N-96-01
II-A-208

EPA 550/9-82-336

DEMONSTRATION OF NOISE CONTROL FOR THE DDA 6V-92TTA HEAVY DUTY TRUCK DIESEL ENGINE

MAY 1982

FINAL REPORT

CONTRACT NO. 68-01-4737

N-96-01
II-A-208

Document is available to the public in limited quantities from the National Technical Information Service, Springfield, VA 22151.

EPA 550/9-82-336

DEMONSTRATION OF NOISE CONTROL FOR THE DDA 6V-92TTA HEAVY DUTY TRUCK DIESEL ENGINE

MAY 1982

FINAL REPORT

Richard G. DeJong

Natan E. Parsons

Jerome E. Manning

Prepared for

Office of Noise Abatement
U.S. Environmental Protection Agency
Washington, D.C. 20460

Contract No. 68-01-4737

Cambridge Collaborative, Inc.
Kendall Square
Cambridge, Massachusetts 02142

Disclaimer Statement

This report has been reviewed by the Office of Noise Abatement and Control, EPA, and approved for publication. Approval does not signify that the contents necessarily reflect the views and policies of the Environmental Protection Agency, nor does mention of trade names or commercial products constitute endorsement or recommendation for use.

Cambridge Collaborative, Inc., the contractor, is responsible for the facts, the accuracy of the data, the analysis, and the views presented in this report. Neither the official views or policies of the Environmental Protection Agency, the sponsor, nor those of the Detroit Diesel Allison Division of General Motors, who consulted in the work, are expressed in this report. With this disclaimer, PERMISSION IS GRANTED TO REPRODUCE THIS MATERIAL WITHOUT FURTHER CLEARANCE.

Preface

The work reported herein has been performed by Cambridge Collaborative, Inc. under Contract No. 68-01-4737 from the U.S. Environmental Protection Agency Office of Noise Abatement. The authors wish to acknowledge the Massachusetts Institute of Technology which served as a subcontractor on the project, and the Detroit Diesel Allison Division of General Motors Corporation. The cooperation and assistance by these organizations has been an essential ingredient to this project.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (cont.)

I. INTRODUCTION

This report presents the results of engineering studies and designs which identified and demonstrated methods of reducing diesel engine block vibration and radiated noise levels. The methods demonstrated were such that noise reductions were achieved without degrading the engine's performance, fuel economy, or emissions. The demonstrated 4 dB reduction in radiated noise was achieved by retrofitting an existing, popular heavy-duty truck diesel engine, specifically the Detroit Diesel Allison 6V-92TTA. The study focused on the noise radiated by the surfaces of the engine and did not include exhaust, intake or cooling fan noise since the noise control for these sources has been demonstrated [1].

The work and the report can be divided into five major phases:

1. Determining the engine noise characteristics.

2. Developing a model which relates internal forces to external vibration or noise levels.

3. Designing noise controls with constraints provided by the basic design and the manufacturer of the engine.

4. Developing a demonstration engine.

5. Testing and evaluating the demonstration engine.

These phases are discussed in the following sections.

Section II presents a discussion of the engine's design and performance features and a detailed description of the engine's noise characteristics. First, the baseline noise characteristics are determined which relate the total sound radiation spectra of the engine to the speed and load of operation. Second, the magnitude of the sound intensity radiating from each of the various engine surfaces is measured in order to rank them according to their importance in generating noise. These measurements determine the basic noise emission characteristics of the engine in its standard configuration and are useful in comparing results, either before and after or with other investigators, particularly the manufacturer.

Section III presents the development of a noise generation model for the engine. First, the levels of various internal sources (such as combustion pressure, piston impacts, injectors, bearings, gears, pumps, and air flow through valves) are determined through direct or indirect measurements. Second, the characteristics of the vibration transmission through the engine structure, from each source location to the external surfaces, are determined from vibration response measurements on the disassembled engine. Third, the sound radiation characteristics of each vibrating surface are determined by relating the sound intensity measurements to the vibration levels of the engine measured during the operation of the engine in its standard configuration. Combining the results of these three procedures gives a means of estimating quantitatively the contribution of each internal source to the total radiated noise. The major sources of noise are discussed in detail. These included piston-slap, injectors, and combustion.

Section IV of this report presents the development of engine design modifications which will reduce the radiated noise in a truck by 5 dB. First, the noise generation model is used to develop analytical models which relate quantitative changes in the engine design parameters to quantitative changes in the engine noise. These models are used to identify values of engine design parameters which will give the desired noise reduction. Second, prototype designs are developed for the engine components which will achieve the specified design parameters. These prototype designs are modified and improved based on the constraints of other factors such as durability and performance.

Section V describes the fabrication and assembly of the demonstration engine. Both the noise control features and the necessary redesign of cooling passages and head bolts to accommodate the noise control designs are described.

Results for two demonstration engines are presented in Section VI. The final Section presents conclusions of the study.

II. ENGINE NOISE CHARACTERISTICS

A. Engine Design and Performance

The DDA 6V-92TTA is a constant horsepower, high torque rise, turbocharged, after-cooled, two-stroke diesel engine with six cylinders in a vee-configuration. It has the following specifications:

bore (inches)	4.84
stroke (inches)	5.00
compression ratio	17:1
total displacement (in^3)	552
no. of main bearings	4
firing order	1L, 3R, 3L, 2R, 2L, 1R

The engine is rated at a constant 270 brake hp over a speed range from 1500-1950 RPM. At the 270 hp rating the torque rise is 31.6% overall, with a rise rate of 6.7% per 100 RPM reduction in engine speed. The designation of the engine, 6V-92TTA, refers to the following engine parameters:

- 6V - six cylinders in a vee construction
- 92 - cubic displacement per cylinder
- TTA - turbocharged, tailored-torque, after-cooled

The tailored-torque feature indicates the engine's performance in constant horsepower over a significant speed range. This characteristic was established for two reasons: improved fuel economy, and a reduction in the number of gear shifts that the driver has to make while driving uphill. The tailored-torque is achieved through a simple change to the governor operation with no other changes to the engine compared to the 6V-92TA model. The tailored-torque configuration encourages drivers to operate in a lower RPM range which results in better fuel economy in the range of 20% over standard TA engines at the maximum RPM setting but with a minimum hp penalty.

As a result of the engine being a two-stroke engine, air intake is introduced through air ports midway up the liner height. Above the air ports the liner is water-cooled directly, and under the air ports the liner is cooled indirectly through a cast sleeve (see Figure 1).

The fuel system of this engine is of a unit injector type with a pump/injector for each cylinder located in the head activated by the camshaft through a push rod, rocker arm assembly.

B. Baseline Noise and Vibration Characteristics

The noise radiated from the 6V-92TTA engine was measured in one-third octave bands in the frequency range from 100 to 10,000 Hz. A reverberant test cell calibrated for sound power level measurements of noise sources with puretones according to American National Standard ANSI S1.21 and International Standard ISO 3742 was used for the measurements. The instrumentation required for the measurement is shown schematically in Figure 2. Using the rotating microphone boom, the sound pressure in the room was measured and averaged over time and space. The sound power level is computed from the average sound pressure level by comparing it to the measured sound pressure level of a standard noise source with a constant sound power output (a centrifugal fan made by ILG Manufacturers, Chicago was used for this project). The standard noise source was measured after each engine measurement sequence to account for changes in the room calibration with changes in the temperature and humidity of the air.

The vibration levels of the engine structure were measured at various points with accelerometers attached to the engine surface. The noise and vibration signals were analyzed in one-third octave bands using a Nicolet 444 Spectrum Analyzer interfaced with both analog and digital outputs. Selected test measurements were also tape recorded on a 2-channel Nagra IV-SJ Tape Recorder for storage and more detailed analysis.

Series 92

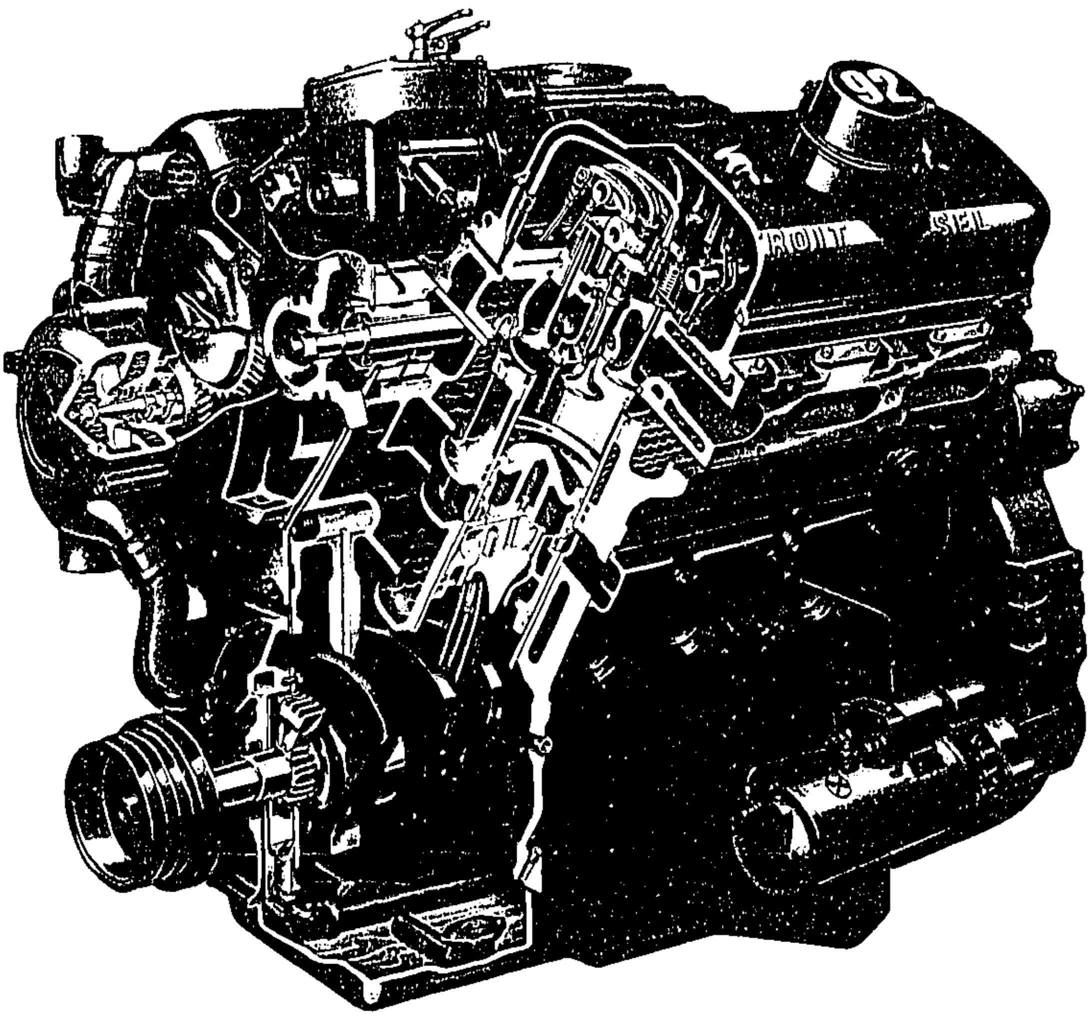

Figure 1. DDA 6V-92TTA Engine

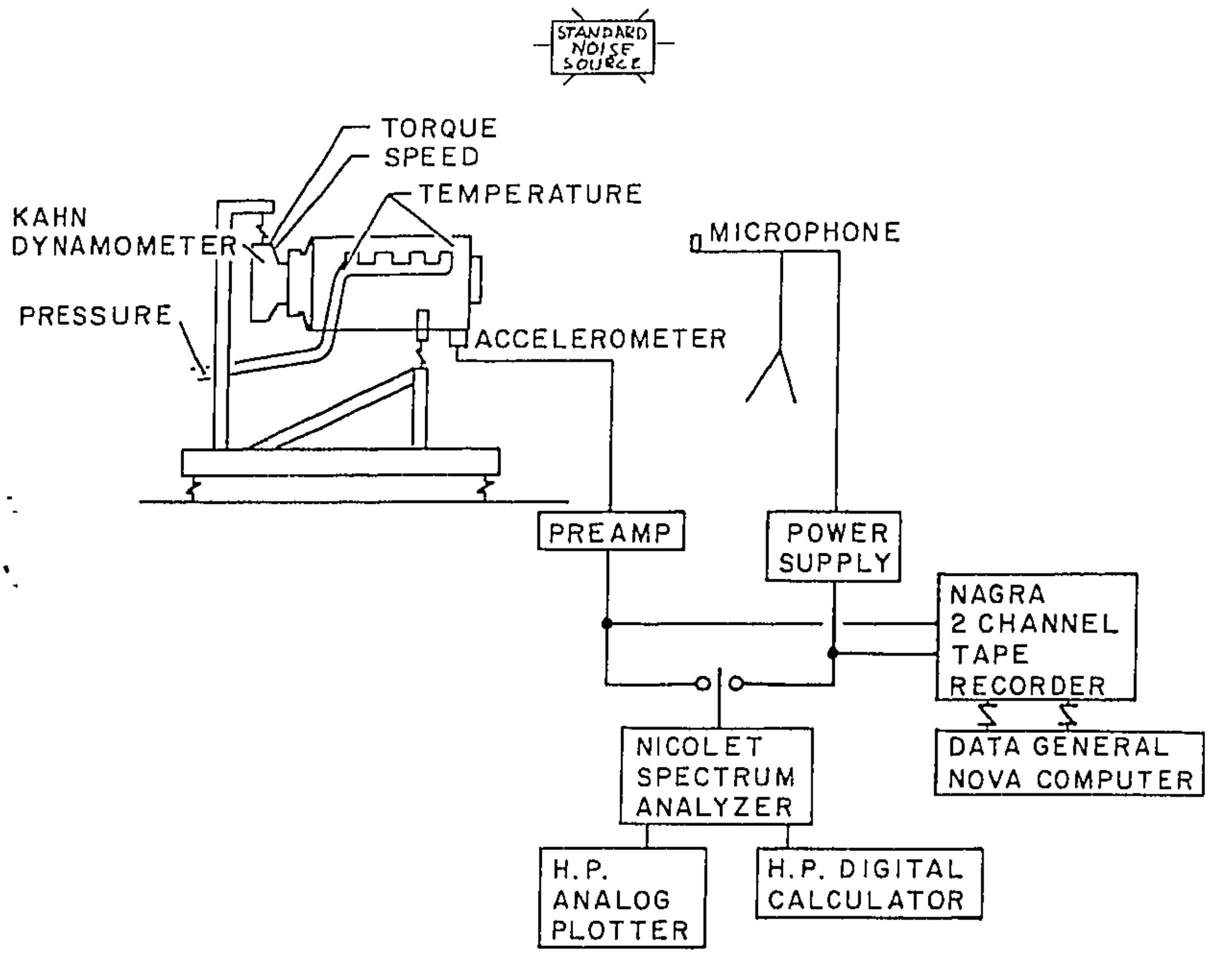

Figure 2. Instrumentation Set-Up

A load was applied to the engine with a KAHN 301 Waterbrake Dynamometer attached to the clutch housing. The dynamometer indicates both speed and torque during the operation of the engine. Also during the tests on the engine other parameters such as temperature and pressures were monitored to assure that the engine was operating properly.

A photograph of the reverberant room facility is shown in Figure 3. A photograph of the engine mounted in the facility is shown in Figure 4. In order to reduce the intake and exhaust noise, the intake and exhaust pipes were wrapped with fiberglass and lead during the noise test. The exhaust was cooled with a water spray. Also the dynamometer and stand were wrapped to reduce their radiated noise. The engine was run without a cooling fan to eliminate that noise source.

The total sound power radiated by the engine was measured as a function of speed and load.

Figure 5 shows a summary of the noise emissions of the diesel engine. The A-weighted sound power level is plotted versus speed for 50 and 100% of the full load at each speed. The noise emissions show a dependence on both speed and load. The sound power level is approximately proportional to 25 log N where N is the engine speed in RPM. A more detailed representation of the noise emissions is given in the next two figures. Figure 6 shows the frequency spectra of the sound power levels at 1950 RPM for the two load conditions. At higher frequencies the increase in sound level with load is small but fairly uniform with frequency. Figure 7 shows the frequency spectra of the sound power levels at full load for three speeds. Except at low frequency the increase in sound level with speed is fairly smooth with frequency.

C. Noise Radiation Characteristics

A more detailed analysis was made of the engine noise by measuring the amount of sound power radiated by individual surfaces of the engine using the acoustic intensity technique. The engine was divided into nine surfaces: left block side,

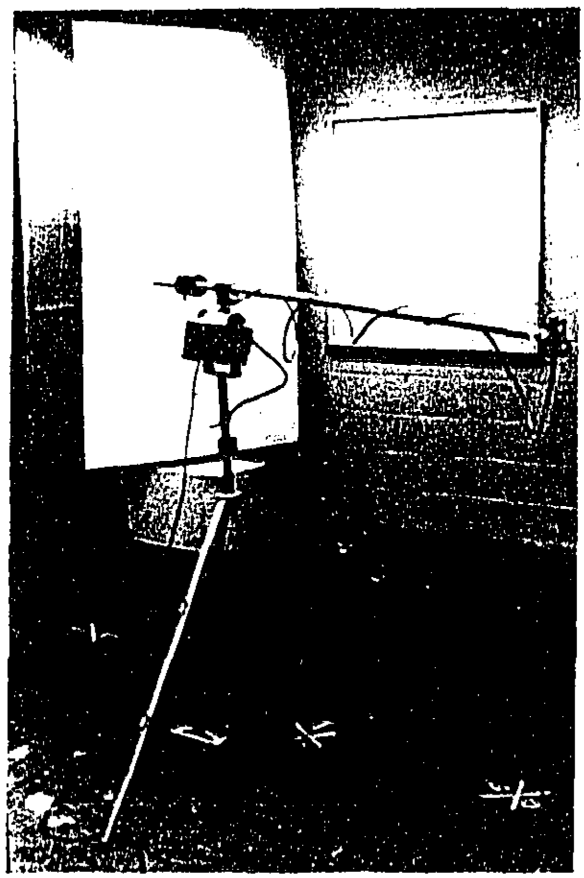

Figure 3. Reverberant Room Facility

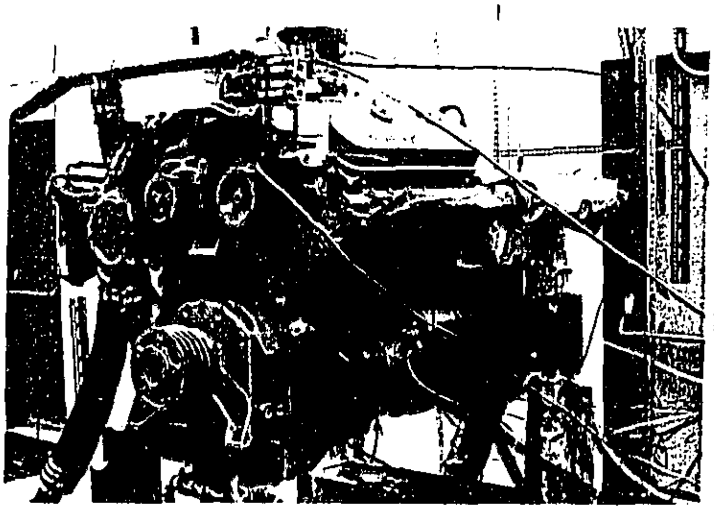

Figure 4. DDA 6V-92TTA Engine in Test Cell

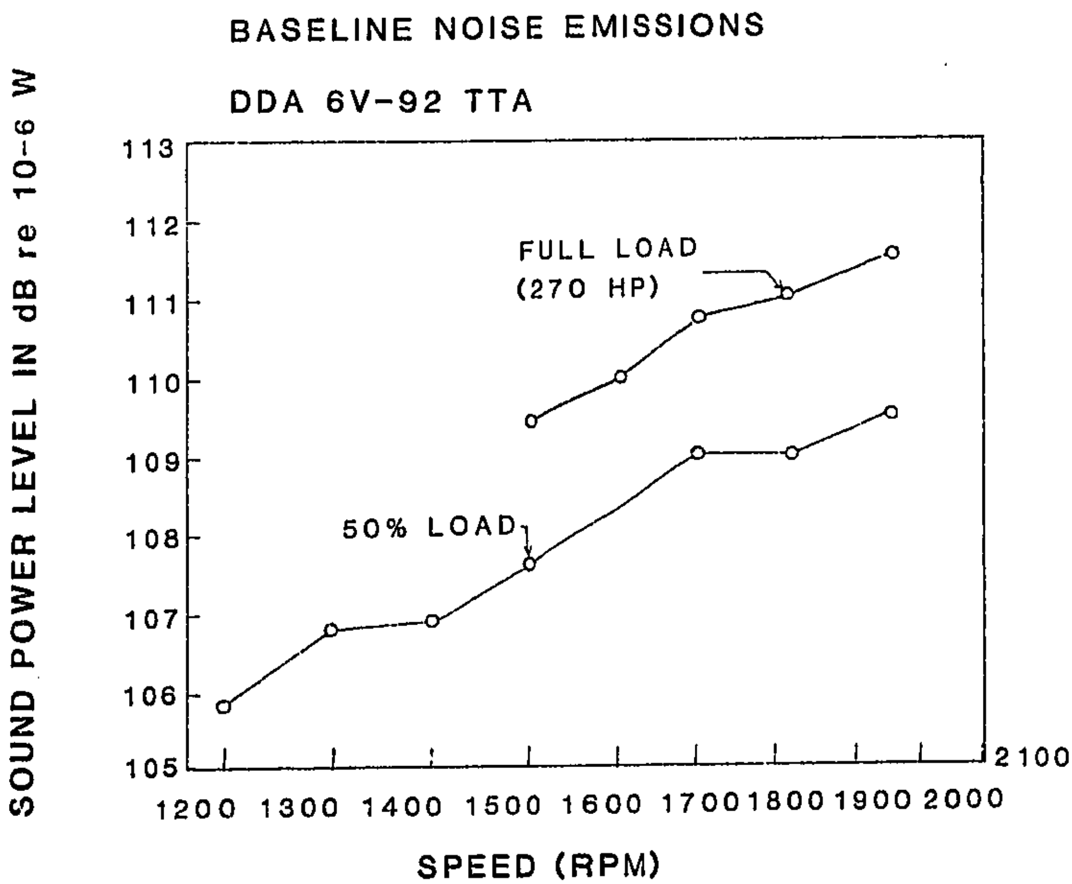

Figure 5. Noise Emissions vs. Engine Speed at Two Load Conditions

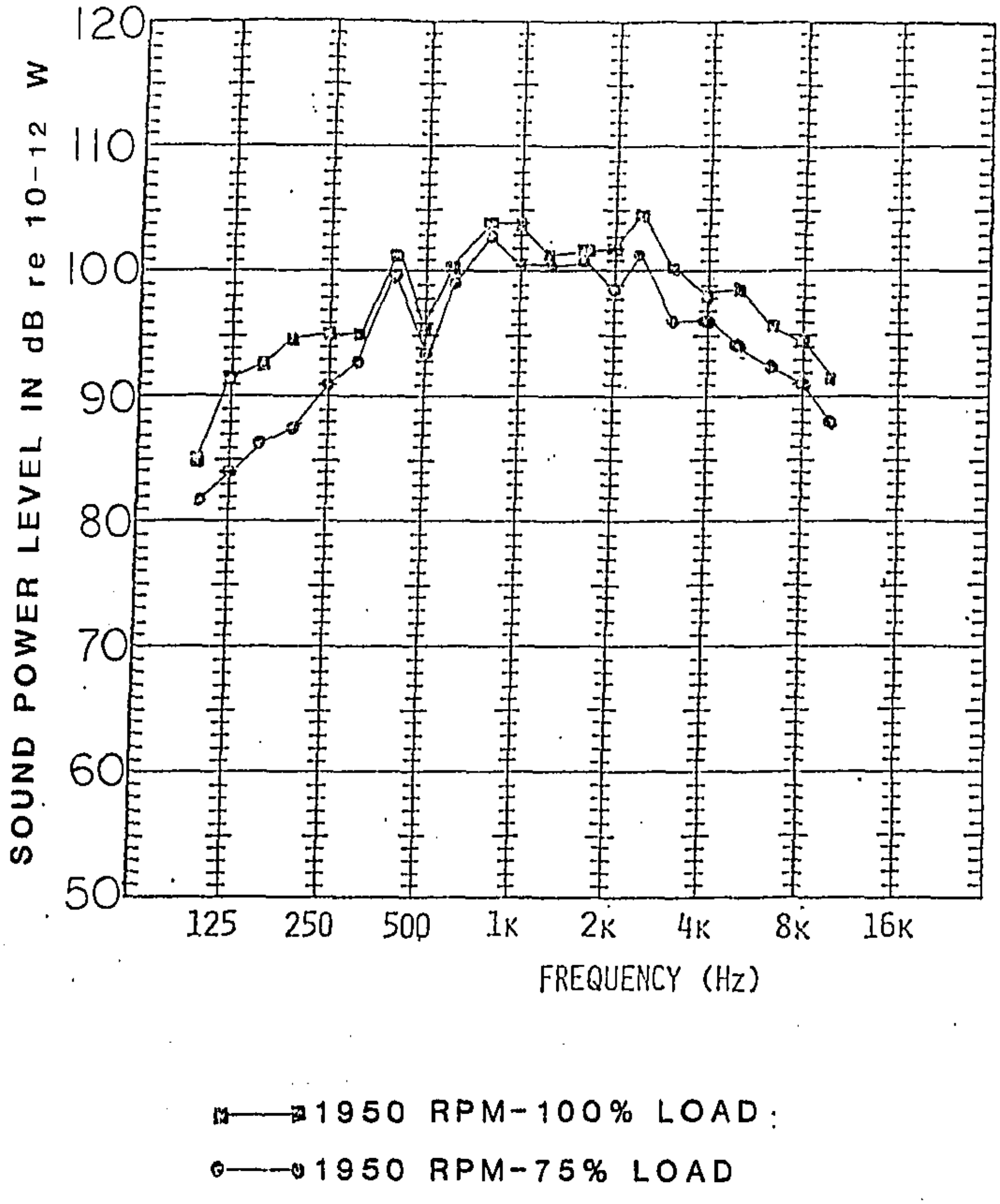

Figure 6. Frequency Spectra of Sound Power Levels of DDA 6V-92TTA Engine at 1950 RPM, Two Load Conditions

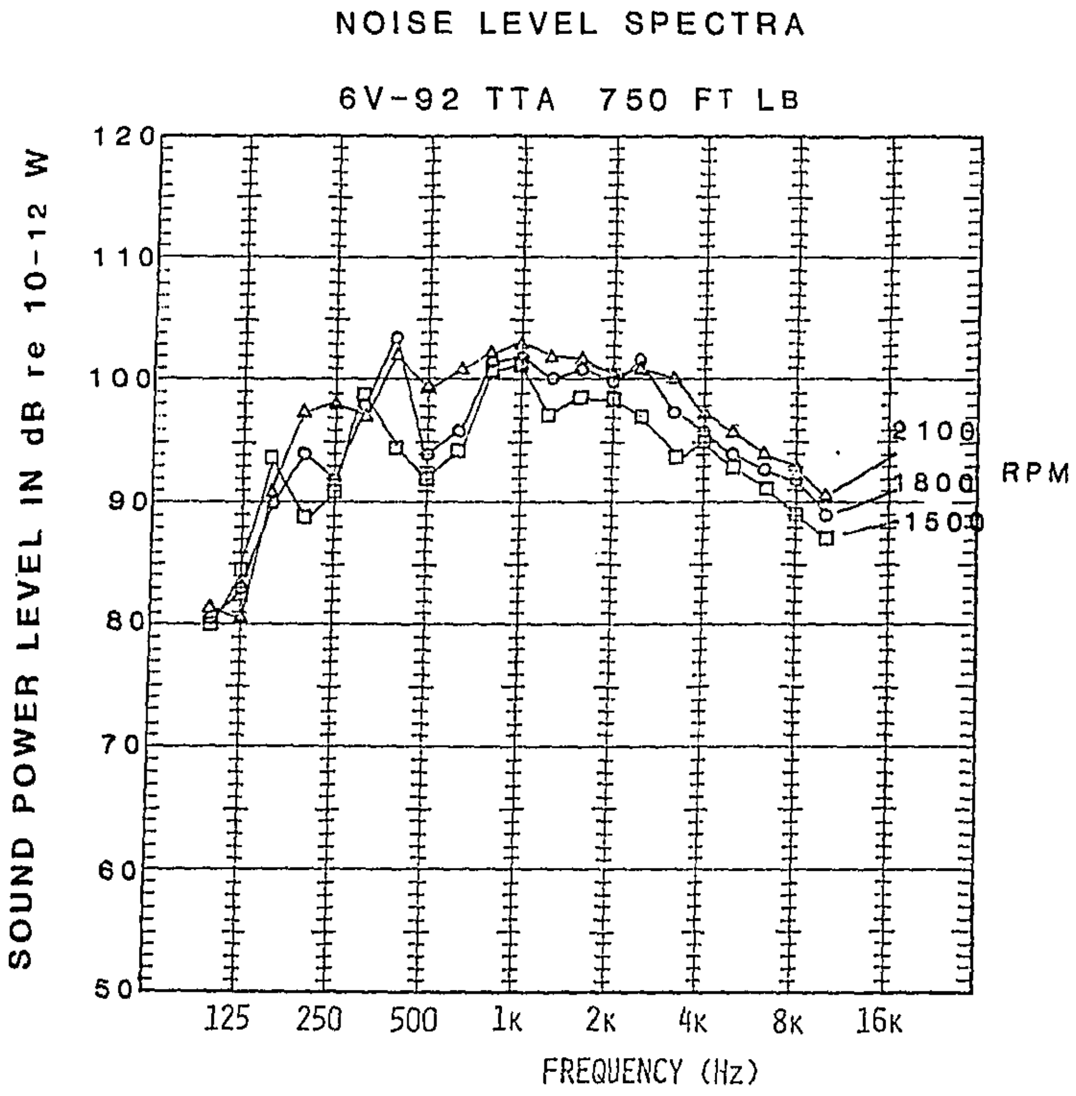

Figure 7. Frequency Spectra of Sound Power Levels of DDA 6V-92TTA Engine for Three Speeds

right block side, oil pan, valve cover, rear cover, front cover, blower housing, exhaust manifolds, and turbocharger. The sound power radiated by each surface was measured using two microphones spaced a small distance d apart [2]. The acoustic intensity in a direction parallel to the line between the two microphones is given by:

$$I = \frac{Q_{12}}{\omega \rho d} \tag{1}$$

where ω is the frequency in radians per second, Q_{12} is the imaginary part of the cross spectrum of the two microphone pressure signals, and ρ is the density of air.

Figure 8 shows the results of the source identification of the DDA 6V-92TTA engine at 1950 RPM, full load. The right block side is the strongest noise source of this engine. The engine rear, engine front, and turbocharger with pipes are all equal in strength, approximately 2 dB less than the right block side. The sum of the individual source levels is also shown for comparison with the total sound power level measured in the reverberant room.

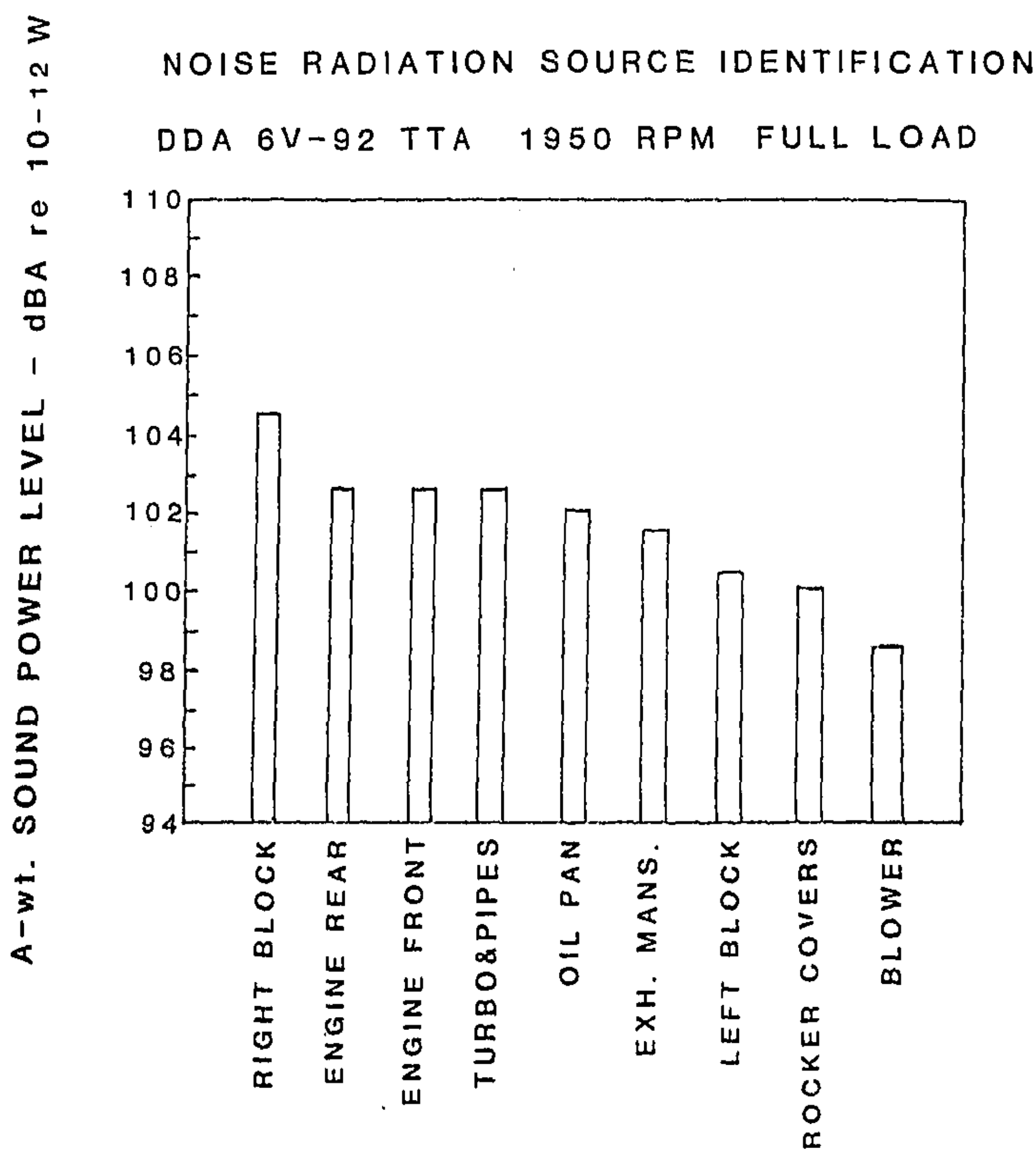

Figure 8. Source Identification of DDA 6V-92TTA Engine at 1950 RPM, Full Load

III. NOISE SOURCES AND VIBRATION TRANSMISSION PATHS

In the previous section of this report the surfaces of the engine that contribute most to the overall noise radiation were identified. At this point it would be possible to add covers and shields to reduce the radiation from these surfaces. It would also be possible to add a damping treatment to these surfaces in an attempt to reduce the vibration levels and subsequent radiated noise. However, the addition of damping is not generally effective because of the relatively high value of damping already present in a built-up engine structure. The addition of covers and shields can be effective in reducing noise, but is not desirable because of the potential damage to these items in a service environment, the interference with maintenance, and the added weight.

To proceed with a practical demonstration of engine noise control we have focused on changes to the internal structure of the engine which reduce the vibration generated by internal sources, such as combustion or piston-slap, or reduce the vibration transmitted to the radiating surfaces of the engine. To accomplish this objective we have continued the modeling work by developing methods to identify the sources of vibration in the engine and the different paths by which that vibration is transmitted to the engine surfaces.

The noise generating process in engines can be divided into four steps:

1. Dynamic pressures and forces are produced within the engine by internal sources such as combustion, piston slap, fuel injection, gear mesh, etc.;
2. The pressures and forces act on the internal engine structure and cause a local vibration;
3. The engine structure transmits the vibration to external surfaces of the engine;
4. The vibrating external surfaces of the engine produce radiated noise.

Three internal sources were chosen for study based on previous test results on Vee and in-line 6-71 engines: combustion, piston-slap, and injectors. For each source the level of the excitation and the vibration transmission to the engine surfaces have been measured. The results have been combined into a noise generation model which predicts the noise radiated by these engine sources and can be compared to the measured noise levels of the engine. The following paragraphs outline the procedures for studying each of these three sources and their associated vibration transmission paths. A more detailed discussion of the procedure is presented elsewhere [3].

A. Combustion Pressure

The source level for combustion noise was measured directly during operation of the engine with a pressure transducer in the combustion chamber of one of the cylinders. The transducer was water-cooled and flush-mounted in the cylinder head. The measured pressure signal and its frequency spectrum are shown in Figure 9. The cylinder pressure exerts a force both on the piston crown and on the cylinder head surface, indicating that there are two paths of vibration transmission. The first transmission path is through the piston, connecting rod, crank shaft, main bearings, and into the block. The second path is through the head and into the block.

The vibration transmission characteristics of the engine structure are determined from measurements in a non-running engine of the transfer mobilities of the various components along each vibration transmission path. The transfer mobility is a transfer function which measures the ratio of the vibration velocity at one point on the structure to the force exerted at another point. The mobility is measured with the instrumentation shown in Figure 10.

A comparison of the two transfer mobilities for the paths by which combustion noise is transmitted is given in Figure 11. These measured mobilities are one-third octave band averages of mobilities measured between eight points on the block and three

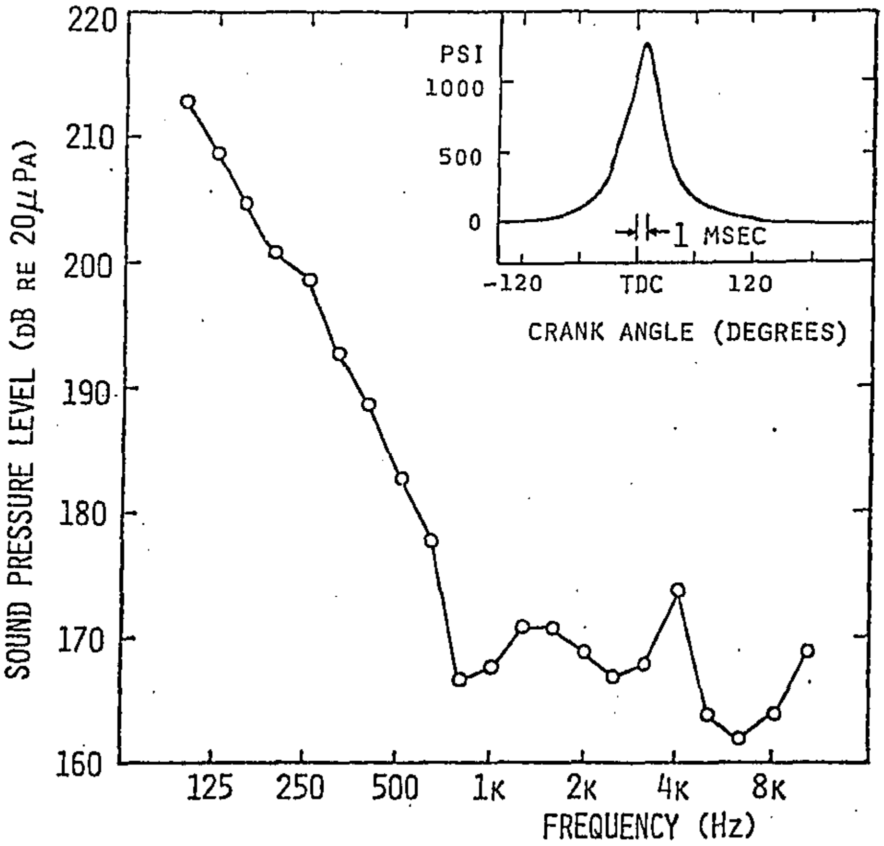

Figure 9. Cylinder Pressure of DDA 6V-92TTA Engine at 1950 RPM, Full Load

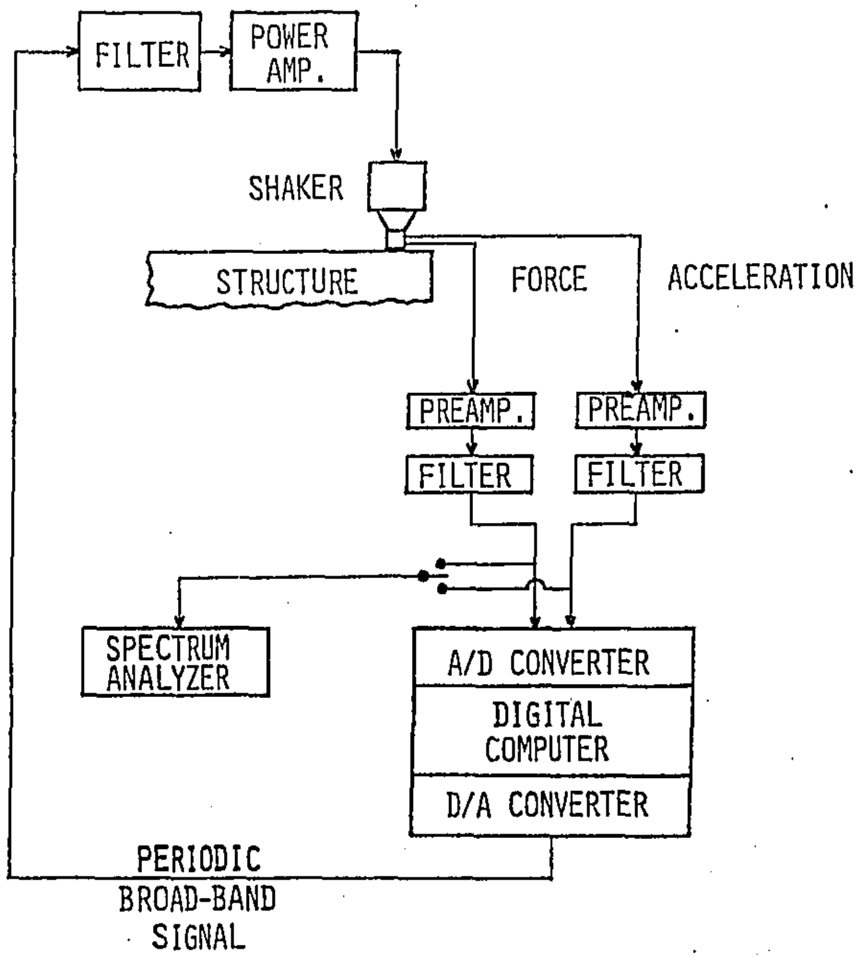

Figure 10. Instrumentation for Mobility Measurements

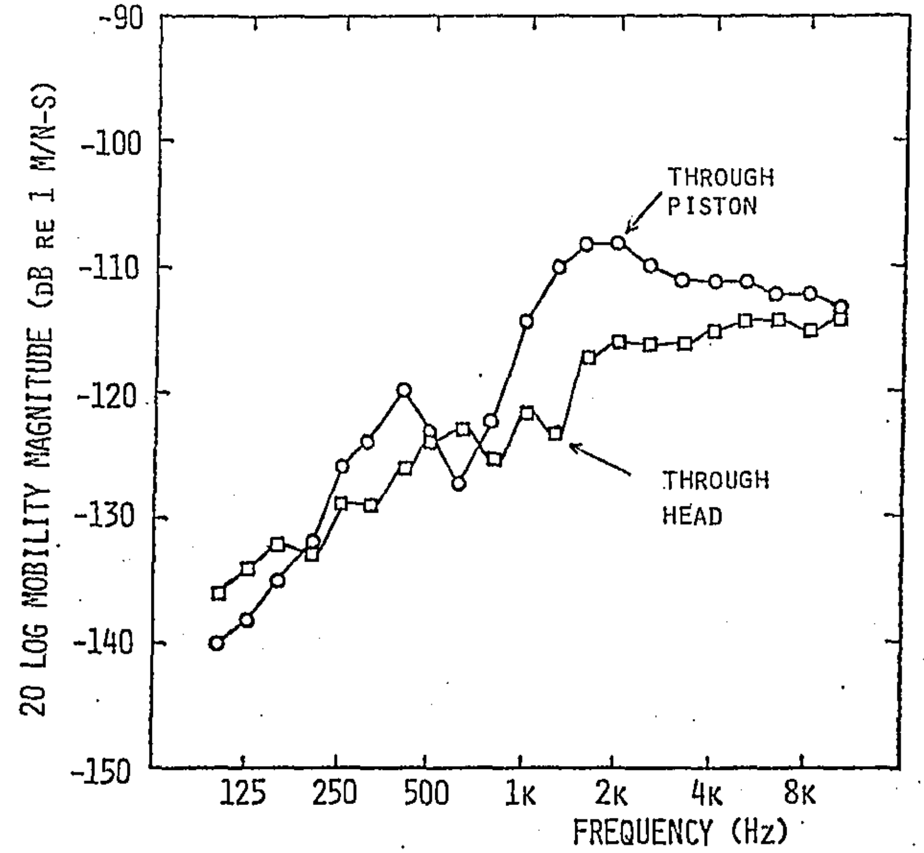

Figure 11. Transfer Mobilities for Combustion Force to Block Vibration on DDA 6V-92TTA Engine

cylinders. A prediction of the block vibration due to combustion when the engine is running is obtained by multiplying the total mobilities for each path by the combustion force (pressure times piston area) and adding the contributions from each path. A comparison of this prediction of block vibration due to combustion with the measured vibration during operation of the engine is shown is Figure 12. The comparison shows that combustion is not a major source of block vibration and therefore is not a major source of noise for this engine.

B. Piston-Slap

The source level of piston-slap cannot be measured directly by any known methods. Therefore, an indirect measurement procedure has been used to determine the magnitude of the force generated. This was done by measuring the local vibration response of the cylinder liner at a point on the liner where piston-slap is believed to occur. Figure 13 shows a typical time response of the liner vibration during operation of the engine and the corresponding velocity spectrum. A second experiment was conducted to measure the drive-point mobility at the same point on the liner where the vibration response was measured. Since the drive-point mobility is the ratio of the velocity to the applied force at the point where the force is applied, the force generated by piston-slap can be determined by dividing the measured velocity level by the drive-point mobility. Results are shown in Figures 14 and 15.

The vibration transmission path from piston-slap to the engine surfaces is primarily through the liner and block side walls for the 6V-92TTA. The transfer mobility between the cylinder liner and the block side walls is shown in Figure 16.

Combining the force excitation level of the piston-slap with the measured transfer mobility gives a prediction of the engine block vibration due to piston-slap. Figure 17 compares this prediction with the measured vibration levels for the right block side, which is the major thrust side and the major radiating

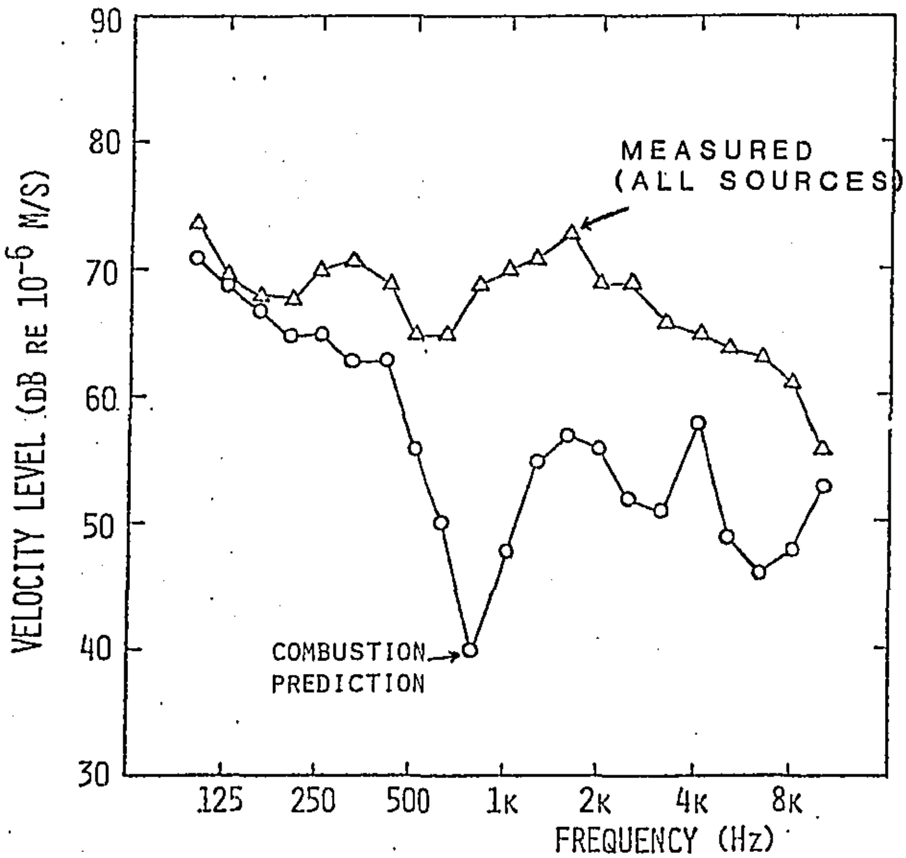

Figure 12. Average Vibration Levels of Engine Block on DDA 6V-92TTA Engine at 1950 RPM, Full Load

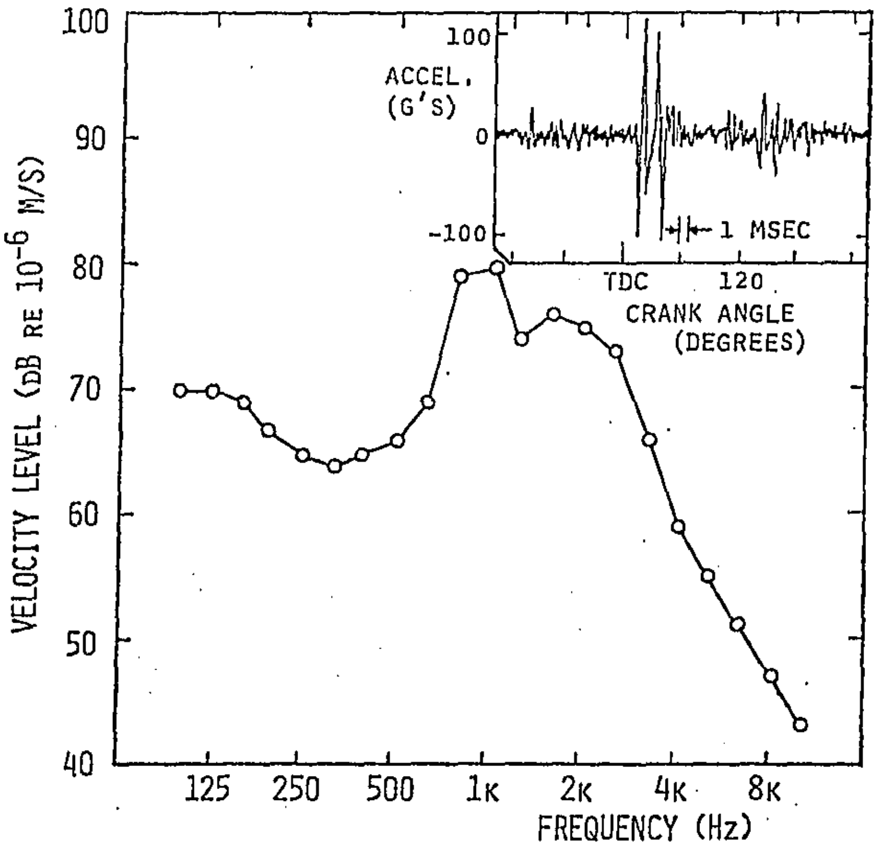

Figure 13. Cylinder Liner Vibration of DDA 6V-92TTA Engine at 1950 RPM, Full Load

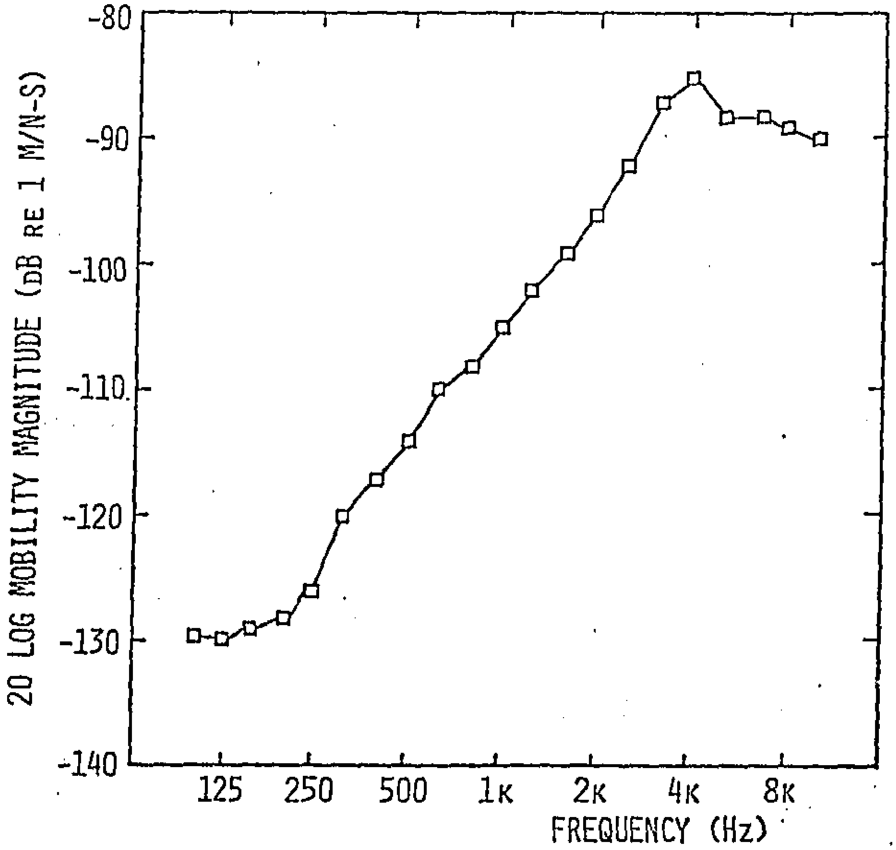

Figure 14. Drive-Point Mobility of Cylinder Liner on DDA 6V-92TTA Engine

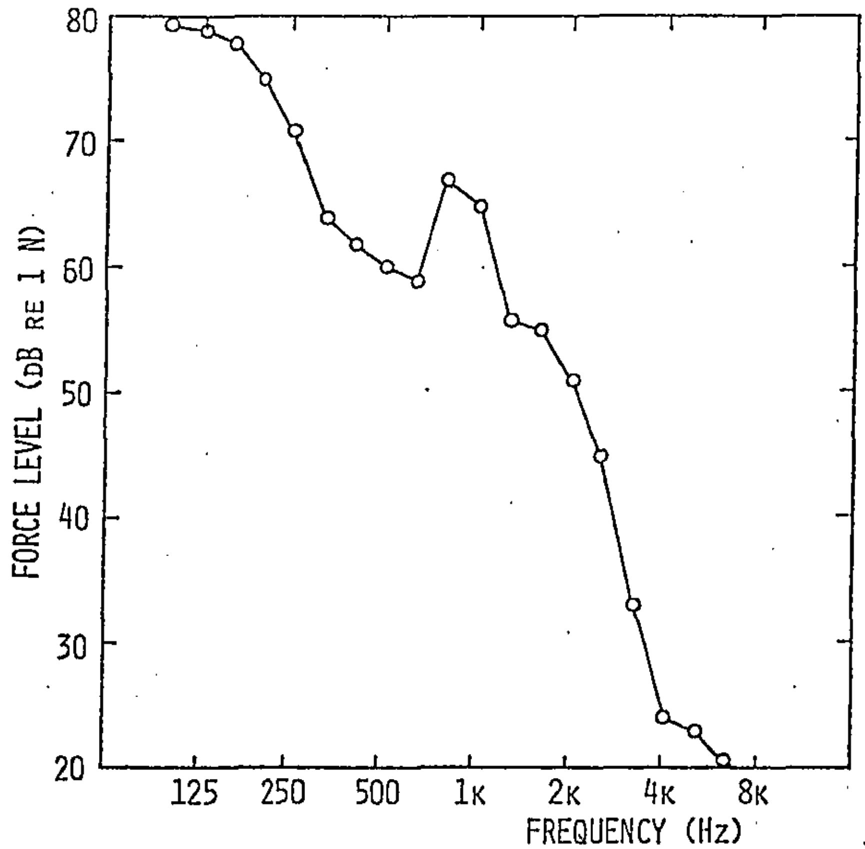

Figure 15. Estimated Force of Piston-Slap in DDA 6V-92TTA Engine at 1950 RPM, Full Load

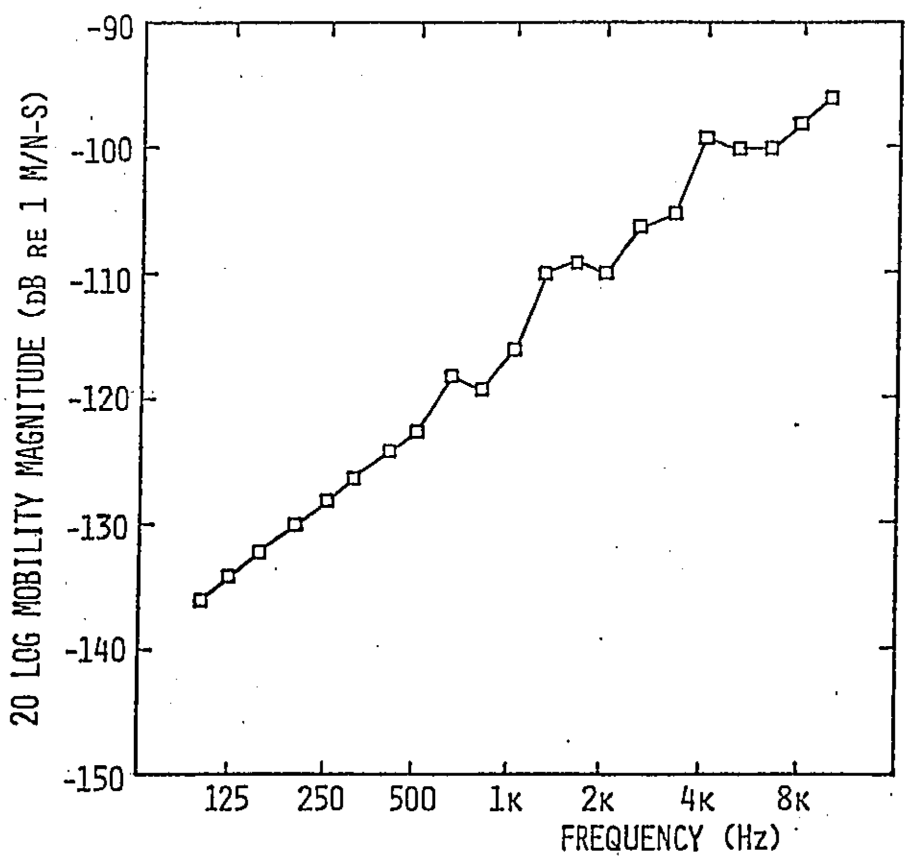

Figure 16. Transfer Mobility of Piston-Slap Force to Block Vibration on DDA 6V-92TTA Engine

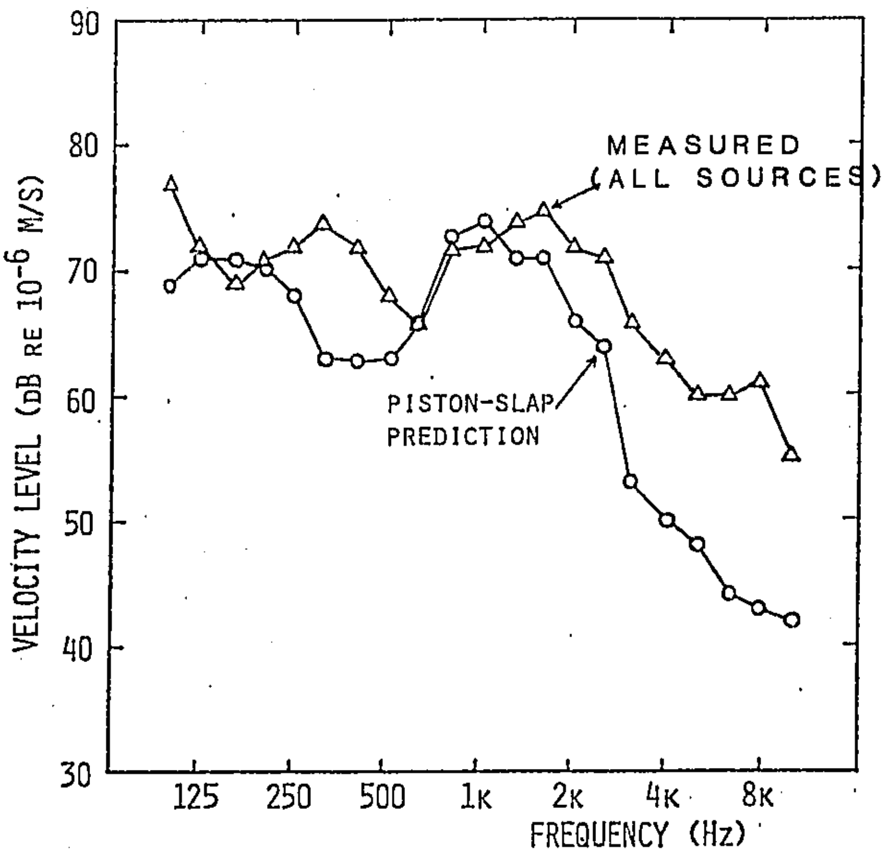

Figure 17. Average Vibration Levels of Right Engine Block on DDA 6V-92TTA Engine at 1950 RPM, Full Load

surface. This comparison shows that piston-slap is a major source of block vibration, and therefore radiated noise, in the frequency range 500 to 2000 Hz.

C. Injector Forces

The 6V-92TTA is equipped with unit injectors, each of which injects a specific amount of fuel into a cylinder by means of a cam-driven plunger. A direct measurement of the dynamic force generated during injection is difficult. Therefore, an indirect measurement of the injector source level was obtained in a manner similar to that done for piston-slap. The velocity of the injector plunger was measured during the operation of the engine and is shown in Figure 18. The designed velocity level of the injector cam follower is also shown by the dashed line. It can be seen that the plunger exhibits a large vibration near the end of injection with a large 1000 Hz component in the spectrum.

The force exerted on the injector plunger was obtained by dividing the velocity spectrum of the plunger by the measured mobility of the plunger in its fully assembled condition. Results are shown in Figures 19 and 20.

Forces generated by the injectors are transmitted to the engine block by two paths: the first path is through the injector casing to the head and the engine block, and the second path is through the cam push-rods, cam-shaft, bearings and cam gears, to the block. The vibration transmission through each path was determined by physically disconnecting the injector plunger from the cam drive train and exciting the structure with a shaker attached to one of the other of the disconnected members. The resulting measured transfer mobilities to the block are shown in Figure 21. The path through the camshaft dominates the vibration transmission above 2000 Hz.

Combining the force excitation level of the injector and the measured transfer functions to the block gives a prediction of the engine block vibration due to the injectors. Figure 22 compares this prediction with the measured block vibration levels

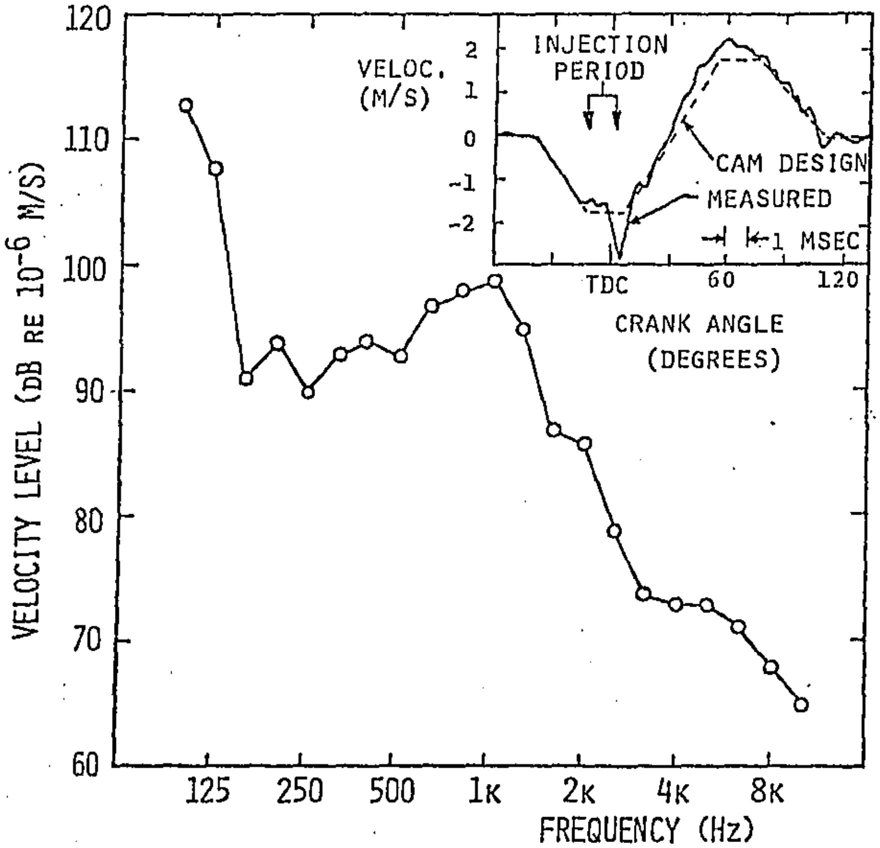

Figure 18. Vibration of Injector Plunger in DDA 6V-92TTA Engine at 1950 RPM, Full Load

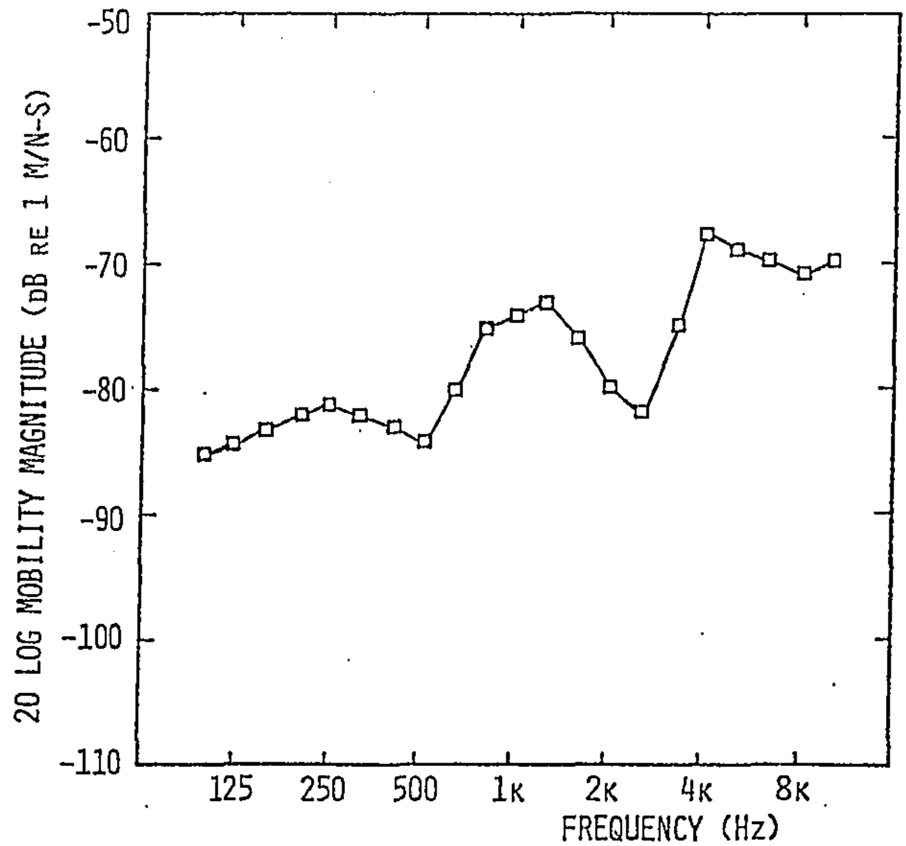

Figure 19. Mobility of Injector on DDA 6V-92TTA Engine

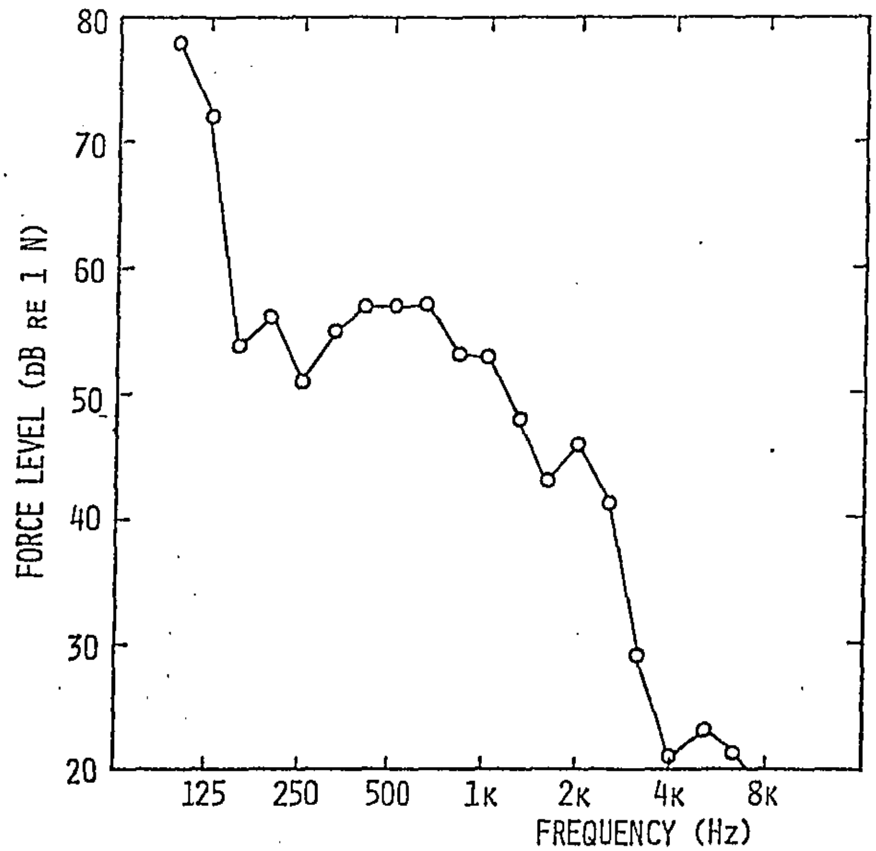

Figure 20. Estimated Injector Force Levels in DDA 6V-92TTA Engine at 1950 RPM, Full Load

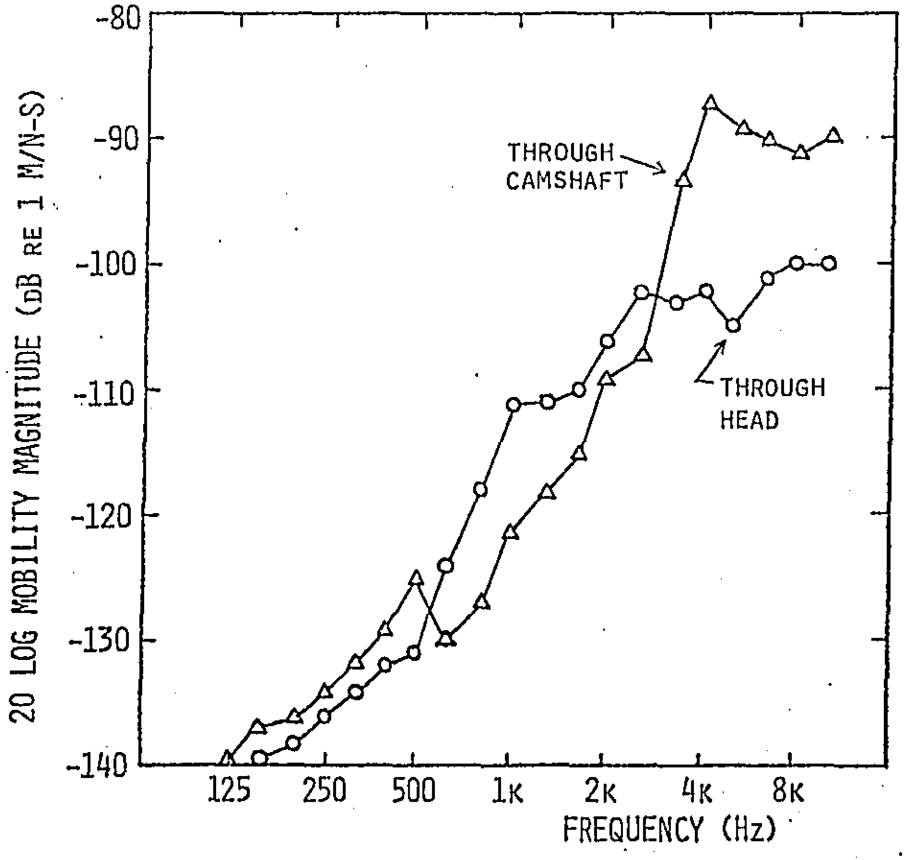

Figure 21. Transfer Mobilities of Injector Forces to Block Vibration on DDA 6V-92TTA Engine

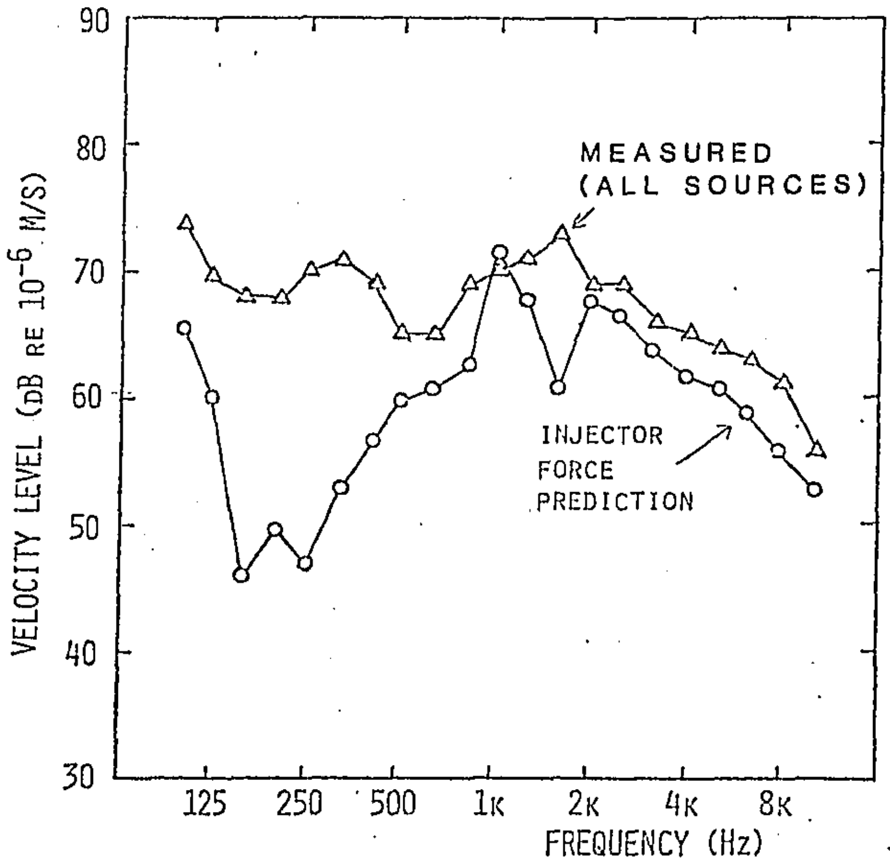

Figure 22. Average Vibration Levels of Engine Block on DDA 6V-92TTA Engine at 1950 RPM, Full Load

during operation of the engine. The comparison shows the injectors are a major source of vibration at and above 1000 Hz.

D. Vibration Transmission of Non-Load Bearing Covers

By summing the contributions to block vibration from the major sources, piston-slap and injectors, a nearly complete model of the engine vibration can be obtained for this specific engine. In order to complete the model it is necessary to relate the vibration of non-load bearing covers, such as the oil pan, valve covers, and front cover, to the block vibration since these covers can be major radiating surfaces. Following the mobility approach the vibration transmission from the block to covers has been determined by combining measured mobilities on a non-running engine with measured block vibration levels. Examples of the mobilities for the oil pan are shown in Figure 23. A prediction of the ratio of velocities of the oil pan and the block is shown in Figure 24 along with a ratio determined from measured data. The comparison shows that the oil pan has higher vibration levels over most of the frequency range of interest.

E. Noise Radiation

To complete the modeling, the noise radiated by each engine surface must be related to the predicted velocity. Our approach was to determine a radiation efficiency by measuring simultaneously the surface vibration and the sound power radiated using the acoustic intensity technique. The measured values of radiation efficiency are shown in Figure 25. Using these values of radiation efficiency the prediction of block and cover vibration can be used to predict the noise radiated by the engine. Results are shown in Figure 26 and compared to the total measured sound power levels. There is good agreement between prediction and measurement except in the 300 to 500 Hz range. In this range the engine noise is thought to be dominated by the Roots blower based

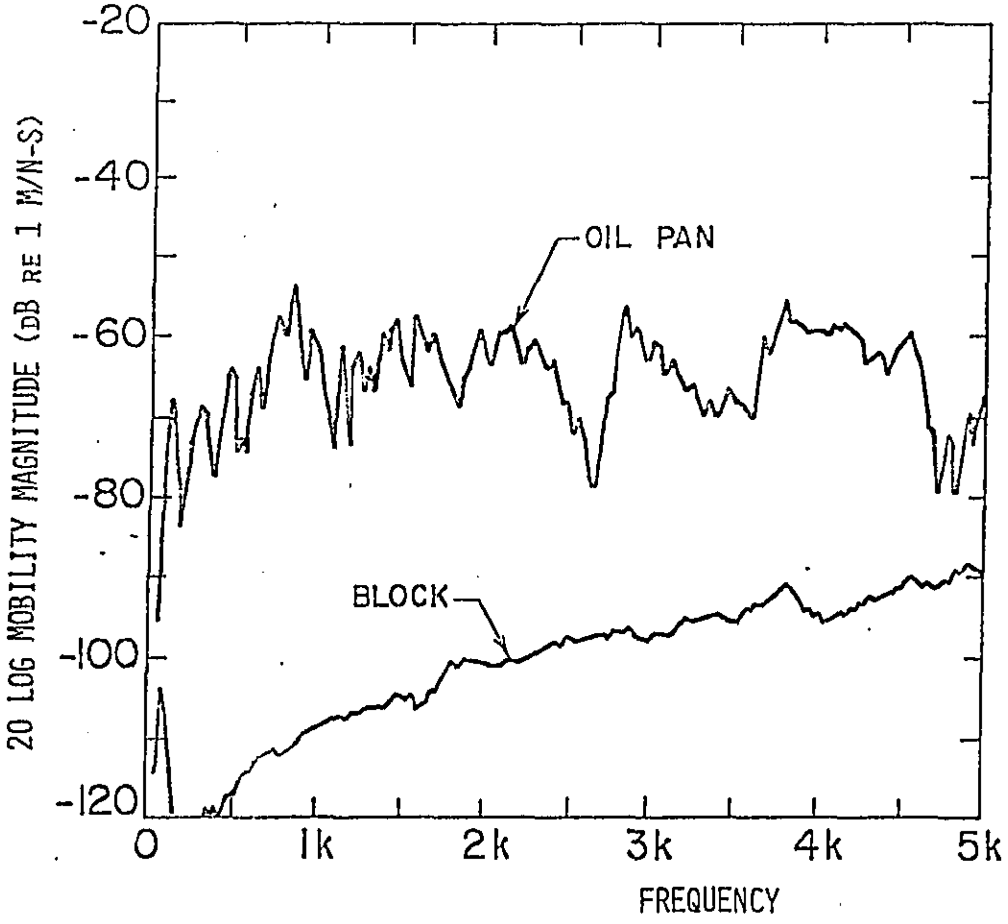

Figure 23. Measured Mobilities of Oil Pan and Block Connection on DDA 6V-92TTA Engine

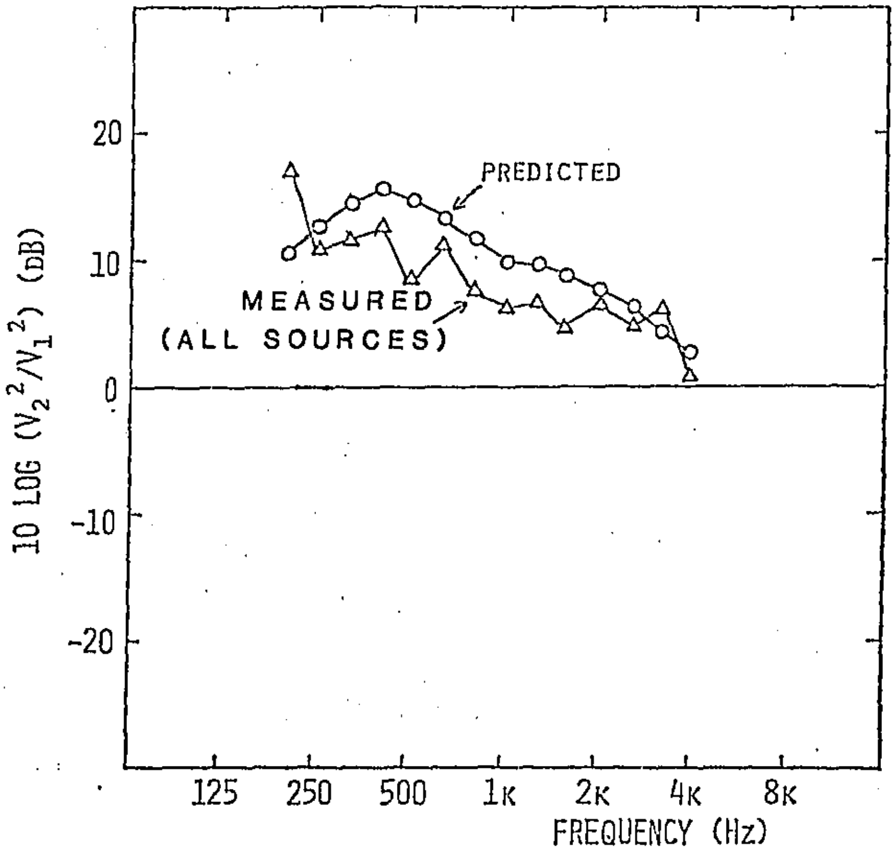

Figure 24. Average Velocity Ratio of Oil Pan to Block on DDA 6V-92TTA Engine

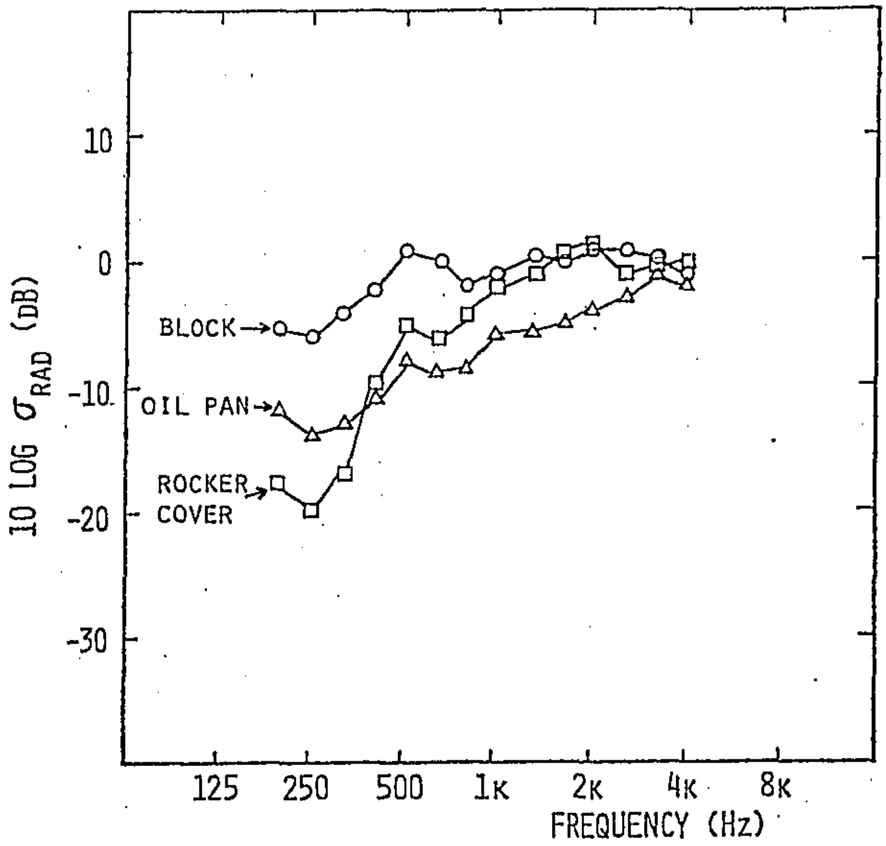

Figure 25. Measured Radiation Efficiencies of Surface Vibration on DDA 6V-92TTA Engine

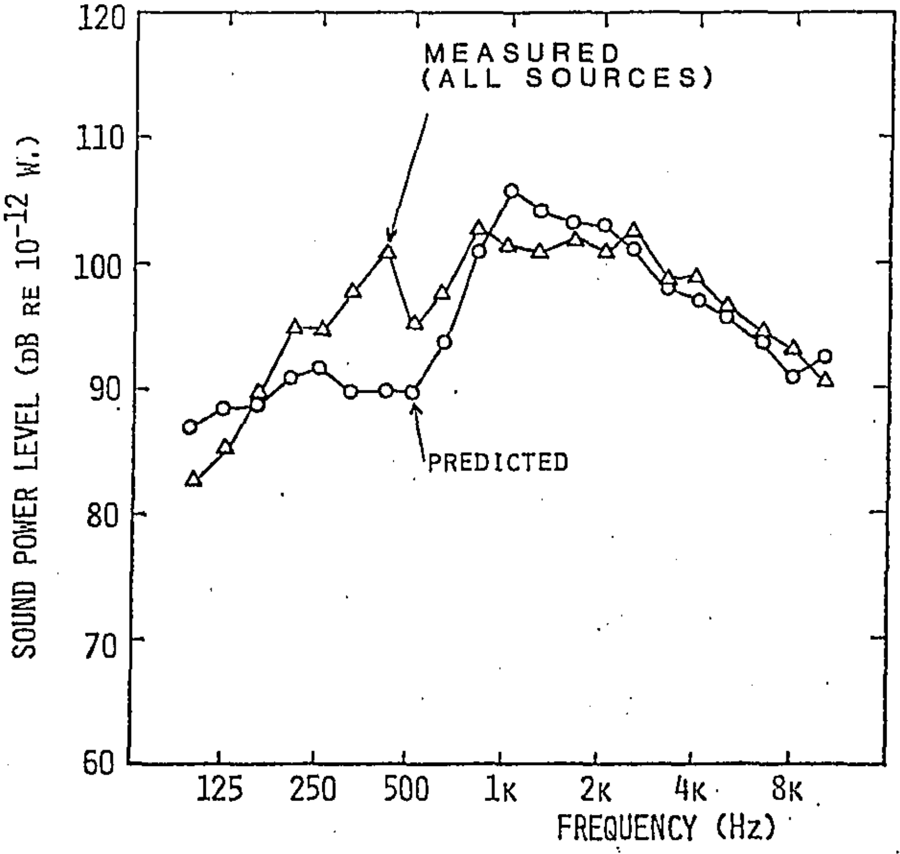

Figure 26. Sound Power Levels of DDA 6V-92TTA Engine at 1950 RPM, Full Load

on the following observations: (1) this frequency range matches the blade passage rate in the blower, and (2) the peak frequency of this source changes with engine speed while the amplitude of the source changes with engine load. The Roots blower is required in the 6V-92TTA since it is a two-stroke engine which requires a positive intake pressure at all operating conditions. The turbocharger provides positive intake pressure at high speeds and steady operating conditions, but cannot achieve this at low speeds and under certain accelerating conditions.

IV. NOISE REDUCTION DESIGNS

The objective of the work presented in this report is to demonstrate a method of achieving a 5 dB reduction of in-truck, A-weighted noise. Toward this end, we have developed new and improved modeling procedures that allow us to predict the major sources of high frequency engine vibration and to understand the paths by which that vibration is transmitted to the surfaces of the engine and radiated as noise.

The major sources of high frequency vibration for the DDA 6V-92TTA engine are piston slap and injection. These two sources are so dominant that a 5 dB reduction in overall noise can be achieved by reducing the noise resulting from these sources alone. Our approach in reducing the noise from these sources is to modify the engine structure to reduce the vibration transmission. Other methods for achieving the desired noise reduction include: (a) modifying the sources to reduce the impulsive forces generated during injection and piston slap, (b) use of engine covers and enclosures to reduce the noise radiation from the vibrating engine surfaces, and/or (c) use of damping and vibration isolation treatments to reduce the vibration and noise radiation from non-load bearing covers and shields. These other techniques are valid noise control procedures, and have received a great deal of attention by others working in the area of engine noise reduction. In our work we have not eliminated these techniques from consideration, but have not spent any time developing them because of the work already done by others and because of our belief that modification of the engine structure is the most cost-effective means of achieving our 5 dB noise reduction goal.

A. Injector Noise Reduction

The DDA 6V-92TTA engine uses unit injectors, shown schematically in Figure 27. Requirements for fuel economy and low emissions put stringent requirements on injector design. The rate of fuel injection must be very accurately controlled and an abrupt shut-off of injection is required. Because of the high pressures involved and the abrupt shut-off, impulsive forces are generated and transmitted to the engine structure. Two transmission paths are important: one through the rocker arm pedestal support to the head and to the engine block; and one through the rocker arm to the pushrods, cam, camshaft bearings, and to the block.

The measurements of the injector assembly vibration characteristics indicate that there are two important frequency ranges in the injector noise with different characteristics. Around 1,000 Hz the force excitation is controlled by a resonance in the cam drive train structure which produces a peak in the rocker arm vibration response and a peak in the radiated noise due to the injectors as shown in Figure 28. This force is transmitted well through the injector casing to the cylinder head, and to the block. Above 1600 Hz the force excitation is due to an impulse at the end of injection. The vibration from this impulsive force is transmitted primarily through the camshaft and its driving gears to the block. Two different approaches are needed to reduce the injector source contributions in these two frequency ranges.

For the 1000 Hz frequency range, a design has been developed which changes the frequency of the injector assembly resonance. A closer investigation of the vibration pattern of that resonance around 1000 Hz indicated that it was controlled by the stiffness of the pushrod and the inertias of the rocker arm and cam lobes. The most successful parameter modification came from an increase in the pushrod diameter.

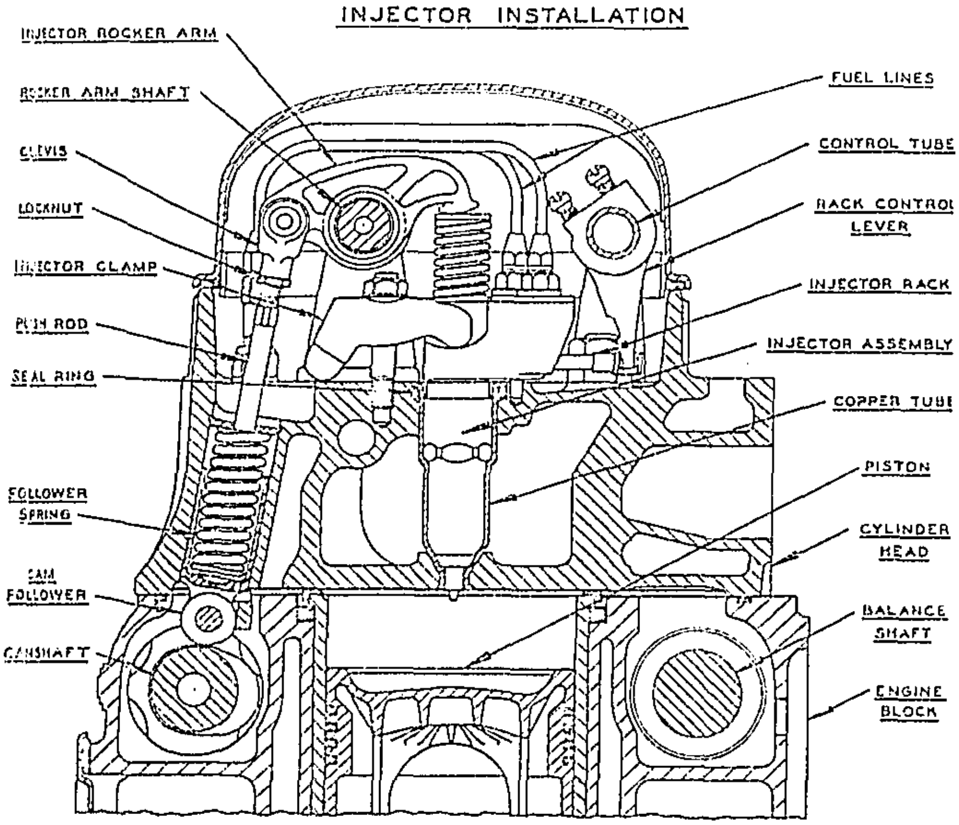

Figure 27. DDA 6V-92TTA Unit Injector Mechanisms

RADIATED NOISE LEVELS

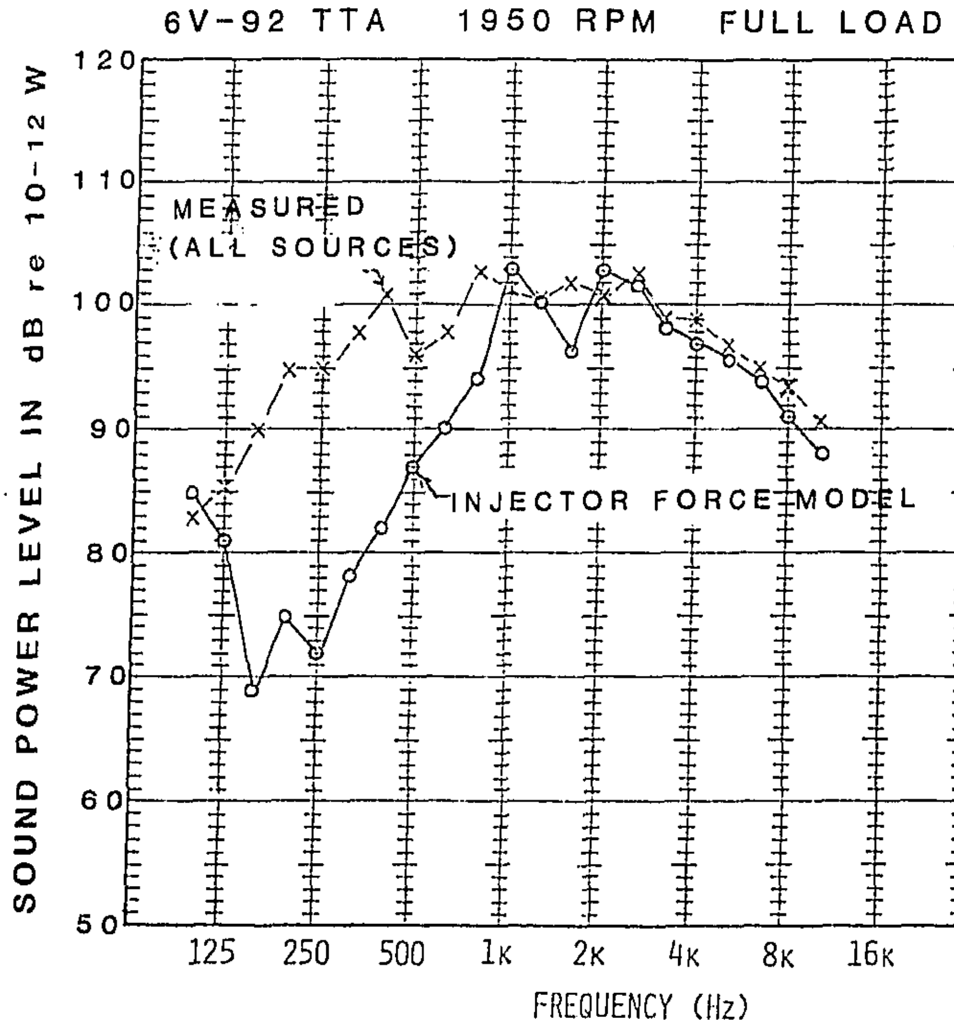

Figure 28. Overall Radiated Noise Levels in DDA 6V-92TTA Engine vs. Injector Force Model

The pushrod design modification was developed by measurements of mobility on a static, non-running engine. The mobility measured by driving the rocker arm, Figure 29, shows two peaks near 1,000 Hz. This measurement supports our conclusion that the peak in the injector noise at 1,000 Hz is due to resonances in the transmission path rather than a peak in the excitation force spectrum. Since the spectrum of the injector force decreases significantly above 1,000 Hz (Figure 20), we concluded that increasing the resonance frequencies of the injector-rocker-pushrod-camshaft assembly would reduce radiated noise. Lowering the resonance frequencies was also considered, since the transfer mobility from the injector to the block is much lower at low frequencies, Figure 21. However, this approach was ruled out because decreasing the stiffness of the assembly or increasing its mass would degrade injector performance. On the one hand, decreasing the stiffness would increase the response time for injection and cause retardation problems in the injector performance curve. On the other hand, increasing the mass of the assembly would increase forces on the cam, increase inertia, and possibly introduce a bounce of the cam follower.

Further mobility measurements showed that the resonance at 1,000 Hz could be most easily increased by increasing pushrod stiffness. Clearance allowed the pushrod cross-sectional area (and its stiffness) to be increased by a factor of three. The resonance frequency was expected to increase from 1,000 Hz to 1,700 Hz due to this increased stiffness. The measured increase was somewhat less than $\sqrt{3}$ as might be expected, because of the added pushrod mass. The measured mobility for the modification design is shown in Figure 29. As expected, the mobility is reduced at 1,000 Hz and increased in the region of 1,600 Hz due to the shift in resonance frequency.

Further reduction of injector noise requires that the vibration at high frequencies above 1,600 Hz be reduced. Changes to the injectors themselves were considered and are believed to have a long term potential for reducing noise. However, because of the stringent demands on injector performance, we decided that

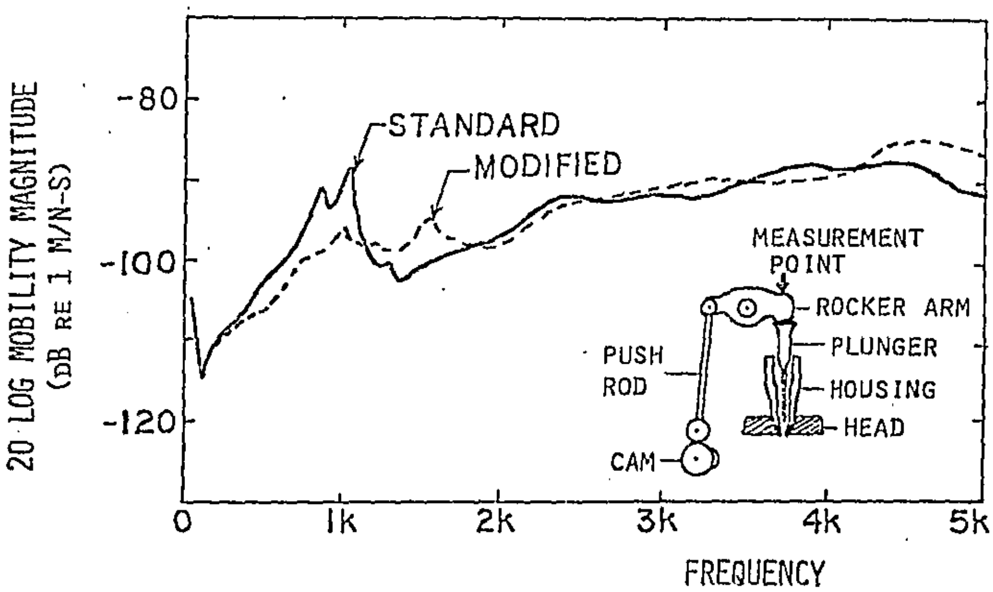

Figure 29. Vibration Response of Injector Assembly on DDA 6V-92TTA Engine

changes to the vibration transmission path were more feasible. The best way to reduce the vibration transmission is to introduce an impedance mismatch at the camshaft support bearings by means of resilient materials. Resilient bearings have been previously shown to be effective in reducing vibration transmission on a non-running engine and we decided to pursue this approach [5].

The vibration transmission across a bearing is proportional to the coupling loss factor η_{CB} given by

$$\eta_{CB} = \frac{\mathrm{Re}\ [Y_{BLOCK}]/(2\pi f M_C)}{|Y_{CAMSHAFT} + Y_{BEARING/BLOCK}|^2} \qquad (2)$$

where the mobilities, Y, are a function of frequency. In an engine structure the block mobility at high frequencies increases with frequency and can be modeled by a spring. The bearing mobility can also be modeled by a spring. The camshaft mobility is somewhat irregular due to camshaft resonances but generally decreases with frequency and can be modeled as a beam. To be effective in reducing vibration transmission, the bearing mobility must be increased so that it is greater than the sum of the block and camshaft mobilities over the frequency range in which vibration reduction is to be achieved.

For the purpose of reducing injector noise in the 6V-92TTA we set the frequency at which the bearing mobility becomes greater than the sum of the block and camshaft mobilities to be 1,600 Hz. A design meeting this requirement is shown in Figure 30. In this design the layers of the sandwiched construction are pinned together to prevent rotation of any layer. This is necessary to insure that the oil ports maintain alignment. Static calculations were done to determine the effect of the resilient bearing on injection timing. The resilience of the bearing was found to retard timing by 1/6 of a crank degree. This retardation is not significant.

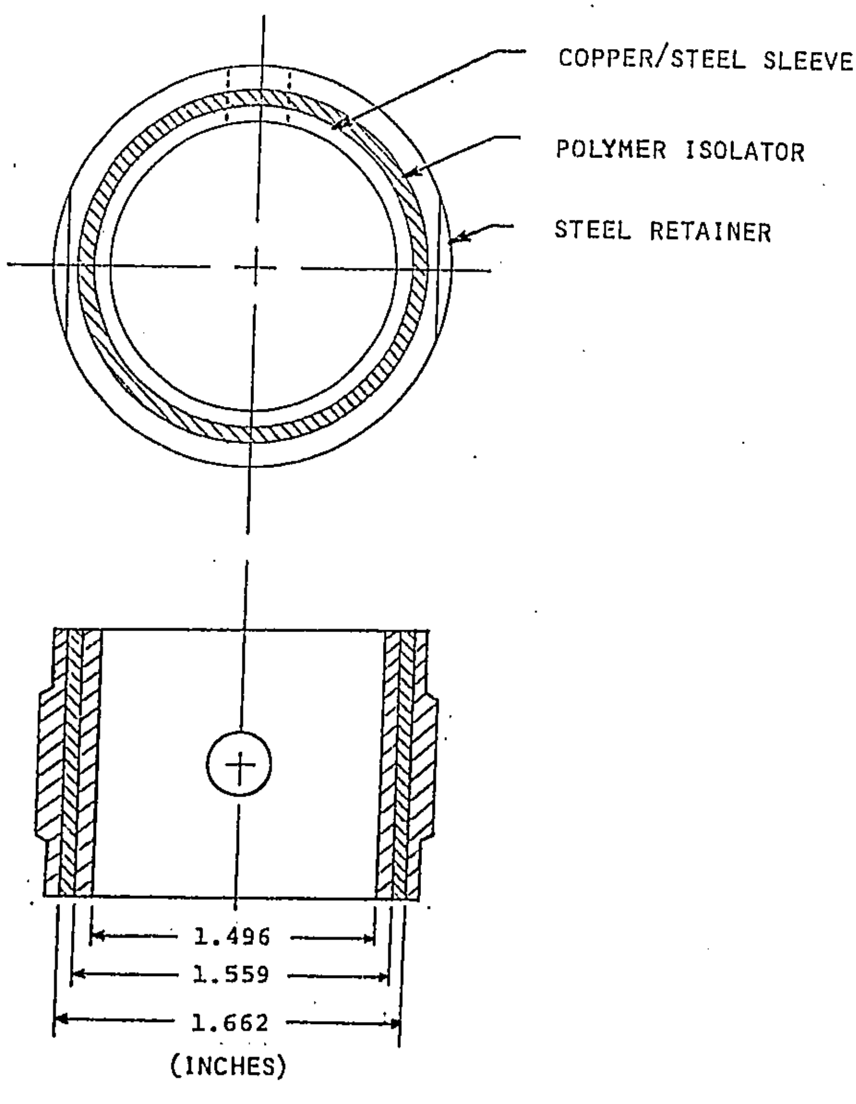

Figure 30. Resilient Cam Bearing

B. Piston-Slap Noise Reduction

The DDA 6V-92TTA engine uses cast iron cylinder liners. A slip-fit is maintained between the lower section of the liner and the block. A lip on the upper rim of the liner is clamped between the head and the block. A high compressive load is applied in order to maintain a combustion seal. The liners are wet liners in that they are directly cooled by the engine coolant.

The piston-slap noise is a result of the impact of the piston on the major thrust side of the liner shortly after TDC. The forces generated by this impact can be reduced by decreasing the clearance between the piston and the liner so that the distance traveled by the piston as it crosses from one side of the liner to the other is reduced. This technique has been implemented by DDA and others using tin-coated tight pistons. We have incorporated tight pistons in our first experimental engine and have found that the combined effect of tight pistons and the injector noise control design described in the previous section results in a 3.5 dB reduction of A-weighted noise. However, the use of tight-pistons in truck engines may not be acceptable because of the very long engine life required and because of requirements for duration of engine operation after loss of coolant.

The use of off-centered piston pins has also been suggested as a means of noise control. However, there are data showing that this technique has an adverse effect on fuel economy, emissions, and wear [4].

The use of piston coatings and insert pads to reduce the force generated by piston impact has also been shown to have benefits in reducing piston slap noise. However, the limited life of the materials used eliminate this technique for truck engines [4].

The piston-slap noise model developed in the previous section indicates that a reduction in noise can be achieved by changes to the cylinder liner. Increasing the thickness of the liner should decrease its mobility and thereby reduce the vibratory power input to the liner. Based on an analytical prediction of liner mobility a doubling of the liner wall thickness should result in a 3 dB reduction in drive point mobility for the liner and a corresponding 3 dB reduction in power input. Within the requirement to maintain a water passage between the liner and the block and within the limit on the maximum block bore due to cylinder to cylinder spacing, the upper section of the liner could be increased from 0.149 in. to 0.362 in. - a factor of 2.4 increase which should result in a 4 dB decrease in power input to the liner.

A further reduction of piston-slap noise can be achieved by reducing the vibration transmission from the liner to the block. This is done by lowering the support so that the liner is midway supported by the block rather than being supported at the upper lip of the liner where piston impact occurs. By increasing the length of the transmission path from the point of piston impact on the liner to the support point the vibration transmission from the liner to the block should be reduced. The transmission path through the head is also altered by making the connection more of a simple-support than a clamped joint. The final liner design is shown in Figure 31 where it can be compared with the original design. It can be seen from Figure 31 that the modified liner has a thicker wall section at the bottom of the piston impact and a lower support point in the block as compared with the original design. This lower support point is in a stiffer region of the block than in the original design which also contributes to a reduction in the vibration transmission from the liner to the block.

To verify the effectiveness of the design in reducing piston-slap noise, a prototype liner was constructed and fitted into one cylinder of a modified non-running block. Drive point and transfer mobility measurements were taken on both a standard and the modified liner. These measurements, shown in Figure 32, support the design and show that a significant reduction in piston-slap noise is expected at high frequencies. The increased mobility at lower frequencies is not of immediate concern because piston-slap is not the dominant source at these frequencies.

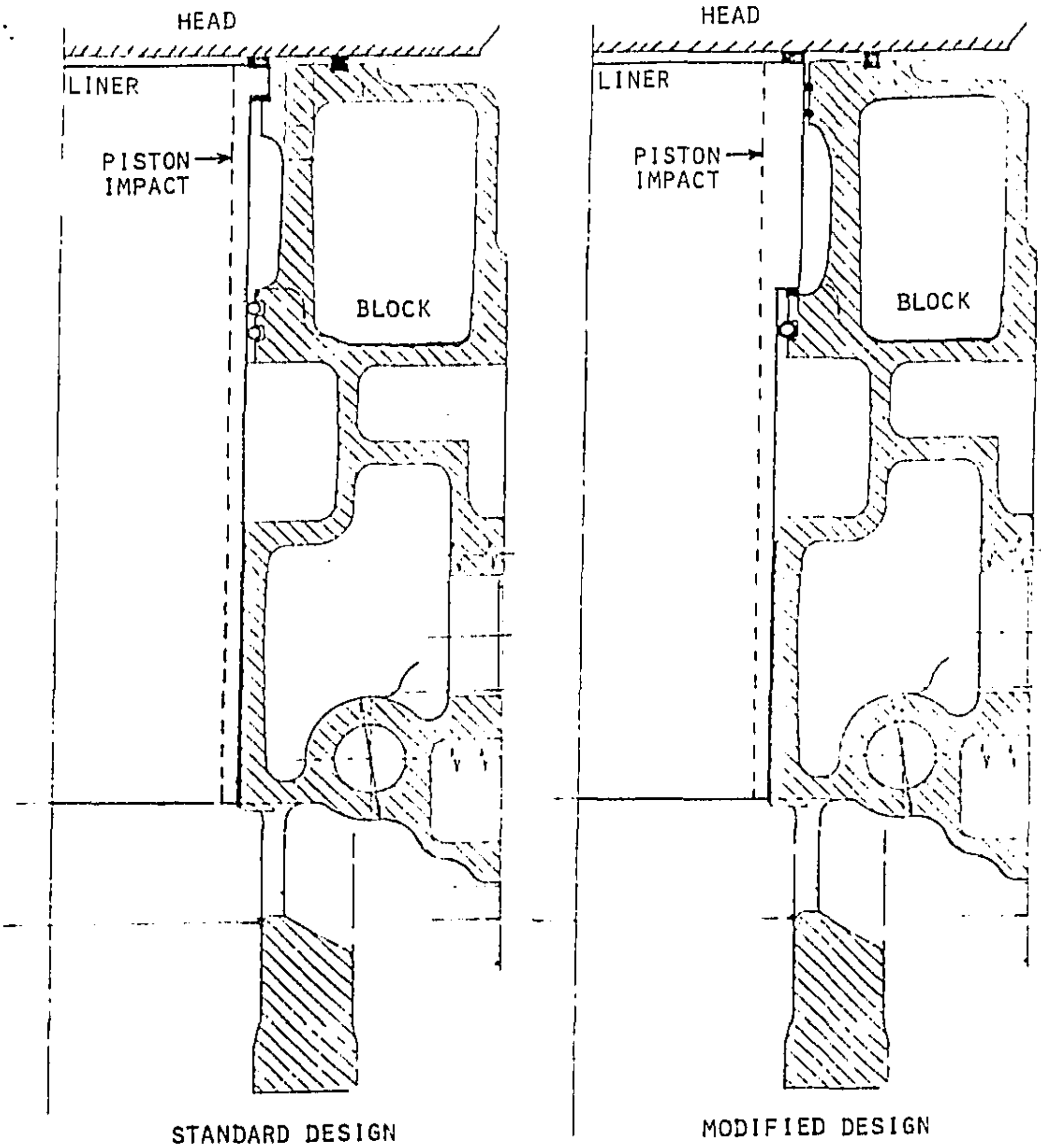

Figure 31. Piston-Slap Noise Reduction Design

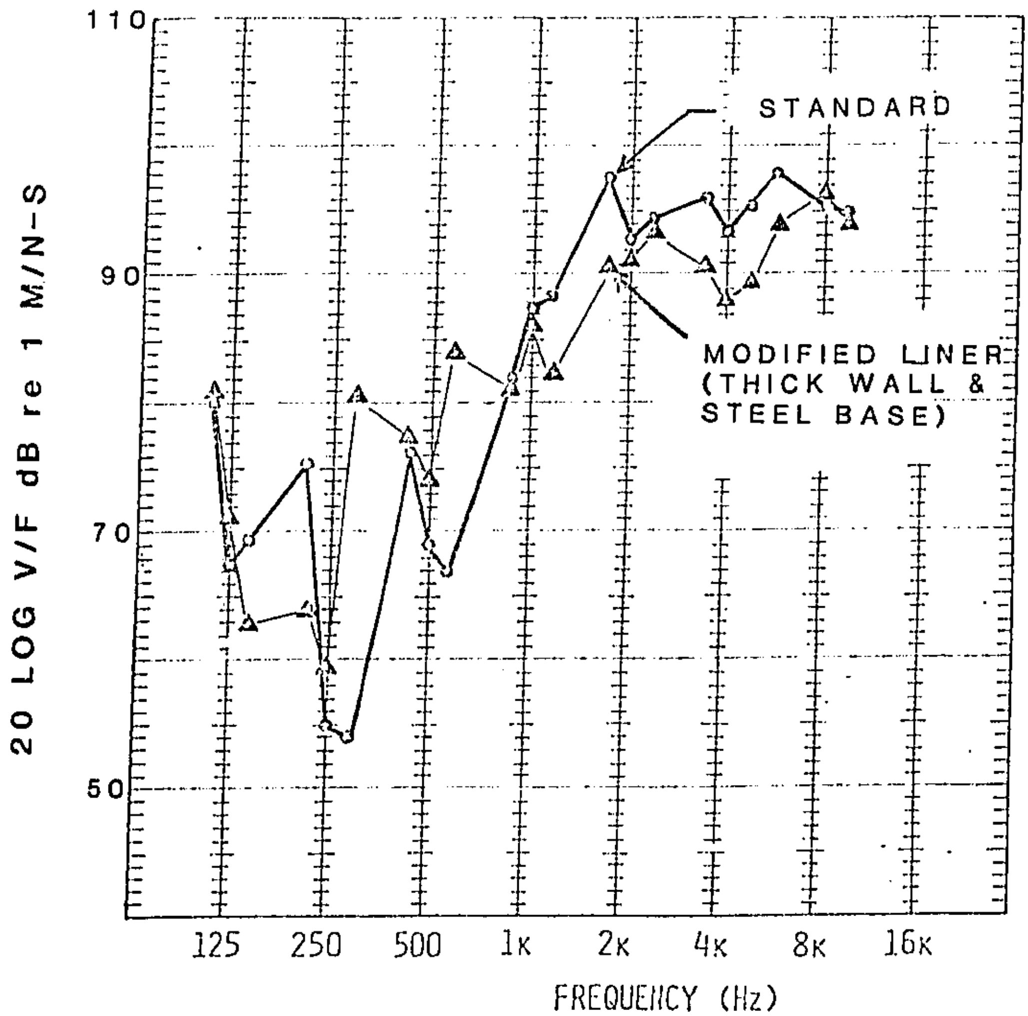

Figure 32. Measured Transfer Mobility: TDC Cylinder Liner to Block Side

V. CONSTRUCTION OF A DEMONSTRATION ENGINE

A. Demonstration Engine #1

To verify the validity of the injector noise control design we decided to incorporate it in a first demonstration engine that did not incorporate the more complicated piston slap-liner design change. Design requirements for the injection system were sent to DDA. Starting with a standard block, they bored the bearing supports to accommodate the larger diameter resilient bearings and constructed the required bearings. A copper/steel sleeve was used as a bushing to give the same wear characteristics as the standard bearings. Nylon 6 x 6 was used as the polymer isolator. Grooves in the bushing for oil passage were hand ground in the bushing.

The pistons of this first demonstration engine were replaced with tin-coated, low clearance pistons to reduce piston-slap noise. Although the practicality of tight-fitting pistons as a long term solution to reduce piston slap noise is open to question because of durability, their effectiveness in reducing noise has been shown on other similar engines [4]. By reducing piston slap, we were better able to evaluate the effectiveness of the resilient bearing design.

The first demonstration engine was shipped from Detroit to our test facility in Cambridge where we carried out noise and vibration measurements. Tests were stopped after approximately 15 minutes when an oil seal blew. The engine was disassembled for inspection and it was found that the bushings had fused to the camshaft. We concluded that the hand-ground oil passages were not adequate to maintain oil flow and caused bearing failure.

To proceed with the development we refinished the camshaft, constructed new resilient bearings, and reassembled the engine in our test facility. The engine was run up to full load at 1950 RPM and noise measurements were taken. During the measurements we noticed that the high frequency noise level was slowly increasing. Tests were stopped and the engine was disassembled for inspection. Again one of the camshaft bearings had failed. We concluded that the nylon 6 x 6 could not withstand the engine environment. However, the noise measurements taken for this first demonstration engine show a reduction in radiated noise (seen in Figure 33) and support the validity of the resilient bearing design for noise reduction.

B. Demonstration Engine #2

In the development process of the second demonstration engine, two major tasks were undertaken. The first task was to improve the durability of the resilient bearings used for the camshaft and idler gear so that they would sustain the engine environment. The second task was to design and fabricate modified liners based on the criteria established in the transfer path analysis of piston slap while maintaining the performance required by other design criteria. The following paragraphs describe the approach taken to perform these two tasks.

Resilient Bearing Modifications

Based on the information generated by the running tests of the first demonstration engine, it appeared that three problems needed to be resolved in the resilient bearing designs.

1) Since several different materials having different values of thermal expansion and creep rate were used, it was necessary to determine more carefully the clearances in the bearing design.

2) In order to improve the dynamic behavior of the bearing isolation it became necessary to go to a design which was not limited by the present size of the bearing housings and block supports.

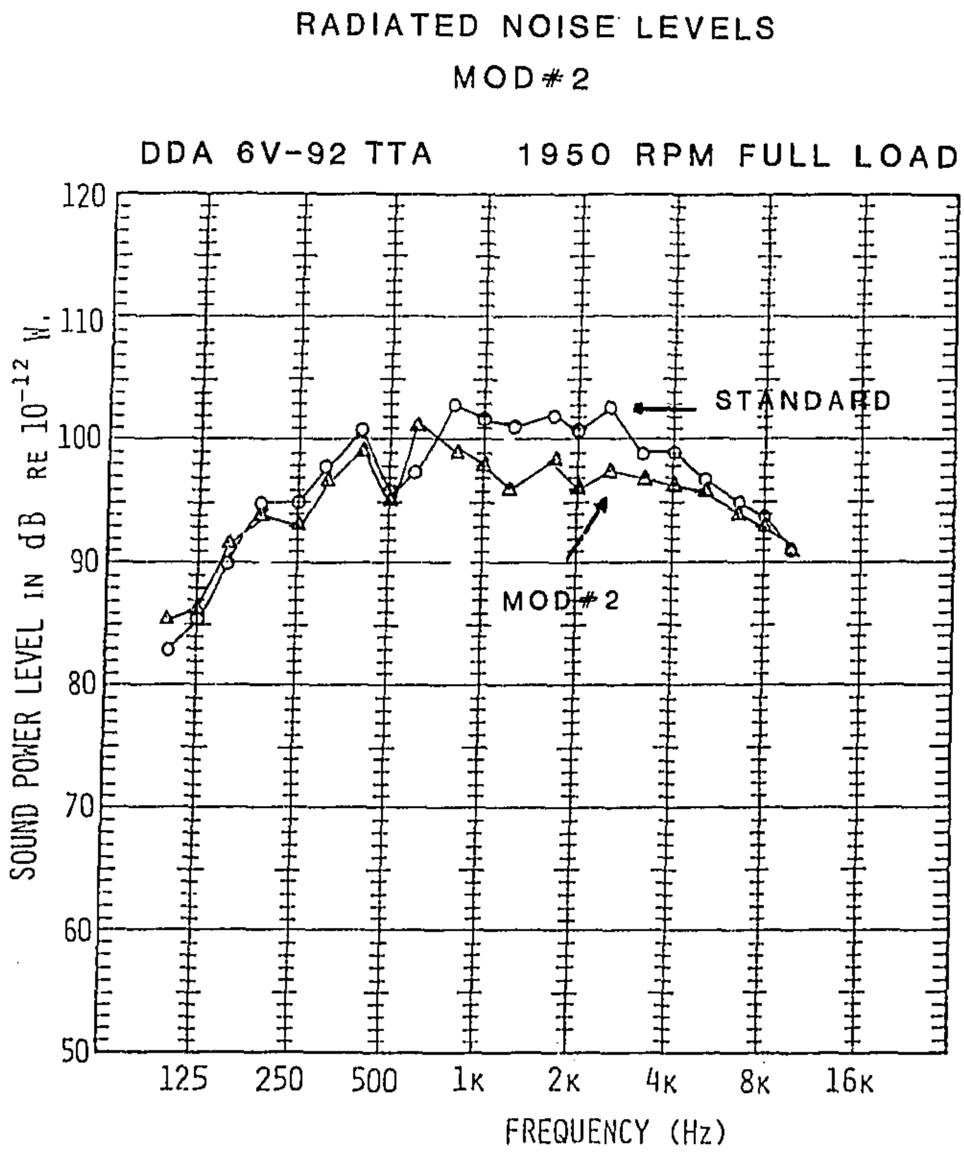

Figure 33. Overall Noise Radiation of Demonstration Engine #1 vs. Standard DDA 6V-92TTA Engine

3) It was necessary to find a better plastic for the second demonstration engine than the nylon 6 x 6 used in the first demonstration engine.

To solve the above problems, a more detailed evaluation of the structural requirements of the bearing was made in order to establish more exactly the bearing tangential loads, rotational loads, temperatures, and lubrication requirements. After this assessment a material search was carried out to find a material meeting these requirements. A polymer was chosen.

Following the selection of the material, a new design was developed which incorporated the necessary clearances and load bearing capabilities associated with the parameters of this material. Included in this design was a completely new bearing housing with enlarged cam bearing block boxes.

The new bearing housings and bearings had to be constructed from raw stock since no castings were available in the right size. This work was performed by a machine shop in the Cambridge area (RB Machine). The increased cam bearing bores were machined by DDA along with the other modifications to the block required for the second demonstration engine.

Modified Liner Design and Fabrication

Based on the conclusions derived from the transfer path analysis of piston slap, a modified liner design was developed to reduce the transmission of vibration from the liner to the block (see Figure 31). This design incorporates a thicker liner wall in the location of the piston impact and a lower attachment point of the liner to the block.

In order to accommodate this modified liner, several changes needed to be made to the engine block. First, the block bore had to be increased in order for the thicker liner walls to fit in. Second, a new mounting location for the liner had to be machined just above the air box in the block. Third, as a result of the lower mounting points of the liner in the block, the head bolts had to be extended below this point in order to prevent the combustion loads from creating tensile stresses in the block material. (Previous experience with cast iron blocks indicates that even moderate levels of tensile stresses will cause fatigue cracks to occur).

Fourth, as a result of lowering the head bolts, another problem arose. The water passages from the bottom water jacket to the top water jacket are located right in line with the axis of the head bolts. By extending the head bolts the new thread locations blocked these water passages. In order to overcome this problem, an external water passage was designed to connect the lower and upper water jackets. The lower block was accessed through the freeze plugs adjacent to each liner, and the upper block was accessed through a hole drilled through the side walls of the block. These water passages were designed to have the same cross-sectional area as in the standard design in order to handle the same flow rates.

Fifth, as a result of the external water passage design another problem arose with the cooling of the upper liner section. In the standard engine design, the upper water jacket is supplied by four smaller passages spaced evenly around the circumference of the liner. But in the modified design the upper block has only one water supply on the external block wall side. This creates the possibility of uneven water flow and cooling patterns resulting in undesirable temperature distributions in the liner and distortions in the cylinder shape. In order to solve this problem, the upper water jacket was designed to have a changing radius which enabled the control of the water flow to achieve a sufficiently uniform water flow around the liner and maintain a uniform heat rejection from the liner to the block.

After the design was completed, four blocks were removed from the production line at DDA at a specified location where most of the standard machining had been done except for the liner and camshaft bores. Two of these blocks were machined further by DDA according to the design specifications for the new liner and camshaft bores. In the process one of the blocks was rejected as a result of a machining error. All four blocks were then purchased from DDA.

Simultaneously, the new liner design was given to a centrifugal casting company (Dana Perfect Circle Co.) which produced the modified castings. They were then transported to RB Machine for final machining to specifications. The new cam bearings were also machined by RB Machine.

After all the parts were complete, the second demonstration engine was assembled by CC and tested. Figure 33 illustrates the overall noise radiation of the first demonstration engine versus a standard 6V-92TTA engine measured with the engine running at 1950 RPM, full load, after several hours of operation. Figure 34 depicts the overall noise radiation of the second demonstration engine versus a standard 6V-92TTA measured under the same conditions.

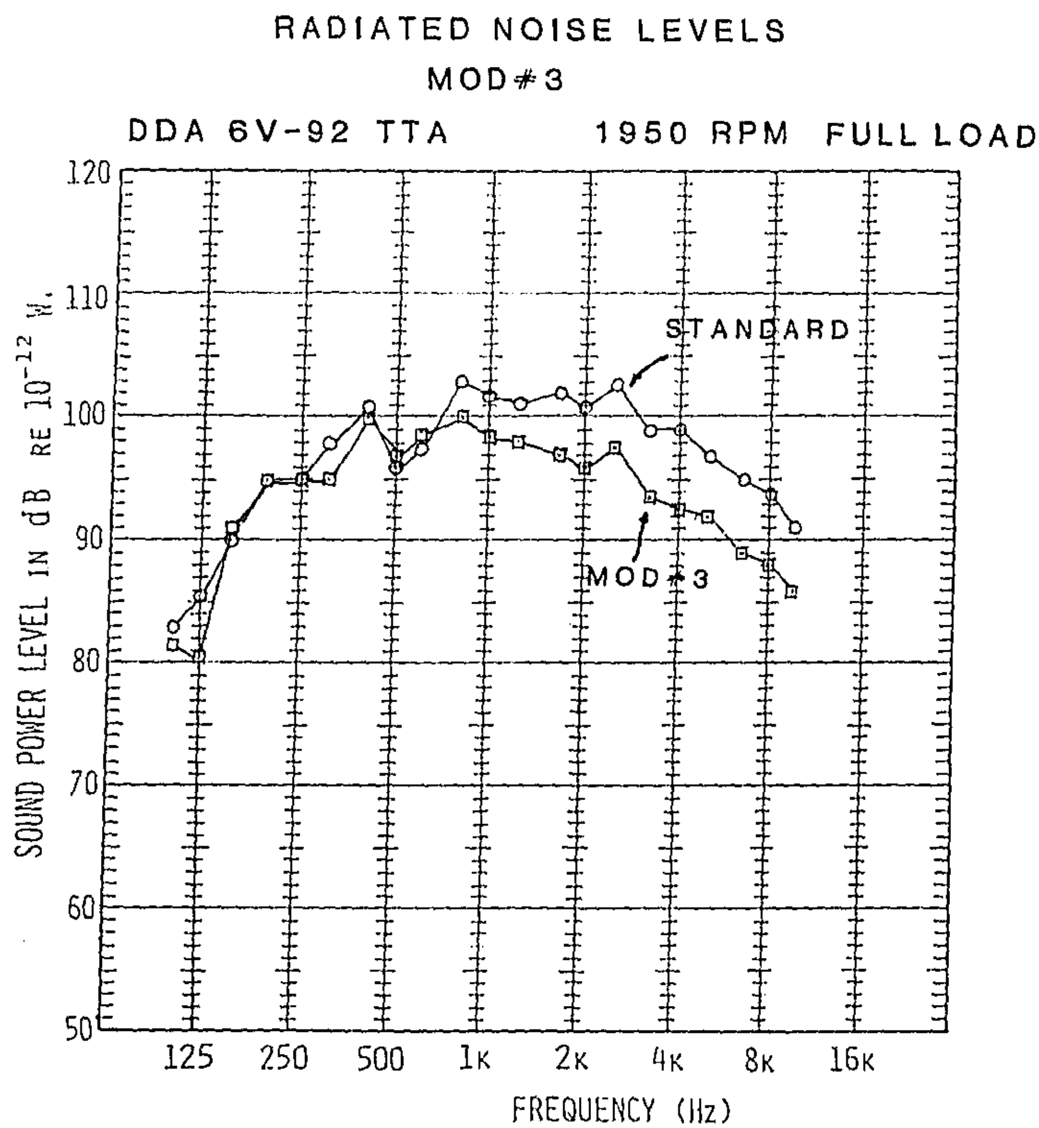

Figure 34. Overall Noise Radiation of Demonstration Engine #2 vs. Standard DDA 6V-92TTA Engine

VI. SUMMARY OF RESULTS

The noise generation model developed for the DDA 6V-92TTA heavy duty truck engine indicates that injectors and piston-slap are the two major noise sources. Noise reduction designs that reduce the noise from these sources have been developed and implemented in two running demonstration engines. These designs do not alter the basic combustion or fuel injection processes, nor do they influence piston/liner wear. They involve changes to the vibration transmission paths.

A first demonstration engine was constructed to demonstrate the effectiveness of an injector noise reduction design and tight pistons. The tight pistons are known to reduce piston-slap noise but cannot be used in truck engines because of limited life. The injector noise reduction design included increased diameter push-rods, resilient camshaft bearings, and resilient cam gear bearings.

The first demonstration engine was assembled and run at full RPM and power. A 3 dB reduction in overall engine noise was obtained, which supports the validity of the injector noise control design. The resilient bearings in the experimental engine failed after a short time. A new material for the bearing was selected and used in the demonstration engine, which was later constructed.

A second demonstration engine was constructed which incorporated both the injector noise control design and the piston-slap noise reduction design. The design to reduce piston-slap included thicker cylinder liners and redesign of the liner support.

This demonstration engine has been assembled and run at full power for several hours. A 4 dB reduction in engine noise was obtained and no engine failures have occurred. Figures 33 and 34 compare the overall radiated noise spectra for the demonstration engines and the original engine. The high frequency noise has been significantly reduced. Piston-slap and injectors are no longer the major sources. It is believed that further noise reductions will require the treatment of the noise produced by the Roots blower.

VII. CONCLUSIONS

This report has described a demonstration of certain methods of reducing noise from the DDA 6V-92TTA heavy duty truck diesel engine. The specific noise reduction designs have resulted in a 4 dB reduction in A-weighted noise without compromises to fuel economy or emissions. The effect of the design changes on engine life have not been determined because of the limited running time for the demonstration engine.

It should be noted that the design modifications for the 6V-92TTA have been specifically designed for that engine. Other engines may have different sources and vibration transmission paths so that different noise control techniques may be required.

The major conclusion of the study is that engine noise can be significantly reduced through changes to the vibration transmission paths from internal sources to the radiating surfaces of the engine block. The study has demonstrated the value of an improved technique for identifying vibration sources and the significant paths of vibration transmission. This technique allows new noise control designs to be evaluated on non-running engines. Elimination of the need to implement the design in a running engine in order to evaluate its effectiveness results in a major saving of effort and money.

VIII. REFERENCES

[1] "A Baseline Study of the Parameters Affecting Diesel Engine Intake and Exhaust Silencer Design," Final Report to DOT, prepared by the Donaldson Co., Inc., April 1973.

[2] Chung, J.Y., J. Pope and D.A. Feldmaier, "Application of Acoustic Intensity Measurements to Engine Noise Evaluation," SAE Paper No. 790502, 1979.

[3] DeJong, R.G. and J.E. Manning, "Modeling of Vibration Transmission in Engines to Achieve Noise Reduction," SAE Paper No. 790360, 1979.

[4] Rohrle, Manfred D., "Affecting Diesel Engine Noise by the Piston," SAE Paper No. 750799, 1975.

[5] Lyon, R.H. and DeJong, R.G., "Designing Diesel Engines for Reduced Noise," Proceedings of the EPA-University Seminar, Purdue Univ., Oct. 18-20, 1976.

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

1. REPORT NO.	2.	3. RECIPIENT'S ACCESSION NO.
EPA 550/9-82-336		
4. TITLE AND SUBTITLE Demonstration of Noise Control for the DDA 6V-92TTA Heavy Duty Truck Diesel Engine		5. REPORT DATE May 1982 6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S) Richard G. DeJong, Natan E. Parsons, and Jerome E. Manning		8. PERFORMING ORGANIZATION REPORT NO.
9. PERFORMING ORGANIZATION NAME AND ADDRESS Cambridge Collaborative, Inc. P.O. Box 74, Kendall Square Cambridge, MA 02142		10. PROGRAM ELEMENT NO. 11. CONTRACT/GRANT NO. 68-01-4737
12. SPONSORING AGENCY NAME AND ADDRESS Office of Noise Abatement U.S. Environmental Protection Agency Washington, D.C. 20460		13. TYPE OF REPORT AND PERIOD COVERED Final 14. SPONSORING AGENCY CODE EPA-ONA
15. SUPPLEMENTARY NOTES		

16. ABSTRACT

This report presents the results of an engineering study to design and demonstrate methods of reducing diesel engine block vibration and radiated noise. The Detroit Diesel Allison 6V-92TTA heavy duty diesel truck engine was selected for the demonstration. This engine was structurally modified to reduce vibration and noise due to the unit injectors and piston-slap. The modifications were designed so that the noise reduction was achieved without degrading engine performance, fuel economy, or emissions. A 4 dBA reduction of overall engine noise reduction was demonstrated.

17. KEY WORDS AND DOCUMENT ANALYSIS		
a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Noise Vibration Engine Truck Diesel		
18. DISTRIBUTION STATEMENT Release to Public	19. SECURITY CLASS (This Report) Unclassified	21. NO. OF PAGES 68
	20. SECURITY CLASS (This page) Unclassified	22. PRICE

EPA Form 2220-1 (9-73)