 1100 lb, and the maximum torque is 375 ft/lb, as specified by the manufacturer. The mounting points are 10 in. away from the center of the transmission; therefore, the maximum torque load on a mount is

$$\frac{1100 + 375 (12/10)}{4} = 388 \text{ lb.}$$

The shock loading is more difficult to estimate. However, if we assume a shock loading factor of 3, the maximum shock level is

$$\frac{1100 \times 3}{4} = 825 \text{ lb.}$$

These estimates were used to select the Huntington M700 vibration mount, which has a nominal load capacity of 900 lb. The mounts were installed on the transmission. Figure 40 shows a top view of the mounting structure. The transmissability obtained by taking the difference between accelerations above and below the mount during the chipping hammer test, for the vertical, longitudinal,* and transverse directions is shown in Fig. 41.

*The longitudinal direction is along the axis of the machine.

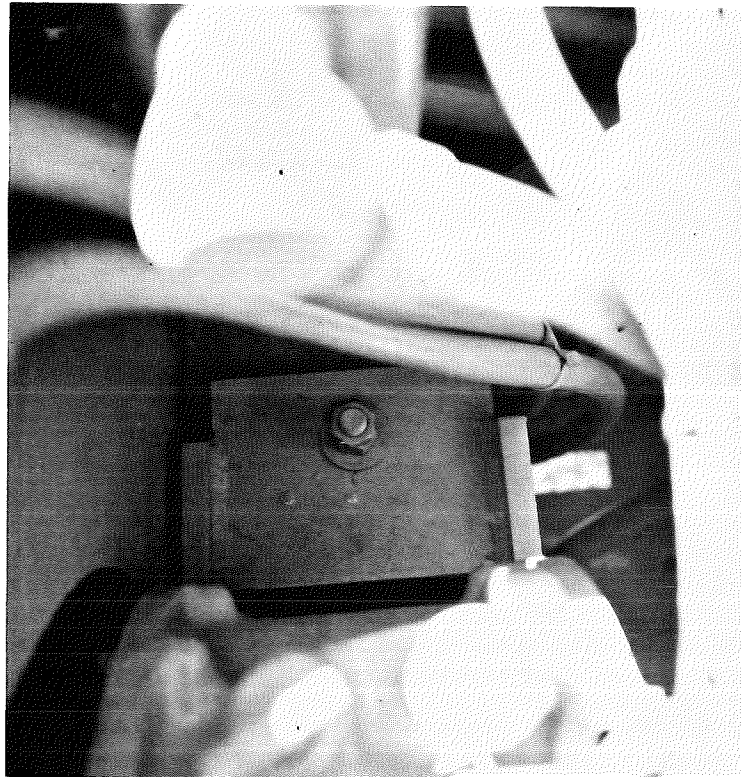


FIG. 40. TRANSMISSION VIBRATION ISOLATOR MOUNTING STRUCTURE.

Applying these reductions to the structureborne sound estimate of Fig. 22 gives a reduction of 11 dB(A). Therefore, the final value of structureborne sound from the transmission is estimated to be 85 dB(A).

In addition to these treatments, a number of components had to be modified or relocated. Examples of the relocating of the components are: (1) air cleaner brackets were altered and moved outboard, (2) air precleaners were attached directly to the cleaners, (3) combustion air intake pipe was moved to the other end of intake manifold, (4) the headlights were moved to the previous air intake holes, (5) a new engine hood was constructed, (6) several electrical components and the engine speed governor

were moved from cooling air intake grille to outside engine enclosures, (7) air compressor discharge tubing was modified and moved, (8) the oil filler tube and dipstick were moved to make them more accessible, (9) three bulkhead fittings were installed for several hoses to allow a good seal, (10) several compressed-air hoses were moved or rerouted, (11) remote handles were put on exhaust scrubber drains to allow scrubbers to be drained (daily) without opening engine enclosure, (12) the primary air tank was moved to allow better air flow, (13) the left exhaust pipe was moved to allow better air flow, and (14) a large number of small holes were plugged to prevent sound from escaping the compartment.

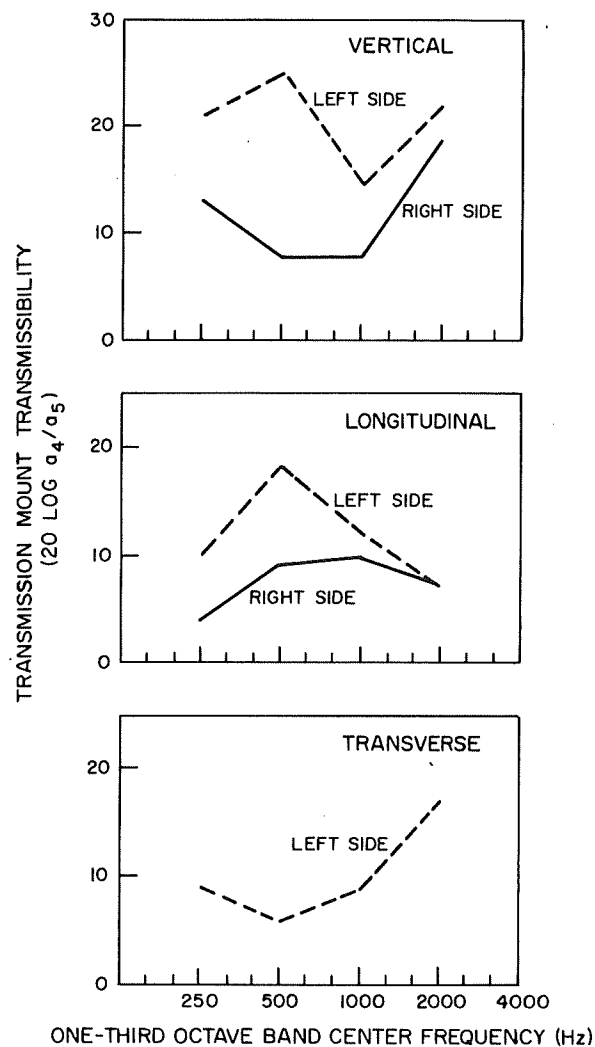


FIG. 41. TRANSMISSABILITY OF TRANSMISSION MOUNTS.

4.3 Summary of Noise Control Treatments

A summary of the noise control treatments is presented in Tables 4, 5, and 6. Treatments which include absorptive materials are presented in Table 4; Table 5 presents sealing and miscellaneous treatments; and Table 6 presents vibration isolation

TABLE 5. ABSORPTIVE MATERIALS USED IN NOISE CONTROL OF WAGNER ST-5A.

Location	Material Type	Thick-ness (In.)	Area Ft ²	Film Covering				Protective Covering		\$ Cost Labor and Materials
				Type	Thickness	No. Layers	Spacing	Type	Percent Open Area	
Engine hood	*	1	13.6	*	.0005 in.	1	None	***	51	242
Engine side enclosure (operator's side)	*	2	6.8	*	.0005 in.	1	None	None	NA	151
Engine side enclosure (other side)	*	2	7.4	*	.0005 in.	1	None	None	NA	151
Cooling air exhaust duct	*	1	9.4	*	.0005 in.	1	None	***	51	583
Transmission hood	*	1	6.4	*	.0005 in.	1	None	***	51	73
Front of transmis-sion cover	**	5/8	5.6	None	NA	NA	NA	None	NA	130
Cooling fan area	*	1	11.2	*	.0005 in.	1	None	***	51	473
Converter compart-ment and cover	*	1&2	22.4	*	.0005 in.	1	None	None	NA	398
Water and fuel tank cover	**	5/8	18.6	None	NA	NA	NA	None	NA	202

*Polyurethane foam — density 2 lbs/ft³ — with .0005-in.-thick aluminized mylar facing — 1-in.-thick material. NRC .62.

**Mass loaded plastic with 1/2 in. polyurethane foam decoupler — 5/8 in. thick. Total 1.7 lb/ft².

***Perforated sheet steel — 22 gal, 3/16-in. diam. holes on 1/4 in. staggered center — 51% open.

TABLE 6. SEALING AND MISCELLANEOUS TREATMENTS USED IN NOISE CONTROL OF WAGNER ST-5A.

Location	\$ Cost
Steering column cover	43
Motor mount gusset	1
Light mounting bracket	2
Air intake heat shield	24
Transmission belly pan	92
Miscellaneous installation & materials required*	1294

*Plugging on miscellaneous holes in chassis, operator's compartment, engine compartment, plumbing changes, relocation of filters, air cleaners, air tanks, etc.

TABLE 7. VIBRATION ISOLATORS USED IN NOISE CONTROL OF WAGNER ST-5A.

Location	Number	Supplier	Catalog No.	\$ Cost
Transmission isolators only	4	Huntington	M-700	
Transmission isolator mount	2 sets	Wagner		2988
Operator's compartment — transmission mount cover	1	Wagner		62

treatments. The final column represents the manufacturer's cost estimate for materials: \$6914 for noise control treatments applied to a new machine.

5. PERFORMANCE OF NOISE CONTROL TREATMENTS

The final configuration of the noise-controlled LHD is chosen in Fig. 42. The picture was taken underground in the National Gypsum Mine just prior to placement of the machine in service. A comparison of the spectrum of the noise measured at high idle outdoors at Wagner prior to the noise reduction with that measured outdoors at National Gypsum, following the noise reduction, is shown in Fig. 43. The overall noise reduction was 7 dB(A).

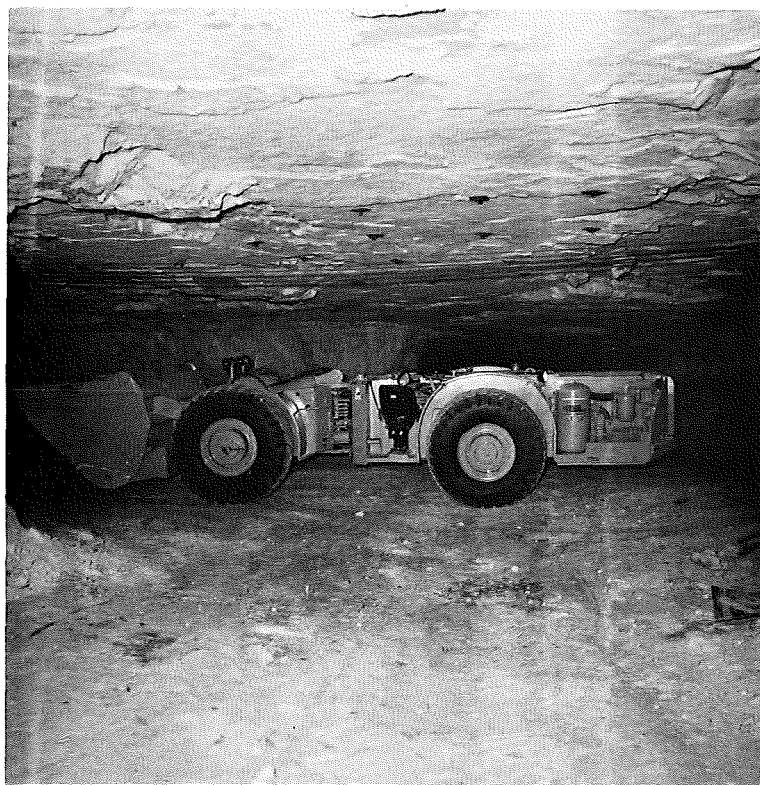


FIG. 42. MACHINE LOCATION FOR NOISE CONTROL TREATMENT PERFORMANCE EVALUATION UNDERGROUND AT NATIONAL GYPSUM MINE.

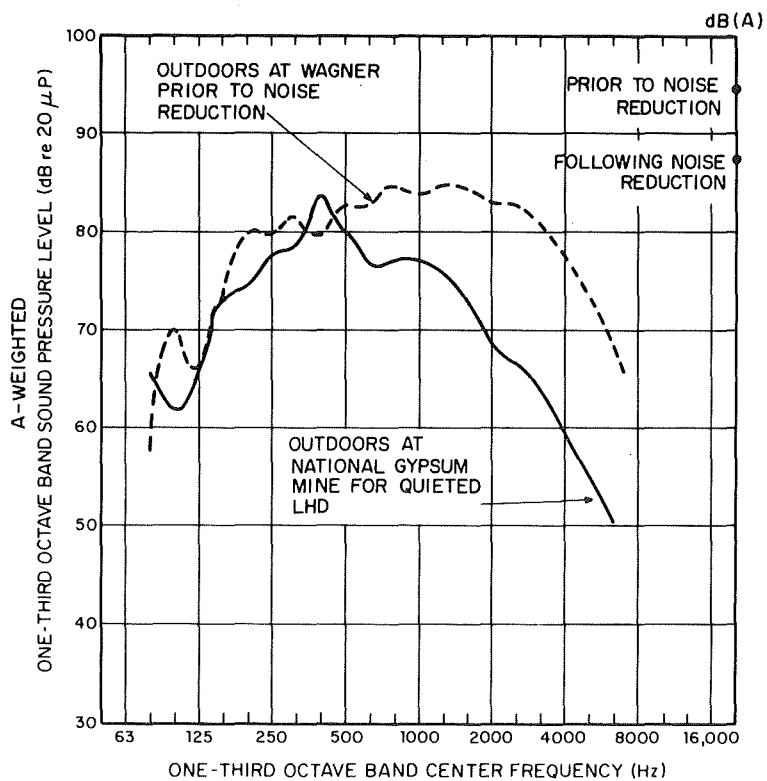


FIG. 43. COMPARISON OF ABOVEGROUND AND UNDERGROUND SOUND PRESSURE LEVELS MEASURED ON WAGNER ST-5A LHD.

5.1 Underground Evaluation

The machine was placed underground on 31 August 1976. In order to evaluate the treatments a number of performance tests were conducted underground wherein the individual treatments were removed and noise measurements were made at the operator station. Testing was conducted with the machine parked at an intersection, as shown in Fig. 42. The results of these tests are given in Table 7. The tests with the hood raised indicate that the engine enclosure is an extremely effective noise control treatment, giving 5 dB(A) of attenuation. The removal of the side covers while the hood was raised had negligible effect on the operator

TABLE 8. PERFORMANCE TESTING UNDERGROUND CONDUCTED AT NATIONAL GYPSUM.

Engine Mode	Test Condition	SPL at Operator's Ear dB(A)	Increase Over Baseline dB(A)
HI	With all treatments in place	92.5	-
TCS	With all treatments in place	93	-
HI	Engine hood up	97.5	5
TCS	Engine hood up	97.5	4.5
HI	Engine hood up and side covers off	97.5	5
TCS	Engine hood up and side covers off	98	5
HI	Engine hood down with side covers off	95	2.5
TCS	Engine hood down with side covers off	95.5	2.5
HI	Transmission cover up	94	1.5
TCS	Transmission cover up	95	2
HI	Fan grille up	92.5	0
TCS	Fan grille up	94.5	1.5

HI = High Idle

TCS = Torque Converter Stall

exposure, as expected, since the path through the hood is direct. This result is verified by the decrease in sound level at the operator position with the sides removed and the hood down. The tests with the transmission cover up are not truly representative

of the performance of the treatments in this area, because the transmission does not operate in either the high idle or torque converter stall mode. Since it was not possible, when the machine was in the mine, to raise it on jacks to repeat the diagnostic tests, the evaluation of the transmission treatments will have to be inferred from the dosimeter readings discussed later. The fan grille up tests indicate only the performance of the baffle and not the other treatments, since they were not removed underground. There is an indication, however, that the barrier does provide attenuation, at least in the torque converter stall mode.

A repeat set of acoustic measurements on the machine with all noise treatments installed was performed underground approximately 60 days after it was placed in service. The sound pressure level at the operator station was 92.5 in high idle and 93 in torque converter stall, indicating no degradation in performance of the acoustical treatments.

5.2 Operator Exposure Measurements

Exposure measurements using noise dosimeters were performed on the machine operator by MESA on two separate occasions. The first measurement, performed on December 3, 1976,* indicated an exposure index of 1.03, and the second, performed on January 31, 1977,* indicated an exposure index of 0.89.

5.3 In-the-Field Experience

Discussions with the management of the National Gypsum Mine in June 1977 have indicated that all of the noise control treatments are still intact and are performing as expected. One problem expressed by the mine mechanics at the 60-day visit is that

*Jack Gill, Supervisory Mining Engineer, MESA, Vincennes, IN.

the fixed portions of the side enclosures (see Sec. 4.2.1) would make the transmission oil coolers difficult to clean, possibly resulting in degradation in transmission or other component life. This problem has not as yet materialized.

6. CONCLUSIONS AND RECOMMENDATIONS

The LHD was identified as a principal contributor to the noise exposure of underground miners. The present program involved a noise source analysis of the LHD and the selection and implementation of noise control treatments which were designed to bring the LHD into compliance with the Health and Safety Act. The program's successes were two:

- It achieved a noise reduction which, under the operating scenario for the selected machine, was acceptable according to dosimeter measurements made by MESA;
- The noise reduction treatments are practical and are enduring in the mine environment.

Although this effort was successful in meeting the noise exposure goals for this particular mining environment, there may be cases where these treatments are not sufficient to bring into compliance a particular machine under a particular work scenario. For those cases, we have prepared the following table of noise sources with additional treatments which will further reduce the noise.

TABLE 9. ADDITIONAL NOISE CONTROL TREATMENTS FOR LHD's.

Source	Additional Treatments Possible
Transmission Structureborne	• Less conservative vibration mount design • Additional stiffening of mounting bulkhead
Engine Structureborne Airborne	• More stiffening of frame rails • More absorptive material on cooling air ducts
Cooling Fan	• Modify baffle to get smoother air inflow
Exhaust	• Manifold muffler
Air Intakes	• Intake silencer

The absorptive material used on the LHD was polyurethane foam. Although this material does meet the acoustic requirements it does emit toxic fumes (up to 10% of weight) when burned, see Ref. 2. It is therefore recommended that the polyurethane be replaced by a material less susceptible to this problem (e.g., fiberglass) in future applications.

REFERENCES

1. Patterson, W.N., Huggins, G.G., Galaitsis, A.G., "Noise of Diesel Powered Underground Mining Equipment: Impact Prediction and Control," Final Report on USBM Contract No. HO346046 (March 1975).
2. Wooley, W.D., Fardel, P.J., "Prediction of Combustion Products," *Fire Research Journal*, Vol. 1, pp. 11-21 (1977), The Netherlands.

APPENDIX A: RESULTS OF COOLING TESTS FOR ENCLOSED DIESEL ENGINE

A critical problem regarding the partial engine enclosure for the ST-5A is the potential overheating of components within the engine compartment. Of specific concern are the alternator and air compressor, which have had overheating problems in the past.

To quantify the problem, a test was conducted in which temperatures at a number of locations within the engine compartment were measured under two conditions:

1. With the machine in its normal configuration,
2. With an engine enclosure aerodynamically the same as envisioned for quieting purposes.

For the first condition, a sheet of plywood was placed against the bottom of the engine compartment to simulate the belly pan. For the second condition, the configuration contained the following elements:

1. Lead sheeting was used to cover the two exposed sides of the engine compartment (see Figs. 8 and 9).
2. The rear face of the engine compartment was sealed around the cooling air intake cowling, as shown in Fig. A1 (again from the earlier tests). This procedure not only prevented engine noise from escaping, but also minimized hot air recirculation back into the cooling fan.
3. The plywood sheet on the engine bottom was cut to allow about a 4-in. gap on either side, running the length of the engine compartment. This provides the minimum cross-sectioned area for discharge ducting recommended by Deutz for the F8L714. See footnote, Sec. 4.2.1.

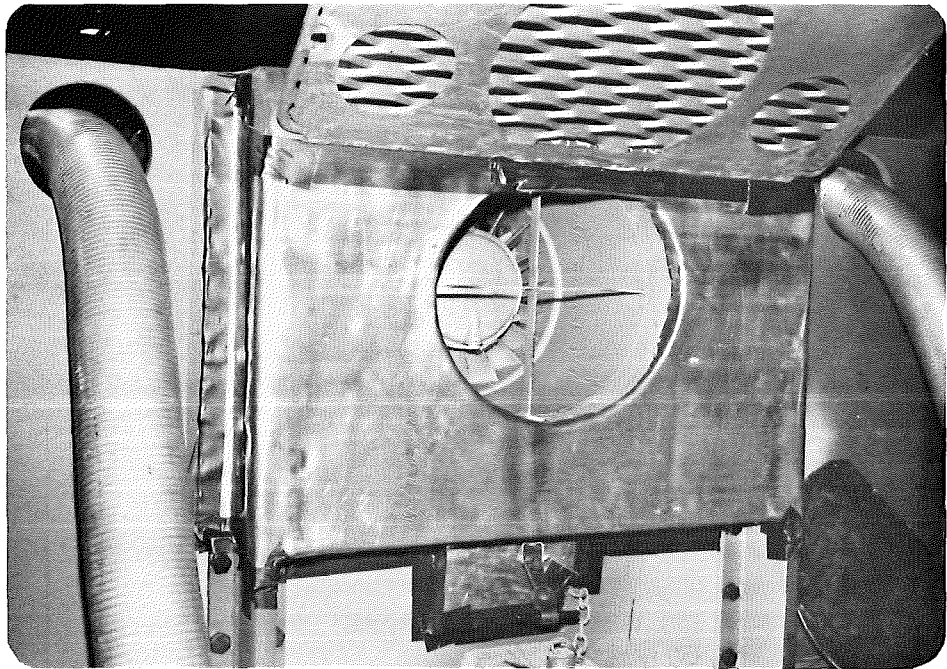


FIG. A1. SEALING AROUND COOLING FAN FOR TEMPERATURE TESTS.

4. A 2 to 3-in.-wide gap around the torque converter housing in the front fire wall was sealed.

A comprehensive set of temperature measurements was made, including the temperature of cooling and lubricating oils, engine compartment air, intake air, and skin temperatures of several components. Air pressure data were also taken in the air tunnel, exhaust, and combustion air intake.

The test procedure followed in each case was to allow the engine to warm up at high idle (until temperatures stabilized), then run at torque converter stall until the converter oil gauge temperature reached 220°F. With the engine under load, data were recorded. The machine was outdoors. Therefore, conditions could not be precisely controlled, since ambient temperature varied from in the 60's to the 80's during the tests.

In all tests, the machine was backed up with the rear of the machine (cooling fan end) about 2 to 3 ft away from the large door to the shop. In this configuration, some warm air exiting from the bottom of the engine compartment was trapped between the machine and the door and was recirculated back into the fan. The primary before-and-after comparisons have been made with this arrangement.

The test results contrasting the original and enclosed conditions (with the shop door closed) are given in Table A1. Item 10 shows the average intake temperature to be the same in each case, about 95°F. Note, however, that in the original condition, the temperature at the lower left-hand corner is at least 20° higher than at other positions. This increase was caused by some warm air escaping the engine compartment and entering the cooling fan at this corner. For the enclosed configuration, less recirculation occurred and a smaller temperature variation was measured around the inlet.

Item 4 shows that cylinder fin temperatures increased an average 6°. Note that the left bank of cylinders is actually slightly warmer in the before condition which can probably be traced to the high inlet temperature in the lower left-hand corner.

The results of air temperature measurements near the torque converter oil coolers (Item 13) are inconclusive. The data show that the left side increases a negligible amount and the right side increases some 30°. These results are not completely reliable, however, because the temperature probes were moved between the two tests and were in somewhat different positions.

Similar comments apply to Items 5 and 6, temperatures of engine tunnel cover and the oil pan. The probes were moved

TABLE A1. TEMPERATURE AT VARIOUS LOCATIONS ON DIESEL ENGINE WITH SIMULATED ENCLOSURE COMPARED TO ENGINE WITH NO ENCLOSURE.

Location	Original Configuration	Enclosed Configuration
1. Converter Oil		
a. In	207°F	223°F
b. Out	170	207
2. Engine Oil	173	144
3. Fuel Oil	80	88
4. Engine Fins		
a. Left front	145	138
b. Left rear	180 Avg	180 Avg
c. Right front	150 159	164 165
d. Right rear	162	178
5. Engine Tunnel Cover		
a. Left	64(?)	98
b. Right	70	112
6. Engine Oil Pan	155	187
7. Compressor Oil	312	340
8. Compressor Skin	-	233
9. Combustion Air Intake		
a. Left	95	93
b. Right	73	85
10. Cooling Air Intake		
a. Top	94 Avg	95 Avg
b. Operator side 120° from top	75 95	92 96
c. Exhaust side 120° from top	115 ±20	102 +6 -4
11. Alternator		
a. Top	107	-
b. Engine side	116	160
12. Hydraulic	145	165
13. Air Near Converter Oil Cooler		
a. Top left	210	215
b. Bottom left	211	215
c. Top right	158	196
d. Bottom right	172	195
14. Air Near Engine Oil Cooler	179	190
15. Engine Compartment		
a. Left front	162	115
b. Left rear	132	123
c. Right rear	147	130
d. Right rear	121	105

between tests, with the possibility of being relocated in different positions.

Converter oil and fuel oil temperatures each increased about 10%. Engine oil temperature inexplicably decreased from 173° to 144°.

The temperatures of two components of concern — the alternator and the air compressor — both increased. The alternator went from 116° to 160°, and the compressor exhaust air temperature increased from 312° to 340°.

We contacted Motorola on the alternator and Bendix and Deutz on the compressor. At a recommended maximum rpm of 5000 — 6000 for the alternator,* the desirable operating temperature quoted by Motorola[†] is 212°F. The lowest component failure temperature is 275°F, for the hypalon wire. With these values, a measured temperature of 160° would seem to be no problem.

We were unable to obtain recommended temperatures for the air compressor. A Bendix spokesman noted, however, that they recommend copper tubing on the discharge, with no bends near the compressor. The copper will withstand higher temperatures than the existing teflon tubing and also dissipate heat better. He suggested that cooling fins may be installed. It was felt that the temperature rise was not severe and, therefore, no modifications were made. In-the-field experience later proved this to be a correct assumption

Finally, note the differences in air temperature at the four upper corners of the engine compartment (Item 15) in Table A1.

*The alternator in its present configuration operates at about 4200 rpm at the rated engine speed of 2400 rpm.

[†]The Motorola Automotive Department in Chicago was consulted for Model 85C2001R, 12 volt, 88 amp.

The temperature dropped an average 15° when the engine was enclosed. This seems to be an indication that the air flow within the engine compartment is more controlled, being forced to wash down over the cylinder heads and out the two long slits at the bottom of the engine compartment.